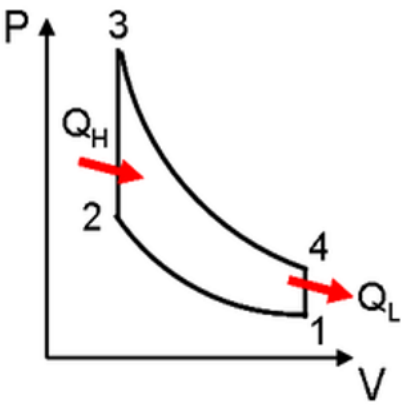
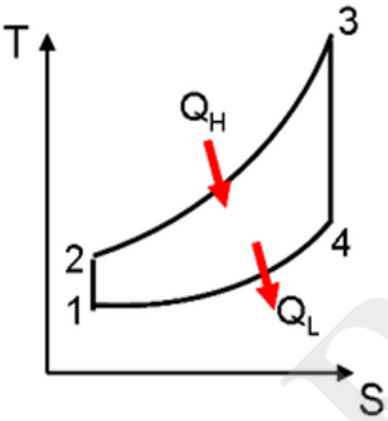
KONGUNADU COLLEGE OF ENGINEERING AND TECHNOLOGY

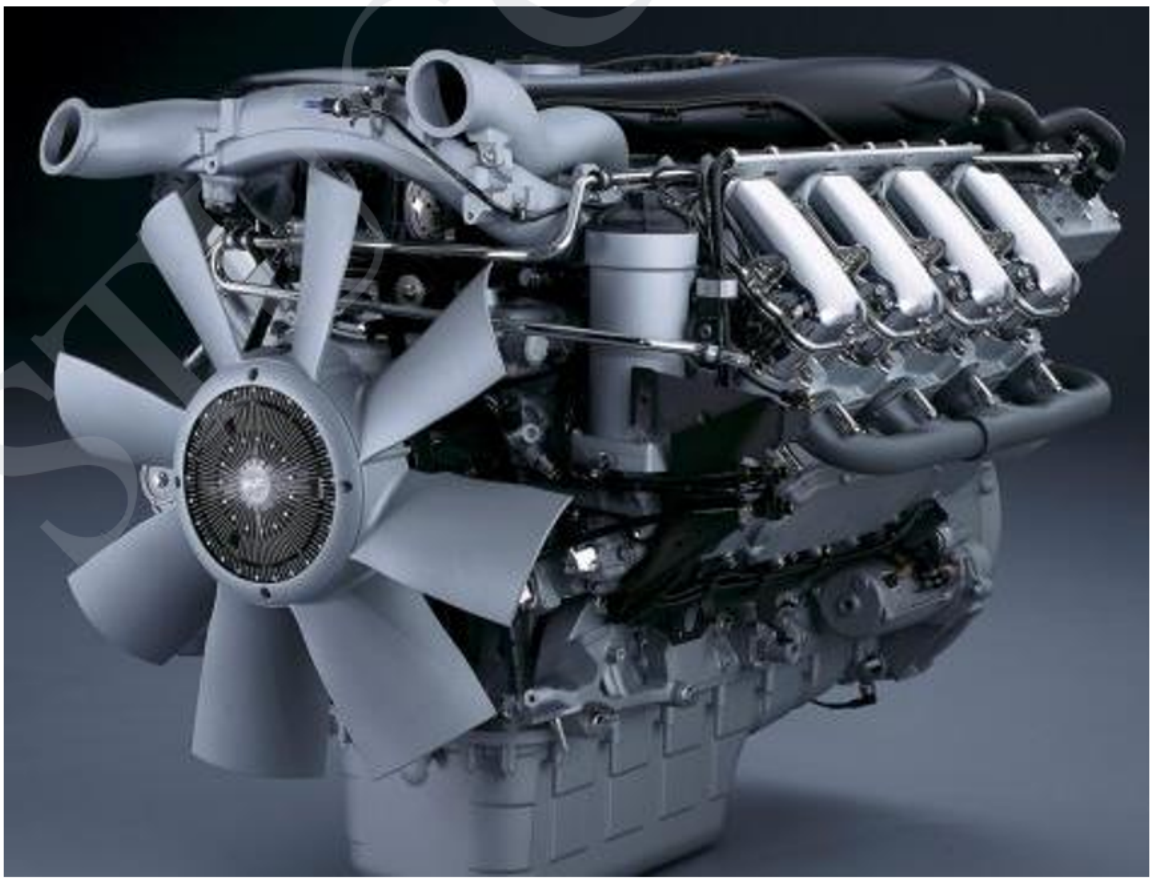
DEPARTMENT OF MECHANICAL ENGINEERING

ME8493-Thermal Engineering I

UNIT I - GAS AND STEAM POWER CYCLES







CLASSIFICATION OF CYCLES:

The purpose of a thermodynamic cycle is either to produce power, or to produce refrigeration/pumping of heat. Therefore, the cycles are broadly classified as follows:

(a) Heat engine or power cycles.

(b) Refrigeration/heat pump cycles.

The following are illustrations of heat engines operating on open cycle:

- Petrol and diesel engines in which the air and fuel are taken into the engine from a fuel tank and products of combustion are exhausted into the atmosphere.
- Steam locomotives in which the water is taken in the boiler from a tank and steam is exhausted into the atmosphere.

The power cycles are accordingly classified into two groups as:

- Vapour power cycles in which the working fluid undergoes a phase change during the cyclic process.
- Gas power cycles in which the working fluid does not undergo any phase change.

AIR STANDARD CYCLE:

Assumption made in Air standard cycle.

(i) Air is the working fluid and it obeys the perfect gas laws

(ii) The engine operates in a closed cycle. The cylinder is filled with constant amount of working substance and the same fluid is used repeatedly and hence mass remains constant.

(iii) The working fluid is homogeneous throughout at all times and no chemical reaction takes place, inside the cylinder.

(iv) The compression and expansion processes are assumed to be adiabatic.

(v) The values of specific heat (C_p and C_v) of the working fluid remains constant.

(vi) All processes are internally reversible and no mechanical or frictional losses to occur throughout the process.

(vii) Combustion is replaced by heat addition process and exhaust is replaced by heat rejection process.

Air standard efficiency

It is defined as the efficiency produced by the ideal engine with air as a working medium.

Compression ratio

It is defined as the ratio of the stroke volume of cylinder to the clearance volume

Mean effective pressure

It is defined as the average pressure acting on the piston during the entire power stroke that would produce the same amount of net work output during the actual cycle. It is also defined as the ratio of work-done per cycle to swept volume.

Clearance Volume

It is the minimum volume occupied by the fluid in the cylinder when the piston reaches the top dead centre position.

When compression ratio is kept constant, what is the effect of cut-off ratio on the efficiency of diesel cycle

The diesel cycle efficiency decreases with increase in cut-off ratio at constant compression ratio.

Effect of regeneration in the efficiency of Brayton /Joule cycle

The efficiency of Brayton cycle is increased by regeneration. The large quantity of heat energy possessed by the exhaust gases leaving the turbine will be utilized to heat up the air leaving the compressor. This heating is done in a heat exchanger called a regenerator.

Improve the efficiency of Gas turbine cycle

(i) Reheating (ii) Regeneration (iii) Inter cooling (iv) Combination of the above three

Deviation of Actual Cycles from Air Standard Cycles

The actual Otto and Diesel engines show marked deviations form the air-standard cycles described above. Above fig shows p-v-diagram for a high-speed diesel engine would be very similar in appearance. The main differences between the actual and theoretical cycles are as follows.

- Compression and expansion are not friction less adiabatic processes. A Certain amount of friction is always present and there is considerable heat transfer between the gases and cylinder wall.
- Combustion does not occur either at constant volume or at constant pressure.
- The thermodynamics properties of the gases after combustion are different than those of the fuel-air mixture before combustion..
- The combustion may be incomplete.
- The specific heats of the working fluid are not constant but increases with temperature.
- The cylinder pressure during exhaust process is higher than the atmosphere. As a result, more work has to be done by the piston on the gases to expel them out of the cylinder, than work done by the gases on the piston during the intake stroke. This difference in work, called pumping work, is represented by the pumping loop shown by hatched area. Note that this work is negative and represents loss of work called pumping loss.

Differentiate any four major difference between otto and diesel cycle.

S.NO.	OTTO CYCLE	DIESEL CYCLE
1	Constant volume cycle	Constant pressure cycle
2	Heat is added at constant volume process	Heat is added at constant pressure process
3	$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}}$	$\eta_{Diesel} = 1 - \frac{1}{\gamma . r^{\gamma-1}} . \frac{r_c^{\gamma} - 1}{r_c - 1}$
4	Efficiency is high	Efficiency is low

RANKINE CYCLE:

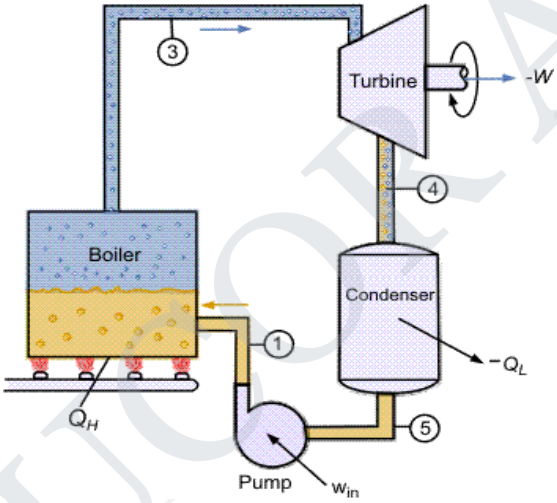
A simple Rankine cycle has the following processes for the boiler, turbine, condenser and pump of a steam power cycle:

Process 1–2: Isentropic expansion of the working fluid through the turbine from saturated vapour at state 1 to the condenser pressure.

Process 2–3: Heat transfer from the working fluid as it flows at constant pressure through the condenser with saturated liquid at state 3.

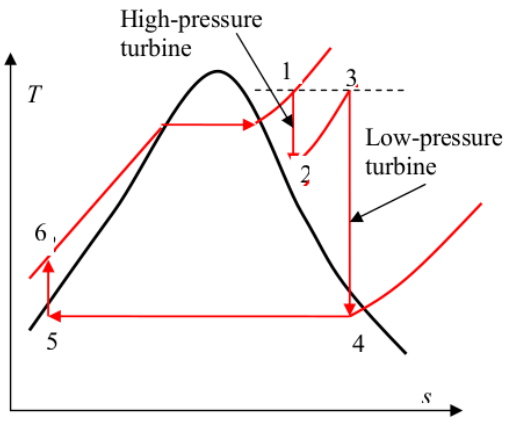
Process 3–4: Isentropic compression in the pump to state 4 in the compressed liquid region.

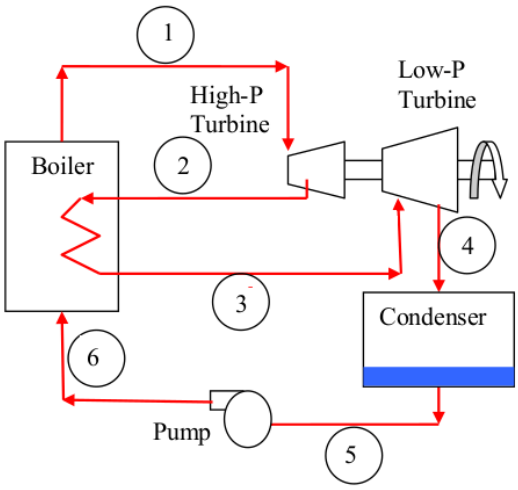
Process 4–1: Heat transfer to the working fluid as it flows at constant pressure through the boiler to complete the cycle.



REHEAT RANKINE CYCLE:

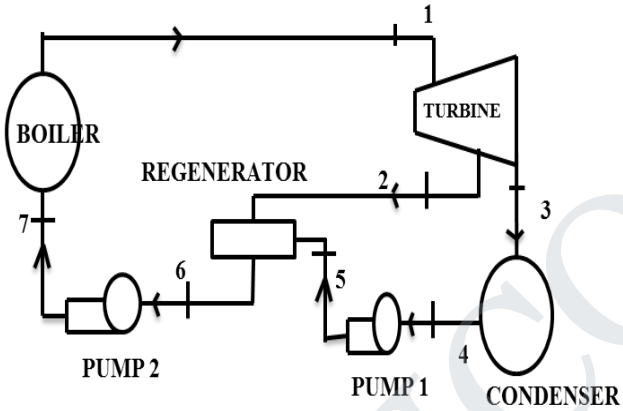
Increasing the boiler pressure can increase the thermal efficiency of the Rankine cycle, but it also increases the moisture content at the exit of the turbine to an unacceptable level. To correct this side effect, the simple Rankine cycle is modified with a reheat process. The schematic of an ideal reheat Rankine cycle with its T-s diagram. In this reheat cycle, steam is expanded isentropically to an intermediate pressure in a high-pressure turbine (stage I) and sent back to the boiler, where it is reheated at constant pressure to the inlet temperature of the high-pressure turbine. Then the steam is sent to a low-pressure turbine and expands to the condenser pressure (stage II) . The total heat input and total work output is

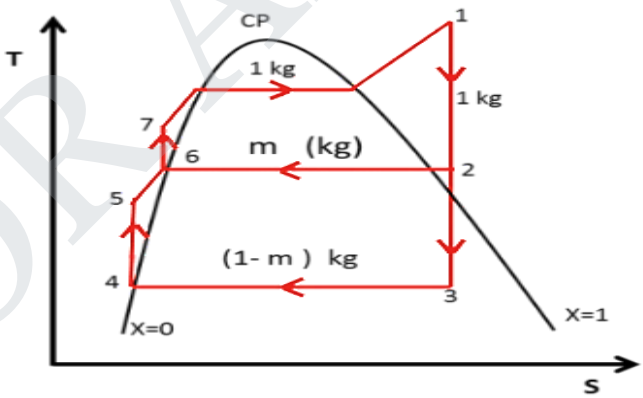




REGENERATIVE RANKINE CYCLE:

In regenerative Rankine cycle, some amount of steam is bled off during expansion in the turbine and mixed with feed water before it enters the boiler to reduce the heat input. In the below both reheat and regeneration (Two feed water heaters) are incorporated.





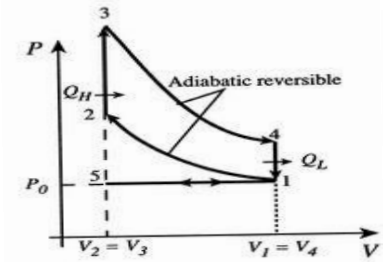
RANKINE CYCLE IS MODIFIED?

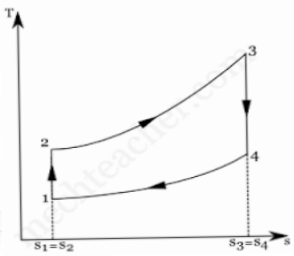
The work obtained at the end of the expansion is very less. The work is too inadequate to overcome the friction. Therefore the adiabatic expansion is terminated at the point before the end of the expansion in the turbine and pressure decreases suddenly, while the volume remains constant.

IMPROVE THERMAL EFFICIENCY OF THE RANKINE CYCLE.

- ✓ Reheating of steam
- ✓ Regenerative feed water heating
- ✓ By water extraction
- ✓ Using binary vapour

Otto Cycle





Compression Ratio (r_c) $= \frac{v_1}{v_2} = \frac{v_4}{v_3}$

Pressure Ratio (r_p) $= \frac{P_3}{P_2} = \frac{P_4}{P_1}$

1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1}$, $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma}$

2-3 Process: Constant volume heat addition

$Q_{2-3} = c_v(T_3 - T_2)$ **(kJ / kg)** , $\frac{P_3}{P_2} = \frac{T_3}{T_2}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = r_c^{\gamma-1}$, $\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = r_c^{\gamma-1}$

4-1 Process: Constant Volume heat rejection

$Q_{4-1} = c_v(T_4 - T_1)$ **(kJ / kg)**

Work done

$W = Q_{2-3} - Q_{4-1}$ **(kJ / kg)**

Efficiency

$$\eta = \frac{\text{Work done}}{\text{Heat Supplied}} = \frac{W}{Q_{2-3}}$$

$$\eta = 1 - \frac{1}{(r_c)^{\gamma-1}}$$

Mean Effective Pressure

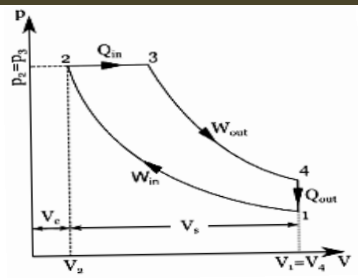
$$P_m = \frac{P_1 r_c\left[(r_c^{\gamma-1} - 1)(r_p - 1)\right]}{(\gamma - 1)(r_c - 1)}$$

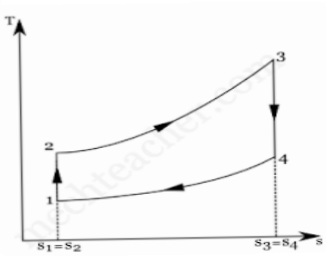
$$P_m = \frac{m \times W}{V_S} = \frac{\text{Work done}}{V_1 - V_2} = \frac{kg \times (kJ/kg)}{m^3}$$

Note:

1. Find V_1 if not given, from $p_1V_1 = mRT_1$
2. Assume per kg of mass if not given mass
3. Find V_2 if not given, from Compression ratio $(r_c) = \frac{v_1}{v_2}$

Diesel Cycle





Compression Ratio (r_c) $= \frac{v_1}{v_2}$

Cut-off Ratio (ρ) $= \frac{v_3}{v_2}$

Expansion Ratio (r_e) $= \frac{v_4}{v_3} = \frac{r_c}{\rho}$

1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1}$, $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c$

2-3 Process: Constant pressure heat addition

$Q_{2-3} = c_p(T_3 - T_2)$ **(kJ / kg)**, $\frac{v_3}{v_2} = \frac{T_3}{T_2} = \rho$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1}$, $\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma}$

4-1 Process: Constant Volume heat rejection

$Q_{4-1} = c_v(T_4 - T_1)$ **(kJ / kg)**

Work done

$W = Q_{2-3} - Q_{4-1}$ **(kJ / kg)**

Efficiency

$$\eta = \frac{\text{Work done}}{\text{Heat Supplied}} = \frac{W}{Q_{2-3}}$$

$$\eta = 1 - \frac{1}{\gamma(r_c)^{\gamma-1}}\left[\frac{\rho^{\gamma} - 1}{\rho - 1}\right]$$

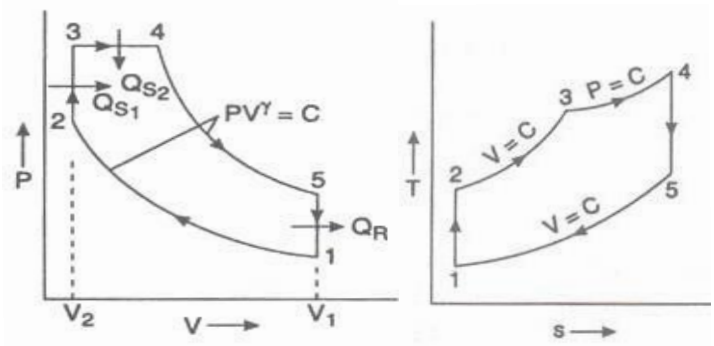
Mean Effective Pressure

$$\frac{\gamma[\gamma(\rho - 1) - r_c^{\ 1-\gamma}(\rho^{\gamma} - 1)]}{(\gamma - 1)(r_c - 1)}$$

Note:

1. Use above formulae if mass is not given or mention per kg or Q is given in kJ/kg
2. If mass is given, multiple with heat supplied and rejected. For ex: $Q_{4-1} = m\, c_v(T_4 - T_1)$
3. In process 2-3, any one parameter will be given which is ρ, T_3, Q_{2-3}

Duel Cycle



Compression Ratio (r_c) $= \frac{v_1}{v_2}$

Cut-off Ratio (ρ) $= \frac{v_4}{v_3}$

pressure or explosion ratio $= \frac{p_3}{p_2}$

Expansion Ratio (r_p) $= \frac{v_5}{v_4} = \frac{r_c}{\rho}$

1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1}, \qquad \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma}$$

2-3 Process: Constant volume heat addition

$$Q_{2-3} = c_v(T_3 - T_2) \ \left(\frac{kJ}{kg}\right), \qquad \frac{p_3}{p_2} = \frac{T_3}{T_2} = r_p$$

3-4 Process: Constant pressure heat addition

$$Q_{3-4} = c_p(T_4 - T_3) \ \left(\frac{kJ}{kg}\right), \qquad \frac{v_4}{v_3} = \frac{T_4}{T_3} = \rho, \ P_3 = P_4$$

4-5 Process: Adiabatic Expansion process

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1}, \qquad \frac{p_4}{p_5} = \left(\frac{v_5}{v_4}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma}$$

5-1 Process: Constant Volume heat rejection

$$Q_{5-1} = c_v(T_1 - T_5) \ \left(\frac{kJ}{kg}\right), \qquad \frac{p_5}{p_1} = \frac{T_5}{T_1} = r_p$$

Work done

$$W = Q_{2-3} + Q_{3-4} - Q_{5-1} \quad \left(\frac{kJ}{kg}\right)$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W}{Q_{2-3} + Q_{3-4}}$$

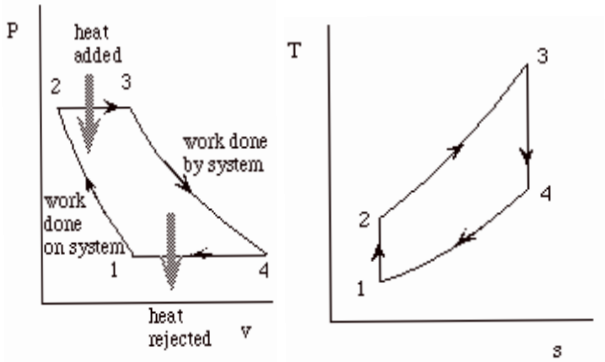
(Or)

$$\eta = 1 - \frac{1}{(r_c)^{\gamma-1}}\left[\frac{r_p\rho^{\gamma} - 1}{(r_p - 1) + r_p\gamma(\rho - 1)}\right]$$

Mean Effective Pressure

$$P_m = \frac{P_1 r_c^{\gamma}\left[r_p\gamma(\rho - 1) + (r_p - 1) - r_c^{\ 1-\gamma}(r_p\rho^{\gamma} - 1)\right]}{(\gamma - 1)(r_c - 1)}$$

Brayton Cycle



1-2 Process: Adiabatic compression process

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

Compressor Work $(W_C) = mc_p(T_2 - T_1)$ **kJ**

2-3 Process: Constant pressure heat addition

$Q_S = mc_p(T_3 - T_2)$ **kJ**

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\ \frac{\gamma-1}{\gamma}}$$

Turbine Work $(W_T) = mc_p(T_3 - T_4)$ **kJ**

4-1 Process: Constant pressure Heat Rejection

$Q_R = mc_p(T_4 - T_1)$ **kJ**

Work done

$W = Q_s - Q_R$ **kJ**

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W}{Q_S}$$

Efficiency

$$\eta = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

Turbine Work Ratio

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

Note: for diesel and dual cycle

Fuel cut off $= \frac{\rho - 1}{r_c - 1}$

For ex: Fuel cut off at 8% of the stroke in the diesel cycle.

$$V_3 - V_2 = 0.08\,(V_1 - V_2) \rightarrow 0.08 = \frac{\rho - 1}{r_c - 1}$$

PART - B

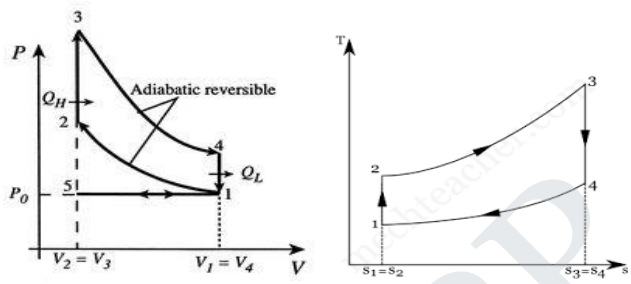
OTTO CYCLE:

1. An engine works on otto cycle. The initial pressure and temperature of the air is 1bar and 40°C. 825 kJ of heat is supplied per kg of air at the end of compression find the temperature and pressure at all salient points if the compression ratio is 6. Also find the efficiency and MEP for the cycle. Assume air as the working fluid & take all ideal conditions.

GIVEN:

$p_1 = 1\text{bar}, T_1 = 40°C + 273$

$Q_s = 825\ \text{kJ/kg}$



COMPRESSION RATIO

$$(r_c) = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 6$$

PRESSURE RATIO

$$(r_p) = \frac{P_3}{P_2} = \frac{P_4}{P_1}$$

1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\gamma-1} \Rightarrow T_2 = r_c^{\gamma-1} \times T_1 = 6^{1.4-1} \times 313 \Rightarrow \mathbf{T_2 = 640.93K}$$

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\gamma} \Rightarrow P_2 = r_c^{\gamma} \times P_1 = 6^{1.4} \times 1 \Rightarrow \mathbf{P_2 = 12.29\ bar}$$

2-3 Process: Constant volume heat addition

$$Q_S = c_v(T_3 - T_2) \Rightarrow 825 = 0.714(T_3 - 640) \Rightarrow \mathbf{T_3 = 1795.46\ K}$$

$$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow P_3 = P_2 \times \frac{T_3}{T_2} = 12.29 \times \frac{1795.46}{640.93} \Rightarrow \mathbf{P_3 = 34.43\ bar}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = r_c^{\gamma-1} \Rightarrow T_4 = \frac{T_3}{r_c^{\gamma-1}} = \frac{1795.46}{6^{1.4-1}} \Rightarrow \mathbf{T_4 = 876.83\ K}$$

$$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = r_c^{\gamma} \Rightarrow P_4 = \frac{P_3}{r_c^{\gamma}} = \frac{34.43}{6^{1.4}} \Rightarrow \mathbf{P_4 = 2.8\ bar}$$

4-1 Process: Constant Volume heat rejection

$$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.714(876.83 - 313) \Rightarrow \mathbf{Q_R = 402.57\ kJ/kg}$$

Work done

$$W = Q_S - Q_R \Rightarrow W = 825 - 402.57 \Rightarrow \mathbf{W = 422.43\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{\text{Work done}}{\text{Heat Supplied}} = \frac{422.43}{825} \Rightarrow \mathbf{\eta = 51.2\%}$$

Mean Effective Pressure

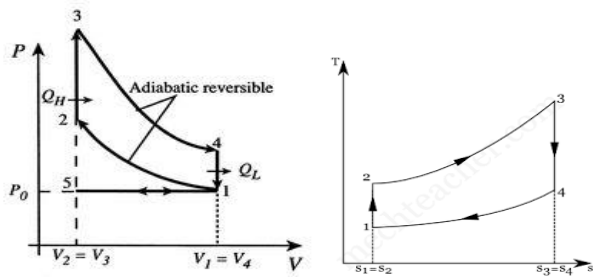
$$P_m = \frac{P_1 r_c[(r_c^{\gamma-1}-1)(r_p-1)]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1\times 6\left[(6^{1.4-1}-1)\left(\frac{34.43}{12.29}-1\right)\right]}{(1.4-1)(6-1)} \Rightarrow \mathbf{P_m = 5.66\ bar}$$

2. An Otto cycle has an compression ratio of 7. The initial pressure and temperature at the beginning of compression stroke is 1 bar and 40o C. The heat supplied is 2510 kJ/kg. Find (i) Maximum temperature and pressure, (ii) Work done per kg of air, (iii) Cycle efficiency and (iv) Mean effective pressure. Assume, Cp and Cv , R and γ suitably.

GIVEN:

$p_1 = 1\text{bar},\ T_1 = 40^\circ\text{C} + 273$

$Q_s = 2510\ \text{kJ/kg}$



COMPRESSION RATIO

$(r_c) \quad = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 7$

PRESSURE RATIO

$(r_p) \quad = \frac{P_3}{P_2} = \frac{P_4}{P_1}$

1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 7^{1.4-1} \times 313 \Rightarrow \mathbf{T_2 = 681.68K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 7^{1.4} \times 1 \Rightarrow \mathbf{P_2 = 15.25bar}$

2-3 Process: Constant volume heat addition

$Q_S = c_v(T_3 - T_2) \Rightarrow 2510 = 0.714(T_3 - 681.68) \Rightarrow \mathbf{T_3 = 4197.08\ K}$

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow P_3 = P_2 \times \frac{T_3}{T_2} = 15.25 \times \frac{4197.08}{681.68} \Rightarrow \mathbf{P_3 = 93.89\ bar}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = r_c^{\gamma-1} \Rightarrow T_4 = \frac{T_3}{r_c^{\ \gamma-1}} = \frac{4197.08}{7^{1.4-1}} \Rightarrow \mathbf{T_4 = 1927.12\ K}$

$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = r_c^{\gamma} \Rightarrow P_4 = \frac{P_3}{r_c^{\ \gamma}} = \frac{93.89}{7^{1.4}} \Rightarrow \mathbf{P_4 = 6.16\ bar}$

4-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.714(1927.12 - 313) \Rightarrow \mathbf{Q_R = 1152.48\ kJ/kg}$

Work done

$W = Q_S - Q_R \Rightarrow W = 2510 - 1152.48 \Rightarrow \mathbf{W = 1357.52\ kJ/kg}$

Thermal Efficiency

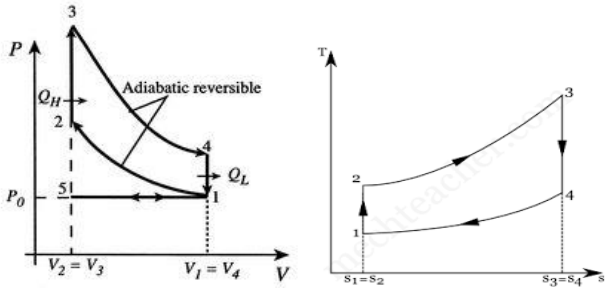
$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{1357.52}{2510} \Rightarrow \boldsymbol{\eta = 54.08\%}$

Mean Effective Pressure

$P_m = \frac{P_1 r_c[(r_c^{\gamma-1}-1)(r_p-1)]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1\times7\left[(7^{1.4-1}-1)\left(\frac{93.89}{15.25}-1\right)\right]}{(1.4-1)(7-1)} \Rightarrow \mathbf{P_m = 18.83\ bar}$

3. **In an Otto cycle air at 17° C and 1 bar is compressed adiabatically until the pressure is 15bar. Heat is added at constant volume until the pressure rises to 40bar. Calculate the air standard efficiency, the compression ratio and the mean effective pressure for the cycle. Assume Cv=0.717kJ/kg.K and R=8.314kJ/Kmol K**

Given:



$p_1 = 1\text{bar}, \quad T_1 = 17^\circ C + 273 = 290K$

$P_2 = 15$ bar, $P_3 = 40$ bar, Cv=0.717kJ/kg.K

R_u=8.314kJ/Kmol K

Compression Ratio $(r_c) = \frac{v_1}{v_2} = \frac{v_4}{v_3}$ **Pressure Ratio** $(r_p) = \frac{P_3}{P_2} = \frac{P_4}{P_1}$

Molecular Weight of air:

$M = 21\%O_2 + 79\%N_2 \Rightarrow M = (0.21 \times 32) + (0.79 \times 28) \Rightarrow \mathbf{M = 28.84\ kmol/Kg}$

$R = \frac{R_U}{M} \Rightarrow R = \frac{8.314}{28.84} \Rightarrow \mathbf{R = 0.288\ \frac{kJ}{kg}.K}$

$R = C_P - C_V \Rightarrow R = \left(\frac{C_P}{C_V} - 1\right) C_V \Rightarrow R = (\gamma - 1)C_V \Rightarrow 0.288 = (\gamma - 1)0.717 \Rightarrow \boldsymbol{\gamma = 1.4}$

1-2 Process: Adiabatic compression process

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma = r_c{}^\gamma \Rightarrow r_c = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \Rightarrow r_c = \left(\frac{15}{1}\right)^{\frac{1}{1.4}} \Rightarrow \mathbf{r_c = 6.9}$

$\frac{T_2}{T_1} = r_c{}^{\gamma-1} \Rightarrow T_2 = r_c{}^{\gamma-1} \times T_1 \Rightarrow T_2 = 6.9^{1.4-1} \times 290 \Rightarrow \mathbf{T_2 = 628\ K}$

2-3 Process: Constant volume heat addition

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} \Rightarrow T_3 = 628 \times \frac{40}{15} \Rightarrow \mathbf{T_3 = 1674.7K}$

$Q_S = c_v(T_3 - T_2) \Rightarrow Q_S = 0.717(1674.7 - 628) \Rightarrow \mathbf{Q_S = 750.5\ kJ/kg}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = r_c{}^{\gamma-1} \Rightarrow T_4 = \frac{T_3}{r_c{}^{\gamma-1}} \Rightarrow T_4 = \frac{1674.7}{6.9^{1.4-1}} \Rightarrow \mathbf{T_4 = 773.4\ K}$

$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^\gamma = r_c{}^\gamma \Rightarrow P_4 = \frac{P_3}{r_c{}^\gamma} \Rightarrow P_4 = \frac{40}{6.9^{1.4}} \Rightarrow \mathbf{P_4 = 2.7\ bar}$

4-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.717(773.4 - 290) \Rightarrow \mathbf{Q_R = 346.6\ kJ/kg}$

Work done

$W = Q_S - Q_R \Rightarrow W = 750.5 - 346.6 \Rightarrow \mathbf{W = 403.9\ kJ/kg}$

Thermal Efficiency

$\eta = 1 - \frac{1}{(r_c)^{\gamma-1}} \Rightarrow \eta = 1 - \frac{1}{(6.9)^{1.4-1}} \Rightarrow \boldsymbol{\eta = 53.82\%}$

Mean Effective Pressure

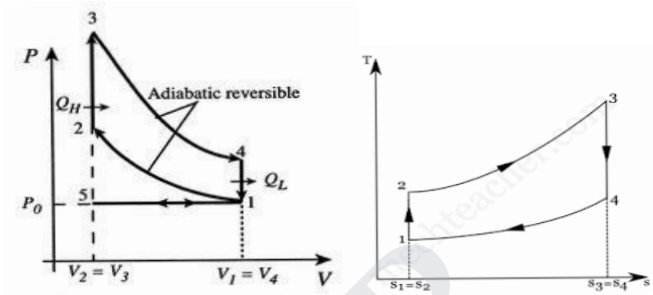
$P_m = \frac{P_1 r_c[(r_c{}^{\gamma-1}-1)(r_p-1)]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1\times 6.9\left[(6.9^{1.4-1}-1)\left(\frac{40}{15}-1\right)\right]}{(1.4-1)(6.9-1)} \Rightarrow \mathbf{P_m = 5.68\ bar}$

4. **An engine working on Otto cycle has compression ratio of 8.5. The initial pressure and temperature at the beginning of compression stroke is 0.93 bar and 93° C. The Maximum pressure at cycle is 38bar. Find (i) Maximum temperature and pressure, (ii) Work done per kg of air, (iii) Cycle efficiency and (iv) Mean effective pressure. Assume, Cp and Cv , R and γ suitably.**

GIVEN:

$p_1 = 0.93\text{bar}, \quad T_1 = 93^\circ\text{C} + 273$

$P_3 = 38\text{ bar}$



COMPRESSION RATIO

$(r_c) \quad = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 8.5$

PRESSURE RATIO

$(r_p) \quad = \frac{P_3}{P_2} = \frac{P_4}{P_1}$

1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 8.5^{1.4-1} \times 366 \Rightarrow \mathbf{T_2 = 861.4K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 8.5^{1.4} \times 0.93 \Rightarrow \mathbf{P_2 = 18.6\text{ bar}}$

2-3 Process: Constant volume heat addition

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 861.4 \times \frac{38}{18.6} \Rightarrow \mathbf{T_3 = 1759.85K}$

$Q_S = c_v(T_3 - T_2) \Rightarrow Q_S = 0.714(1759.85 - 681.68) \Rightarrow \mathbf{Q_S = 644.9\text{ kJ/kg}}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = r_c^{\gamma-1} \Rightarrow T_4 = \frac{T_3}{r_c^{\ \gamma-1}} = \frac{1759.85}{8.5^{1.4-1}} \Rightarrow \mathbf{T_4 = 747.75\text{ K}}$

$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = r_c^{\gamma} \Rightarrow P_4 = \frac{P_3}{r_c^{\ \gamma}} = \frac{38}{8.5^{1.4}} \Rightarrow \mathbf{P_4 = 1.9\text{ bar}}$

4-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.714(747.75 - 366) \Rightarrow \mathbf{Q_R = 274.1\text{ kJ/kg}}$

Work done

$W = Q_S - Q_R \Rightarrow W = 644.9 - 274.1 \Rightarrow \mathbf{W = 370.8\text{ kJ/kg}}$

Thermal Efficiency

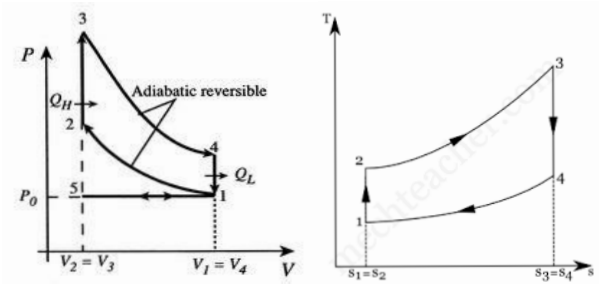
$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{370.8}{644.9} \Rightarrow \mathbf{\eta = 57.5\%}$

Mean Effective Pressure

$P_m = \frac{P_1 r_c[(r_c^{\gamma-1}-1)(r_p-1)]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1\times8.5\left[\left(8.5^{1.4-1}-1\right)\left(\frac{38}{18.6}-1\right)\right]}{(1.4-1)(8.5-1)} \Rightarrow \mathbf{P_m = 3.72\text{ bar}}$

5. Fuel supplied to an SI engine has a calorific value 42000 kJ/kg. the pressure in the cylinder at 30% and 70% of the compression stroke are 1.3 bar and 2.6 bar respectively. Assuming the compression follows the law PV1.3=C. Find the compression ratio. If the relative efficiency of the engine compared with the air standard efficiency is 50%. Calculate the fuel consumption in kg/kW h.

GIVEN:



For 30% of compression 1.3bar

For 70% of compression 2.6 bar

CV=42000kJ/kg

$PV^{1.3}=C$

Relative Efficiency is 50%

$p_1 = 0.98\text{bar}, p_2 = 3.25\text{ bar}$

COMPRESSION RATIO

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c{}^{\gamma} \Rightarrow r_c = \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} = \left(\frac{3.25}{0.98}\right)^{\frac{1}{1.3}} \Rightarrow \mathbf{r_c = 2.51}$$

AIR STANDARD EFFICIENCY

$$\eta = 1 - \frac{1}{(r_c)^{\gamma-1}} = 1 - \frac{1}{2.51^{1.3-1}} \Rightarrow \boldsymbol{\eta = 24.13\%}$$

ACTUAL EFFICIENCY

$$\eta_{rel} = \frac{\eta_{act}}{\eta_{ideal}} \Rightarrow \eta_{act} = \eta_{rel} \times \eta_{ideal} = 0.5 \times 0.2413 \Rightarrow \boldsymbol{\eta_{act} = 12.06\%}$$

SPECIFIC FUEL CONSUMPTION

$$SFC = \frac{\dot{m}_f}{IP} =$$

Where

$$\eta_{IP} = \frac{IP}{Q_s} = \frac{IP}{\frac{\dot{m}_f \times CV}{3600}} \Rightarrow IP = \frac{\eta_{IP} \times \dot{m}_f \times CV}{3600}$$

$$SFC = \frac{\dot{m}_f}{IP} = \frac{\dot{m}_f}{\frac{\eta_{IP} \times \dot{m}_f \times CV}{3600}} \Rightarrow SFC = \frac{3600}{CV \times \eta_{IP}} = \frac{3600}{42000 \times 0.1206} \Rightarrow \mathbf{SFC = 0.71\ kg/kWh}$$

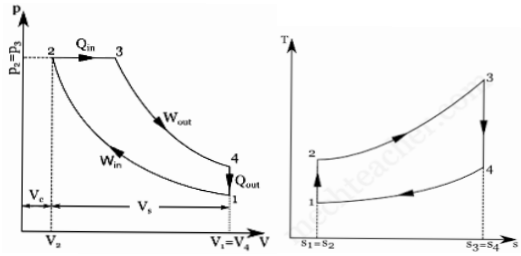
DIESEL CYCLE:

6. Derive an expression for the air-standard efficiency of diesel cycle.Explain why the efficiency of Otto cycle is greater than that of the diesel cycle for the same compression ratio.

COMPRESSION RATIO $= \frac{v_1}{v_2}$

CUT-OFF RATIO $= \frac{v_3}{v_2}$

EXPANSION RATIO $= \frac{v_4}{v_3} = \frac{r_c}{\rho}$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1$$

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_v(T_3 - T_2)\ kJ/kg$$

$$\frac{v_3}{v_2} = \frac{T_3}{T_2} = \rho \Rightarrow T_3 = T_2 \times \rho = r_c^{\ \gamma-1} \times T_1 \times \rho$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_4 = r_c^{\ \gamma-1} \times T_1 \times \rho \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} \Rightarrow T_4 = T_1 \times \rho$$

4-1 Process: Constant Volume heat rejection

$$Q_R = c_v(T_4 - T_1)$$

Work done

$$W = Q_{2-3} - Q_{4-1}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W}{Q_{2-3}} \Rightarrow \eta = \frac{Q_{2-3} - Q_{4-1}}{Q_{2-3}} \Rightarrow \eta = 1 - \frac{1}{\gamma(r_c)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right]$$

The efficiency of the diesel cycle is $\eta = 1 - \frac{1}{\gamma(r_c)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right]$

The efficiency of the diesel cycle for the constant compression ratio depends upon the factor

$$K = \frac{1}{\gamma(r_c)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right]$$

For the value of $\gamma = 1.4$, the value of the factor K for different cut- off ratios is given under

ρ	=	3	2.5	2	1.5
K	=	1.31	1.24	1.17	1.092

Thus we see that the value of is always greater than unity and thus we can deduce that for the compression ratio, the efficiency of the Otto cycle is greater than that of the Diesel cycle.

7. An engine with 200 mm cylinder diameter and 300 mm stroke works on theoretical diesel cycle. The intial pressure and temperature of air used are 1 bar and 27 °C. The cut-off is 8% if the stroke. Determine:i) pressures and temperature at all salient points, ii) Theoretical air standard efficiency, iii) Mean effective pressure, iv) power of the engine if the working cycles per minute are 380. Assume the compression ratio is 15 and working fluid is air.

GIVEN:

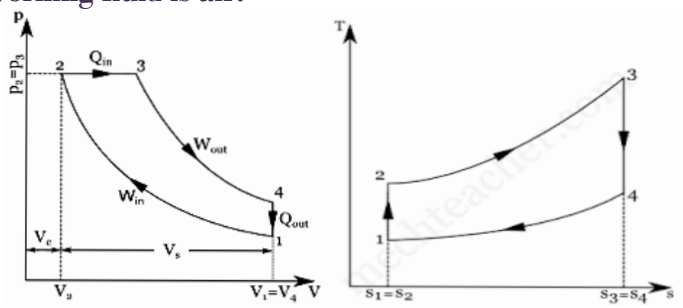
$p_1 = 1\text{bar},\ T_1 = 27^\circ\text{C} + 273$

CUT-OFF RATIO $(\rho) = \frac{v_3}{v_2}$

COMPRESSION RATIO $(r_c) = \frac{v_1}{v_2} = 15$

$V_3 - V_2 = 0.08\,(V_1 - V_2)$

$0.08 = \frac{\rho - 1}{r_c - 1} \Rightarrow \rho = 2.12$



1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 15^{1.4-1} \times 300 \Rightarrow \mathbf{T_2 = 886.25K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 15^{1.4} \times 1 \Rightarrow \mathbf{P_2 = 44.31\ bar}$

2-3 Process: Constant pressure heat addition

$Q_S = c_v(T_3 - T_2) \Rightarrow Q_S = 1.005(1878.85 - 886.25) \Rightarrow \mathbf{Q_S = 997.56\ kJ/kg}$

$\frac{v_3}{v_2} = \frac{T_3}{T_2} = \rho \Rightarrow T_3 = T_2 \times \rho = 886.25 \times 2.12 \Rightarrow \mathbf{T_3 = 1878.85\ K}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_4 = T_3 \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 1878.85 \times \left(\frac{2.12}{15}\right)^{1.4-1} \Rightarrow \mathbf{T_4 = 858.99K}$

$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma} \Rightarrow P_4 = p_3 \times \left(\frac{\rho}{r_c}\right)^{\gamma} = 44.31 \times \left(\frac{2.12}{15}\right)^{1.4} \Rightarrow \mathbf{P_4 = 2.86\ bar}$

4-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.714(858.99 - 300) \Rightarrow \mathbf{Q_R = 399.12\ kJ/kg}$

Work done

$W = Q_S - Q_R \Rightarrow W = 997.56 - 399.12 \Rightarrow \mathbf{W = 598.44 kJ/kg}$

Thermal Efficiency

$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{598.44}{997.56} \Rightarrow \boldsymbol{\eta = 59.99\%}$

Mean Effective Pressure

$P_m = \frac{P_1 r_c^{\gamma}\left[\gamma(\rho-1) - r_c^{\ 1-\gamma}(\rho^{\gamma}-1)\right]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1\times15^{1.4}\left[1.4(2.12-1) - 15^{1-1.4}(2.12^{1.4}-1)\right]}{(1.4-1)(15-1)} \Rightarrow \mathbf{P_m = 7.42\ bar}$

$V_S = \frac{\pi d^2}{4} \times L \Rightarrow V_S = \frac{\pi 0.2^2}{4} \times 0.3 \Rightarrow \boldsymbol{V_S = 9.42 \times 10^{-3} m^3}$,

$\dot{V} = V_S \times N \Rightarrow \dot{V} = 9.42 \times 10^{-3} \times \frac{380}{60} \Rightarrow \boldsymbol{\dot{V} = 0.0597\ m^3/s}$

$\dot{m} = \frac{P \times \dot{V}}{RT} \Rightarrow \dot{m} = \frac{1\times10^2\times0.0597}{0.287\times300} \Rightarrow \boldsymbol{\dot{m} = 0.0693\ kg/s}$

Power

$\dot{W} = \dot{m} \times w \Rightarrow \dot{W} = 0.0693 \times 598.44 \Rightarrow \boldsymbol{\dot{W} = 41.49\ kW}$

8. **Air standard Diesel cycle has a compression ratio of 8. The pressure at the beginning of the compression stroke is 1 bar and the temperature is 300K.The heat supplied is 1800 kJ/kg. Determine: (i) Air Standard efficiency, (ii) Pressure and temperature at salient points, (iii) Mean effective pressure. Assume, Cp = 1.005 , R = 0.287.**

GIVEN:

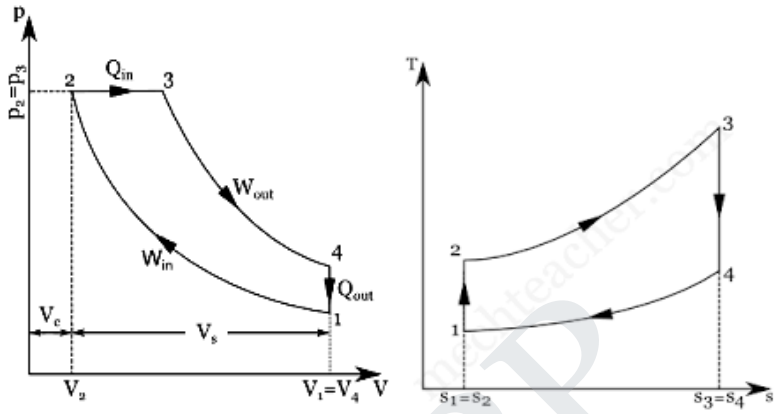
$p_1 = 1\text{bar},\ T_1 = 300\ K$

$Q_S = 1800 kJ/kg$

COMPRESSION RATIO

$(r_c) = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 8$

$C_p = 1.005$, $R = 0.287$



1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 8^{1.4-1} \times 300 \Rightarrow \mathbf{T_2 = 689.21K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 8^{1.4} \times 1 \Rightarrow \mathbf{P_2 = 18.38\ bar}$

2-3 Process: Constant pressure heat addition

$Q_S = c_v(T_3 - T_2) \Rightarrow 1800 = 1.005(T_3 - 689.21) \Rightarrow \mathbf{T_3 = 2480.25\ K}$

$\frac{v_3}{v_2} = \frac{T_3}{T_2} = \rho \Rightarrow \rho = \frac{T_3}{T_2} = \frac{2480.25}{689.21} \Rightarrow \boldsymbol{\rho = 3.59}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_4 = T_3 \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 2480.25 \times \left(\frac{3.59}{8}\right)^{1.4-1} \Rightarrow \mathbf{T_4 = 1801.84\ K}$

$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma} \Rightarrow P_4 = p_3 \times \left(\frac{\rho}{r_c}\right)^{\gamma} = 18.38 \times \left(\frac{3.59}{8}\right)^{1.4} \Rightarrow \mathbf{P_4 = 5.99\ bar}$

4-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.714(1801.84 - 300) \Rightarrow \mathbf{Q_R = 1072.32\ kJ/kg}$

Work done

$W = Q_S - Q_R \Rightarrow W = 1800 - 1072.32 \Rightarrow \mathbf{W = 727.68\ kJ/kg}$

Thermal Efficiency

$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{727.68}{1800} \Rightarrow \boldsymbol{\eta = 40.42\%}$

Mean Effective Pressure

$P_m = \frac{P_1 r_c^{\gamma}\left[\gamma(\rho-1) - r_c^{\ 1-\gamma}(\rho^{\gamma}-1)\right]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1 \times 8^{1.4}\left[1.4(3.59-1) - 8^{1-1.4}(3.59^{1.4}-1)\right]}{(1.4-1)(8-1)} \Rightarrow \mathbf{P_m = 9.56\ bar}$

9. **Air standard Diesel cycle has a compression ratio of 18. The pressure at the beginning of the compression stroke is 1 bar and the temperature is 30 °C.The heat supplied is 1800 kJ/kg.Determine: (i) Thermal efficiency, (ii) Pressure and temperature at salient points, (iii) Heat rejected, (iv) Mean effective pressure. Assume, Cp and Cv , R and γ suitably.**

GIVEN:

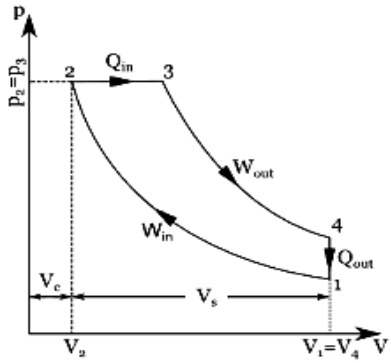
$p_1 = 1\text{bar},\ T_1 = 30^\circ\text{C} + 273$

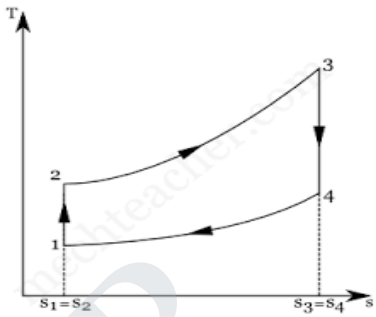
$Q_S = 1800 kJ/kg$

COMPRESSION RATIO

$(r_c) = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 18$

$C_p = 1.005$, $R = 0.287$





1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 18^{1.4-1} \times 303 \Rightarrow \mathbf{T_2 = 962.83K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 18^{1.4} \times 1 \Rightarrow \mathbf{P_2 = 57.19\ bar}$

2-3 Process: Constant pressure heat addition

$Q_S = c_v(T_3 - T_2) \Rightarrow 1800 = 1.005(T_3 - 962.83) \Rightarrow \mathbf{T_3 = 2753.87\ K}$

$\frac{v_3}{v_2} = \frac{T_3}{T_2} = \rho \Rightarrow \rho = \frac{T_3}{T_2} = \frac{2753.87}{962.83} \Rightarrow \boldsymbol{\rho = 2.86}$

3-4 Process: Adiabatic Expansion process

$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_4 = T_3 \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 2753.87 \times \left(\frac{2.86}{18}\right)^{1.4-1} \Rightarrow \mathbf{T_4 = 1319.41\ K}$

$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma} \Rightarrow P_4 = p_3 \times \left(\frac{\rho}{r_c}\right)^{\gamma} = 57.19 \times \left(\frac{2.86}{18}\right)^{1.4} \Rightarrow \mathbf{P_4 = 4.35\ bar}$

4-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_4 - T_1) \Rightarrow Q_R = 0.714(1319.41 - 303) \Rightarrow \mathbf{Q_R = 725.72\ kJ/kg}$

Work done

$W = Q_S - Q_R \Rightarrow W = 1800 - 725.72 \Rightarrow \mathbf{W = 1074.28\ kJ/kg}$

Thermal Efficiency

$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{1074.28}{1800} \Rightarrow \boldsymbol{\eta = 59.68\%}$

Mean Effective Pressure

$P_m = \frac{P_1 r_c^{\gamma}\left[\gamma(\rho-1) - r_c^{\ 1-\gamma}(\rho^{\gamma}-1)\right]}{(\gamma-1)(r_c-1)} \Rightarrow P_m = \frac{1 \times 18^{1.4}\left[1.4(2.86-1) - 18^{1-1.4}(2.86^{1.4}-1)\right]}{(1.4-1)(18-1)} \Rightarrow \mathbf{P_m = 13.02\ bar}$

DUEL CYCLE:

10. The swept volume of a diesel engine working on a dual cycle is 0.053 m³ and clearance volume is 0.0035 m³. The maximum pressure is 65 bar. Fuel injection ends at 5% of the stroke. The temperature and pressure at the start of compression are 80 °C and 0.9 bar. Determine the air standard efficiency of the cycle. Take γ=1.4 for air.

GIVEN:

$p_1 = 0.9\text{bar}$

$T_1 = 80°C + 273$

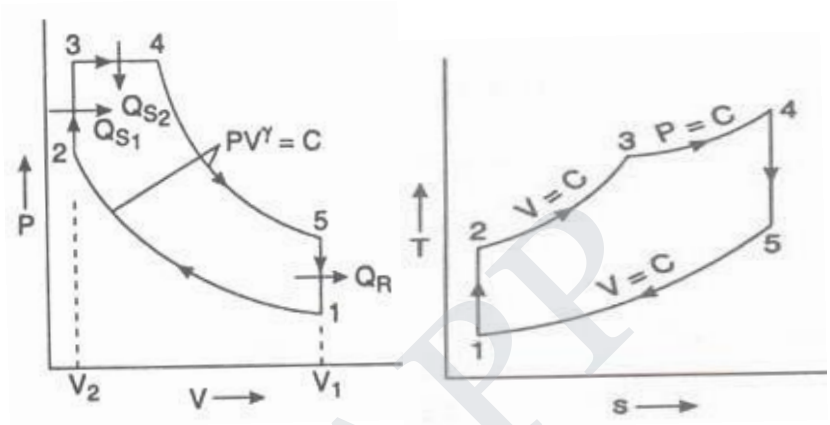
$p_3 = 65\text{bar}$

COMPRESSION RATIO

$(r_c) = \frac{v_1}{v_2} = \frac{v_c+v_s}{v_c} = 16.14$

$V_4 - V_3 = 0.08\,(V_1 - V_2)$

$0.05 = \frac{\rho-1}{r_c-1} \Rightarrow \rho = 1.76$



1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 16^{1.4-1} \times 353 \Rightarrow \mathbf{T_2 = 1070.09K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 16^{1.4} \times 0.9 \Rightarrow \mathbf{P_2 = 48.5\ bar}$

2-3 Process: Constant volume heat addition

$Q_{S1} = c_v(T_3 - T_2) \Rightarrow Q_{S1} = 0.714(1434.05 - 1070.09) \Rightarrow \mathbf{Q_{S1} = 259.87\ kJ/kg}$

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 1070.09 \times \frac{65}{48.5} \Rightarrow \mathbf{T_3 = 1434.05\ K}$

3-4 Process: Constant pressure heat addition

$Q_{S2} = c_p(T_4 - T_3) \Rightarrow Q_{S2} = 1.005(2523.93 - 1434.05) \Rightarrow \mathbf{Q_{S2} = 1095.33\ kJ/kg}$

$\frac{v_4}{v_3} = \frac{T_4}{T_3} = \rho \Rightarrow T_4 = \rho \times T_3 = 1.76 \times 1434.05 \Rightarrow \mathbf{T_4 = 2523.93K}$

4-5 Process: Adiabatic Expansion process

$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_5 = T_4 \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 2523.93 \times \left(\frac{1.76}{16}\right)^{1.4-1} \Rightarrow \mathbf{T_5 = 1043.84\ K}$

$\frac{p_4}{p_5} = \left(\frac{v_5}{v_4}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma} \Rightarrow P_5 = p_4 \times \left(\frac{\rho}{r_c}\right)^{\gamma} = 65 \times \left(\frac{1.76}{16}\right)^{1.4} \Rightarrow \mathbf{P_4 = 2.96\ bar}$

5-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_5 - T_1) \Rightarrow Q_R = 0.714(1043.84 - 353) \Rightarrow \mathbf{Q_R = 493.26\ kJ/kg}$

Work done

$W = Q_{S1} + Q_{S1} - Q_R \Rightarrow W = 259.87 + 1095.33 - 493.26 \Rightarrow \mathbf{W = 861.94\ kJ/kg}$

Thermal Efficiency

$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{861.94}{259.87+1095.33} \Rightarrow \mathbf{\eta = 63.6\%}$

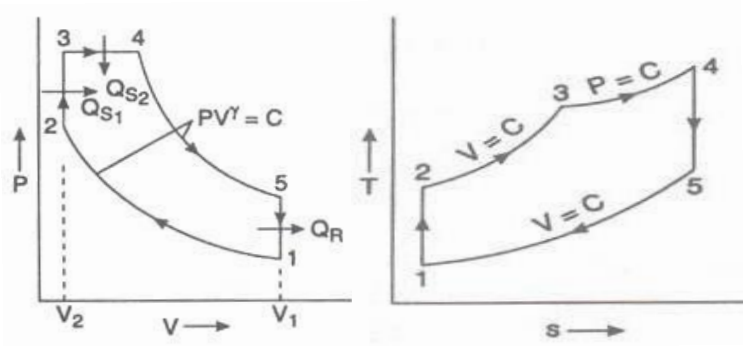
11. Derive the expression for air standard efficiency of duel combustion cycle.

COMPRESSION RATIO $(r_c) = \frac{v_1}{v_2}$

CUT-OFF RATIO $(\rho) = \frac{v_4}{v_3}$

PRESSURE RATIO $(r_p) = \frac{p_3}{p_2}$

EXPANSION RATIO $(r_c) = \frac{v_5}{v_4} = \frac{r_c}{\rho}$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1$$

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1$$

2-3 Process: Constant volume heat addition

$$Q_S = c_v(T_3 - T_2)\ \text{kJ/kg}$$

$$\frac{p_3}{p_2} = \frac{T_3}{T_2} = r_p \Rightarrow T_3 = r_p \times T_2 \Rightarrow T_3 = r_p \times r_c^{\ \gamma-1} \times T_1$$

3-4 Process: Constant pressure heat addition

$$Q_S = c_p(T_4 - T_3)\ \text{kJ/kg}$$

$$\frac{v_4}{v_3} = \frac{T_4}{T_3} = \rho \Rightarrow T_4 = \rho \times T_3 \Rightarrow T_4 = \rho \times r_p \times r_c^{\ \gamma-1} \times T_1$$

4-5 Process: Adiabatic Expansion process

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_5 = \rho^{\gamma} \times r_p \times T_1$$

5-1 Process: Constant Volume heat rejection

$$Q_R = c_v(T_1 - T_5)\ \textbf{kJ / kg}, \qquad \frac{p_5}{p_1} = \frac{T_5}{T_1} = r_p$$

Work done

$$W = Q_{2-3} + Q_{3-4} - Q_{5-1}\ \textbf{kJ / kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W}{Q_{2-3}+Q_{3-4}} \Rightarrow \eta = 1 - \frac{1}{(r_c)^{\gamma-1}}\left[\frac{r_p\rho^{\gamma}-1}{(r_p-1)+r_p\gamma(\rho-1)}\right]$$

Mean Effective Pressure

$$P_m = \frac{P_1 r_c^{\gamma}\left[r_p\gamma(\rho-1) + (r_p-1) - r_c^{\ 1-\gamma}(r_p\rho^{\gamma}-1)\right]}{(\gamma-1)(r_c-1)}$$

12. An air-standard dual cycle has a compression ratio of 10. The pressure and temperature at the beginning of compression are 1 bar and 27 °C. The maximum pressure reached is 42 bar and maximum temperature is 1500°C. Determine (i) the temperature at the end of constant volume heat addition (ii)cut-off ratio (iii) work done per kg of air and (iv) the cycle efficiency. Assume Cp = 1.004 kJ/kg K, Cv = 0.717 kJ/kg K for air.

GIVEN:

$p_1 = 1\text{bar},\ T_1 = 27^\circ\text{C} + 273$

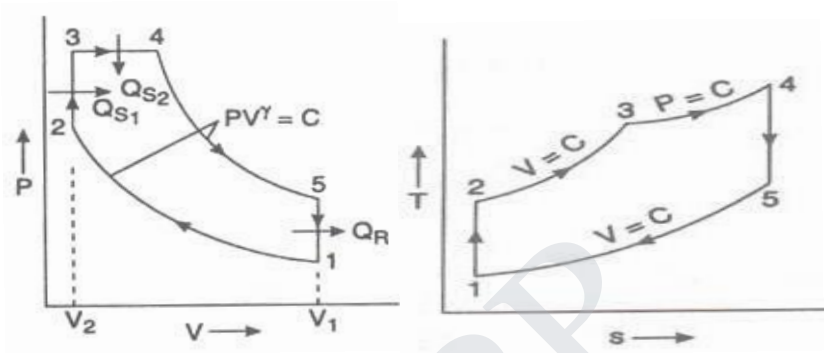
$p_3 = 42\text{bar},$

$T_4 = 1500^\circ\text{C} + 273$

COMPRESSION RATIO

$(r_c) \quad = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 10$

$C_p = 1.005$, $R = 0.287$



1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 10^{1.4-1} \times 300 \Rightarrow \mathbf{T_2 = 753.57\ K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 10^{1.4} \times 1 \Rightarrow \mathbf{P_2 = 25.12\ bar}$

2-3 Process: Constant volume heat addition

$Q_{S1} = c_v(T_3 - T_2) \Rightarrow Q_{S1} = 0.714(1259.95 - 753.57) \Rightarrow \mathbf{Q_{S1} = 361.56\ kJ/kg}$

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 753.57 \times \frac{42}{25.12} \Rightarrow \mathbf{T_3 = 1259.95\ K}$

3-4 Process: Constant pressure heat addition

$Q_{S2} = c_p(T_4 - T_3) \Rightarrow Q_{S2} = 1.005(1773 - 1259.95) \Rightarrow \mathbf{Q_{S2} = 515.62\ kJ/kg}$

$\frac{v_4}{v_3} = \frac{T_4}{T_3} = \rho \Rightarrow \rho = \frac{T_4}{T_3} = \frac{1773}{1259.95} \Rightarrow \boldsymbol{\rho = 1.41}$

4-5 Process: Adiabatic Expansion process

$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_5 = T_4 \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 1773 \times \left(\frac{1.41}{10}\right)^{1.4-1} \Rightarrow \mathbf{T_5 = 809.19\ K}$

$\frac{p_4}{p_5} = \left(\frac{v_5}{v_4}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma} \Rightarrow P_5 = p_4 \times \left(\frac{\rho}{r_c}\right)^{\gamma} = 42 \times \left(\frac{1.41}{10}\right)^{1.4} \Rightarrow \mathbf{P_4 = 2.7\ bar}$

5-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_5 - T_1) \Rightarrow Q_R = 0.714(809.19 - 300) \Rightarrow \mathbf{Q_R = 363.56\ kJ/kg}$

Work done

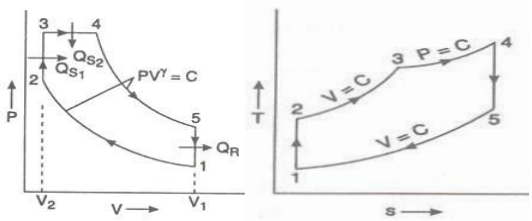
$W = Q_{S1} + Q_{S1} - Q_R \Rightarrow W = 361.56 + 515.62 - 363.56 \Rightarrow \mathbf{W = 513.62\ kJ/k}$

Thermal Efficiency

$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{513.62}{361.56+515.62} \Rightarrow \boldsymbol{\eta = 58.55\%}$

13. **The compression ratio for a single-cylinder engine operating on dual cycle is 9. The maximum pressure in the cycle limited to 60 bar. The pressure and temperature of the air at the beginning of the cycle are 1 bar and 30°C. Heat is added during constant pressure process upto 4% of the stroke. Assuming the cylinder diameter and stroke length as 250 mm and 300 mm respectively, Determine (i) The air standard efficiency of the cycle (ii) The power developed. If the number of working cycles is 3 per second.**

GIVEN:



$p_1 = 1\text{bar},\ T_1 = 30^{\circ}\text{C} + 273,\ p_3 = 60\text{bar},$

$(r_c = \frac{v_1}{v_2} = \frac{v_4}{v_3} = 9$

$V_4 - V_3 = 0.08\,(V_1 - V_2)$

$0.04 = \frac{\rho - 1}{r_c - 1} \Rightarrow \boldsymbol{\rho = 1.32}$

1-2 Process: Adiabatic compression process

$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\ \gamma-1} \Rightarrow T_2 = r_c^{\ \gamma-1} \times T_1 = 9^{1.4-1} \times 303 \Rightarrow \boldsymbol{T_2 = 729.69\ K}$

$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r_c^{\ \gamma} \Rightarrow P_2 = r_c^{\ \gamma} \times P_1 = 9^{1.4} \times 1 \Rightarrow \boldsymbol{P_2 = 21.67\ bar}$

2-3 Process: Constant volume heat addition

$Q_{S1} = c_v(T_3 - T_2) \Rightarrow Q_{S1} = 0.714(2020.37 - 729.69) \Rightarrow \boldsymbol{Q_{S1} = 921.55\ kJ/kg}$

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 729.69 \times \frac{60}{21.67} \Rightarrow \boldsymbol{T_3 = 2020.37\ K}$

3-4 Process: Constant pressure heat addition

$Q_{S2} = c_p(T_4 - T_3) \Rightarrow Q_{S2} = 1.005(2666.89 - 2020.37) \Rightarrow \boldsymbol{Q_{S2} = 649.75\ kJ/kg}$

$\frac{v_4}{v_3} = \frac{T_4}{T_3} = \rho \Rightarrow T_4 = \rho \times T_3 = 1.32 \times 2000 \Rightarrow \boldsymbol{T_4 = 2666.89\ K}$

4-5 Process: Adiabatic Expansion process

$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r_c}{\rho}\right)^{\gamma-1} \Rightarrow T_5 = T_4 \times \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 2666.89 \times \left(\frac{1.32}{9}\right)^{1.4-1} \Rightarrow \boldsymbol{T_5 = 1237.48\ K}$

$\frac{p_4}{p_5} = \left(\frac{v_5}{v_4}\right)^{\gamma} = \left(\frac{r_c}{\rho}\right)^{\gamma} \Rightarrow P_5 = p_4 \times \left(\frac{\rho}{r_c}\right)^{\gamma} = 60 \times \left(\frac{1.32}{9}\right)^{1.4} \Rightarrow \boldsymbol{P_4 = 4.08\ bar}$

5-1 Process: Constant Volume heat rejection

$Q_R = c_v(T_5 - T_1) \Rightarrow Q_R = 0.714(1237.48 - 303) \Rightarrow \boldsymbol{Q_R = 667.22\ kJ/kg}$

Work done

$W = Q_{S1} + Q_{S1} - Q_R \Rightarrow W = 921.55 + 649.75 - 667.22 \Rightarrow \boldsymbol{W = 904.08\ kJ/k}$

Thermal Efficiency

$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{904.08}{921.55 + 649.75} \Rightarrow \boldsymbol{\eta = 57.54\%}$

Power:

$V_S = \frac{\pi d^2}{4} \times L \Rightarrow V_S = \frac{\pi 0.25^2}{4} \times 0.3 \Rightarrow \boldsymbol{V_S = 0.0147 m^3},$

$\dot{V} = V_S \times N \Rightarrow \dot{V} = 0.0147 \times 3 \Rightarrow \boldsymbol{\dot{V} = 0.0441\ m^3/s}$

$\dot{m} = \frac{P \times \dot{V}}{RT} \Rightarrow \dot{m} = \frac{1 \times 10^2 \times 0.0597}{0.287 \times 300} \Rightarrow \boldsymbol{\dot{m} = 0.0507\ kg/s}$

Power: $\dot{W} = \dot{m} \times w \Rightarrow \dot{W} = 0.0507 \times 904.08 \Rightarrow \boldsymbol{\dot{W} = 45.84\ kW}$

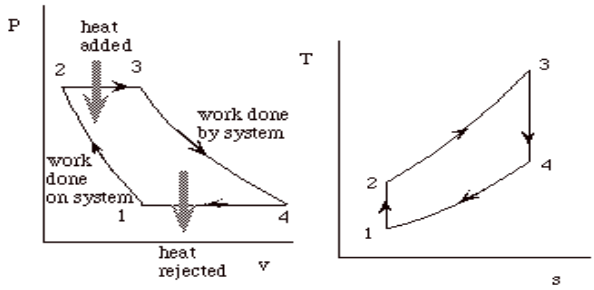
GAS TURBINE PLANT OR BRAYTON CYCLE

14. A gas turbine on works air standard brayton cycle. The intial condition of air is 25 °C and 1 bar the maximum pressure and temperature are limited to 3bar and 650°C. Determine i) Cycle efficiency ii)Heat supplied and rejected per kg of air iii)Work output iv) Exhaust temperature.

GIVEN:

$p_1 = 1\text{bar},\ T_1 = 25°C + 273$

$p_3 = 3\text{bar}, T_3 = 650°C + 273$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 3^{\frac{1.4-1}{1.4}} \times 298 \Rightarrow \mathbf{T_2 = 407.88\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(407.88 - 298) \Rightarrow \mathbf{W_C = 110.43\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(923 - 407.88) \Rightarrow \mathbf{Q_S = 517.7\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{923}{3^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 674.34\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(923 - 674.34) \Rightarrow \mathbf{W_T = 249.9\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(674.34 - 298) \Rightarrow \mathbf{Q_R = 378.22\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 517.7 - 378.22 \Rightarrow \mathbf{W_{net} = 139.48\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{139.48}{517.7} \Rightarrow \boldsymbol{\eta = 26.94\%}$$

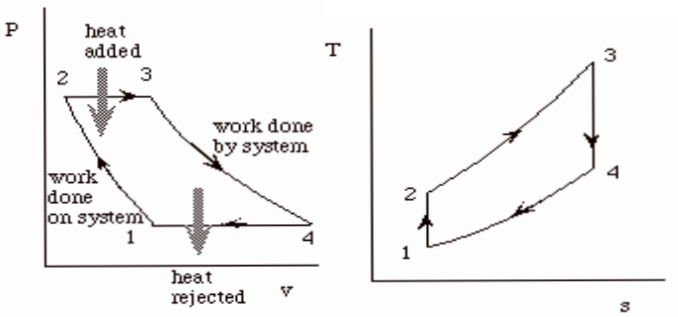
15. Air enters the compressor of a gas turbine plant operating on brayton cycle at 1 bar, 27 °C. The pressure ratio in the cycle is 6. If Wt = 2.5 Wc where Wt and Wc are the turbine and compressor work respectively, calculate the maximum temperature and the cycle efficiency.

GIVEN:

$p_1 = \mathbf{1bar},\ T_1 = \mathbf{27^\circ C + 273}$

$r_p = \mathbf{6bar},$

$W_t = 2.5\ W_c$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 6^{\frac{1.4-1}{1.4}} \times 300 \Rightarrow \mathbf{T_2 = 500.55\ K}$$

$$W_t = 2.5\ W_c \Rightarrow c_p(T_3 - T_4) = 2.5c_p(T_2 - T_1) \Rightarrow \left(T_3 - \frac{T_3}{1.67}\right) = 501.38 \Rightarrow \mathbf{T_3 = 1249.7\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(500.55 - 300) \Rightarrow \mathbf{W_C = 201.55\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(1249.7 - 500.55) \Rightarrow \mathbf{Q_S = 752.89\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{1249.7}{6^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 748.32\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(1249.7 - 748.32) \Rightarrow \mathbf{W_T = 503.89\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(748.32 - 300) \Rightarrow \mathbf{Q_R = 450.56\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 752.89 - 450.56 \Rightarrow \mathbf{W_{net} = 301.99\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{301.99}{752.89} \Rightarrow \boldsymbol{\eta = 40.11\%}$$

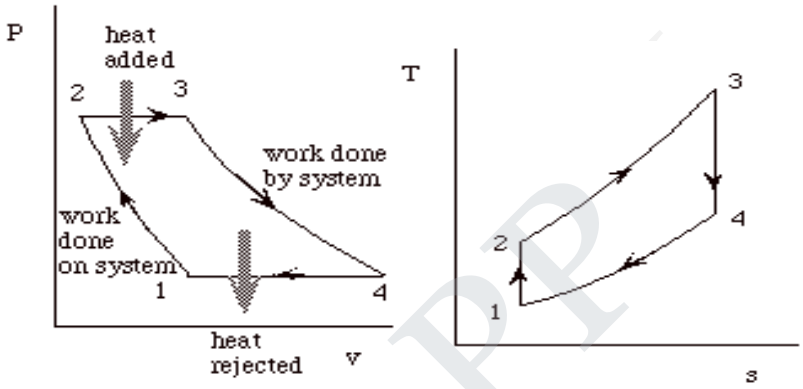
16. Consider an air standard cycle in which the air enters the compressor at 1.0 bar and 20°C. The pressure of air leaving the compressor is 3.5 bar and the temperature at turbine inlet is 600°C. Determine per kg of air, (i)Efficiency of the cycle, (ii) Heat supplied to air, (iii) Work available at the shaft,(iv) Heat rejected in the cooler and (v) Temperature of air leaving the turbine. For air γ = 1.4 and Cp = 1.005 kJ/kg K.

GIVEN:

$p_1 = 1\text{bar},\ T_1 = 20^\circ\text{C} + 273$

$p_3 = 3.5\text{bar},$

$T_3 = 600^\circ\text{C} + 273$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 3.5^{\frac{1.4-1}{1.4}} \times 293 \Rightarrow \mathbf{T_2 = 419.09\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(419.09 - 293) \Rightarrow \mathbf{W_C = 126.72\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(873 - 419.09) \Rightarrow \mathbf{Q_S = 456.18\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{873}{3.5^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 610.33\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(873 - 610.33) \Rightarrow \mathbf{W_T = 263.98\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(610.33 - 293) \Rightarrow \mathbf{Q_R = 318.92\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 456.18 - 318.92 \Rightarrow \mathbf{W_{net} = 137.26\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{137.26}{456.18} \Rightarrow \boldsymbol{\eta = 30.08\%}$$

17. A closed cycle ideal gas turbine plant operates between temperature limits of 800°C and 30°C and produces a power of 100 kW. The plant is designed such that there is no need for a regenerator. A fuel of calorific value 45000 kJ/kg is used. Calculate the mass flow rate of air through the plant and rate of fuel consumption. Assume Cp = 1 kJ/kg K and γ = 1.4.

GIVEN:

$T_1 = 30^\circ C + 273$

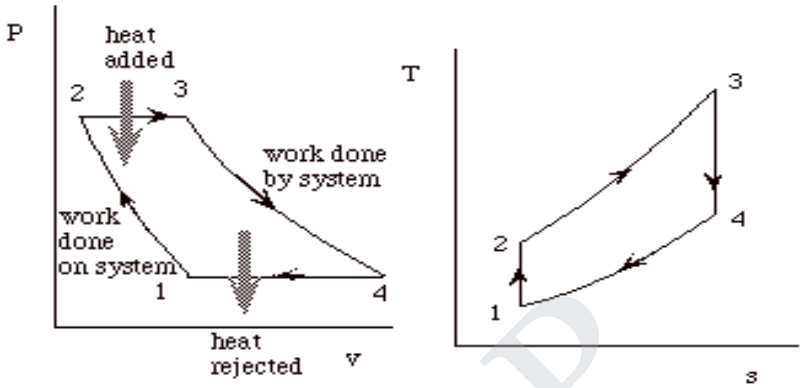
$T_3 = 800^\circ C + 273$

Assume $r_p = 6$

Power=100kW

CV=45000kJ/kg

$C_p = 1$ kJ/kg K, $\gamma = 1.4$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 6^{\frac{1.4-1}{1.4}} \times 303 \Rightarrow \mathbf{T_2 = 505.56\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(505.56 - 303) \Rightarrow \mathbf{W_C = 203.57\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(1073 - 505.56) \Rightarrow \mathbf{Q_S = 570.28\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{1073}{6^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 643.09\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(1073 - 643.09) \Rightarrow \mathbf{W_T = 432.06\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(643.09 - 303) \Rightarrow \mathbf{Q_R = 341.79\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 570.28 - 341.79 \Rightarrow \mathbf{W_{net} = 228.49\ kJ/kg}$$

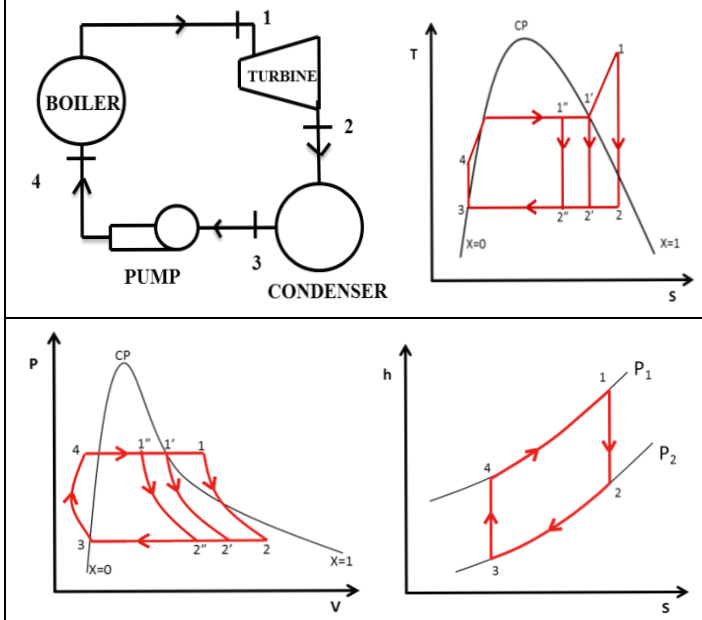
Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{228.49}{570.28} \Rightarrow \boldsymbol{\eta = 40.06\%}$$

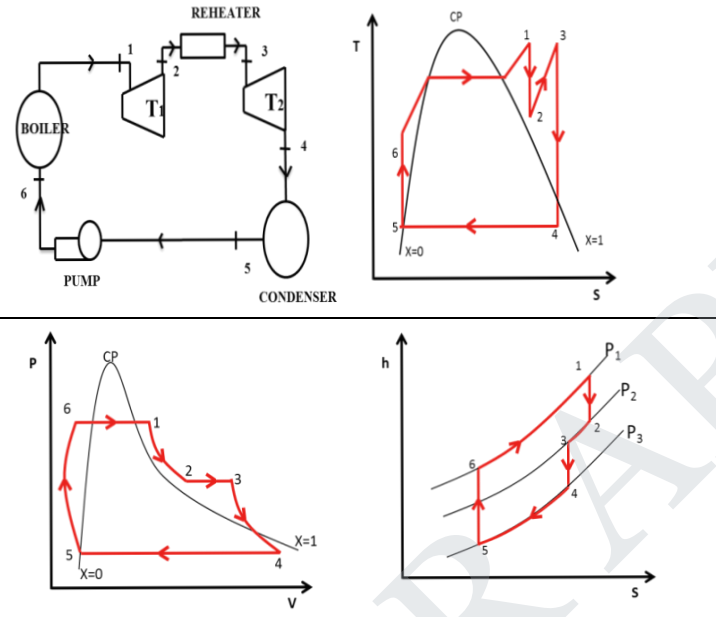
mass flow rate

$$\dot{m} = \frac{\text{Power}}{\text{Work done}} = \frac{100}{228.49} \Rightarrow \dot{m} = 0.438\ \text{kg/s}$$

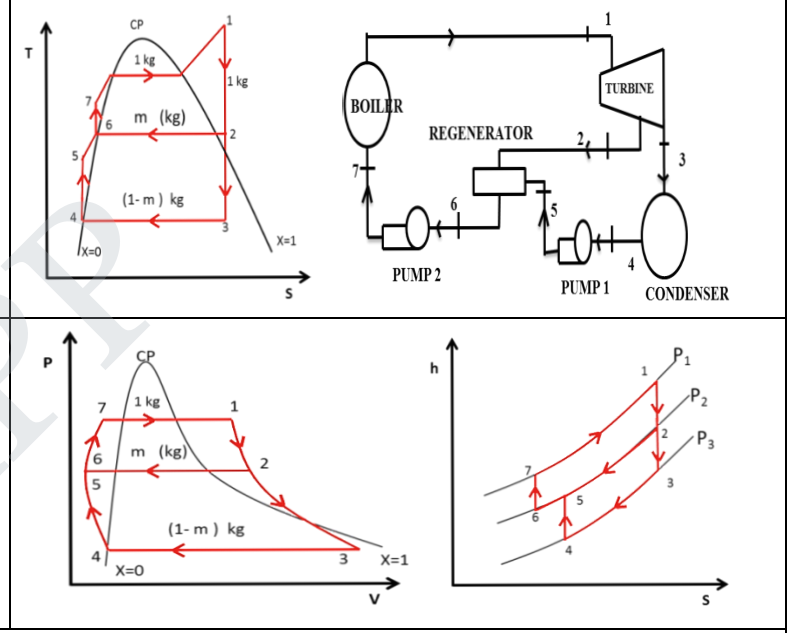
SIMPLE RANKINE CYCLE



REHEAT RANKINE CYCLE



REGENERATION RANKINE CYCLE



COMMON CONDITIONS TO THREE CYCLES

NOTE:

1. Boiler oulet steam condition may be Dry saturated or Super heated steam which enters to the Turbine.(Use saturated (h_g) or superheated steam table (Table 3,4,5) to find out properties.)
2. Turbine exit steam condition is wet steam which enters to the Condenser. (Use formulas $[s = s + (xs_{fg})]$ to find out Dryness Fraction).
3. **For Reheater and Regenerative cycle**, Turbine exit at state 2, steam condition may be Dry Saturated Or Superheated Condition.
4. Condenser outlet condition is Saturated liquid always which enters to the Pump.(at a given Pressure, the enthalpy h_f)
5. Pump outlet condition is subcooled liquid which enters to the boiler. (Ex: Simple Rankine Cycle, Use formulas $[W_p = V_{f3}(P_4 - P_3)]$ to find out pump work and for enthalpy $h_4 = W_p + h_3\ (\frac{kJ}{kg})$. (not possible to use steam table at this condition.)

NOTATIONS IN THE STEAM TABLE:

f – Liquid water
g – Dry or sautated vapour

fg – During the phase change from liquid to vapor

NOTE: (FOR REHEAT AND REGENERATION AT STATE 2)

Turbine inlet entropy = Turbine exit entropy

$(s_1 = s_2)$

at a given turbine exit pressure find entropy in the saturated pressure table s_{g2}

if $s_{g2} = s_2$ then the steam is dry saturated steam, $s_{g2} < s_2$ then the steam is superheated steam, $s_{g2} > s_2$ then the steam is wet steam

SIMPLE RANKINE CYCLE

Process 1-2: Reversible adiabatic expansion $(s_1 = s_2)$

$$W_T = h_1 - h_2 \ (\frac{kJ}{kg})$$

h_1 – Turbine inlet enthalpy

h_2 – Turbine exit and condenser intet enthalpy

Process 2-3:Constant pressure heat rejection $(P_2 = P_3)$

$$Q_R = h_2 - h_3 \ (\frac{kJ}{kg})$$

$h_3 = h_{f2} = h_{f3}$ Condenser exit and pump inlet enthalpy

Process 3-4: Reversible adiabatic pumping $(s_3 = s_4)$

$$W_p = h_4 - h_3 \ (\frac{kJ}{kg})$$

h_4 – Pump exit & boiler inlet enthalpy

$$W_p = V_{f3}(P_4 - P_3) \ (\frac{kJ}{kg})$$

$$W_p = V_{f3}(P_1 - P_2) \ (\frac{kJ}{kg})$$

$$h_4 = W_p + h_3 \ (\frac{kJ}{kg})$$

Process 4-1: Constant pressure heat supplied $(P_4 = P_1)$

$$Q_s = h_1 - h_4 \ (\frac{kJ}{kg})$$

Efficiency :

$$\eta = \frac{W_T - W_p}{Q_s} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

REHEAT RANKINE CYCLE

Process 1-2: Reversible adiabatic expansion $(s_1 = s_2)$

$$W_{T1} = h_1 - h_2 \ (\frac{kJ}{kg})$$

h_1 – Turbine1 inlet enthalpy

h_2 – Turbine exit and reheater intet enthalpy

Process 2-3: Constant pressure reheating $(P_2 = P_3)$

$$Q_{s1} = h_3 - h_2 \ (\frac{kJ}{kg})$$

h_3 – Turbine1 inlet enthalpy

Process 3-4: Reversible adiabatic expansion $(s_3 = s_4)$

$$W_{T2} = h_3 - h_4 \ (\frac{kJ}{kg})$$

h_4 – Turbine exit and condenser intet enthalpy

Process 4-5:Constant pressure heat rejection $(P_4 = P_5)$

$$Q_R = h_4 - h_5 \ (\frac{kJ}{kg})$$

$h_5 = h_{f4} = h_{f5}$ Condenser exit *enthalpy*

Process 5-6:Reversible adiabatic pumping $(s_5 = s_6)$

$$W_p = h_6 - h_5 \ (\frac{kJ}{kg})$$

h_6 – Pump exit & boiler inlet *enthalpy*

$$W_p = V_{f5}(P_6 - P_5) \ (\frac{kJ}{kg})$$

$$h_6 = W_p + h_5 \ (\frac{kJ}{kg})$$

Process 6-1:Constant pressure heat supplied $(P_6 = P_1)$

$$Q_{s2} = h_1 - h_6 \ (\frac{kJ}{kg})$$

Efficiency :

$$\eta = \frac{W_{T1} + W_{T2} - W_p}{Q_{s1} + Q_{s2}}$$

$$\eta = \frac{(h_1 - h_2) + (h_4 - h_3) - (h_6 - h_5)}{(h_3 - h_2) + (h_1 - h_6)}$$

REGENERATION RANKINE CYCLE

Turbine Work: Reversible adiabatic expansion $(s_1 = s_2 = s_3)$

$$W_T = 1kg(h_1 - h_2) + (1 - m)(h_2 - h_3)$$

h_1 – Turbine1 inlet enthalpy

h_2 – bypass regeneration enthalpy

Process 3-4: Constant pressure heat rejection $(P_3 = P_4)$

$$Q_R = h_3 - h_4 \ (\frac{kJ}{kg})$$

h_3 – Turbine exit and condenser intet enthalpy

$h_4 = h_{f3} = h_{f4}$ Condenser exit *enthalpy*

To find bypass steam mass: energy balance

$$m((h_2 - h_6) = (1 - m)(h_6 - h_5)$$

Process 4-5: Reversible adiabatic pumping $(s_4 = s_5)$

$$W_{p1} = (1 - m)h_4 - h_5 \ (\frac{kJ}{kg})$$

Process 6-7: Reversible adiabatic pumping $(s_6 = s_7)$

$$W_{p2} = 1kg \ (h_7 - h_6) \ (\frac{kJ}{kg})$$

$$W_p = V_{f6}(P_7 - P_6) \ (\frac{kJ}{kg})$$

$$h_7 = W_p + h_6 \ (\frac{kJ}{kg})$$

Process 7-1: Constant Pressure Heat Supplied $(P_7 = P_1)$

$$Q_{s1} = 1kg \ (h_1 - h_7) \ (\frac{kJ}{kg})$$

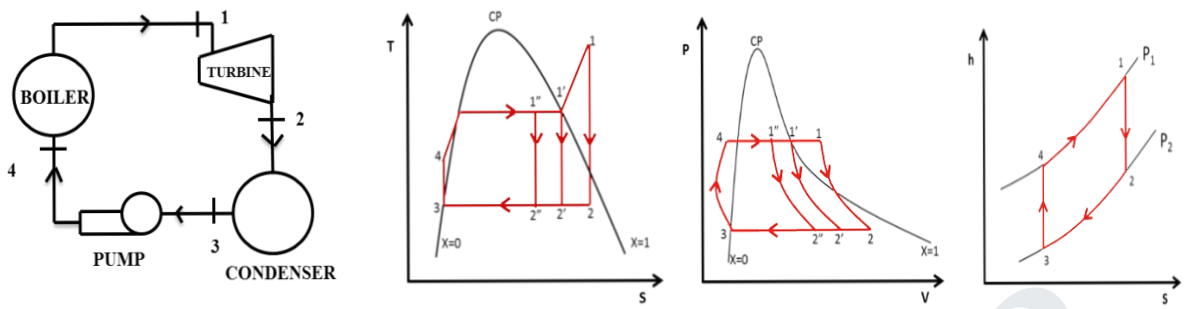
Efficiency :

$$\eta = \frac{W_T}{Q_s} = \frac{1kg(h_1 - h_2) + (1 - m)(h_2 - h_3)}{1kg(h_1 - h_7)}$$

Note: Here Pump work is negligible

RANKINE CYCLE

1. **Draw the P-v, T-s, h -s, diagrams and theoretical lay out for Rankine cycle and hence deduce the expression for its efficiency.**



Process 1-2: Turbine Work : Reversible adiabatic expansion ($s_1 = s_2$)

$$W_T = h_1 - h_2 \ (\frac{kJ}{kg})$$

h_1 – Turbine inlet (**wet or dry or superheated steam**) enthalpy

h_2 – Turbine exit and condenser intet (**wet or dry steam**) enthalpy

Process 2-3: Condenser heat rejection : Constant pressure heat rejection (wet or dry steam convert into liquid) ($P_2 = P_3$)

$$Q_R = h_2 - h_3 \ (\frac{kJ}{kg})$$

$h_3 = h_{f2} = h_{f3}$ Condenser exit and pump inlet (saturated liquid) *enthalpy*

Process 3-4: Pump Work: Reversible adiabatic pumping ($s_3 = s_4$)

$$W_p = h_3 - h_4 \ (\frac{kJ}{kg})$$

h_4 – Pump exit & boiler inlet (subcooled liquid) *enthalpy*

$$W_p = V_{f3}(P_4 - P_3) \ (\frac{kJ}{kg})$$

$$h_4 = W_p + h_3 \ (\frac{kJ}{kg})$$

Process 4-1: Boiler heat supplied (Constant pressure heat supplied) : Subcooled liquid to dry or superheated steam ($P_4 = P_1$)

$$Q_s = h_1 - h_4 \ (\frac{kJ}{kg})$$

Efficiency :

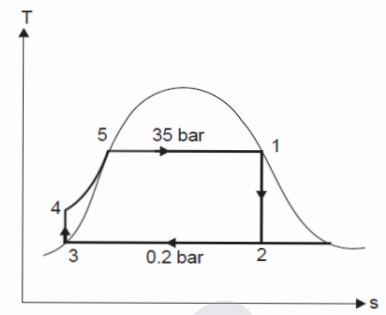
$$\eta = \frac{W_T - W_p}{Q_s}$$

$$= \frac{(h_1 - h_2) - (h_3 - h_4)}{h_1 - h_4}$$

2. **In a Rankine cycle, the steam at inlet to turbine is saturated at a pressure of 35 bar and the exhaust pressure is 0.2 bar. Determine : (i) The turbine work (ii) The condenser heat flow, (iii) The pump work, (iv) Heat Supplied to the boiler, (v) Network done, (vi)The Rankine efficiency, (vii) The dryness at the end of expansion, (viii) Carnot efficiency (ix) SSC, (x)Work ratio, (xi) Turbine power output if flow rate of 9.5 kg/s.**

GIVEN: $p_1 = 35$ bar, $p_2 = 0.2$ bar

FIND : (i) W, (ii) η



SOLUTION:

State 1

From steam tables at 35 bar

$s_1 = s_g = 6.1228$ kJ/kgK, $h_1 = h_g = 2802$ kJ/kg

State 2

From steam tables at 0.2 bar

$s_{f2} = 0.8321$ kJ/kgK , $s_{fg2} = 7.0773$kJ/kgK ,

$h_{f2} = 251.5$ kJ/kg $h_{fg2} = 2358.4$ kJ/kg ,

$v_{f2} = 0.001017$ m^3/kg

Enthalpy of wet steam (after expansion) $(s_1 = s_2)$

$s_1 = s_2 = s_{f2} + x_2 s_{fg2} \Rightarrow 6.1228 = 0.8321 + (x_2 \times 7.0773) \Rightarrow \mathbf{x_2 = 0.747}$

$h_2 = h_{f2} + x_2 h_{fg2} \Rightarrow h_2 = 251.5 + (0.747 \times 2358.4) \Rightarrow \mathbf{h_2 = 2013 \frac{kJ}{kg}}$

State 3

$h_3 = h_{f2} = h_{f3} = \mathbf{251.5 \frac{kJ}{kg}}$

State 4

$W_p = V_{f3}(P_4 - P_3) \Rightarrow W_p = 0.001017\,(35 - 0.2) \times 10^2 \Rightarrow \mathbf{W_p = 3.54\ kJ/kg}$

$h_4 = W_p + h_3 \Rightarrow h_4 = 3.54 + 251.5 \Rightarrow \mathbf{h_4 = 255.04\ kJ/kg}$

PROCESS 1-2: TURBINE WORK : REVERSIBLE ADIABATIC EXPANSION

$W_T = (h_1 - h_2) \Rightarrow W_T = (2802 - 2013) \Rightarrow \mathbf{W_T = 789 \frac{kJ}{kg}}$

PROCESS 2-3: CONDENSER HEAT REJECTION: CONSTANT PRESSURE HEAT REJECTION $(P_2 = P_3)$

$Q_R = h_2 - h_3 \Rightarrow Q_R = 2013 - 251.5 \Rightarrow \mathbf{Q_R = 1761.5 \frac{kJ}{kg}}$

PROCESS 3-4: PUMP WORK: REVERSIBLE ADIABATIC PUMPING $(s_3 = s_4)$

$W_p = V_{f3}(P_4 - P_3) \Rightarrow W_p = 0.001017\,(35 - 0.2) \times 10^2 \Rightarrow \mathbf{W_p = 3.54\ kJ/kg}$

PROCESS 4-1: BOILER HEAT SUPPLIED: CONSTANT PRESSURE HEAT SUPPLIED $(P_4 = P_1)$

$Q_s = h_1 - h_4 \Rightarrow Q_s = 2802 - 255.04 \Rightarrow \mathbf{Q_s = 2546.96\ kJ/kg}$

NET WORK DONE:

$W_{Net} = W_T - W_p \Rightarrow W_{Net} = 789 - 3.54 \Rightarrow \mathbf{W_{Net} = 785.46\ kJ/kg}$

EFFICIENCY :

$$\eta = \frac{W_{Net}}{Q_s} \Rightarrow \eta = \frac{785.46}{2546.96} \Rightarrow \boldsymbol{\eta = 30.84\%}$$

CARNOT EFFICIENCY:

$$\eta = 1 - \frac{T_2}{T_1} \Rightarrow \eta = 1 - \frac{333.9}{515.5} \Rightarrow \boldsymbol{\eta = 35\%}$$

SPECIFIC STEAM CONSUMPTION:

$$SSC = \frac{3600}{W_{net}} \Rightarrow SSC = \frac{3600}{785.46} \Rightarrow \boldsymbol{SSC = 4.58\ kg/kWh}$$

WORK RATIO:

$$W_r = \frac{W_{net}}{W_T} \Rightarrow W_r = \frac{785.46}{789} \Rightarrow \boldsymbol{W_r = 0.995}$$

TURNINE POWER OUTPUT:

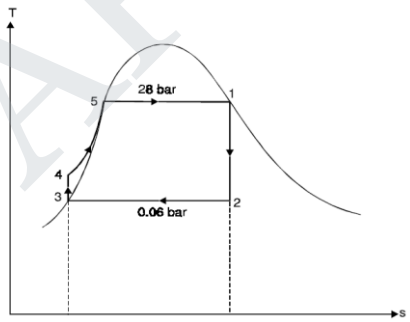
$$P = \dot{m}(h_1 - h_2) \Rightarrow P = 9.5(2802 - 2013) \Rightarrow \boldsymbol{P = 7495.5\ kW}$$

3. **A simple Rankine cycle works between pressures 28 bar and 0.06 bar, the initial condition of steam being dry saturated. Calculate the cycle efficiency, work ratio and specific steam consumption.**

GIVEN: $p_1 = 28$ bar, $p_2 = 0.06$ bar

FIND : (i) W_r , (ii) η, (iii) SSC

SOLUTION:



State 1

From Superheated steam tables at 28 bar

$s_1 = s_g = 6.2104$ kJ/kgK, $h_1 = h_g = 2802$ kJ/kg

State 2

From steam tables at 0.06 bar

$s_{f2} = 0.521$ kJ/kgK , $s_{fg2} = 7.809$ kJ/kgK ,

$h_{f2} = 151.5$ kJ/kg $h_{fg2} = 2415.9$ kJ/kg ,

$v_{f2} = 0.001 m^3/kg$

Enthalpy of wet steam (after expansion) $(\mathbf{s_1 = s_2})$

$$s_1 = s_2 = s_{f2} + x_2 s_{fg2} \Rightarrow 6.2104 = 0.521 + (\mathbf{x_2} \times 7.809) \Rightarrow \mathbf{x_2 = 0.728}$$

$$h_2 = h_{f2} + x_2 h_{fg2} \Rightarrow h_2 = 151.5 + (0.728 \times 2415.9) \Rightarrow \mathbf{h_2 = 1910.27\ \frac{kJ}{kg}}$$

State 3

$$h_3 = h_{f2} = h_{f3} = \mathbf{151.5\ \frac{kJ}{kg}}$$

State 4

$$W_p = V_{f3}(P_4 - P_3) \Rightarrow W_p = 0.001\ (28 - 0.06) \times 10^2 \Rightarrow \mathbf{W_p = 2.79\ kJ/kg}$$

$$h_4 = W_p + h_3 \Rightarrow h_4 = 2.79 + 151.5 \Rightarrow \mathbf{h_4 = 154.29\ kJ/kg}$$

PROCESS 1-2: TURBINE WORK : REVERSIBLE ADIABATIC EXPANSION

$$W_T = (h_1 - h_2) \Rightarrow W_T = (2802 - 1910.27) \Rightarrow \mathbf{W_T = 891.73\ \frac{kJ}{kg}}$$

PROCESS 2-3: CONDENSER HEAT REJECTION: CONSTANT PRESSURE HEAT REJECTION ($P_2 = P_3$)

$Q_R = h_2 - h_3 \Rightarrow Q_R = 1910.27 - 151.5 \Rightarrow \mathbf{Q_R = 1758.77\ \frac{kJ}{kg}}$

PROCESS 3-4: PUMP WORK: REVERSIBLE ADIABATIC PUMPING ($s_3 = s_4$)

$W_p = V_{f3}(P_4 - P_3) \Rightarrow W_p = 0.001\ (28 - 0.06) \times 10^2 \Rightarrow \mathbf{W_p = 2.79\ kJ/kg}$

PROCESS 4-1: BOILER HEAT SUPPLIED: CONSTANT PRESSURE HEAT SUPPLIED ($P_4 = P_1$)

$Q_s = h_1 - h_4 \Rightarrow Q_s = 2802 - 154.29 \Rightarrow \mathbf{Q_s = 2647.71\ kJ/kg}$

NET WORK DONE:

$W_{Net} = W_T - W_p \Rightarrow W_{Net} = 891.73 - 2.79 \Rightarrow \mathbf{W_{Net} = 888.94\ kJ/kg}$

EFFICIENCY :

$\eta = \frac{W_{Net}}{Q_s} \Rightarrow \eta = \frac{888.94}{2647.71} \Rightarrow \boldsymbol{\eta = 33.57\%}$

SPECIFIC STEAM CONSUMPTION:

$SSC = \frac{3600}{W_{net}} \Rightarrow SSC = \frac{3600}{888.94} \Rightarrow \mathbf{SSC = 4.049\ kg/kWh}$

WORK RATIO:

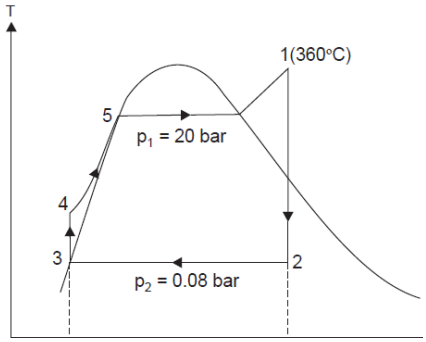
$W_r = \frac{W_{net}}{W_T} \Rightarrow W_r = \frac{888.94}{891.73} \Rightarrow \mathbf{W_r = 0.997}$

4. **In a steam turbine steam at 20 bar, 360°C is expanded to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. Assume ideal processes, find per kg of steam the net work and the cycle efficiency.**

GIVEN: $p_1 = 20$ bar, $T_1 = 360°C$, $p_2 = 0.08$ bar

FIND : (i) W, (ii) η

SOLUTION:



From Superheated steam tables at 20 bar, 360°C

By interpolation

$\frac{h_R - h_b}{h_a - h_b} = \frac{T_R - T_b}{T_a - T_b} \Rightarrow \frac{h_R - 3138.6}{3248.7 - 3138.6} = \frac{360 - 350}{400 - 350} \Rightarrow \mathbf{h_r = 3159.3\ \frac{kJ}{kg} = h_1}$

$\frac{s_R - s_b}{s_a - s_b} = \frac{T_R - T_b}{T_a - T_b} \Rightarrow \frac{s_R - 6.960}{7.130 - 6.960} = \frac{360 - 350}{400 - 350} \Rightarrow \mathbf{s_r = 6.991\ \frac{kJ}{kg.k} = s_1}$

From steam tables at 0.08 bar

$s_{f2} = 0.5926$ kJ/kgK , $s_{fg2} = 7.6361$ kJ/kgK ,

$h_{f2} = 173.88$ kJ/kg $h_{fg2} = 2403.1$ kJ/kg ,

$v_{f2} = 0.001008\ m^3/kg$

Enthalpy of wet steam (after expansion) ($s_1 = s_2$)

$s_2 = s_{f2} + x_2 s_{fg2} \Rightarrow 6.991 = 0.5926 + (\mathbf{x_2} \times 7.6361) \Rightarrow \mathbf{x_2 = 0.838}$

$h_2 = h_{f2} + x_2 h_{fg2} \Rightarrow h_2 = 173.88 + (0.838 \times 2403.1) \Rightarrow \mathbf{h_2 = 2187.68\ \frac{kJ}{kg}}$

PROCESS 1-2: TURBINE WORK : REVERSIBLE ADIABATIC EXPANSION

$W_T = (h_1 - h_2) \Rightarrow W_T = (3159.3 - 2187.68) \Rightarrow \mathbf{W_T} = 971.62\ \frac{kJ}{kg}$

PROCESS 2-3: CONDENSER HEAT REJECTION: CONSTANT PRESSURE HEAT REJECTION $(P_2 = P_3)$

$Q_R = h_2 - h_3 \Rightarrow Q_R = 2187.62 - 173.88 \Rightarrow \mathbf{Q_R = 2013.74\ \frac{kJ}{kg}}$

$h_3 = h_{f2} = h_{f3}$ Condenser exit and pump inlet (saturated liquid) enthalpy

PROCESS 3-4: PUMP WORK: REVERSIBLE ADIABATIC PUMPING $(s_3 = s_4)$

$W_p = V_{f3}(P_4 - P_3) \Rightarrow W_p = 0.00108\ (20 - 0.08) \times 10^2 \Rightarrow \mathbf{W_p = 2.008\ kJ/kg}$

$h_4 = W_p + h_3 \Rightarrow h_4 = 2.008 + 173.88 \Rightarrow \mathbf{h_4 = 175.89\ kJ/kg}$

h_4 – Pump exit & boiler inlet (subcooled liquid) enthalpy

PROCESS 4-1: BOILER HEAT SUPPLIED: CONSTANT PRESSURE HEAT SUPPLIED $(P_4 = P_1)$

$Q_s = h_1 - h_4 \Rightarrow Q_s = 3159.3 - 175.89 \Rightarrow \mathbf{Q_s = 2983.41\ kJ/kg}$

NET WORK DONE:

$W_{Net} = W_T - W_p \Rightarrow W_{Net} = 971.62 - 2.008 \Rightarrow \mathbf{W_{Net} = 969.61\ kJ/kg}$

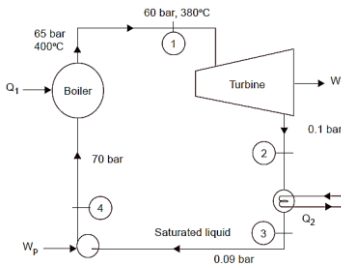
EFFICIENCY :

$\eta = \frac{W_{Net}}{Q_s} \Rightarrow \eta = \frac{969.61}{2983.41} \Rightarrow \mathbf{\eta = 32.5\%}$

5. **The following data refer to a simple steam power plant :**
Calculate :(i) Power output of the turbine, (ii) Heat transfer per hour in the boiler and condenser separately, (iii) Mass of cooling water circulated per hour in the condenser. Choose the inlet temperature of cooling water 20°C and 30°C at exit from the condenser, (iv) Diameter of the pipe connecting turbine with condenser.

S. No.	Location	Pressure	Quality/temp.	Velocity
1.	Inlet to turbine	6 MPa (= 60 bar)	380°C	—
2.	Exit from turbine inlet to condenser	10 kPa (= 0.1 bar)	0.9	200 m/s
3.	Exit from condenser and inlet to pump	9 kPa (= 0.09 bar)	Saturated liquid	—
4.	Exit from pump and inlet to boiler	7 MPa (= 70 bar)	—	—
5.	Exit from boiler Rate of steam flow = 10000 kg/h.	6.5 MPa (= 65 bar)	400°C	—

SOLUTION:



(I) POWER OUTPUT OF THE TURBINE, P :

From Superheated steam tables

At 60 bar, 380°C (380°C is not available in steam table)

By interpolation

$\frac{h_R - h_b}{h_a - h_b} = \frac{T_R - T_b}{T_a - T_b} \Rightarrow \frac{h_R - 3045.8}{3180.1 - 3045.8} = \frac{380 - 350}{400 - 350} \Rightarrow \mathbf{h_r = 3123.5\ \frac{kJ}{kg} = h_1}$

From steam tables at 0.1 bar

$h_{f2} = 191.8\ kJ/kg$

$h_{fg2} = 2392.8\ kJ/kg\ ,\ x_2 = 0.9$

$v_{g2} = 14.67\ m^3/kg$

$h_2 = h_{f2} + x_2 h_{fg2} \Rightarrow h_2 = 191.8 + (0.9 \times 2392.8) \Rightarrow \mathbf{h_2 = 2345.3\ kJ/kg}$

$W_T = m(h_1 - h_2) \Rightarrow W_T = \frac{10000}{3600}(3123.5 - 2392.8) \Rightarrow \mathbf{W_T = 2162kW}$

HEAT TRANSFER PER HOUR IN THE BOILER AND CONDENSER :

From steam tables at 70 bar

$h_{f4} = 1267.4\ kJ/kg$

From Superheated steam tables at 65 bar, 400°C

$h_{a1} = 3167.6\ kJ/kg$ (by interpolation)

HEAT TRANSFER PER HOUR IN THE BOILER,

$Q_1 = m(h_{a1} - h_{f4}) \Rightarrow Q_1 = \frac{10000}{3600}(3167.6 - 1267.4) \Rightarrow \mathbf{Q_1 = 5277.78kW}$

From steam tables at 70 bar

$h_{f3} = 183.3\ kJ/kg$

HEAT TRANSFER PER HOUR IN THE CONDENSER,

$Q_2 = m(h_2 - h_{f3}) \Rightarrow Q_2 = \frac{10000}{3600}(2345.3 - 183.3) \Rightarrow \mathbf{Q_2 = 6000\ kW}$

MASS OF COOLING WATER CIRCULATED PER HOUR IN THE CONDENSER, m_w :

Heat lost by steam = Heat gained by the cooling water

$Q_2 = m_w C_P(t_2 - t_1) \Rightarrow Q_2 = m_w \times 4.18(30 - 20) \Rightarrow \mathbf{m_w = 3100\ kg/s}$

DIAMETER OF THE PIPE CONNECTING TURBINE WITH CONDENSER, d :

$m_s = \rho AC \Rightarrow m_s = \frac{AC}{x_2 v_{g2}} \Rightarrow \frac{m_s x_2 v_{g2}}{C} = \frac{\pi}{4}d^2 \Rightarrow \frac{2.78\times0.9\times14.7}{200} = \frac{\pi}{4}d^2 \Rightarrow \mathbf{d = 483mm}$

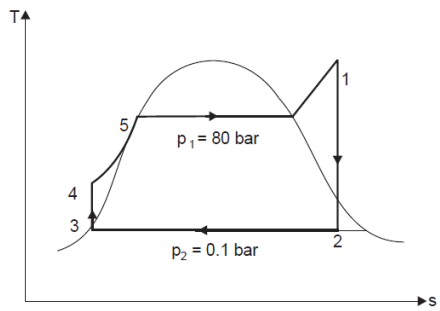
6. A Rankine cycle operates between pressures of 80 bar and 0.1 bar. The maximum cycle temperature is 600°C. If the steam turbine and condensate pump efficiencies are 0.9 and 0.8 respectively, calculate the specific work and thermal efficiency. Relevant steam table extract is given below.

p(bar)	t(°C)	Specific volume (m^3/kg)		Specific enthalpy (kJ/kg)			Specific entropy ($kJ/kg\ K$)		
		v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0.1	45.84	0.0010103	14.68	191.9	2392.3	2584.2	0.6488	7.5006	8.1494
80	295.1	0.001385	0.0235	1317	1440.5	2757.5	3.2073	2.5351	5.7424

80 bar, 600°C	v	$0.486\ m^3/kg$
Superheat	h	$3642\ kJ/kg$
table	s	$7.0206\ kJ/kgK$

GIVEN: $p_1 = 80$ bar, $T_1 = 600°C$, $p_2 = 0.1$ bar

FIND : (i) W, (ii) η



SOLUTION:

State 1

From Superheated steam tables at 80 bar, 600°C

$s_1 = 7.0206\ kJ/kgK,\ h_1 = 3642\ kJ/kg,\ v_{g2} = 0.486\ m^3/kg$

State 2

From steam tables at 0.1 bar

$s_{f2} = 0.6488\ kJ/kgK,\ s_{fg2} = 7.5006\ kJ/kgK,$

$h_{f2} = 191.9\ kJ/kg\ \ h_{fg2} = 2392.3\ kJ/kg,$

$v_{f2} = 0.0010103\ m^3/kg$

Enthalpy of wet steam (after expansion)

$$s_2 = s_{f2} + x_2 s_{fg2} \Rightarrow 7.0206 = 0.6488 + (\mathbf{x_2} \times 7.006) \Rightarrow \mathbf{x_2 = 0.85}$$

$$h_2 = h_{f2} + x_2 h_{fg2} \Rightarrow h_2 = 191.9 + (0.85 \times 2392.3) \Rightarrow \mathbf{h_2 = 2225.36\ \frac{kJ}{kg}}$$

State 3

$$h_3 = h_{f2} = h_{f3} = \mathbf{191.9\ \frac{kJ}{kg}}$$

State 4

$$W_p = V_{f3}(P_4 - P_3) \Rightarrow W_p = 0.486\,(60 - 0.1) \times 10^2 \Rightarrow \mathbf{W_p = 8.072\ kJ/kg}$$

$$\text{Actual pump work} = \frac{8.072}{\eta_{pump}} = \frac{8.072}{0.8} \Rightarrow \mathbf{W_p = 10.09\ kJ/kg}$$

$$h_4 = W_p + h_3 \Rightarrow h_4 = 10.09 + 191.9 \Rightarrow \mathbf{h_4 = 201.99\ kJ/kg}$$

PROCESS 1-2: TURBINE WORK : REVERSIBLE ADIABATIC EXPANSION

$$W_T = (h_1 - h_2) \Rightarrow W_T = (3642 - 2225.36) \Rightarrow \mathbf{W_T = 1275\ \frac{kJ}{kg}}$$

PROCESS 2-3: CONDENSER HEAT REJECTION: CONSTANT PRESSURE HEAT REJECTION ($P_2 = P_3$)

$$Q_R = h_2 - h_3 \Rightarrow Q_R = 2225.36 - 191.9 \Rightarrow \mathbf{Q_R = 2033.76\ \frac{kJ}{kg}}$$

PROCESS 3-4: PUMP WORK: REVERSIBLE ADIABATIC PUMPING ($s_3 = s_4$)

$$\mathbf{W_p = 10.09\ kJ/kg}$$

PROCESS 4-1: BOILER HEAT SUPPLIED: CONSTANT PRESSURE HEAT SUPPLIED ($P_4 = P_1$)

$$Q_s = h_1 - h_4 \Rightarrow Q_s = 3642 - 201.99 \Rightarrow \mathbf{Q_s = 3440.01\ kJ/kg}$$

NET WORK DONE:

$$W_{Net} = W_T - W_p \Rightarrow W_{Net} = 1275 - 10.09 \Rightarrow \mathbf{W_{Net} = 1264.91\ kJ/kg}$$

EFFICIENCY :

$$\eta = \frac{W_{Net}}{Q_s} \Rightarrow \eta = \frac{1264.91}{3440.01} \Rightarrow \boldsymbol{\eta = 36.8\%}$$

SPECIFIC STEAM CONSUMPTION:

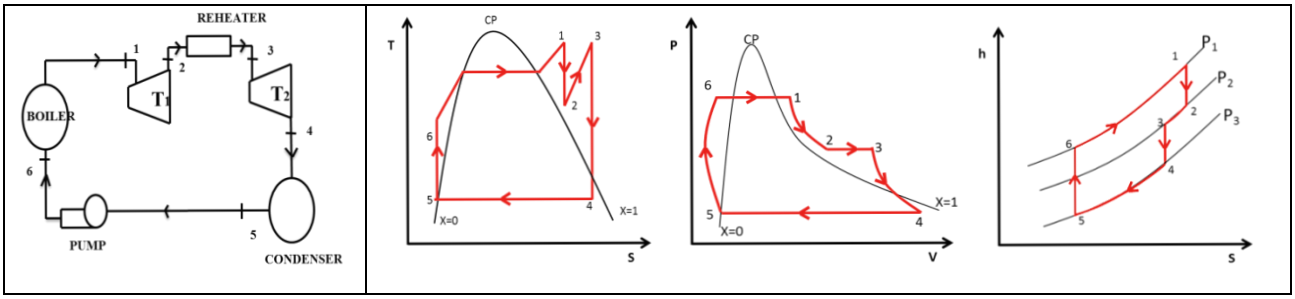
$$SSC = \frac{3600}{W_{net}} \Rightarrow SSC = \frac{3600}{1264.91} \Rightarrow \mathbf{SSC = 2.85\ kg/kWh}$$

WORK RATIO:

$$W_r = \frac{w_{net}}{W_T} \Rightarrow W_r = \frac{1264.91}{1275} \Rightarrow \boldsymbol{W_r = 0.992}$$

7. **Draw the P-V, T-S, h -s, diagrams and theoretical lay out for Reheat Rankine cycle and hence deduce the expression for its efficiency.**

SOLUTION:



PROCESS 1-2: TURBINE WORK : REVERSIBLE ADIABATIC EXPANSION $(s_1 = s_2)$

$$W_{T1} = h_1 - h_2 \left(\frac{kJ}{kg}\right)$$

h_1 – Turbine1 inlet (**dry or superheated steam**) enthalpy

h_2 – Turbine exit and reheater intet (**wet or dry steam**) enthalpy

PROCESS 2-3: CONSTANT PRESSURE REHEATING $(P_2 = P_3)$

$$Q_{s1} = h_3 - h_2 \left(\frac{kJ}{kg}\right)$$

h_3 – Turbine1 inlet (**dry or superheated steam**) enthalpy

PROCESS 3-4: TURBINE WORK2 : REVERSIBLE ADIABATIC EXPANSION $(s_3 = s_4)$

$$W_{T2} = h_3 - h_4 \left(\frac{kJ}{kg}\right)$$

h_4 – Turbine exit and condenser intet (**wet or dry steam**) enthalpy

PROCESS 4-5: CONDENSER HEAT REJECTION: CONSTANT PRESSURE HEAT REJECTION $(P_4 = P_5)$

$$Q_R = h_4 - h_5 \left(\frac{kJ}{kg}\right)$$

$h_5 = h_{f4} = h_{f5}$ Condenser exit (saturated liquid) enthalpy

PROCESS 5-6: PUMP WORK: REVERSIBLE ADIABATIC PUMPING $(s_5 = s_6)$

$$W_p = h_6 - h_5 \left(\frac{kJ}{kg}\right)$$

h_6 – Pump exit & boiler inlet (subcooled liquid) enthalpy

$$W_p = V_{f5}(P_6 - P_5) \left(\frac{kJ}{kg}\right)$$

$$h_6 = W_p + h_5 \left(\frac{kJ}{kg}\right)$$

PROCESS 6-1: BOILER HEAT SUPPLIED: CONSTANT PRESSURE HEAT SUPPLIED $(P_6 = P_1)$

$$Q_{s2} = h_1 - h_6 \left(\frac{kJ}{kg}\right)$$

EFFICIENCY :

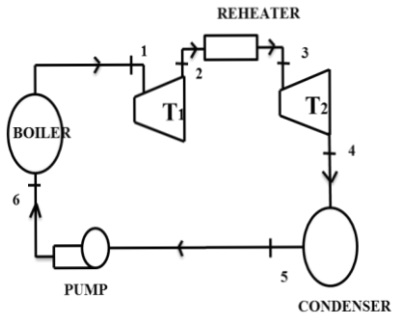
$$\eta = \frac{W_{T1} + W_{T2} - W_p}{Q_{s1} + Q_{s2}} = \frac{(h_1 - h_2) + (h_4 - h_3) - (h_6 - h_5)}{(h_3 - h_2) + (h_1 - h_6)}$$

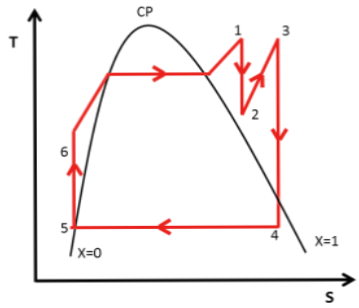
8. **A reheat Rankine cycle receives steam at 35 bar and 0.1 bar. Steam enters the first stage steam turbine 350 °C. If reheating is done at 8 bar and 350 °C, calculate the specific steam consumption and reheat Rankine cycle efficiency.**

GIVEN:

P_1=35 bar, T_1=350 °C, P_3=8 bar,

T_3=350 °C, P_4=0.1 bar





SOLUTION:

PROCESS 1-2: ADIABATIC EXPANSION PROCESS

From the superheated steam table at 35 bar and 350 °C

By interpolation

$$\frac{h_r-h_b}{h_a-h_b}=\frac{p_r-p_b}{p_a-p_a} \Rightarrow \frac{h_r-3104.2}{3108.7-3104.2}=\frac{35-34}{36-34} \Rightarrow \mathbf{h_r = 3106.45\frac{kJ}{kg} = h_1}$$

$$\frac{s_r-s_b}{s_a-s_b}=\frac{p_r-p_b}{p_a-p_a} \Rightarrow \frac{s_r-6.647}{6.679-6.647}=\frac{35-34}{36-34} \Rightarrow \mathbf{s_r = 6.663\frac{kJ}{kg.k} = s_1}$$

From saturated steam table at 8 bar

$$s_g = 6.660\frac{kJ}{kg.K}$$

Where, $\mathbf{s_1 = s_2} = s_g$ **,** so the exit of turbine is saturated steam

From saturated steam table at 8 bar

$$\mathbf{h_2 = 2767.4\frac{kJ}{kg} = h_1}$$

TURBINE WORK

$$w_{T1} = h_1 - h_2 \Rightarrow w_{T1} = 3106.45 - 2767.4 \Rightarrow \mathbf{w_{T1} = 339.05 kJ/kg}$$

PROCESS 2-3: CONSTANT PRESSURE HEAT ADDITION

From superheated steam table at 8 bar and 350 °C

$$\mathbf{h_3 = 3162.4\frac{kJ}{kg}}, \quad \mathbf{s_3 = 7.411\frac{kJ}{kg.K}}$$

HEAT SUPPLIED TO REHEATER

$$q_{s2} = h_3 - h_2 \Rightarrow q_{s2} = 3162.4 - 2767.4 \Rightarrow \mathbf{q_{s2} = 395\frac{kJ}{kg.K}}$$

PROCESS 3-4: ADIABATIC EXPANSION PROCESS IN TURBINE 2: $s_3 = s_4$

From saturated steam table at 0.1 bar

$$s_{g4} = 8.151\frac{kJ}{kg.K}$$

$\mathbf{s_3 = s_4} < s_{g4}$, so the exit of turbine is wet steam

From saturated steam table at 0.1 bar

$$s_{f4} = 0.649\frac{kJ}{kg.K}, \quad s_{fg4} = 7.502\frac{kJ}{kg.K}, \quad h_{f4} = 191.8\frac{kJ}{kg}, \quad h_{fg4} = 2392.8\frac{kJ}{kg}$$

$$s_4 = s_{f4} + x_4 s_{fg4} \Rightarrow 7.411 = 0.649 + (\mathbf{x_4} \times 7.502) \Rightarrow \mathbf{x_4 = 0.9}$$

$h_4 = h_{f4} + x_4 h_{fg4}$ ⇨ $h_4 = 191.8 + (0.9 \times 2392.8)$ ⇨ $\mathbf{h_4 = 2345.32 \frac{kJ}{kg}}$

$w_{T2} = h_3 - h_4$ ⇨ $w_{T2} = 3162.4 - 2345.32$ ⇨ $\mathbf{w_{T2} = 817.08 kJ/kg}$

PROCESS 4-5: CONSTANT PRESSURE HEAT REJECTION

$$\mathbf{h_{f4} = h_{f5} = h_5 = 191.8\ kJ/kg}$$

PROCESS 5-6: ADIABATIC PUMPING

$W_p = v_{f4}(P_6 - P_5)$ ⇨ $W_p = 0.001010 \times (35 - 0.1) \times 100$ ⇨ $\mathbf{W_p = 3.53\ kJ/kg}$

$W_p = h_6 - h_5$ ⇨ $3.53 = h_6 - 191.8$ ⇨ $\mathbf{h_6 = 195.33\ kJ/kg}$

PROCESS 5-6: CONSTANT PRESSURE HEAT ADDITION

$q_{s1} = h_1 - h_6$ ⇨ $q_{s1} = 3106.45 - 195.33$ ⇨ $\mathbf{q_{s1} = 2911.12 \frac{kJ}{kg.K}}$

RANKINE CYCLE EFFICIENCY

$\eta = \frac{W_{T1} + W_{T2} - W_p}{Q_{s1} + Q_{s2}} = \frac{339.05 + 817.08 - 3.53}{395 + 2911.12}$ ⇨ $\boldsymbol{\eta = 34.86\%}$

SPECIFIC STEAM CONSUMPTION

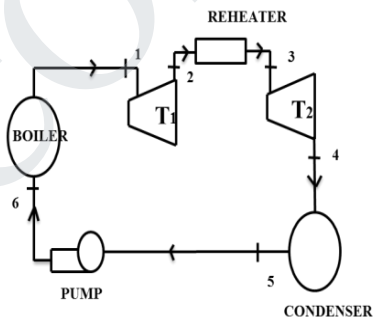
$SSC = \frac{3600}{W_{net}}$ ⇨ $SSC = \frac{3600}{339.05 + 817.08}$ ⇨ $\mathbf{SSC = 3.12\ kg/kWh}$

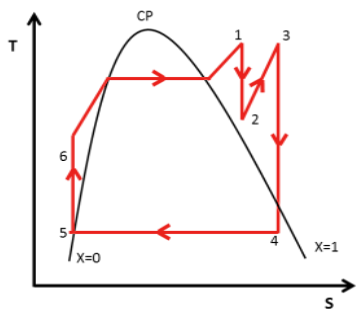
9. A steam power plant operates on a theoretical reheat cycle. Steam at boiler at 150bar, 550 °C expands through the high pressure turbine. It is reheated at a constant pressure of 40 bar to 550 °C and expands through the low pressure turbine to a condenser at 0.1 bar. Draw T-s and h-s diagram. Find (i) Quality of steam at turbine exhaust (ii) Cycle efficiency (iii) Steam Rate in kg/kWh.

GIVEN:

P_1=150 bar, T_1=550 ºC, P_3=40 bar,

T_3=550 ºC, P_4=0.1 bar





SOLUTION:

PROCESS 1-2: ADIABATIC EXPANSION PROCESS

State 1

From the superheated steam table at 150 bar and 550 ºC

$\mathbf{h_1 = 3445.2 \frac{kJ}{kg}}$, $\mathbf{s_1 = 6.5125 \frac{kJ}{kg.k}}$

State 2

From saturated steam table at 40 bar

$$s_g = 6.069 \frac{kJ}{kg.K}$$

Where, $\mathbf{s_1 = s_2} > s_g$, so the exit of turbine is Superheated steam

From saturated steam table at $\mathbf{s_2 = 6.5125 \frac{kJ}{kg.k}}$ **and 40 bar**

By interpolation

$\frac{T_r-T_b}{T_a-T_b}=\frac{s_r-s_b}{s_a-s_b}$ ⇒ $\frac{T_r-300}{350-300}=\frac{6.5125-6.364}{6.587-6.364}$ ⇒ $\mathbf{T_r = 332°C = T_2}$

$\frac{h_r-h_b}{h_a-h_b}=\frac{T_r-T_b}{T_a-T_b}$ ⇒ $\frac{h_r-2962}{3095.1-2962}=\frac{332-300}{350-300}$ ⇒ $\mathbf{h_r = 3047.18\,\frac{kJ}{kg} = h_2}$

TURBINE WORK

$w_{T1} = h_1 - h_2$ ⇒ $w_{T1} = 3445.2 - 3047.18$ ⇒ $\mathbf{w_{T1} = 398.02\ kJ/kg}$

PROCESS 2-3: CONSTANT PRESSURE HEAT ADDITION

From superheated steam table at 8 bar and 350 °C

$\mathbf{h_3 = 3558.9\,\frac{kJ}{kg}}$, $\mathbf{s_3 = 7.229\,\frac{kJ}{kg.K}}$

HEAT SUPPLIED TO REHEATER

$q_{s2} = h_3 - h_2$ ⇒ $q_{s2} = 3558.9 - 3047.18$ ⇒ $\mathbf{q_{s2} = 511.72\,\frac{kJ}{kg.K}}$

PROCESS 3-4: ADIABATIC EXPANSION PROCESS IN TURBINE 2: $s_3 = s_4$

From saturated steam table at 0.1 bar

$$s_{g4} = 8.151\,\frac{kJ}{kg.K}$$

$\mathbf{s_3 = s_4} < s_{g4}$, so the exit of turbine is wet steam

From saturated steam table at 0.1 bar

$s_{f4} = 0.649\,\frac{kJ}{kg.K}$, $s_{fg4} = 7.502\,\frac{kJ}{kg.K}$

$h_{f4} = 191.8\,\frac{kJ}{kg}$, $h_{fg4} = 2392.8\,\frac{kJ}{kg}$

$s_4 = s_{f4} + x_4 s_{fg4}$ ⇒ $7.411 = 0.649 + (\mathbf{x_4} \times 7.502)$ ⇒ $\mathbf{x_4 = 0.9}$

$h_4 = h_{f4} + x_4 h_{fg4}$ ⇒ $h_4 = 191.8 + (0.9 \times 2392.8)$ ⇒ $\mathbf{h_4 = 2345.32\,\frac{kJ}{kg}}$

$w_{T2} = h_3 - h_4$ ⇒ $w_{T2} = 3558.9 - 2345.32$ ⇒ $\mathbf{w_{T2} = 1213.58\ kJ/kg}$

PROCESS 4-5: CONSTANT PRESSURE HEAT REJECTION

$$\mathbf{h_{f4} = h_{f5} = h_5 = 191.8\ kJ/kg}$$

PROCESS 5-6: ADIABATIC PUMPING

$W_p = v_{f4}(P_6 - P_5)$ ⇒ $W_p = 0.001010 \times (150 - 0.1) \times 100$ ⇒ $\mathbf{W_p = 15.14\ kJ/kg}$

$W_p = h_6 - h_5$ ⇒ $15.14 = h_6 - 191.8$ ⇒ $\mathbf{h_6 = 206.94\ kJ/kg}$

PROCESS 5-6: CONSTANT PRESSURE HEAT ADDITION

$q_{s1} = h_1 - h_6$ ⇒ $q_{s1} = 3445.2 - 206.94$ ⇒ $\mathbf{q_{s1} = 3238.26\,\frac{kJ}{kg.K}}$

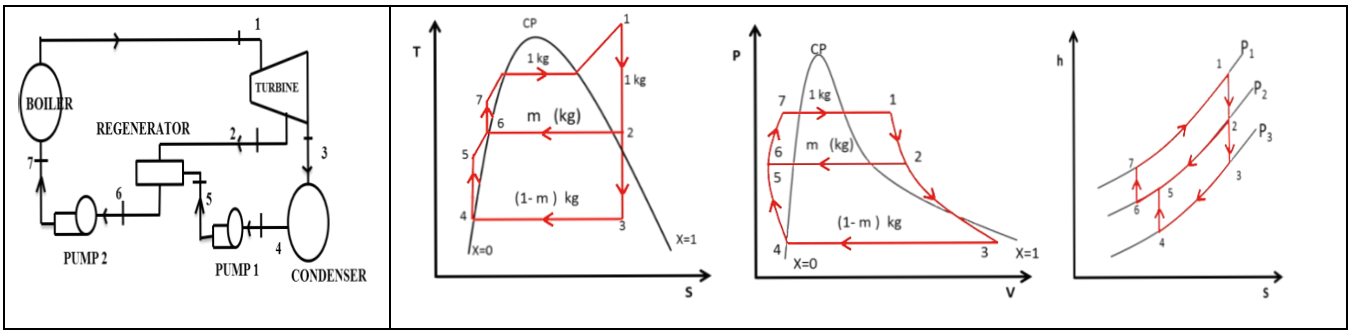
RANKINE CYCLE EFFICIENCY

$\eta = \frac{W_{T1}+W_{T2}-W_p}{Q_{s1}+Q_{s2}} = \frac{398.02+1213.58-15.14}{511.72+3238.26}$ ⇒ $\boldsymbol{\eta = 42.57\%}$

SPECIFIC STEAM CONSUMPTION

$SSC = \frac{3600}{W_{net}}$ ⇒ $SSC = \frac{3600}{398.02+1213.58-15.14}$ ⇒ $\mathbf{SSC = 2.254\ kg/kWh}$

10.Draw the P-V, T-S, h -s, diagrams and theoretical lay out for Regeneration Rankine cycle and hence deduce the expression for its efficiency.



PROCESS 1-2 & 1-3 TURBINE WORK

$$W_T = 1kg(h_1 - h_2) + (1 - m)(h_2 - h_3)$$

h_1 – Turbine1 inlet enthalpy

h_2 – bypass regeneration enthalpy

PROCESS 3-4: CONDENSER HEAT REJECTION: CONSTANT PRESSURE HEAT REJECTION $(P_3 = P_4)$

$$Q_R = h_3 - h_4 \left(\frac{kJ}{kg}\right)$$

h_3 – Turbine exit and condenser intet enthalpy

$h_4 = h_{f3} = h_{f4}$ Condenser exit enthalpy

TO FIND BYPASS STEAM MASS: ENERGY BALANCE

$$m((h_2 - h_6) = (1 - m)(h_6 - h_5)$$

PROCESS 4-5: PUMP WORK1: REVERSIBLE ADIABATIC PUMPING $(s_4 = s_5)$

$$W_{p1} = (1 - m)h_4 - h_5 \left(\frac{kJ}{kg}\right)$$

PROCESS 6-7: PUMP WORK2: REVERSIBLE ADIABATIC PUMPING $(s_6 = s_7)$

$$W_{p2} = 1kg\,(h_7 - h_6)\left(\frac{kJ}{kg}\right)$$

$$W_p = V_{f6}(P_7 - P_6)\left(\frac{kJ}{kg}\right)$$

$$h_7 = W_p + h_6 \left(\frac{kJ}{kg}\right)$$

PROCESS 7-1: BOILER HEAT SUPPLIED: CONSTANT PRESSURE HEAT SUPPLIED $(P_7 = P_1)$

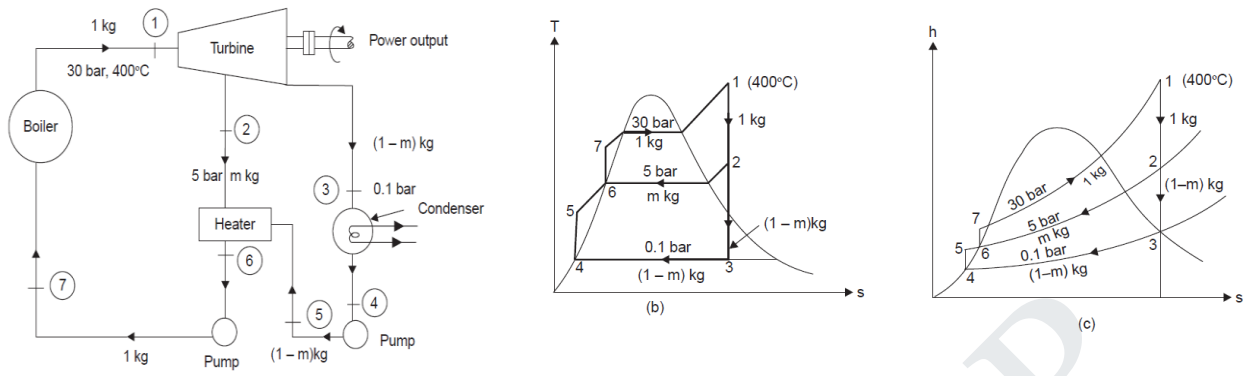
$$Q_{s1} = 1kg\,(h_1 - h_7)\left(\frac{kJ}{kg}\right)$$

EFFICIENCY

$$\eta = \frac{W_T}{Q_s} = \frac{1kg(h_1 - h_2) + (1 - m)(h_2 - h_3)}{1kg(h_1 - h_7)}$$

Note: Here Pump work is negligible

11.In a single-heater regenerative cycle the steam enters the turbine at 30 bar, 400°C and the exhaust pressure is 0.10 bar. The feed water heater is a direct contact type which operates at 5 bar. Find : (i) The efficiency and the steam rate of the cycle. (ii) The increase in mean temperature of heat addition, efficiency and steam rate as compared to the Rankine cycle (without regeneration). Pump work may be neglected.



PROCESS 1-2: ADIABATIC EXPANSION PROCESS

State 1

From the superheated steam table at 30 bar and 400 ºC

$$\mathbf{h_1 = 3232.5\,\frac{kJ}{kg}} \quad , \quad \mathbf{s_1 = 6.925\,\frac{kJ}{kg.k}}$$

State 2

From saturated steam table at 5 bar

$$s_g = 6.819\,\frac{kJ}{kg.K}$$

Where, $\mathbf{s_1 = s_2} > s_g$ **,** so the exit of turbine is Superheated steam

From saturated steam table at $\mathbf{s_2 = 6.925\,\frac{kJ}{kg.k}}$ **and 5 bar**

By interpolation

$$\frac{T_r - T_b}{T_a - T_b} = \frac{s_r - s_b}{s_a - s_b} \Rightarrow \frac{T_r - 151.8}{200 - 151.8} = \frac{6.925 - 6.819}{7.059 - 6.819} \Rightarrow \mathbf{T_r = 173.08°C = T_2}$$

$$\frac{h_r - h_b}{h_a - h_b} = \frac{T_r - T_b}{T_a - T_b} \Rightarrow \frac{h_r - 2747.5}{2855.1 - 2747.5} = \frac{173.08 - 151.8}{200 - 151.8} \Rightarrow \mathbf{h_r = 2795.02\,\frac{kJ}{kg} = h_2}$$

State 3

From saturated steam table at 0.1 bar

$$s_{g3} = 8.151\,\frac{kJ}{kg.K}$$

$\mathbf{s_1 = s_3} < s_{g3}$, so the exit of turbine is wet steam

From saturated steam table at 0.1 bar

$$s_{f3} = 0.649\,\frac{kJ}{kg.K} \quad , \quad s_{fg3} = 7.502\,\frac{kJ}{kg.K}$$

$$h_{f3} = 191.8\,\frac{kJ}{kg} \quad , \quad h_{fg3} = 2392.8\,\frac{kJ}{kg}$$

$$s_3 = s_{f3} + x_3 s_{fg3} \Rightarrow 6.925 = 0.649 + (\mathbf{x_3} \times 7.502) \Rightarrow \mathbf{x_3 = 0.837}$$

$$h_3 = h_{f3} + x_3 h_{fg3} \Rightarrow h_3 = 191.8 + (0.837 \times 2392.8) \Rightarrow \mathbf{h_3 = 2194.57\,\frac{kJ}{kg}}$$

State 4 and 6

Since pump work is neglected

$h_4 = h_{f4} = 191.8$ kJ/kg $= h_5$, $h_6 = h_{f6} = 640.1$ kJ/kg (at 5 bar) $= h_7$

ENERGY BALANCE FOR HEATER GIVES

$$m (h_2 - h_6) = (1 - m) (h_6 - h_5)$$

$$m (2796 - 640.1) = (1 - m) (640.1 - 191.8) = 448.3 (1 - m)$$

$$2155.9 m = 448.3 - 448.3 m$$

$$\therefore m = 0.172 \text{ kg}$$

TURBINE WORK,

$W_T = (h_1 - h_2) + (1 - m) (h_2 - h_3)$ ⇨ $W_T = (3230.9 - 2796) + (1 - 0.172) (2796 - 2192.2)$

$W_T = 434.9 + 499.9$ **$W_T = 934.8$ kJ/kg**

HEAT SUPPLIED,

$Q_1 = h_1 - h_{f6}$ ⇨ $Q_1 = 3230.9 - 640.1$ **$Q_1 = 2590.8$ kJ/kg.**

EFFICIENCY OF CYCLE, η_{cycle} :

$\eta_{cycle} = \frac{W_T}{Q_1}$ ⇨ $\eta_{cycle} = \frac{934.8}{2590.8}$ **$\eta_{cycle} = 0.3608$ or 36.08%**

STEAM RATE

$SSC = \frac{3600}{W_T}$ ⇨ $SSC = \frac{3600}{934.8}$ **SSC= 3.85 kg/kWh**

THE INCREASE IN MEAN TEMPERATURE OF HEAT ADDITION

$T_{m1} = \frac{h_1 - h_{f7}}{s_1 - s_7}$ ⇨ $T_{m1} = \frac{3230.9 - 191.8}{6.921 - 0.649}$ **$T_{m1} = 484.5$ K** or 211.5°C.

INCREASE IN T_{m1} DUE TO REGENERATION

Increase in $T_{m1} = 238.9 - 211.5$ **Increase in $T_{m1} = 27.4$°C**

WITHOUT REGENERATION:

TURBINE WORK

$W_T = h_1 - h_3$ ⇨ $W_T = 3230.9 - 2192.2$ **$W_T = 1038.7$ kJ/kg**

STEAM RATE WITHOUT REGENERATION

$SSC = \frac{3600}{W_T}$ ⇨ $SSC = \frac{3600}{1038.7}$ **SSC= 3.46 kg/kWh**

INCREASE IN STEAM RATE DUE TO REGENERATION

Increase in steam rate = 3.85 – 3.46 **Increase in steam rate = 0.39 kg/kWh**

EFFICIENCY OF CYCLE, η_{cycle} :

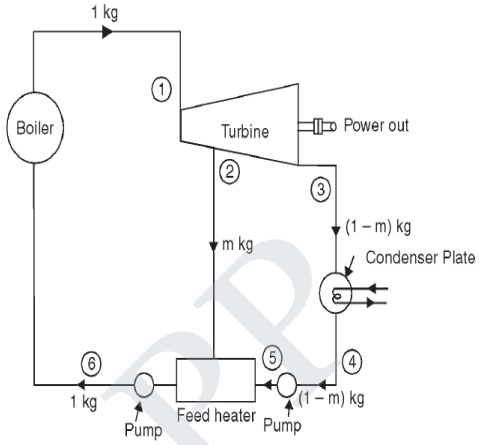
$\eta_{cycle} = \frac{h_1 - h_3}{h_1 - h_{f4}}$ **$\eta_{cycle} = 0.3418$ or 34.18%**

INCREASE IN CYCLE EFFICIENCY DUE TO REGENERATION

$\eta_{cycle} = 36.08 - 34.18$ **$\eta_{cycle} = 1.9$%**

12. A steam turbine is fed with steam having an enthalpy of 3100 kJ/kg. It moves out of the turbine with an enthalpy of 2100 kJ/kg. Feed heating is done at a pressure of 3.2 bar with steam enthalpy of 2500 kJ/kg. The condensate from a condenser with an enthalpy of 125 kJ/kg enters into the feed heater. The quantity of bled steam is 11200 kg/h. Find the power developed by the turbine. Assume that the water leaving the feed heater is saturated liquid at 3.2 bar and the heater is direct mixing type. Neglect pump work.

SOLUTION:



From the steam table at 3.2 bar

h_{f6} = 570.9 kJ/kg.

Consider **m** kg out of 1 kg is taken to the feed heater

Energy balance for the feed heater is written as :

$$(m \times h_2) + (1 - m)\, h_{f5} = 1 \times h_{f6}$$

$$m \times 2100 + (1 - m) \times 125 = 1 \times 570.9$$

$$2100\, m + 125 - 125\, m = 570.9$$

$$1975\, m = 570.9 - 125$$

m = 0.226 kg per kg of steam supplied to the turbine

STEAM SUPPLIED TO THE TURBINE PER HOUR

$$= \frac{11200}{0.226}$$

= 49557.5 kg/h

NET WORK DEVELOPED PER KG OF STEAM

$$= 1 \text{ kg } (h_1 - h_2) + (1 - m)(h_2 - h_3)$$

$$= (3100 - 2500) + (1 - 0.226)(2500 - 2100)$$

$$= 600 + 309.6$$

= 909.6 kJ/kg

POWER DEVELOPED BY THE TURBINE

= 12521.5 kW

13. Steam enters the turbine at 3 Mpa and 400°C and is condensed at 10kPa. Some quantity of steam leaves the turbine at 0.5 Mpa and enters feed water heater. Compute the fraction of the steam extracted per kg of steam and cycle thermal efficiency.

From steam tables :

At 30 bar, 400°C : h_1 = 3230.9 kJ/kg,
s_1 = 6.921 kJ/kg K = s_2 = s_3,

At 5 bar : s_f = 1.8604,
s_g = 6.8192 kJ/kg K,
h_f = 640.1 kJ/kg

Since $s_2 > s_g$, the state 2 must lie in the superheated region. From the table for superheated steam

$t_2 = 172°C,$

$h_2 = 2796$ kJ/kg.

At 0.1 bar : $s_f = 0.649,$

$s_{fg} = 7.501,$

$h_f = 191.8,$

$h_{fg} = 2392.8$

Now, $s_2 = s_3$

i.e., $6.921 = s_{f3} + x_3\, s_{fg3}$

$x_3 = 0.836$

$h_3 = h_{f3} + x_3\, h_{fg3}$

$= 2192.2$ kJ/kg

Since pump work is neglected

$h_{f4} = 191.8$ kJ/kg $= h_{f5}$

$h_{f6} = 640.1$ kJ/kg (at 5 bar) $= h_{f7}$

Energy balance for heater gives

$m\,(h_2 - h_{f6}) = (1 - m)\,(h_{f6} - h_{f5})$

$\therefore m = 0.172$ kg

$\therefore$ Turbine work, $W_T = (h_1 - h_2) + (1 - m)\,(h_2 - h_3)$

$= 934.8$ kJ/kg

Heat supplied, $Q_1 = h_1 - h_{f6}$

$= 2590.8$ kJ/kg.

Efficiency of cycle, η_{cycle} :

$$\eta_{cycle} = \frac{W_T}{Q_1}$$

$= 0.3608$ or 36.08%. (Ans.)

Steam rate = 3.85 kg/kWh. (Ans.)

14. In a steam generator compressed liquid water at 10 MPa, 30°C enters a 30 m diameter tube at the Rate of 3 litres/sec. Steam at 9MPa, 400°C exits the tube. Find the rate of heat transfer to the water.

GIVEN:

In boiler Pressurized water enters at section 1 (p_1=100 bar, 30°C)

At section 2 it exits the tube as Steam at 90 bar, 400°C.

Check the condition of water at 1. (find $v1$)

From volume flow rate find mass flow rate of water

and find h_1

Check the condition of water at 2. (find h_2)

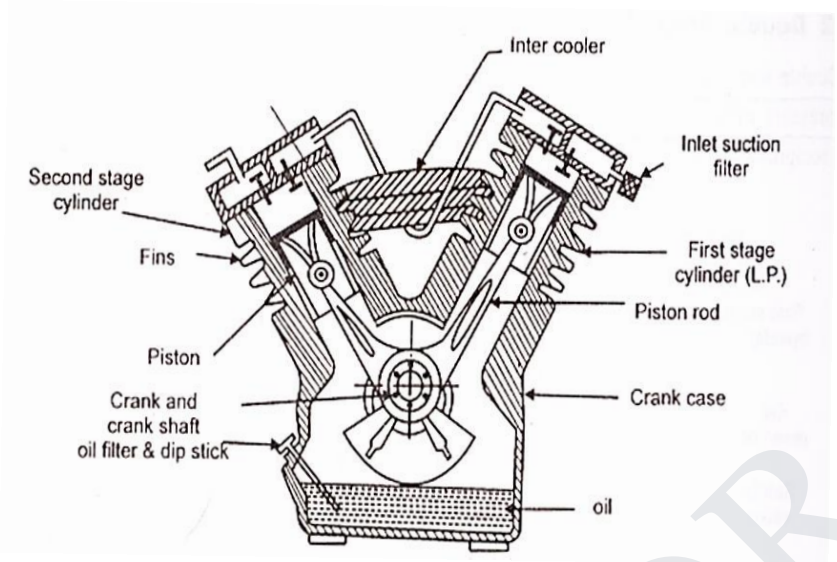
Find H.T rate $Q = m(h_2 - h_1)$ in kJ/min

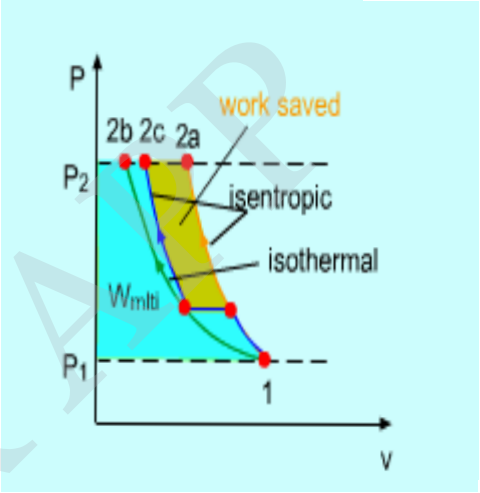
Given:
In boiler Pressurized water enters at section 1 (p_1=100 bar, 30°C)
At section 2 it exits the tube as Steam at 90 bar, 400°C.

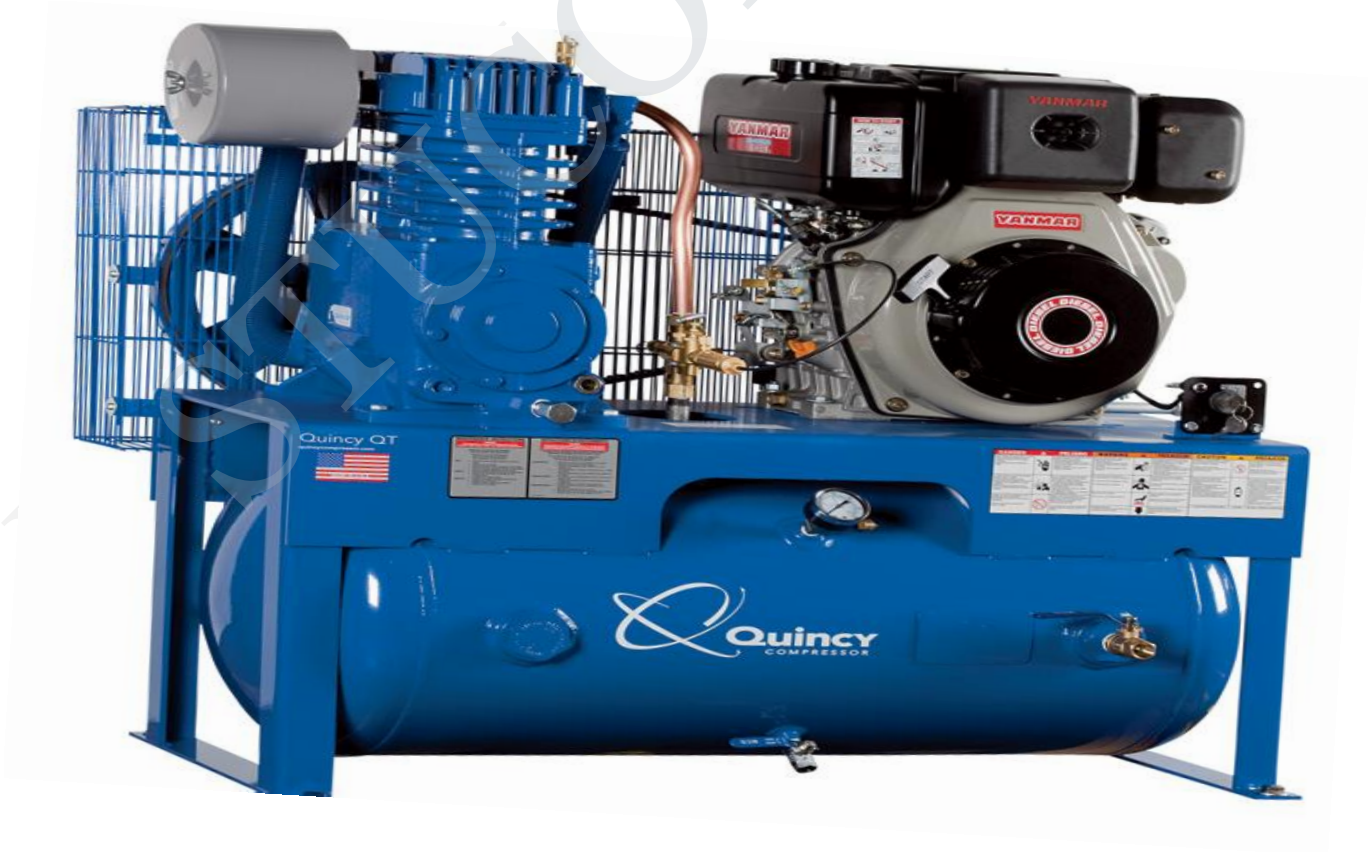
Check the condition of water at 1. (find $v1$)
From volume flow rate find mass flow rate of water
and find h_1
Check the condition of water at 2. (find h_2)
Find H.T rate $Q = m(h_2 - h_1)$ in kJ/min

THERMAL ENGINEERING 1

UNIT II – RECIPROCATING AIR COMPRESSORS







AIR COMPRESSOR

Air compressor is a device which is used to increase the pressure of gas from low pressure to high pressure.

CLASSIFICATION COMPRESSORS

According to number of stages:

- Single stage compressor
- Multi stage compressor

According to number of cylinders:

- Single cylinder compressor
- Multi cylinder compressor

According to method of cooling:

- Air cooled compressor
- Water cooled compressor

According to working:

- Reciprocating compressor
- Rotary compressor

LIST THE TYPES OF ROTARY AIR COMPRESSOR.

Positive displacement type

- (a) Screw compressor.
- (b) Vane compressor.

Steady flow type

- (a)Centrifugal compressor.
- (b)Axial flow compressor.

POSITIVE DISPLACEMENT COMPRESSORS

Positive displacement compressor is one in which air is compressed adiabatically. The air is entrapped in between two sets of engaging surfaces. The pressure rise is either by back flow of air (as in roots blower) or both by variation in the flow and back flow (as in vane blower).

NON-POSITIVE DISPLACEMENT COMPRESSORS

In non-positive displacement compressor, air is not trapped in specific boundaries but it flows continuously and steadily through the machine (as in centrifugal compressor and axial flow compressor).

ISOTHERMAL EFFICIENCY

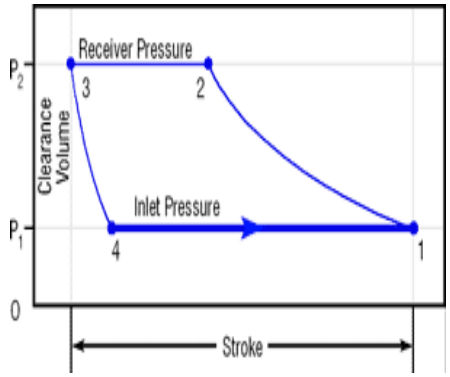
Isothermal efficiency of the compressor is defined as the ratio of isothermal work input to actual work input during compression.

ISENTROPIC EFFICIENCY

Isentropic efficiency of the compressor is defined as the ratio of isentropic work input to the actual work input.

THEORETICAL & ACTUAL INDICATOR DIAGRAM FOR RECIPROCATING AIR COMPRESSOR

a.Theoretical indicator diagram



b. Actual indicator diagram

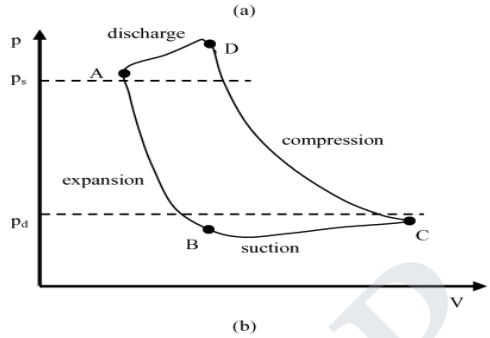


Figure 1. Schematic of a reciprocating compressor and its indicator diagram.

FAD

Free air delivered (FAD) means the actual volume of air delivered by the compressor under normal temperature and pressure.

INFLUENCE OF PRESSURE RATIO ON THE VOLUMETRIC EFFICIENCY

The volumetric efficiency of the compressor also depends on the compressor ratio. If the difference between the suction and the discharge pressure is higher, the compressed air will remain trapped inside the clearance volume for longer time and prevent the opening of the suction valve. Thus as the compression ratio of the compressor and the discharge pressure is increases, its volumetric efficiency and the capacity reduces.

(i) To reduce the temperature of compression from each stage.

(ii) To increase compressor efficiency.

(iii) To condensate from air.

SLIP FACTOR

Slip factor is the ratio of whirl velocity of static pressure to tip velocity

PRESSURE COEFFICIENT

Pressure coefficient is the ratio of isentropic work of the compressor to the Euler work

VARIOUS FACTORS AFFECTING THE DELIVERY PRESSURE OF THE RECIPROCATING COMPRESSOR?

The size of the cylinder will be too large for very high pressure.

Due to compression, there will be a rise in the temperature of the air. So the delivery pressure is limited, so that rise in temperature of air is not going beyond limit and size of cylinder is not too large.

VOLUMETRIC EFFICIENCY OF THE COMPRESSOR AND HOW ITS REDUCES?

Volumetric efficiency of an air compressor is the ratio of free air delivered to the displacement of the compressor. Volumetric efficiency = Free air delivered / Displacement volume

As the piston moves downwards the high-pressure gas in the clearance volume reduces due to expansion, and only when the pressure reaches a certain level does the suction-valve open. Therefore, this gas used in the piston's suction stroke goes unused and reduces the volumetric efficiency of the compressor.

INTERCOOLER, PERFECT INTERCOOLING AND ITS PURPOSE

An intercooler is any mechanical device used to cool a fluid, including liquids or gases, between stages of a multi-stage heating process, typically a heat exchanger that removes waste heat in a gas compressor.

When temperature of the air leaving the intercooler is equal to the original temperature of atmospheric air temperature, then the intercooling is known as perfect intercooling.

MERITS OF MULTISTAGE COMPRESSION?

(i) For each stage, pressures during suction and delivery remain constant.

(ii) The index (n) in polytropic law is same in each stage of compression.

(iii) Intercooling in each stage is done at constant pressure.

(iv) Low pressure and high pressure cylinder handle same mass of air.

(v) There is no inter-stage pressure drop

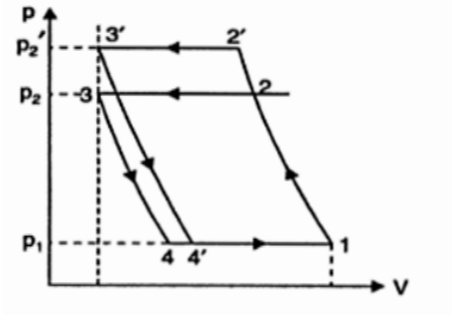
WHY CLEARANCE VOLUME IS NECESSARY AND EXPLAIN ITS IMPORTANCE?

In actual compressor, the clearance volume is provided to give cushioning effect otherwise the piston will strike the other end of the cylinder. It is generally expressed as percentage of piston displacement.

- To give cushioning effect to the piston.
- To provide space for valve movement.
- The maximum pressure may also be controlled by clearance volume.
- If clearance volume is more, it reduces the volumetric efficiency.

EFFECT OF PRESSURE RATIO ON VOLUMETRIC EFFICIENCY WITH NEAT P-V DIAGRAM.

Volumetric efficiency increases with decrease in pressure ratio in compressor.



INTERCOOLER

An intercooler is any mechanical device used to cool a fluid, including liquids or gases, between stages of a multi-stage heating process, typically a heat exchanger that removes waste heat in a gas compressor.

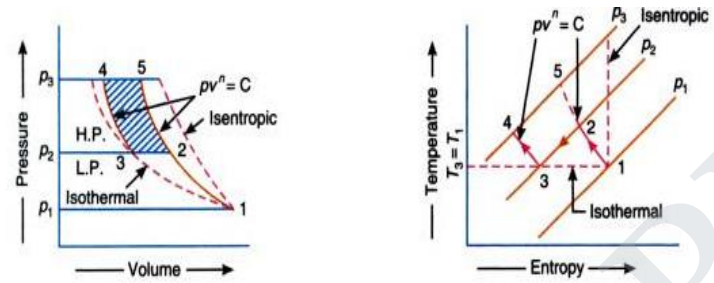
PURPOSE OF INTERCOOLING

(i) To reduce the temperature of compression from each stage.

(ii) To increase compressor efficiency.

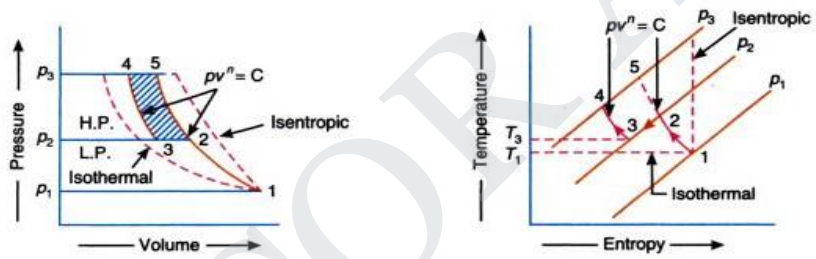
(iii) To condensate from air.

P-V & T-S DIAGRAM FOR PERFECT/COMPLETE INTERCOOLING IN MULTISTAGE COMPRESSOR



(a) p-v diagram. (b) T-s diagram.

P-V & T-S DIAGRAM FOR IMPERFECT/INCOMPLETE INTERCOOLING IN MULTISTAGE COMPRESSOR



(a) p-v diagram. (b) T-s diagram.

CONDITION FOR MINIMUM WORK DONE ON MULTISTAGE COMPRESSOR FOR 'Z'STAGES.

(i) The pressure ratio of each stage should be the same.

(ii) The pressure ratio of any stage is the square root of overall pressure ratio for a two stage compressor.

(iii) Air after compression in each stage should be cooled to initial temperature of air intake.

(iv) The work input to each stage is same.

DIFFERENTIATE CENTRIFUGAL AND AXIAL FLOW COMPRESSORS

Centrifugal Compressor	Axial flow Compressor
Flow of air is perpendicular to the axis of compressor.	The flow of air is parallel to the axis of compressor.
It has low manufacturing and running cost.	It has high manufacturing and running cost.
It requires low starting torque.	It requires high starting torque.
It is not suitable for multi staging.	It is suitable for multi staging.

FACTORS AFFECTING THE DELIVERY PRESSURE OF THE RECIPROCATING CPMPRESSOR

The size of the cylinder will be too large for very high pressure.

Due to compression, there will be a rise in the temperature of the air. So the delivery pressure is limited, so that rise in temperature of air is not going beyond limit and size of cylinder is not too large.

SINGLE STAGE AIR COMPRESSOR :

INDICATED POWER OR POLYTROPHIC WORK:

$$\dot{W} = \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ kW}$$

$$\dot{W} = \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ kW}$$

Where,

FOR COMPRESSOR WITHOUT CLEARANCE

$$\dot{V} = \dot{V}_1$$

FOR COMPRESSOR WITH CLEARANCE

$$\dot{V} = \dot{V}_1 - \dot{V}_4$$

ISOTHERMAL WORK

$$\dot{W} = P_1\dot{V}\ln\left(\frac{P_2}{P_1}\right) \text{ kW}$$

ISOTHERMAL EFFICIENCY:

$$\eta_{iso} = \frac{\text{Isothermal Power}}{\text{Polytropic Work}}$$

ISENTROPIC WORK:

$$\dot{W} = \frac{\gamma-1}{\gamma}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ kW}$$

ISENTROPIC EFFICIENCY:

$$\eta_{isen} = \frac{\text{Isentropic Work}}{\text{Polytropic work}}$$

POWER INPUT TO THE COMPRESSOR:

$$W_C = \frac{\text{Indicated Power}}{\text{Mechanical Efficiency}}$$

POWER INPUT TO THE MOTOR:

$$W_{motor} = \frac{\text{Power input}}{\text{Transmission or motor Efficiency}}$$

Where ,

n = Index of Compression,

$P_1\dot{V} = \dot{m}RT_1$

P_1 - Inlet Pressure of the air (kN/m^2),

$\dot{V}$ – Volume flow rate of air ($\frac{m^3}{Sec}$)

$\dot{m}$ – Mass flow rate of air ($\frac{kg}{Sec}$)

R – Gas Constant ($R = 287\frac{J}{kgK}$)

T_1- Inlet Temperature of the air (K)

SWEPT OR STROKE VOLUME:

$$V_S = \frac{\pi D^2}{4} \times L,$$

FIND $\dot{V}$, V_S AND N: (WITHOUT CLEARANCE VOLUME)

$$\dot{V} = V_S \times N \times \text{Number of Acting}$$

Where,

N – Speed of Compressor (rpm)

FIND $\dot{V}$, V_S AND N : (WITH CLEARANCE VOLUME)

$$\dot{V} = \frac{\dot{V}_{Act}}{\eta_{vol}}$$

$$\dot{V}_{Act} = V_S \times N \times \text{Number of Acting}$$

CLEARANCE RATIO:

$$C = \frac{V_C}{V_S}$$

VOLUMETRIC EFFICIENCY:

$$\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

$$\eta_{vol} = \frac{\text{Volume of air delivered}}{\text{Volume of air filled in Swept Volume}}$$

$$\eta_{vol} = \frac{\dot{V}}{V_s \times N \times \text{Number of Acting}}$$

VOLUMETRIC EFFICIENCY IF CONSIDER FAD:

$$\eta_{vol} = \frac{\text{FAD}}{V_s \times N \times \text{Number of Acting}}$$

$$\text{FAD} = (\dot{V}_1 - \dot{V}_4)\frac{T_a}{T_1}\frac{P_1}{P_a} \qquad \text{FAD} = \dot{V}\frac{T_a}{T_1}\frac{P_1}{P_a}$$

Where,

P_1 - Pressure of the air at FAD(kN/m^2),

T_1 - Temperature of the air at FAD (K)

P_a - Inlet or Suction Pressure of the air (kN/m^2),

T_a- Inlet or Suction Temperature of the air (K)

MULTI-STAGE AIR COMPRESSOR:

POWER REQUIRED TO DRIVE THE COMPRESSOR:

$$\dot{W} = Z \times \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_{Z+1}}{P_1}\right)^{\frac{n-1}{z\times n}} - 1\right] \textbf{ kW}$$

Where,

Z = number of stages

P_1- Inlet Pressure of the air (kN/m^2)

P_{Z+1}- Delivery Pressure of the air (kN/m^2)

FOR LP COMPRESSOR:

$$\dot{W}_{LP} = \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

FOR LP COMPRESSOR:

$$\dot{W}_{HP} = \frac{n}{n-1}P_2\dot{V}_2\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

INTERMEDIATE PRESSURE:

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \left(\frac{P_{Z+1}}{P_1}\right)^{\frac{1}{Z}}$$

TO FIND PRESSURE OR TEMPERATURE: IF NOT GIVEN

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

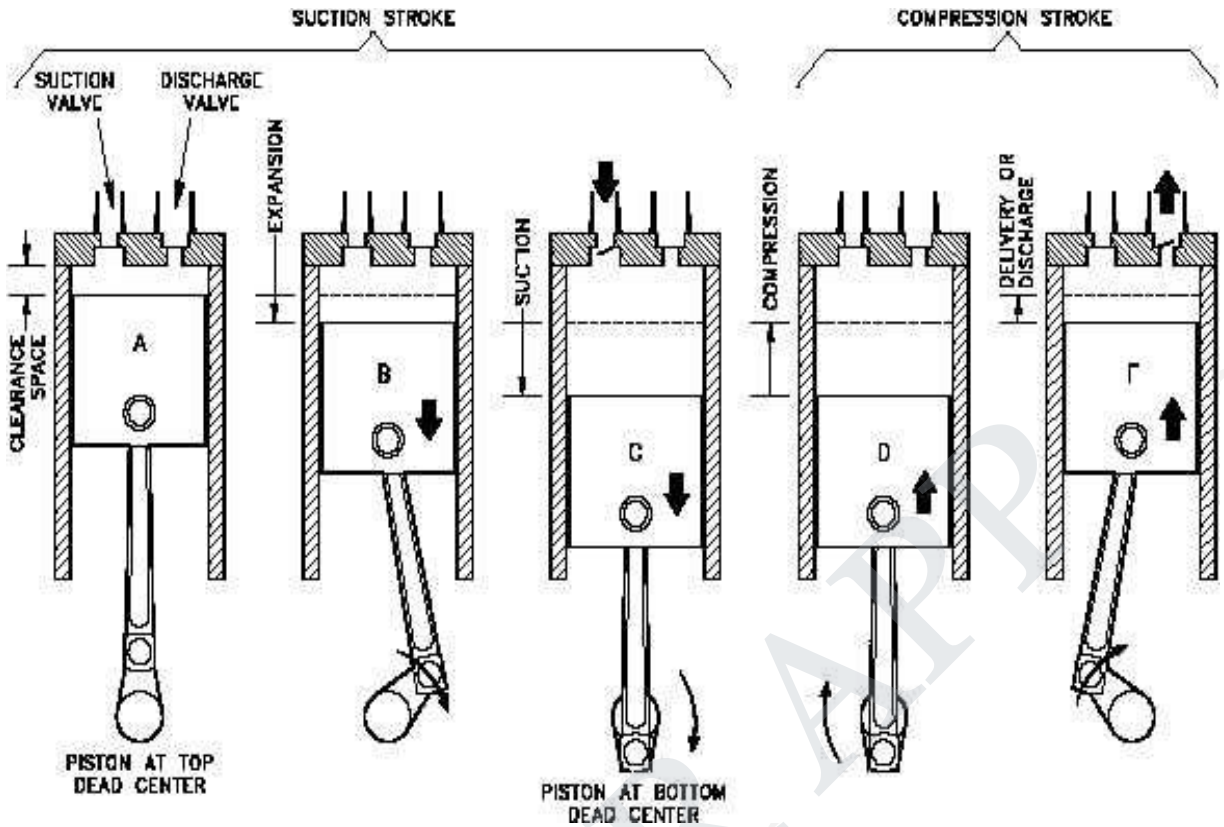
$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1} = (r_c)^{n-1}$$

COMPRESSION RATIO:

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

PART – B

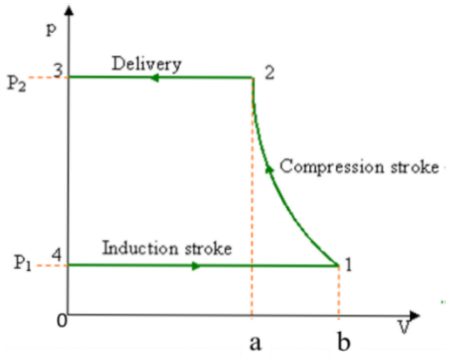
1. With a neat sketch, describe the construction of single stage reciprocating air compressor .



- In single stage reciprocating air compressor the entire compression is carried out in a single cylinder.
- If the compression is affected in one end of the piston & cylinder then it is known as single acting & if the compression is affected in both ends of piston & cylinder then it is known as double acting reciprocating air compressor.
- The opening & closing of simple check valve (plate or spring valve) is depend upon difference in pressure.
- if mechanically operated valves are used for suction & discharge then their functioning is controlled by cams.
- The weight of air in the cylinder will be zero when the piston is at top dead center, if we neglect clearance volume.
- When piston starts moving downwards, the pressure inside the cylinder falls below atmospheric pressure& suction valve/inlet valve opens. The air is drawn into the cylinder through suction filter element. This operation is known as suction stroke.
- When piston moves upwards, compresses the air in cylinder & inlet valve closes when pressure reaches to atmospheric pressure.
- Further compression follows as the piston moves towards the top of its stroke until, when the pressure in the cylinder exceeds that in the receiver. This is compression stroke of compressor. At the end of this stroke discharge/delivery valve opens & air is delivered to receiver.
- When it is double acting reciprocating air compressor, suction stroke is in process at one end of piston while at same time discharge stroke is in process at other end of piston. In simple word we can say that suction & compression took place on both end of piston & cylinder in double acting reciprocating air compressor.

2. Derive the work done for single cylinder reciprocating air compressor with and without clearance.

COMPRESSOR WITHOUT CLEARANCE:



Process 1-2 : Compression

Process 2-3 : Deliver (During compression delivery valve opens)

Process 4-1 : Suction

$(P_1 = P_4, \quad P_2 = P_3)$

WORK DONE FOR SINGLE STAGE AIR COMPRESSOR WITHOUT CLEARANCE VOLUME:

$W = \text{Process } (1234 + 230a - 140b)$

$$W = \frac{P_2V_2 - P_1V_1}{n-1} + (P_2V_2 - P_3V_3) - (P_1V_1 - P_4V_4)$$

Where, $V_3 = V_4 = 0$

$$W = \frac{P_2V_2 - P_1V_1}{n-1} + (P_2V_2 - P_1V_1)$$

$$W = (P_2V_2 - P_1V_1)\left[\frac{1}{n-1} + 1\right]$$

$$\mathbf{W} = \frac{n}{n-1}(P_2V_2 - P_1V_1)$$

$$\mathbf{W} = \frac{n}{n-1}\dot{m}\,R(T_2 - T_1)$$

$$\mathbf{W} = \frac{n}{n-1}\dot{m}\,RT_1\left(\frac{T_2}{T_1} - 1\right)$$

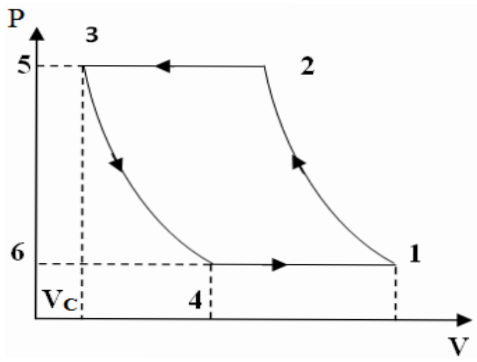
Where,

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

$$\mathbf{W} = \frac{n}{n-1}\dot{m}\,RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\dot{\mathbf{W}} = \frac{\mathbf{n}}{\mathbf{n-1}}\mathbf{P_1}\dot{V}\left[\left(\frac{\mathbf{P_2}}{\mathbf{P_1}}\right)^{\frac{\mathbf{n-1}}{\mathbf{n}}} - \mathbf{1}\right] \quad \mathbf{kW}$$

COMPRESSOR WITH CLEARANCE VOLUME:



Process 1-2 : Compression

Process 2-3 : Deliver (During compression delivery valve opens)

Process 3-4 : Expansion of high pressure air in the clearance volume

Process 4-1 : Suction

WORK DONE FOR SINGLE STAGE AIR COMPRESSOR WITH CLEARANCE VOLUME:

$W = \text{Process } (1256 - 3564)$

$$W = \frac{n}{n-1}\mathbf{P_1}\dot{V}_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] - \frac{n}{n-1}\mathbf{P_4}\dot{V}_4\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

Where,

$\dot{m}_1 = \dot{m}_2, \dot{m}_3 = \dot{m}_4$, $P_1 = P_4$, $\frac{P_2}{P_1} = \frac{P_3}{P_2}$

$$W = \frac{n}{n-1}\mathbf{P_1}(\dot{V}_1 - \dot{V}_4)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\mathbf{W} = \frac{n}{n-1}(\dot{m}_1 - \dot{m}_4)\,RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\mathbf{W} = \frac{n}{n-1}P_1(\dot{V}_1 - \dot{V}_4)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \mathbf{kW}$$

Note:

P_1 - Inlet Pressure of the air $(\mathbf{kN/m^2})$

$\dot{V} = (\dot{V}_1 - \dot{V}_4)$ – Volume flow rate of air $(\frac{\mathbf{m^3}}{\mathbf{Sec}})$

$\dot{m} = (\dot{m}_1 - \dot{m}_4)$ – Mass flow rate of air $\left(\frac{\mathbf{kg}}{\mathbf{Sec}}\right)$

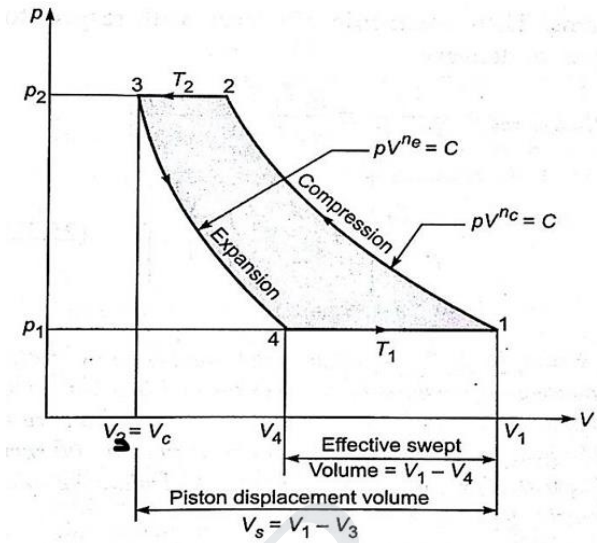
R – Gas Constant $(\mathbf{R = 0.287}\,\frac{\mathbf{kJ}}{\mathbf{kgK}})$

T_1 - Inlet Temperature of the air **(K)**

3. Derive the expression for volumetric efficiency of air compressor.

The actual volume sucked in the cylinder during the suction stroke is always less than the swept volume. It is due to

- ✓ The resistance offered by inlet valve to incoming air
- ✓ Temperature of incoming air
- ✓ Back pressure of residual gas left in the clearance volume



In terms of volume ratio:

The volumetric efficiency of air compressor is defined as the volume of gas delivered as measured at atmospheric pressure and temperature divided by swept volume of the cylinder

In terms of mass ratio:

The volumetric efficiency of air compressor is defined as the mass of the gas delivered divided by mass of gas which would fill the swept volume at atmospheric pressure and temperature.

$$\eta_{vol} = \frac{\text{Mass of the gas delivered}}{\text{mass of gas which would fill the swept volume at atm cond.}}$$

In an indicator diagram for reciprocating air compressor showing effective swept volume and piston displacement volume.

The volumetric efficiency can be expressed in terms of effective volume and piston displacement volume.

$$\eta_{vol} = \frac{V_1 - V_4}{V_1 - V_C} \Rightarrow \eta_{vol} = \frac{V_s + V_c - V_4}{V_s + V_c - V_C} \Rightarrow \eta_{vol} = \frac{V_s + V_c - V_4}{V_s} \Rightarrow \eta_{vol} = 1 + \frac{V_c}{V_s} - \frac{V_4}{V_s}$$

Multiply and divide by V_C at last right term, Introducing $C = \frac{V_c}{V_s}$ as clearance ratio and using $V_c = V_3$

$$\eta_{vol} = 1 + \frac{V_c}{V_s} - \frac{V_4}{V_s} \times \frac{V_C}{V_C} \Rightarrow \eta_{vol} = 1 + \frac{V_c}{V_s} - \frac{V_C}{V_s} \times \frac{V_4}{V_c} \Rightarrow \eta_{vol} = 1 + C - C\left(\frac{V_4}{V_3}\right)$$

For expansion of gas in clearance volume

$$\frac{P_3}{P_4} = \left(\frac{V_4}{V_3}\right)^n \Rightarrow \frac{V_4}{V_3} = \left(\frac{P_3}{P_4}\right)^{\frac{1}{n}}$$, where $P_3 = P_2$ and $P_4 = P_1$

Then, $$\eta_{vol} = 1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \Rightarrow \boldsymbol{\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]}$$

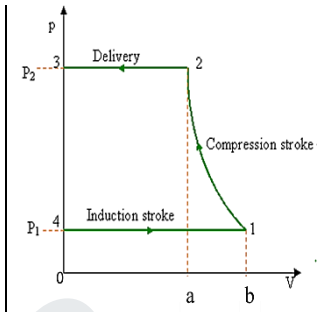
The volumetric efficiency decreases with pressure ratio $\left(\frac{P_2}{P_1}\right)$ increases in the compressor,

PROBLEMS ON SINGLE CYLINDER RECIPROCATING AIR COMPRESSOR WITHOUT CLEARANCE VOLUME

1. **A single stage single acting reciprocating air compressor takes 1 m^3 per minute of air at 1.013 bar and 15 °C and delivers at 7 bar assume that pv^n=C where n=1.35. Calcuate Work done or Indicated Power. In which clearance is negligible.**

GIVEN:

$\dot{V}_1 = 1\ \frac{m^3}{min}$ $P_1 = 1.013$ bar, $T_1 = 15°C$, $P_2 = 7$ bar, n=1.35



SOLUTION:

Work Done

$$\dot{W} = \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.35}{1.35-1} \times 1.013 \times 10^2 \times \frac{1}{60}\left[\left(\frac{7}{1.013}\right)^{\frac{1.35-1}{1.35}} - 1\right]$$

$$\Rightarrow \dot{W} = 4.2145\ kW$$

2. **A single acting reciprocating air compressor has cylinder diameter and stroke of 200 mm and 300 mm respectively. The compressor sucks air at 1 bar and 27 °C and delivers at 8 bar. Speed of the compressor is 100 rpm. Find (i) Indicated power, (ii) Mass of the air delivered per min and (iii) Temperature of the air delivered. The compressor follow the law $PV^{1.25}$=C.**

GIVEN:

D=200mm, L=300mm, $P_1 = 1$ bar, $T_1 = 27°C$, $P_2 = 8$ bar, N=100rpm, n=1.25

SOLUTION:

Work Done

$$\dot{W} = \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.25}{1.25-1} \times 1 \times 10^2 \times 0.01 \times \left[\left(\frac{8}{1.}\right)^{\frac{1.25-1}{1.25}} - 1\right] \Rightarrow \dot{W} = 2.58\ kW$$

Stroke Volume:

$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow V_S = \frac{\pi \times 0.2^2}{4} \times 0.3 \Rightarrow V_S = 9.42 \times 10^{-3} m^3$$

Volume of air sucked:

$$\dot{V} = V_S \times N \Rightarrow \dot{V}_a = 9.42 \times 10^{-3} \times 100 \Rightarrow \dot{V}_a = 0.9\ \frac{m^3}{min} \Rightarrow \dot{V}_a = 0.01\ \frac{m^3}{Sec}$$

Mass of the air delivered per min:

$$P_1 \dot{V} = \dot{m}_1 R T_1 \Rightarrow \dot{m}_1 = \frac{P_1 \dot{V}}{R T_1} \Rightarrow \dot{m}_a = \frac{1 \times 10^2 \times 0.9}{0.287 \times 300} \Rightarrow \dot{m}_a = 1.05\ \frac{kg}{min}$$

Temperature of the air delivered:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \Rightarrow T_2 = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \times T_1 = \left(\frac{8}{1}\right)^{\frac{1.25-1}{1.25}} \times 300 \Rightarrow T_2 = 454.7K$$

3. **A single stage single acting reciprocating air compressor has air entering at 1 bar, 20°C and compression occurs following polytrophic process with index 1.2 up to the delivery pressure of 12 bar. The compressor runs at the speed of 240 rpm and has L/D ratio of 1.8. The compressor has mechanical efficiency of 0.88. Determine the isothermal efficiency and cylinder dimensions. Also find out the rating of drive required to run the compressor which admits 1 m^3 of air per minute.**

GIVEN:

$P_1 = 1$ bar, $T_1 = 20°C$, n=1.2, $P_2 = 12$ bar, N=240rpm, $\frac{L}{D} = 1.8$, $\eta_{mech} = 0.88$, $\dot{V}_1 = 1\ \frac{m^3}{min}$

SOLUTION:

Isothermal Work:

$\dot{W}_{iso} = P_1\dot{V}\ln\left(\frac{P_2}{P_1}\right) \Rightarrow \dot{W}_{iso} = 1 \times 10^2 \times \frac{1}{60}\ln\left(\frac{12}{1}\right) \Rightarrow \mathbf{\dot{W}_{iso} = 4.14kW}$

Work Done

$\dot{W} = \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.2}{1.2-1} \times 1 \times 10^2 \times \frac{1}{60} \times \left[\left(\frac{12}{1}\right)^{\frac{1.2-1}{1.2}} - 1\right] \Rightarrow \mathbf{\dot{W} = 5.13\ kW}$

Isothermal Efficiency:

$\eta_{iso} = \frac{\dot{W}_{iso}}{\dot{W}} \Rightarrow \eta_{iso} = \frac{4.14}{5.13} \Rightarrow \mathbf{\eta_{iso} = 80.7\%}$

Capacity to Drive required to run compressor:

$\dot{W}_{Act} = \frac{\dot{W}}{\eta_{mech}} \Rightarrow \dot{W}_{Act} = \frac{5.13}{0.88} \Rightarrow \mathbf{\dot{W}_{Act} = 5.83\ kW}$

Stroke Volume:

$\dot{V} = V_S \times N \Rightarrow V_S = \frac{\dot{V}_a}{N} \Rightarrow V_S = \frac{1}{240} \Rightarrow \mathbf{V_S = 0.0042\ m^3}$

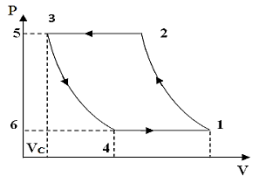
$V_S = \frac{\pi D^2}{4} \times L \Rightarrow 0.042 = \frac{\pi \times D^2}{4} \times 1.8D \Rightarrow \mathbf{D = 0.143\ m}$

$\frac{L}{D} = 1.8 \Rightarrow L = 1.8 \times 0.143 \Rightarrow \mathbf{L = 0.285\ m}$

PROBLEMS ON SINGLE CYLINDER RECIPROCATING AIR COMPRESSOR WITH CLEARANCE VOLUME

1. **A single stage single acting air compressor delivers 0.6 kg of air per minute at 6 bar. The temperature and pressure at the end of suction stroke are 30°C and 1 bar. The bore and stroke of the compressors are 100mm and 150mm respectively. The clearance is 3% of the swept volume. Assuming the index and of compression and expansion to be 1.3, find: i) power required if the mechanical efficiency is 85%, ii) volumetric efficiency of the compressor, and iii) the speed of compressor.**

GIVEN:



$\dot{m}_1 = 0.6\ \frac{kg}{min}$, $P_2 = 6$ bar, $P_1 = 1$ bar, $T_1 = 30°C$, D=100mm, L=150mm, $V_c = 0.03V_S$,

$C = \frac{V_c}{V_S} = 0.03$, n=1.3, $\eta_{mech} = 0.85$,

SOLUTION:

Volume of the air delivered per min:

$P_1\dot{V} = \dot{m}_1RT_1 \Rightarrow \dot{V} = \frac{\dot{m}_1RT_1}{P_1} \Rightarrow \dot{m}_a = \frac{06 \times 0.287 \times 303}{1 \times 10^2} \Rightarrow \mathbf{\dot{V} = 0.522\ \frac{m^3}{min}}$

Work Done

$\dot{W} = \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.3}{1.3-1} \times 1 \times 10^2 \times \frac{0.522}{60} \times \left[\left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right] \Rightarrow \mathbf{\dot{W} = 1.93\ kW}$

Capacity to Drive required to run compressor:

$\dot{W}_{Act} = \frac{\dot{W}}{\eta_{mech}}$ ⇨ $\dot{W}_{Act} = \frac{1.93}{0.85}$ ⇨ $\dot{W}_{Act} = 2.27\ kW$

Volumetric Efficiency of compressor:

$\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$ ⇨ $\eta_{vol} = 1 + 0.03\left[\left(\frac{6}{1}\right)^{\frac{1}{1.3}} - 1\right]$ ⇨ $\eta_{vol} = 91.09\%$

Actual Volume of the air delivered per min:

$\dot{V}_{act} = \frac{\dot{V}}{\eta_{vol}}$ ⇨ $\dot{V}_{act} = \frac{0.522}{0.9109}$ ⇨ $\dot{V}_{act} = 0.573\ \frac{m^3}{min}$

Stroke Volume:

$V_S = \frac{\pi D^2}{4} \times L$ ⇨ $V_S = \frac{\pi \times 0.1^2}{4} \times 0.15$ ⇨ $V_S = 1.18 \times 10^{-3} m^3$

Volume of air sucked:

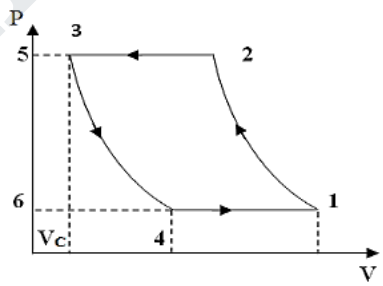
$\dot{V}_{act} = V_S \times N$ ⇨ $N = \frac{0.573}{1.18 \times 10^{-3}}$ ⇨ $N = 485.61\ rpm$

2. **A single stage single acting air compressor delivers 15 m³ of air per minute from 1 bar to 8 bar. The speed of compressor is 300 rpm. Assuming that compression and expansion follow the law PV^1.3=Constant. The clearance is 1/16th of swept volume find the diameter and stroke of the compressor, if L/D=1.5. The temperature and pressure of air at suction is 20°C and 1bar respectively. Also Determine the mean effective pressure and the power required to drive the compressor.**

GIVEN:

$\dot{V} = \dot{V}_1 - \dot{V}_4 = 15\ \frac{m^3}{min}$, $P_1 = 1$ bar, $P_2 = 8$ bar, N=300, $V_c = \frac{1}{16} V_S$,

$C = \frac{V_c}{V_S} = 0.0625$, n=1.3, $\frac{L}{D} = 1.5$, $T_1 = 20°C$,



SOLUTION:

Work Done or Indicated Power

$\dot{W} = \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$ ⇨ $\dot{W} = \frac{1.3}{1.3-1} \times 1 \times 10^2 \times \frac{15}{60} \times \left[\left(\frac{8}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$ ⇨ $\dot{W} = 66.72\ kW$

Volumetric Efficiency of compressor:

$\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$ ⇨ $\eta_{vol} = 1 - 0.0625\left[\left(\frac{8}{1}\right)^{\frac{1}{1.3}} - 1\right]$ ⇨ $\eta_{vol} = 81.55\%$

Actual Volume of the air delivered per min:

$\dot{V}_{act} = \frac{\dot{V}}{\eta_{vol}}$ ⇨ $\dot{V}_{act} = \frac{15}{0.8155}$ ⇨ $\dot{V}_{act} = 18.39\ \frac{m^3}{min}$

Stroke Volume:

$\dot{V}_{act} = V_S \times N$ ⇨ $V_S = \frac{\dot{V}_{act}}{N}$ ⇨ $V_S = \frac{18.39}{300}$ ⇨ $V_S = 0.0613\ m^3$

Cylinder Dimentions:

$V_S = \frac{\pi D^2}{4} \times L$ ⇨ $0.0613 = \frac{\pi \times D^2}{4} \times 1.5D$ ⇨ $D = 0.373m$

$\frac{L}{D} = 1.5$ ⇨ $L = 1.5 \times 0.373$ ⇨ $L = 0.5595\ m$

Mean Effective Pressure

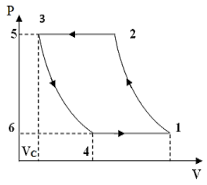
$P_{MEP} = \frac{\dot{W}}{\dot{V}_S}$ ⇨ $P_{MEP} = \frac{\dot{W}}{V_S \times \frac{N}{60}}$ ⇨ $P_{MEP} = \frac{66.72}{0.0613 \times \frac{300}{60}}$ ⇨ $P_{MEP} = 217.68\ \frac{kN}{m^2}$

(OR)

$IP = P_{BMEP} LAnk$ ⇨ $66.72 = P_{MEP} \times 0.5595 \times \frac{\pi \times 0.373^2}{4} \times \frac{300}{60} \times 1$ ⇨ $P_{MEP} = 218.26\ \frac{kN}{m^2}$

3. The free air delivery of a single cylinder, single stage reciprocating air compressor is 2.5 m³/min. The ambient air is at STP condition. The delivery pressure is at 7 bar. The clearance volume is 5 percent of law stroke volume. Both compression and expansion are according to the law $PV^{1.25}$ = constant. Stroke length is 20% more than that of the bore. Compressor runs at 150 rpm. Determine the mass of air per second, indicated power, indicated mean effective pressure, bore and stroke of cylinder.

GIVEN:



$\dot{V} = \dot{V}_1 - \dot{V}_4 = 2.5\ \frac{m^3}{min}$, $P_1 = 1$ bar, $T_1 = 20°C$, $P_2 = 7$ bar, $V_c = 0.05V_S$, $C = \frac{V_c}{V_S} = 0.05$,

n=1.25, N=150rpm, L=1.2D.

SOLUTION:

Work Done or Indicated Power

$$\dot{W} = \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.25}{1.25-1} \times 1 \times 10^2 \times \frac{2.5}{60} \times \left[\left(\frac{7}{1}\right)^{\frac{1.25-1}{1.25}} - 1\right] \Rightarrow \mathbf{\dot{W} = 9.91\ kW}$$

Volumetric Efficiency of compressor:

$$\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_{vol} = 1 - 0.05\left[\left(\frac{7}{1}\right)^{\frac{1}{1.25}} - 1\right] \Rightarrow \mathbf{\eta_{vol} = 81.25\%}$$

Actual Volume of the air delivered per min:

$$\dot{V}_{act} = \frac{\dot{V}}{\eta_{vol}} \Rightarrow \dot{V}_{act} = \frac{2.5}{0.8125} \Rightarrow \mathbf{\dot{V}_{act} = 3.08\ \frac{m^3}{min}}$$

Stroke Volume:

$$\dot{V}_{act} = V_S \times N \Rightarrow V_S = \frac{\dot{V}_{act}}{N} \Rightarrow V_S = \frac{3.08}{150} \Rightarrow \mathbf{V_S = 0.0205\ m^3}$$

Cylinder Dimentions:

$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow 0.0205 = \frac{\pi \times D^2}{4} \times 1.2D \Rightarrow \mathbf{D = 0.279m}$$

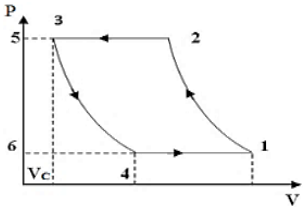
$$\frac{L}{D} = 1.2 \Rightarrow L = 1.2 \times 0.373 \Rightarrow \mathbf{L = 0.335\ m}$$

Mean Effective Pressure

$$P_{MEP} = \frac{\dot{W}}{\dot{V}_S} \Rightarrow P_{MEP} = \frac{\dot{W}}{V_S \times \frac{N}{60}} \Rightarrow P_{MEP} = \frac{9.91}{0.0205 \times \frac{150}{60}} \Rightarrow \mathbf{P_{MEP} = 193.37\ \frac{kN}{m^2}}$$

4. A reciprocating compressor of single stage, double acting type delivers 20 m³/min when measured at free air condition of 1 bar, 27°C. The compressor has compression ratio of 7 and the conditions at the end of suction are 0.97 bar, 35°C. Compressor runs at 240 rpm with clearance volume of 5% of swept volume. The L/D ratio is 1.2. Determine the volumetric efficiency and dimensions of cylinder and isothermal efficiency taking the index of compression and expansion as 1.25. Also show the cycle on P-V diagram.

GIVEN:



$\dot{V} = \dot{V}_1 - \dot{V}_4 = 20\frac{m^3}{min}$, $P_a = 1$ bar, $T_a = 27°C$, $r_c = 7$, N=240rpm,

$V_c = 0.05V_S$, $C = \frac{V_c}{V_S} = 0.05$, L/D=1.2,

$P_1 = 0.95$ bar, $T_1 = 35°C$, n=1.2 , $\eta_{Mech} = 85\%$

SOLUTION:

Deliver pressure:

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n = (r_c)^n \Rightarrow P_2 = (r_c)^n \times P_1 \Rightarrow P_2 = (7)^{1.25} \times 0.95 \Rightarrow \mathbf{P_2 = 10.81\ bar}$$

Volumetric Efficiency of compressor:

$$\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_{vol} = 1 - 0.05\left[\left(\frac{10.81}{1}\right)^{\frac{1}{1.25}} - 1\right] \Rightarrow \boldsymbol{\eta_{vol} = 71.42\%}$$

Actual Volume of the air delivered per min:

$$\dot{V}_{act} = \frac{\dot{V}}{\eta_{vol}} \Rightarrow \dot{V}_{act} = \frac{20}{0.7142} \Rightarrow \boldsymbol{\dot{V}_{act} = 28\frac{m^3}{min}}$$

Stroke Volume:

$$\dot{V}_{act} = V_S \times N \Rightarrow V_S = \frac{\dot{V}_{act}}{N} \Rightarrow V_S = \frac{28}{240} \Rightarrow \mathbf{V_S = 0.117\ m^3}$$

Cylinder Dimensions:

$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow 0.117 = \frac{\pi \times D^2}{4} \times 1.2D \Rightarrow \mathbf{D = 0.498m}$$

$$\frac{L}{D} = 1.2 \Rightarrow L = 1.2 \times 0.477 \Rightarrow \mathbf{L = 0.598\ m}$$

Free Air Deliver:

$$FAD = (\dot{V}_1 - \dot{V}_4) \times \frac{T_a}{T_1} \times \frac{P_1}{P_a} \Rightarrow FAD = 20 \times \frac{300}{308} \times \frac{0.95}{1} \Rightarrow \boldsymbol{FAD = 18.5\frac{m^3}{min}}$$

Actual Volumetric Efficiency Due to FAD:

$$\eta_{vol} = \frac{FAD}{V_S \times N} \Rightarrow \eta_{vol} = \frac{18.5}{0.117 \times 240} \Rightarrow \boldsymbol{\eta_{vol} = 65.88\%}$$

Work Done or Indicated Power

$$\dot{W} = \frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.25}{1.25-1} \times 0.95 \times 10^2 \times \frac{20}{60} \times \left[\left(\frac{10.81}{1}\right)^{\frac{1.25-1}{1.25}} - 1\right] \Rightarrow \boldsymbol{\dot{W} = 95.54\ kW}$$

Isothermal Work:

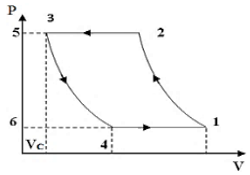
$$\dot{W}_{iso} = P_1\dot{V}\ln\left(\frac{P_2}{P_1}\right) \Rightarrow \dot{W}_{iso} = 0.95 \times 10^2 \times \frac{20}{60}\ln\left(\frac{10.81}{1}\right) \Rightarrow \boldsymbol{\dot{W}_{iso} = 75.38kW}$$

Isothermal Efficiency:

$$\eta_{iso} = \frac{\dot{W}_{iso}}{\dot{W}} \Rightarrow \eta_{iso} = \frac{75.38}{95.54} \Rightarrow \boldsymbol{\eta_{iso} = 78.89\%}$$

5. **A single acting air compressor takes in air at 1 bar 27°C and delivers at 1.4 MPa running at 300 rpm and has cylinder dia of 160 mm and stroke 200 mm. clearance volume is 4% of stroke volume. If the pressure and temperature at the end of the suction stroke is 100 KPa, 47°C and compression and expansion takes places by polytropic process (n=1.2), find 1. Mass of air delivered per minute 2. Volumetric efficiency 3. Driving power required if mechanical efficiency is 85%.**

GIVEN:



$P_1 = 1\text{ bar}, T_a = 27°C,\ P_2 = 1.4\text{MPa} = 14\text{bar}, N=300\text{rpm},$

$V_c = 0.04V_S,\ \ C = \frac{V_c}{V_S} = 0.04,\ L=200\text{mm},\ D=160\text{mm},$

$P_1 = 1\text{ bar}, T_1 = 47°C,\ n=1.2,\ \eta_{Mech} = 85\%$

SOLUTION:

Stroke Volume:

$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow V_S = \frac{\pi \times 0.16^2}{4} \times 0.2 \Rightarrow \mathbf{V_S = 0.004\ m^3}$$

$$\dot{V}_{act} = V_S \times N \Rightarrow \dot{V}_{act} = 0.004 \times 300 \Rightarrow \mathbf{\dot{V}_{act} = 1.21\ \frac{m^3}{min}}$$

Volumetric Efficiency of compressor:

$$\eta_{vol} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_{vol} = 1 - 0.04\left[\left(\frac{14}{1}\right)^{\frac{1}{1.2}} - 1\right] \Rightarrow \mathbf{\eta_{vol} = 67.93\%}$$

Volume of the air delivered per min:

$$\dot{V}_{act} = \frac{\dot{V}}{\eta_{vol}} \Rightarrow \dot{V} = \dot{V}_{act} \times \eta_{vol} \Rightarrow \dot{V} = \dot{V}_{act} \times \eta_{vol} \Rightarrow \dot{V} = \mathbf{0.82\ \frac{m^3}{min}} = (\dot{V}_1 - \dot{V}_4)$$

Mass of the air delivered per min:

$$P_1 \times \dot{V} = \dot{m}_1 R T_1 \Rightarrow \dot{m}_1 = \frac{P_1 \times \dot{V}}{RT_1} \Rightarrow \dot{m}_a = \frac{1\times10^2\times0.769}{0.287\times300} \Rightarrow \mathbf{\dot{m}_a = 0.893\ \frac{kg}{min}}$$

Free Air Deliver:

$$FAD = (\dot{V}_1 - \dot{V}_4) \times \frac{T_a}{T_1} \times \frac{P_1}{P_a} \Rightarrow FAD = 0.82 \times \frac{300}{320} \times \frac{1}{1} \Rightarrow \mathbf{FAD = 0.769\ \frac{m^3}{min}}$$

Actual Volumetric Efficiency of compressor due to FAD:

$$\eta_{vol} = \frac{FAD}{V_S \times N} \Rightarrow \eta_{vol} = \frac{0.769}{0.004\times300} \Rightarrow \mathbf{\eta_{vol} = 64.08\%}$$

Work Done or Indicated Power

$$\dot{W} = \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W} = \frac{1.2}{1.2-1} \times 1 \times 10^2 \times \frac{0.82}{60} \times \left[\left(\frac{14}{1}\right)^{\frac{1.2-1}{1.2}} - 1\right] \Rightarrow \mathbf{\dot{W} = 4.53\ kW}$$

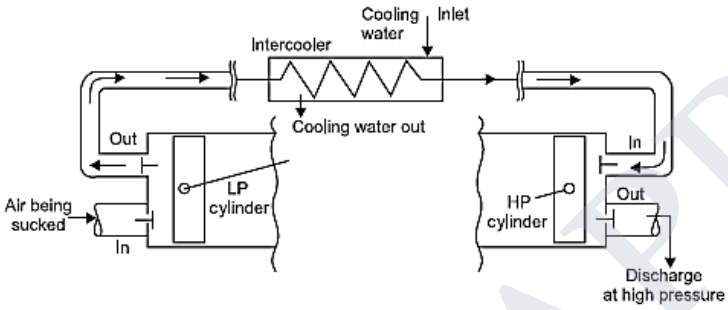
Capacity to Drive required to run compressor:

$$\dot{W}_{Act} = \frac{\dot{W}}{\eta_{mech}} \Rightarrow \dot{W}_{Act} = \frac{4.53}{0.6408} \Rightarrow \mathbf{\dot{W}_{Act} = 7.07\ kW}$$

MULTI-STAGE AIR COMPRESSOR:

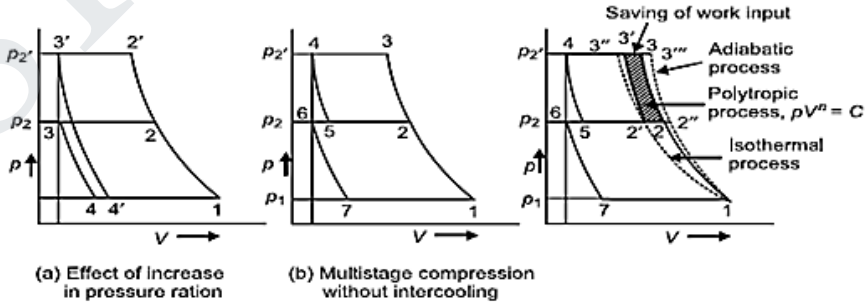
1. Explain with neat sketch the working principle of multistage reciprocating compressor and discuss the perfect cooling with inter cooler.

- The compression of air in two or more cylinders in series is called multistage compression.
- The compressor requires minimum work input with isothermal compression but the delivery temperature increases with pressure ratio and the volumetric efficiency decreases as pressure ratio increases.
- The work input can be reduced to minimum level by compressing the air in more than one cylinders with intercooler between stages for same compression ratio.
- Air cooling between the stages provides the means to achieving an appreciable reduction in the compressor work and maintain the air temperature within safe operating limits.



Advantages of multistage compressor

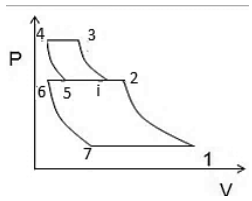
1. The gas can be compressed to a sufficiently high pressure
2. Cooling of air is more sufficient with intercoolers and cylinder walls surface
3. By cooling the air between the stages of compression can be brought to isothermal and power input to the compressor can be reduced considerably
4. By multi-staging, the pressure ratio of each stage is lowered. The air leakage past the piston in the cylinder is also reduced
5. The low pressure ratio in the cylinder improve volumetric efficiency
6. Due to low pressure ratio in stages, the compressor speed could be higher for the same isothermal efficiency,
7. low working temperature in each stage help to sustain better lubrication
8. The low pressure cylinder is made lighter and high pressure cylinder made smaller for reduced pressure ratio in each stage



Apart from the cooling during compression the temperature of air at inlet to compressor can be reduced so as to reduce compression work. In multistage compression the partly compressed air leaving first stage is cooled up to ambient air temperature in intercooler and then sent to subsequent cylinder (stage) for compression Intercoolers when put between the stages reduce the compression work and compression is called intercooled compression. Intercooling is called perfect when temperature at inlet to subsequent stages of compression is reduced to ambient temperature.

2. In a two stage compressor in which inter-cooling is perfect, prove that the work done in the compressor is minimum when the pressure in the inter-cooler is geometric mean between the initial and final pressure. Draw the P-V & T-S Diagram for two stage compression.

The value chosen for the intermediate pressure p_2 influences the work to be done on the gas and its distribution between the stages. The condition for the work done to be a minimum will be proved for two-stage compression but can be extended to any number of stages.



Total work= LP stage work + HP stage work

$$\text{Total power} = \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

It is assumed that intercooling is perfect and therefore the temperature at the start of each stage is $T_1 = T_i$

$$\text{Total power} = \frac{n}{n-1}\dot{m}RT_1\left\{\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]\right\}$$

If P_1, T_1 and P_2 are fixed, then the optimum value of P_i which makes the power a minimum can be obtained by equating $d(\text{power})/dp_i$ to zero, that is optimum value of p_i when

$$\frac{d}{dp_i}\left\{\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]\right\} = 0$$

$$\frac{d}{dp_i}\left\{\left(\frac{1}{P_1}\right)^{\frac{n-1}{n}} P_2^{\frac{n-1}{n}} + P_3^{\frac{n-1}{n}}\left(\frac{1}{P_2}\right)^{\frac{n-1}{n}} - 2\right\} = 0$$

Therefore

$$P_1^{-\left(\frac{n-1}{n}\right)} \times \frac{n-1}{n} P_2^{\left(\frac{n-1}{n}-1\right)} + P_3^{\frac{n-1}{n}} \times \left(-\frac{n-1}{n}\right) P_2^{-\left(\frac{n-1}{n}-1\right)} = 0$$

$$P_1^{-\left(\frac{n-1}{n}\right)} \times \frac{n-1}{n} P_2^{\left(\frac{n-1-n}{n}\right)} + P_3^{\frac{n-1}{n}} \times \left(-\frac{n-1}{n}\right) P_2^{\left(\frac{-n+1-n}{n}\right)} = 0$$

$$P_1^{-\left(\frac{n-1}{n}\right)} \times \frac{n-1}{n} P_2^{\frac{-1}{n}} = P_3^{\frac{n-1}{n}} \times \frac{n-1}{n} P_2^{\left(\frac{1-2n}{n}\right)}$$

Therefore

$$P_2^{\frac{2(n-1)}{n}} = (P_1P_3)^{\frac{n-1}{n}}$$

$$P_2^{\,2} = P_1P_3$$

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

The pressure ratio is same for each stage

Total minimum power = 2 X (power required for one stage)

$$= 2 \text{ X } \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \quad \text{where, } P_2^{\,2} = P_1P_3 \implies P_2 = \sqrt{P_1P_3} = \sqrt{\frac{P_3}{P_1}}$$

$$= 2 \text{ X } \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

This can be shown to extend to Z stages giving in general

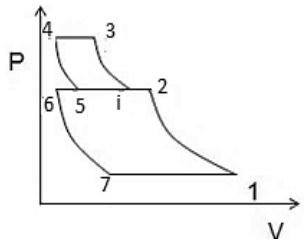
$$\text{Total minimum power} = Z \text{ X } \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{zn}} - 1\right]$$

Also the pressure ratio for each stage $=\left(\frac{P_2}{P_1}\right)^{\frac{1}{Z}}$

Hence the condition for minimum work is that the pressure ratio in each stage is the same and the intercooling is complete.

3. A single acting two stage compressor with complete inter cooling delivers 6 kg/min of air at 16 bar (1.6 MPa). Assuming an intake at 1 bar (100 kPa) and 15°C and compression and expansion with the law $pV^{1.3} = C$. Calculate : (i) Power required to run the compressor (ii) Isothermal efficiency (iii) Free air delivered per sec. (iv) If clearance ratios for LP and HP cylinder are 0.04 and 0.06, Calculate volumetric efficiency and swept volume for each cylinder. Assume R = 0.287 kJ/kg °K, C_v = 0.71 kJ/kg °K.

GIVEN:



$\dot{m} = 0.6\ \frac{kg}{min}$, $P_1 = 1$ bar, $T_1 = 15°C$, $P_2 = 16$bar, $C_{LP} = 0.04$, $C_{HP} = 0.06$,

$pV^{1.3} = C$

R = 0.287 kJ/kg °K

C_v = 0.71 kJ/kg °K

SOLUTION:

Volume of the air delivered per min:

$$P_1\dot{V} = \dot{m}_1 RT_1 \Rightarrow \dot{V} = \frac{\dot{m}_1 RT_1}{P_1} \Rightarrow \dot{V} = \frac{6\times0.287\times288}{1\times10^2} \Rightarrow \dot{V} = 4.96\frac{m^3}{min}$$

Power required to drive the compressor:

$$\dot{W} = Z\times\frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{z\times n}} - 1\right] \Rightarrow \dot{W} = 2\times\frac{1.3}{1.3-1}\times1\times10^2\times\frac{4.96}{60}\left[\left(\frac{16}{1}\right)^{\frac{1.3-1}{2\times1.3}} - 1\right] \Rightarrow \dot{W} = 27kW$$

Isothermal Work:

$$\dot{W}_{iso} = P_1\dot{V}\ln\left(\frac{P_2}{P_1}\right) \Rightarrow \dot{W}_{iso} = 1\times10^2\times\frac{0.496}{60}\times\ln\left(\frac{16}{1}\right) \Rightarrow \dot{W}_{iso} = 22.9kW$$

Isothermal Efficiency:

$$\eta_{iso} = \frac{\dot{W}_{iso}}{\dot{W}} \Rightarrow \eta_{iso} = \frac{2.29}{2.7} \Rightarrow \eta_{iso} = 84.89\%$$

For perfect Intercooler:

$$P_2 = \sqrt{P_1P_3} \Rightarrow P_2 = \sqrt{1\times16} \Rightarrow P_2 = 4\ bar$$

For LP Cylinder:

$$\eta_{VLP} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_{VLP} = 1 - 0.04\left[\left(\frac{4}{1}\right)^{\frac{1}{1.3}} - 1\right] \Rightarrow \eta_{VLP} = 92.38\%$$

Volume of LP cylinder:

$$\eta_{VLP} = \frac{\dot{V}}{V_{SLP}\times N} \Rightarrow V_{SLP} = \frac{FAD}{\eta_{VLP}\times N} \Rightarrow V_{SLP} = \frac{4.96}{0.9238\times300} \Rightarrow V_{SLP} = 0.0179m^3$$

For HP Cylinder:

$$\eta_{VHP} = 1 - C\left[\left(\frac{P_3}{P_i}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_{VHP} = 1 - 0.06\left[\left(\frac{16}{4}\right)^{\frac{1}{1.3}} - 1\right] \Rightarrow \eta_{VHP} = 88.57\%$$

VOLUME FLOWRATE FOR HP CYLINDER:

Volume of the air delivered per min:

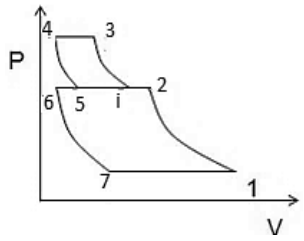
$$P_1\dot{V} = \dot{m}_1 RT_1 \Rightarrow \dot{V} = \frac{\dot{m}_1 RT_1}{P_1} \Rightarrow \dot{V}_{HP} = \frac{6\times0.287\times288}{4\times10^2} \Rightarrow \dot{V}_{HP} = 1.24\frac{m^3}{min}$$

Volume of HP cylinder:

$$\eta_{VHP} = \frac{\dot{V}_{HP}}{V_{SHP}\times N} \Rightarrow V_{SHP} = \frac{\dot{V}_{HP}}{\eta_{VHP}\times N} \Rightarrow V_{SHP} = \frac{1.24}{0.8857\times300} \Rightarrow V_{SHP} = 0.00467m^3$$

4. In a single acting two stage reciprocating air compressor 4.5 kg of air per min are compressed from 1.013 bar and 15°C through a pressure ratio of 9 to 1. Both stages have the same pressure ratio and the law of compression of both compression and expansion follows the law PV $^{1.3}$ = C calculate i) Indicated power ii) the cylinder swept volume required. Assume that the clearance volume of both the cylinder are 5% of their respective swept volume and that the compressors runs at 300 rpm.

GIVEN:



$\dot{m} = 4.5\ \frac{kg}{min}$, $P_1 = 1.013$ bar, $T_1 = 15°C$, $\frac{P_3}{P_1} = 9$ bar, $V_C = 0.05V_S$,

$pV^{1.3} = C$, $R = 0.287$ kJ/kg °K

SOLUTION:

Volume of the air delivered per min:

$$P_1\dot{V} = \dot{m}_1RT_1 \Rightarrow \dot{V} = \frac{\dot{m}_1RT_1}{P_1} \Rightarrow \dot{V} = \frac{4.5\times0.287\times288}{1.013\times10^2} \Rightarrow \dot{V} = 3.672\frac{m^3}{min}$$

Power required to drive the compressor:

$$\dot{W} = Z\times\frac{n}{n-1}P_1\dot{V}\left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{z\times n}} - 1\right] \Rightarrow \dot{W} = 2\times\frac{1.3}{1.3-1}\times1.013\times10^2\times\frac{3.672}{60}\left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{2\times1.3}} - 1\right]$$

$$\Rightarrow \dot{W} = 15.47\ kW$$

For perfect Intercooler:

$$P_2 = \sqrt{P_1P_3} \Rightarrow P_2 = \sqrt{P_1\times P_1\times 9} \Rightarrow \frac{P_2}{P_1} = 3\ bar$$

For LP Cylinder:

$$\eta_v = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_v = 1 - 0.05\left[\left(\frac{3}{1}\right)^{\frac{1}{1.3}} - 1\right] \Rightarrow \eta_v = 93.4\%$$

For LP Cylinder:

Volume of LP cylinder:

$$\eta_{vol} = \frac{\dot{V}}{V_{SLP}\times N} \Rightarrow V_{SLP} = \frac{\dot{V}}{\eta_{vol}\times N} \Rightarrow V_{SLP} = \frac{3.672}{0.934\times300} \Rightarrow V_{SLP} = 0.0131m^3$$

For HP Cylinder:

Volume of the air delivered per min:

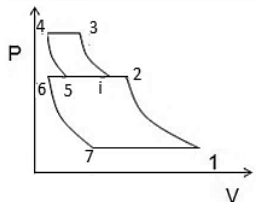
$$P_1\dot{V} = \dot{m}_1RT_1 \Rightarrow \dot{V}_{HP} = \frac{\dot{m}_1RT_1}{P_1} \Rightarrow \dot{V}_{HP} = \frac{4.5\times0.287\times288}{3.039\times10^2} \Rightarrow \dot{V}_{HP} = 1.224\frac{m^3}{min}$$

Volume of HP cylinder:

$$\eta_{vol} = \frac{\dot{V}_{HP}}{V_{SHP}\times N} \Rightarrow V_{SHP} = \frac{\dot{V}_{HP}}{\eta_{vol}\times N} \Rightarrow V_{SHP} = \frac{1.224}{0.934\times300} \Rightarrow V_{SHP} = 0.00437m^3$$

5. A two stage Compressor delivers 2 m^3 of free air per minute. He temperature and pressure of air at suction is 27°C and 1bar. The pressure at the delivery is 50bar. The clearance is 5% of the respective stroke in L.P cylinder as well as H.P cylinder. Assuming perfect inter cooling between the two stages find the minimum power required to run the compressor at 200 rpm. also find the diameter and stroke of the compressor and assuming both are equal for two cylinders. Assume L/D=1.5

GIVEN:



$\dot{V} = 2\ \frac{m^3}{min}$, $P_1 = 1$ bar, $T_1 = 27°C$, $P_2 = 50$bar, $V_C = 0.05V_S$

N=200 rpm, L/D = 1.5

SOLUTION:

Power required to drive the compressor:

$$\dot{W} = Z \times \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{z\times n}} - 1\right] \Rightarrow \dot{W} = 2 \times \frac{1.3}{1.3-1} \times 1 \times 10^2 \times \frac{2}{60}\left[\left(\frac{50}{1}\right)^{\frac{1.3-1}{2\times 1.3}} - 1\right] \Rightarrow \mathbf{\dot{W} = 16.47kW}$$

For perfect Intercooler:

$$P_2 = \sqrt{P_1 P_3} \Rightarrow P_2 = \sqrt{1 \times 50} \Rightarrow \mathbf{P_2 = 7.07\ bar}$$

Volumetric Efficiency:

$$\eta_{VLP} = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_{VLP} = 1 - 0.05\left[\left(\frac{7.07}{1}\right)^{\frac{1}{1.3}} - 1\right] \Rightarrow \mathbf{\eta_{VLP} = 82.5\%}$$

Volume of LP cylinder:

$$\eta_{VLP} = \frac{\dot{V}}{V_{SLP}\times N} \Rightarrow V_{SLP} = \frac{\dot{V}}{\eta_{VLP}\times N} \Rightarrow V_{SLP} = \frac{2}{0.825\times 200} \Rightarrow \mathbf{V_{SLP} = 0.0121m^3}$$

Stroke Volume:

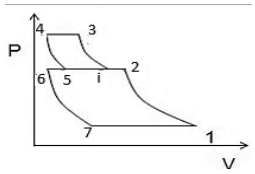
$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow 0.0121 = \frac{\pi \times D^2}{4} \times 1.5D \Rightarrow \mathbf{D = 0.217\ m}$$

$$\frac{L}{D} = 1.5 \Rightarrow L = 1.5 \times 0.217 \Rightarrow \mathbf{L = 0.326}$$

6. A two stage double acting reciprocating air compressor running at 200 rpm has air entering at 1 bar, 25°C. The low pressure stage discharges air at optimum intercooling pressure into intercooler after which it enters at 2.9 bar, 25°C into high pressure stage. Compressed air leaves HP stage at 9 bar. The LP cylinder and HP cylinder have same stroke lengths and equal clearance volumes of 5% of respective cylinder swept volumes. Bore of LP cylinder is 30 cm and stroke is 40 cm. Index of compression for both stages may be taken as 1.2. Determine, (i) the heat rejected in intercooler, (ii) the bore of HP cylinder, (iii) the hp required to drive the HP cylinder.

GIVEN:

N=200rpm, $P_1 = 1$ bar, $T_1 = 25°C$, $P_2 = 2.9$ bar, $T_2 = 25°C$, $P_3 = 9$ bar, $V_C = 0.05V_S$, D=30cm, L=40cm, $pV^{1.2} = C$,



SOLUTION:

Stroke Volume:

$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow V_S = \frac{\pi \times 0.3^2}{4} \times 0.4 \Rightarrow \mathbf{V_S = 0.0283\ m^3}$$

$$\dot{V}_{act} = V_S \times N \times 2 \Rightarrow \dot{V}_{act} = 0.0283 \times (200 \times 2) \Rightarrow \mathbf{\dot{V}_{act} = 11.32\ \frac{m^3}{min}}$$

Volumetric Efficiency:

$$\eta_v = 1 - C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \Rightarrow \eta_v = 1 - 0.05\left[\left(\frac{2.9}{1}\right)^{\frac{1}{1.2}} - 1\right] \Rightarrow \mathbf{\eta_v = 92.85\%}$$

Volume of the air delivered per min:

$$\dot{V}_{act} = \frac{\dot{V}}{\eta_v} \Rightarrow \dot{V} = \dot{V}_{act} \times \eta_v \Rightarrow \dot{V} = 11.32 \times 0.9285 \Rightarrow \mathbf{\dot{V} = \dot{V}_1 - \dot{V}_4 = 10.51\ \frac{m^3}{min}}$$

Mass of the air delivered per min:

$$P_1 \times \dot{V} = \dot{m}_1 R T_1 \Rightarrow \dot{m}_1 = \frac{P_1 \times \dot{V}}{RT_1} \Rightarrow \dot{m}_a = \frac{1\times10^2\times10.51}{0.287\times298} \Rightarrow \dot{m}_a = 12.29\,\frac{kg}{min}$$

Heat rejected to the intercooler:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \Rightarrow T_2 = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \times T_1 \Rightarrow T_2 = \left(\frac{2.9}{1}\right)^{\frac{1.2-1}{1.2}} \times 298 \Rightarrow T_2 = 355.86K$$

$$Q_R = \dot{m}_a C_P (T_2 - T_1) \qquad Q_R = 12.29 \times 1.005(355.86 - 298) \qquad Q_R = 714.65\,\frac{kJ}{min}$$

For HP Cylinder:

Volume of the air delivered per min:

$$P_i\dot{V} = \dot{m}_1 R T_i \Rightarrow \dot{V}_{HP} = \frac{\dot{m}_1 R T_i}{P_i} \Rightarrow \dot{V}_{HP} = \frac{12.29\times0.287\times298}{2.9\times10^2} \Rightarrow \dot{V}_{HP} = 3.62\,\frac{m^3}{min}$$

Volume of HP cylinder:

$$\eta_{vol} = \frac{\dot{V}_{HP}}{V_{SHP}\times N} \Rightarrow V_{SHP} = \frac{\dot{V}_{HP}}{\eta_{vol}\times N} \Rightarrow V_{SHP} = \frac{3.62}{0.9285\times200\times2} \Rightarrow V_{SHP} = 0.00975m^3$$

Bore of HP Cylinder:

$$V_{SHP} = \frac{\pi D^2}{4} \times L \Rightarrow 0.00975 = \frac{\pi\times D^2}{4} \times 0.4 \Rightarrow D = 0.176\,m$$

Power required to drive the HP compressor:

$$\dot{W}_{HP} = \frac{n}{n-1} P_2 \dot{V}\left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right] \Rightarrow \dot{W}_{HP} = \frac{1.2}{1.2-1} \times 2.9\times10^2 \times \frac{3.62}{60}\left[\left(\frac{9}{2.9}\right)^{\frac{1.2-1}{1.2}} - 1\right]$$

$$\Rightarrow \dot{W}_{HP} = 22\,kW$$

7. **A two stage air compressor having 3 cylinders having same bore and stroke. The delivery pressure is 7 bar and free air delivered is 4.3 m³/min. Air is drawn at 1 bar and 15°C and intercool the air at 38°C. The index for compression is 1.3 for all the cylinders. Neglecting clearance find 1. Intermediate pressure, 2. Power required to drive compressor, 3. Isothermal efficiency.**

GIVEN:

$Z=2,\ \dot{V} = 4.3\,\frac{m^3}{min}\ \ P_1 = 1\,bar, T_1 = 15°C,\ \ P_3 = 7\,bar, T_i = 38°C, pV^{1.3} = C,$

SOLUTION:

Intermediate Pressure:

$$\frac{P_2}{P_1} = \left(\frac{P_{Z+1}}{P_1}\right)^{\frac{1}{Z}} \Rightarrow \frac{P_2}{P_1} = \left(\frac{P_3}{P_1}\right)^{\frac{1}{Z}} \Rightarrow P_2 = \left(\frac{7}{1}\right)^{\frac{1}{2}} \times 1 \Rightarrow P_2 = 2.65\,bar$$

Power required to drive the compressor:

$$\dot{W} = Z \times \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_{Z+1}}{P_1}\right)^{\frac{n-1}{Z\times n}} - 1\right] \Rightarrow \dot{W} = 2 \times \frac{1.3}{1.3-1} \times 1\times10^2 \times \frac{4.3}{60}\left[\left(\frac{7}{1}\right)^{\frac{1.3-1}{2\times1.3}} - 1\right]$$

$$\Rightarrow \dot{W} = 15.64\,kW$$

Isothermal Work:

$$\dot{W}_{iso} = P_1\dot{V}\ln\left(\frac{P_3}{P_1}\right) \Rightarrow \dot{W}_{iso} = 1\times10^2 \times \frac{4.3}{60}\ln\left(\frac{7}{1}\right) \Rightarrow \dot{W}_{iso} = 13.95kW$$

Isothermal Efficiency:

$$\eta_{iso} = \frac{\dot{W}_{iso}}{\dot{W}} \Rightarrow \eta_{iso} = \frac{13.95}{15.64} \Rightarrow \eta_{iso} = 89.17\%$$

8. A three stage air compressor with perfect intercooling takes 15 m^3 of air per minute at 95 KPa and 27°C and delivers at 3.5 MPa and compression takes place by polytropic process (n= 1.3), find 1. Power required if mechanical efficiency is 90% 2. Heat rejected in intercoolers per minute 3. Isothermal efficiency.

GIVEN:

$Z=3,\ \dot{V} = 15\ \frac{m^3}{min},\ P_1 = 95KPa, T_1 = 27°C,\ P_{Z+1} = 3.5\ MPa, pV^{1.3} = C,\ \eta_{mach} = 90\%$

SOLUTION:

Power required to drive the compressor:

$$\dot{W} = Z \times \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_{Z+1}}{P_1}\right)^{\frac{n-1}{Z\times n}} - 1\right] \Rightarrow \dot{W} = 3 \times \frac{1.3}{1.3-1} \times 0.95 \times 10^2 \times \frac{15}{60}\left[\left(\frac{35}{0.95}\right)^{\frac{1.3-1}{3\times 1.3}} - 1\right]$$

$$\Rightarrow \dot{W} = 98.72\ kW$$

Capacity to Drive required to run compressor:

$$\dot{W}_{Act} = \frac{\dot{W}}{\eta_{mech}} \Rightarrow \dot{W}_{Act} = \frac{98.72}{0.90} \Rightarrow \dot{W}_{Act} = 109.69\ kW$$

Intermediate Pressure:

$$\frac{P_2}{P_1} = \left(\frac{P_{Z+1}}{P_1}\right)^{\frac{1}{Z}} \Rightarrow \frac{P_2}{P_1} = \left(\frac{P_4}{P_1}\right)^{\frac{1}{Z}} \Rightarrow P_2 = \left(\frac{35}{0.95}\right)^{\frac{1}{3}} \times 0.95 \Rightarrow P_2 = 3.16\ bar$$

$$\frac{P_3}{P_2} = \left(\frac{P_{Z+1}}{P_1}\right)^{\frac{1}{Z}} \Rightarrow \frac{P_3}{P_2} = \left(\frac{P_4}{P_1}\right)^{\frac{1}{Z}} \Rightarrow P_3 = \left(\frac{35}{0.95}\right)^{\frac{1}{3}} \times 3.16 \Rightarrow P_3 = 10.51\ bar$$

Heat rejected to the intercooler:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \Rightarrow T_2 = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \times T_1 \Rightarrow T_2 = \left(\frac{3.16}{0.95}\right)^{\frac{1.3-1}{1.3}} \times 300 \Rightarrow T_2 = 395.89K$$

Mass of the air delivered per min:

$$P_1 \times \dot{V} = \dot{m}_1 R T_1 \Rightarrow \dot{m}_1 = \frac{P_1 \times \dot{V}}{RT_1} \Rightarrow \dot{m}_a = \frac{0.95\times 10^2 \times 15}{0.287\times 300} \Rightarrow \dot{m}_a = 16.55\ \frac{kg}{min}$$

$$Q_{12} = \dot{m}_a C_P (T_2 - T_1) \Rightarrow Q_{12} = 16.55 \times 1.005(395.89 - 300) \Rightarrow Q_{12} = 1594.96\ \frac{kJ}{min}$$

Heat rejected to the intercooler:

$$\frac{T_3}{T_i} = \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} \Rightarrow T_3 = \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} \times T_i \Rightarrow T_3 = \left(\frac{10.51}{3.16}\right)^{\frac{1.3-1}{1.3}} \times 300 \Rightarrow T_3 = 395.88K$$

Mass of the air delivered per min:

$$P_i \times \dot{V} = \dot{m}_i R T_i \Rightarrow \dot{m}_i = \frac{P_i \times \dot{V}}{RT_i} \Rightarrow \dot{m}_a = \frac{3.16\times 10^2 \times 15}{0.287\times 300} \Rightarrow \dot{m}_a = 55.05\ \frac{kg}{min}$$

$$Q_{i3} = \dot{m}_a C_P (T_2 - T_1) \Rightarrow Q_{i3} = 55.05 \times 1.005(395.88 - 300) \Rightarrow Q_{i3} = 5304.8\ \frac{kJ}{min}$$

9. A reciprocating air compressor has four stage compression with 2 m^3/min of air being delivered at 150 bar when initial pressure and temperature are 1 bar, 27°C. Compression occur polytropically following polytropic index of 1.25 in four stages with perfect intercooling between stages. For the optimum intercooling conditions determine the intermediate pressures and the work required for driving compressor.

GIVEN:

$Z=4,\ \dot{V} = 2\ \frac{m^3}{min},\ P_1 = 1\ bar, T_1 = 27°C,\ P_{Z+1} = 150\ bar, pV^{1.25} = C,$

SOLUTION:

Intermediate Pressure:

$$\frac{P_2}{P_1} = \left(\frac{P_{Z+1}}{P_1}\right)^{\frac{1}{Z}} \Rightarrow \frac{P_2}{P_1} = \left(\frac{P_5}{P_1}\right)^{\frac{1}{Z}} \Rightarrow P_2 = \left(\frac{150}{1}\right)^{\frac{1}{4}} \times 1 \Rightarrow \mathbf{P_2 = 3.5\ bar}$$

$$\frac{P_3}{P_2} = \left(\frac{P_{Z+1}}{P_1}\right)^{\frac{1}{Z}} \Rightarrow \frac{P_3}{P_2} = \left(\frac{P_5}{P_1}\right)^{\frac{1}{Z}} \Rightarrow P_3 = \left(\frac{150}{1}\right)^{\frac{1}{4}} \times 3.5 \Rightarrow \mathbf{P_3 = 12.25\ bar}$$

Power required to drive the compressor:

$$\dot{W} = Z \times \frac{n}{n-1} P_1 \dot{V}\left[\left(\frac{P_{Z+1}}{P_1}\right)^{\frac{n-1}{z\times n}} - 1\right] \Rightarrow \dot{W} = 4 \times \frac{1.3}{1.3-1} \times 1 \times 10^2 \times \frac{2}{60}\left[\left(\frac{50}{1}\right)^{\frac{1.25-1}{4\times 1.25}} - 1\right]$$

$$\Rightarrow \mathbf{\dot{W} = 19.018\ kW}$$

10. Difference between Reciprocating and Rotary Compressor.

S. No	Reciprocating Compressor	Rotary Compressor
1.	The maximum delivery pressure may be as high as 1000 bar	The maximum delivery pressure is 10 bar only
2.	The maximum free air discharge is about 300 m^3/min	The maximum free air discharge is as high as 3000 m3/min
3.	They are suitable for low discharge of air at very high pressure	They are suitable for large discharge of air at low pressure
4.	The speed of air compressor is low	The speed of air compressor is high
5.	The air supply is intermittent	The air supply is continuous.
6.	The size of air compressor is large for the given discharge	The size of air compressor is small for the same discharge.
7.	The balancing is a major problem	There is no balancing problem.
8.	The lubricating system is complicated	The lubricating system is simple.
9.	The air delivered is less clean, as it comes in contact with the lubricating oil	The air delivered is more clean, as it does not come in contact with the lubricating oil.
10.	Isothermal efficiency is used for all sorts of calculations	Isentropic efficiency is used for all sorts of calculations.

11. Explain the construction and working principle of Rotary air compressor.

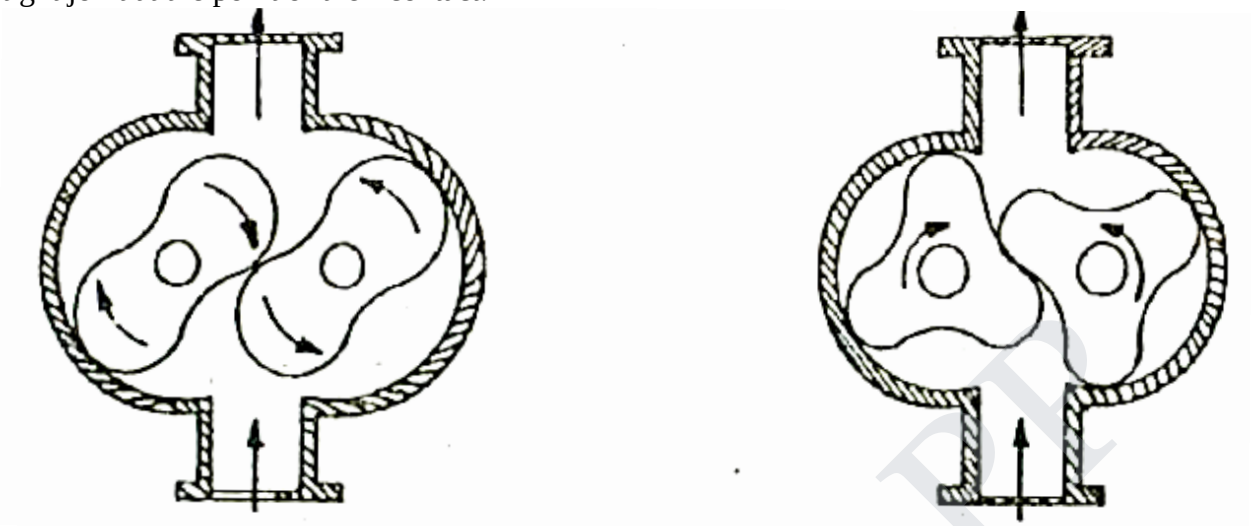
1. Roots blower compressor;
2. Vane blower compressor;
3. Centrifugal blower compressor
4. Axial flow compressor.

The first two compressors are popularly known as positive displacement compressors, whereas the last two as non-positive displacement. We shall discuss all the above mentioned rotary compressors one by one.

Note: The positive displacement compressors (i.e. roots blower and vane blower) are not very popular from the practical point of view. However, they have some academic importance. The only important rotary compressor is the centrifugal blower compressor.

Roots Blower Compressor:

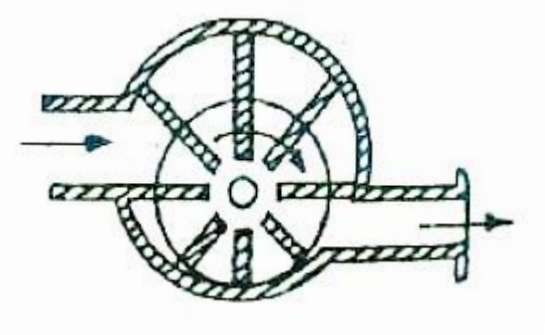
A roots blower compressor, in its simplest form, consists of two rotors with lobes rotating in an air tight casing which has inlet and outlet ports. Its action resembles with that of a gear pump. There are many designs of wheels, but they generally have two or three lobes (and sometimes even more). In all cases, their action remains the same as shown in Fig. The lobes are so designed that they provide an air tight joint at the point of their contact.



The mechanical energy is provided to one of the rotors from some external source, while the other is gear driven from the first. As the rotors rotate, the air, at atmospheric pressure, is trapped in the pockets formed between the lobes and casing. The rotary motion of the lobes delivers the entrapped air into the receiver. Thus more and more flow of air into the receiver increases its pressure. Finally, the air at a higher pressure is delivered from the receiver. It will be interesting to know that when the rotating lobe uncovers the exit port, some air (under high pressure) flows back into the pocket from the receiver. It is known as backflow process. The air, which flows from the receiver to the pocket, gets mixed up with the entrapped air. The backflow of air continues, till the pressure in the pocket and receiver is equalised. Thus the pressure of air entrapped in the pocket is increased at constant volume entirely by the back flow of air. The backflow process is shown in Fig. Now the air is delivered to the receiver by the rotation of the lobes. Finally, the air at a higher pressure is delivered from the receiver.

Vane Blower Compressor

A vane blower, in its simplest form, consists of a disc rotating eccentrically in an air tight casing with inlet and outlet ports. The disc has a number of slots (generally 4 to 8) containing vanes: When the rotor rotates the disc, the vanes are pressed against the casing, due to centrifugal force, and form air tight pockets. The mechanical energy is provided to the disc from some external source. As the disc rotates, the air is trapped in the pockets formed between the vanes and casing.



First of all, the rotary motion of the vanes compresses the air. When the rotating vane uncovers the exit port, some air (under high pressure) flows back into the pocket in the same way as discussed in the ease of roots blower compressor. Thus the pressure of air, entrapped in the pocket, is increased first by decreasing the volume and then by the backflow of air as shown in Fig. Now the air is delivered to the receiver by the rotation of the vanes. Finally, the air at a high pressure is delivered from the receiver.

12. Explain the construction and working principle of centrifugal compressor and axial flow compressor with neat sketches.

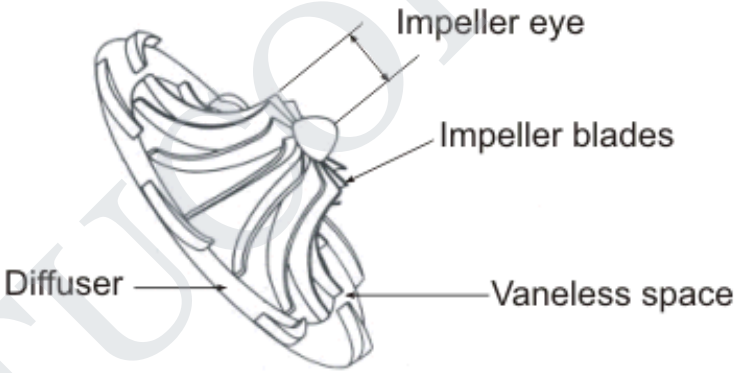
CENTRIFUGAL COMPRESSOR:

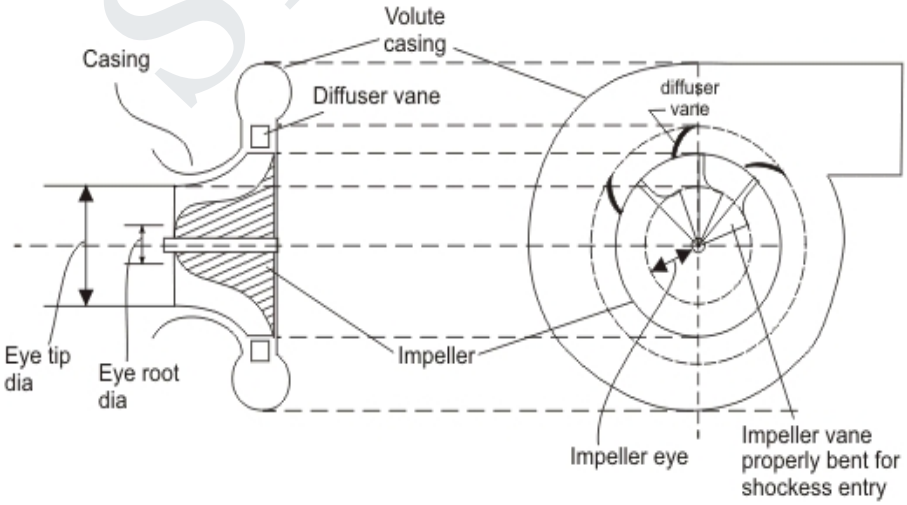
A centrifugal compressor is a radial flow rotodynamic fluid machine that uses mostly air as the working fluid and utilizes the mechanical energy imparted to the machine from outside to increase the total internal energy of the fluid mainly in the form of increased static pressure head.

During the second world war most of the gas turbine units used centrifugal compressors. Attention was focused on the simple turbojet units where low power-plant weight was of great importance. Since the war, however, the axial compressors have been developed to the point where it has an appreciably higher isentropic efficiency. Though centrifugal compressors are not that popular today, there is renewed interest in the centrifugal stage, used in conjunction with one or more axial stages, for small turbofan and turboprop aircraft engines.

A centrifugal compressor essentially consists of three components.

1. A **stationary casing**
2. A **rotating impeller** as shown in Fig. which imparts a high velocity to the air. The impeller may be single or double sided as show in Fig. but the fundamental theory is same for both.
3. A **diffuser** consisting of a number of fixed diverging passages in which the air is decelerated with a consequent rise in static pressure.





Principle of operation: Air is sucked into the impeller eye and whirled outwards at high speed by the impeller disk. At any point in the flow of air through the impeller the centripetal acceleration is obtained by a pressure head so that the static pressure of the air increases from the eye to the tip of the impeller. The remainder of the static pressure rise is obtained in the diffuser, where the very high velocity of air leaving the impeller tip is reduced to almost the velocity with which the air enters the impeller eye.

Usually, about half of the total pressure rise occurs in the impeller and the other half in the diffuser. Owing to the action of the vanes in carrying the air around with the impeller, there is a slightly higher static pressure on the forward side of the vane than on the trailing face. The air will thus tend to flow around the edge of the vanes in the clearing space between the impeller and the casing. These results in a loss of efficiency and the clearance must be kept as small as possible. Sometimes, a shroud attached to the blades as shown in Figure. May eliminate such a loss, but it is avoided because of increased disc friction loss and of manufacturing difficulties.

The straight and radial blades are usually employed to avoid any undesirable bending stress to be set up in the blades. The choice of radial blades also determines that the total pressure rise is divided equally between impeller and diffuser.

Before further discussions following points are worth mentioning for a centrifugal compressor.

(i) The pressure rise per stage is high and the volume flow rate tends to be low. The pressure rise per stage is generally limited to 4:1 for smooth operations.

(ii) Blade geometry is relatively simple and small foreign material does not affect much on operational characteristics.

(iii) Centrifugal impellers have lower efficiency compared to axial impellers and when used in aircraft engine it increases frontal area and thus drag. Multistaging is also difficult to achieve in case of centrifugal machines.

ADVANTAGES OF CENTRIFUGAL COMPRESSOR

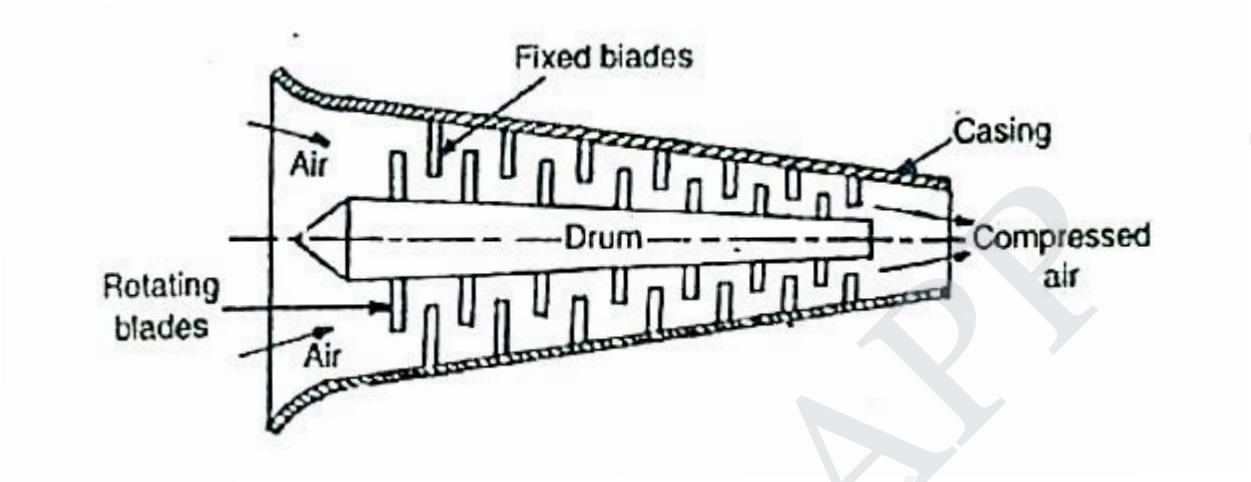
1. Wide Operating ranges
2. Very high reliability
3. Low maintenance costs
4. Low initial cost
5. Medium Capacity storage
6. High efficiency

DISADVANTAGES OF CENTRIFUGAL COMPRESSOR

1. Very unstable when the flow is reduced.
2. Sensitive to changes in gas compositions
3. Limited compression ratios
4. Limited turn downs
5. Periodic replacement is necessary for proper functioning

AXIAL FLOW COMPRESSORS:

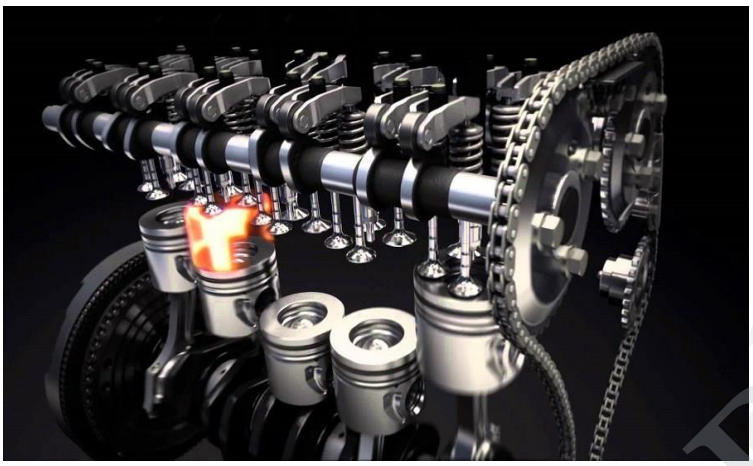
An axial flow compressor, in its simplest form, consists of a number of rotating blade rows fixed to a rotating drum. The drum rotates inside an air tight casing to which are fixed stator blade rows, as shown in Fig. The blades are made of aerofoil section to reduce the loss caused by turbulence and boundary separation.

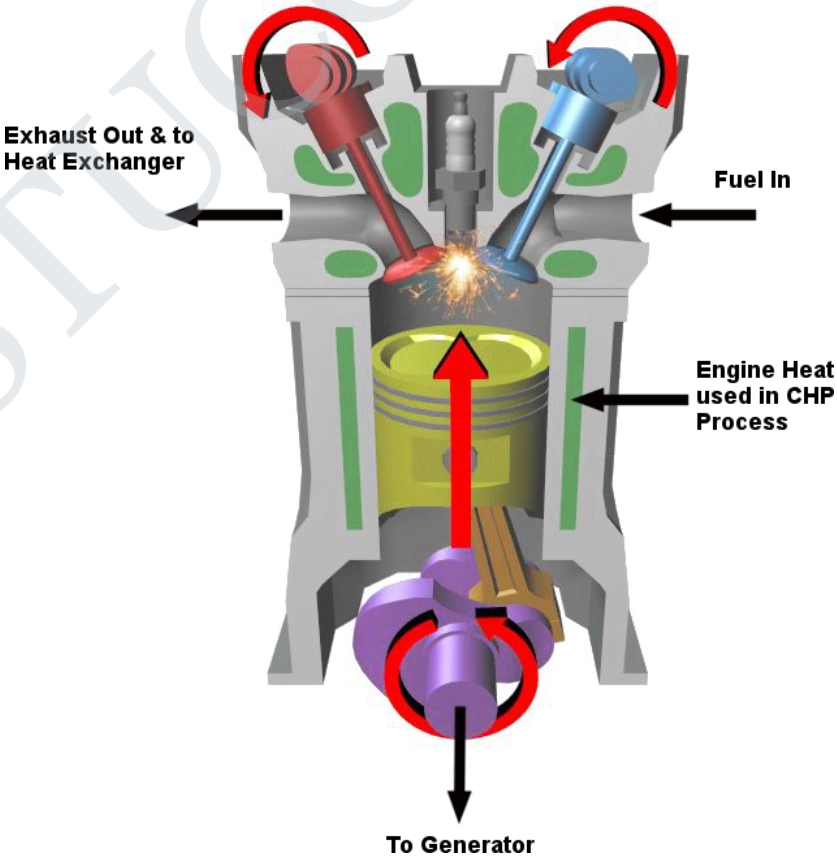


The mechanical energy is provided to the rotating shaft, which rotates the drum. The air enters from the left side of the compressor. As the drum rotates the airflows through the alternately arranged stator and rotor. As the air flows from one set of stator and rotor to another, it gets compressed. Thus successive compression of the air, in all the sets of stator and rotor, the air is delivered at a high pressure at the outlet point.

THERMAL ENGINEERING 1

UNIT III – INTERNAL COMBUSTION ENGINE AND COMBUSTION





ENGINE

- Energy Conversion Device (One Form to the Other)

HEAT ENGINE

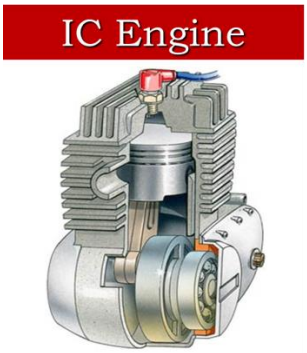
Convert Thermal Energy in fuel into Mechanical Energy for Motion

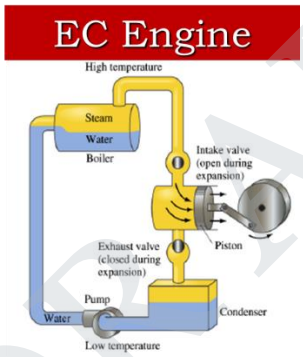
CLASSIFYING ENGINES

Classification is based on:

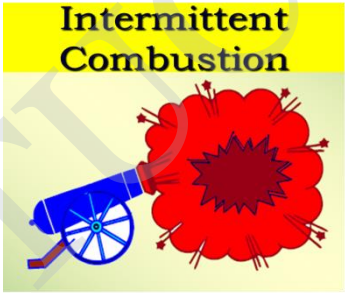
- The location of the combustion
- The type of combustion
- The type of internal motion

THE LOCATION OF THE COMBUSTION

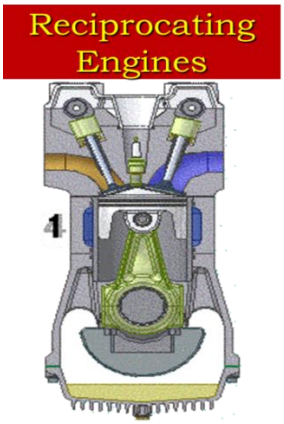


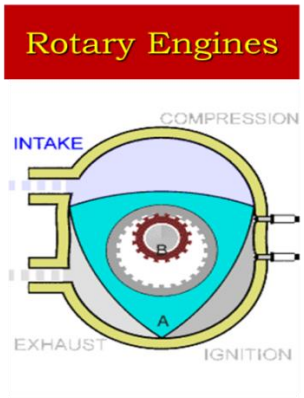


THE TYPE OF COMBUSTION



THE TYPE OF INTERNAL MOTION





IC ENGINE

The combustion of fuel takes place inside the cylinder is known as Internal Combustion Engines

Chemical Energy → **Heat Energy** → **Mechanical Energy**

CLASSIFY IC ENGINE

According to working cycle: a) Four stroke cycle engine b) Two stroke cycle engine

According to the type of fuel used: a) Petrol Engine b) Diesel Engine c) Gas Engine

According to the method of ignition: a) Spark Ignition (SI) b) Compression Ignition (CI)

According to the cooling system: a) Air cooled Engine b) Water cooled Engine

SWEPT VOLUME IN IC ENGINE

The volume swept by the piston during one stroke is called the swept volume (or) piston displacement. In other words, swept volume is the volume covered by the piston while moving from TDC to BDC.

SCAVENGING IN IC ENGINE

The process of pushing out of exhaust gases from the cylinder by admitting the fresh charge into the cylinder is known as scavenging.

SHORT CIRCUITING OF TWO STROKE ENGINE

In two stroke engine at certain speed the air fuel mixture is directly come out from the cylinder without undergoing combustion process is called short circuiting of two stroke engine.

OVERLAP PERIOD

The time duration during which both inlet and exhaust valves remain open is called as overlap period

FUNCTIONS OF A FLYWHEEL

A flywheel is an inertial energy-storage device. It absorb mechanical energy and serves as a reservoir, storing energy during the period when the supply of energy is more than the requirement and releases it during the period when the requirement of energy is more than the supply.

IGNITION DELAY

The period between the start of fuel injection into the combustion chamber and the start of combustion is termed as ignition delay period.

THREE STAGE OF COMBUSTION:

According to Ricardo, There are three stages of combustion in SI Engine

1. Ignition lag stage
2. Flame propagation stage
3. After burning stage

THE FACTORS WHICH AFFECT THE FLAME PROPAGATIONS ARE

1. Air fuel ratio
2. Compression ratio
3. Load on engine
4. Turbulence and engine speed
5. Other factors

EFFECTS OF PRE-IGNITION

- It increase the tendency of denotation in the engine
- It increases heat transfer to cylinder walls because high temperature gas remains in contact with for a longer time
- Pre-ignition in a single cylinder will reduce the speed and power output
- Pre-ignition may cause seizer in the multi-cylinder engines, only if only cylinders have pre-ignition

STAGES OF COMBUSTION IN CI ENGINE

The combustion in CI engine is considered to be taking place in four phases:

- Ignition Delay period /Pre-flame combustion
- Uncontrolled combustion
- Controlled combustion
- After burning

EFFECT OF VARIOUS FACTORS ON DELAY PERIOD IN CI ENGINE

- Many design and operating factors affect the delay period. The important ones are:
- compression ratio
- engine speed
- output
- injection timing
- quality of the fuel
- intake temperature
- intake pressure

KNOCKING

If the delay period of C.I engines is long, more fuel is injected and accumulated in the chamber. When ignition begins, pulsating pressure rise can be noticed and creates heavy noise. This is known as knocking.

THE VARIOUS ENGINE VARIABLES AFFECTING KNOCKING CAN BE CLASSIFIED AS:

- Temperature factors
- Density factors
- Time factors
- Composition factors

PHENOMENON OF "KNOCKING" IN SPARK IGNITED ENGINES.

Auto-ignition of end charge is responsible for knocking in spark ignited engines.

EFFECTS OF KNOCKING

(i) The engine parts gets overheated which may cause damage to the piston.

(ii) It creates heavy vibration of engine and hence louder noise and roughness.

(iii) Decrease in power output and efficiency.

(iv) More heat is lost to the coolant as the dissipation rate is rapid.

EFFECTS OF CO IN GLOBAL WARMING

(a) Rise in global temperature

(b) Rise in sea level

(c) Food shortages and hunger

(d) Climate change

METHODS TO REDUCE NO_x FROM A DIESEL ENGINE

(a) Low self ignition temperature

(b) Reduction of excess air

(c) Use of catalytic converter.

METHODS TO REDUCE HC FROM A DIESEL ENGINE

(a) Complete combustion

(b) Avoiding rapid deceleration

(c) Normal speed running.

PART - B (THEORY)

1. Discuss the Claasification of IC engines.

- A heat engine is a machine, which converts heat energy into mechanical energy.
- The combustion of fuel such as coal, petrol, diesel generates heat.
- This heat is supplied to a working substance at high temperature. By the expansion of this substance in suitable machines, heat energy is converted into useful work.
- Heat engines can be further divided into two types:
  (i) External combustion and
  (ii) Internal combustion.
- In a steam engine the combustion of fuel takes place outside the engine and the steam thus formed is used to run the engine. Thus, it is known as external combustion engine.
- In the case of internal combustion engine, the combustion of fuel takes place inside the engine cylinder itself.

The IC engine can be further classified as:

(i) stationary or mobile,

(ii) horizontal or vertical and

(iii) low, medium or high speed.

The two distinct types of IC engines used for either mobile or stationary operations are:

(i) diesel and (ii) carburettor.

According To :

(i) The Type of Fuel Used

(ii) The Method of Ignition

(iii) The Type of Working Cycle Used

(iv) The Number of Stroke per Cycle

(v) The Number of Cylinders Used

(vi) The Arrangement of Cylinders

(vii) The Valve Location

(viii) The Method of Fuel Injection

(ix) The Type of Cooling System

(x) The Speed of The Engine

(xi) The Method of Governing

(xii) The field of application

Heat Engine

- **External Combustion**
  - **Steam Engine**
- **Internal Combustion**
  - **Reciprocating**
    - **CI Engine**
      - **Two Stroke**
      - **Four Stroke**
    - **SI Engine**
      - **Two Stroke**
      - **Four Stroke**
  - **Wankel Rotary**
  - **Gas Turbine**

Spark Ignition (Carburettor Type) IC Engine

In this engine liquid fuel is atomised, vaporized and mixed with air in correct proportion before being taken to the engine cylinder through the intake manifolds. The ignition of the,mixture is caused by an electric spark and is known as spark ignition.

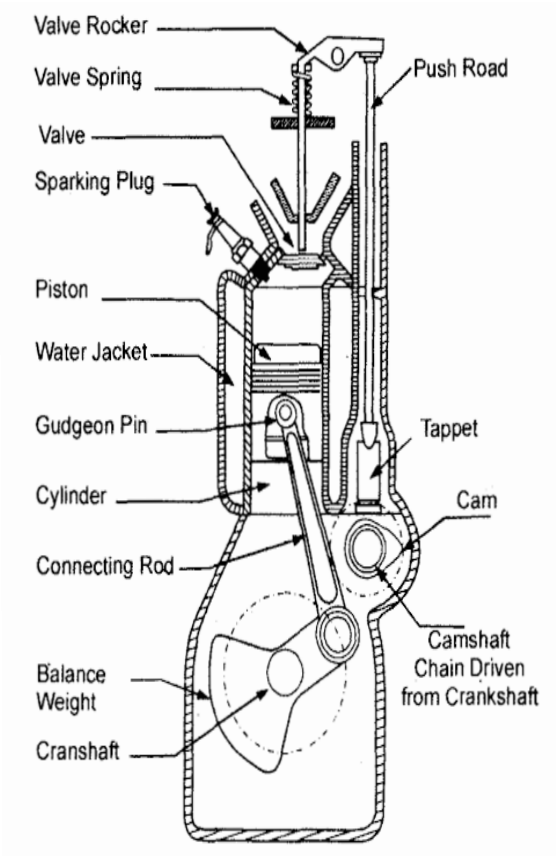
Compression Ignition (Diesel Type) IC Engine

In this only the liquid fuel is injected in the cylinder under high pressure.

2. Discuss the components of IC engines.

Cylinder:

Its major function is to provide space in which the piston can operate to draw in the fuel mixture or air (depending upon spark ignition or compression ignition), compress it, allow it to expand and thus generate power. The cylinder is usually made of high-grade cast iron. In some cases, to give greater strength and wear resistance with less weight, chromium, nickel and molybdenum are added to the cast iron.



Piston:

Piston is the main part of the engine. The main function of the piston is to compress the charge and to transmit the gas force to the connecting rod during the power stroke.

The piston is closed at one end and open on the other end to permit direct attachment of the connecting rod and its free action.

The materials used for pistons are grey cast iron, cast steel and aluminium alloy. However, the modern trend is to use only aluminium alloy pistons in the tractor engine.

Piston Rings:

- The primary function of the piston rings is to retain compression and at the same time reduce the cylinder wall and piston wall contact area to a minimum, thus reducing friction losses and excessive wear.
- The other important functions of piston rings are the control of the lubricating oil, cylinder lubrication, and transmission of heat away from the piston and from the cylinder walls.

There are two types of piston rings

- Compression rings
- Oil scraper rings

The upper rings are the compression rings. They help in sealing and preventing the gas from leaking past the piston into the casing.

- The lower rings are the oil scraper rings. They are provided to remove the oil film from the cylinder walls. It is made of case hardened alloy steel with precision finish. There are three different methods to connect the piston to the connecting rod.
- These are made of cast iron on account of their ability to retain bearing qualities and elasticity indefinitely.

Piston Pin:

The connecting rod is connected to the piston through the piston pin.

Piston rings are circumferential rings that are provided in the piston grooves.

Connecting Rod:

This is the connection between the piston and crankshaft. The end connecting the piston is known as small end and the other end is known as big end. The big end has two halves of a bearing bolted together. The connecting rod is made of drop forged steel and the section is of the I-beam type.

Crankshaft:

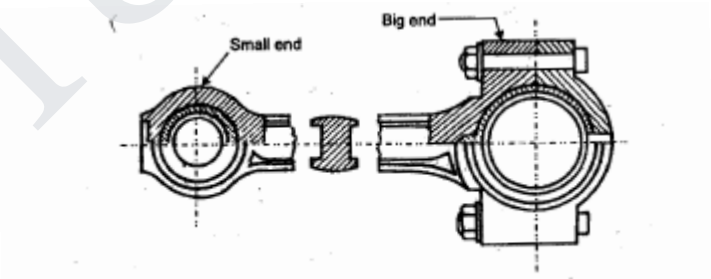


Fig. . Connecting rod.

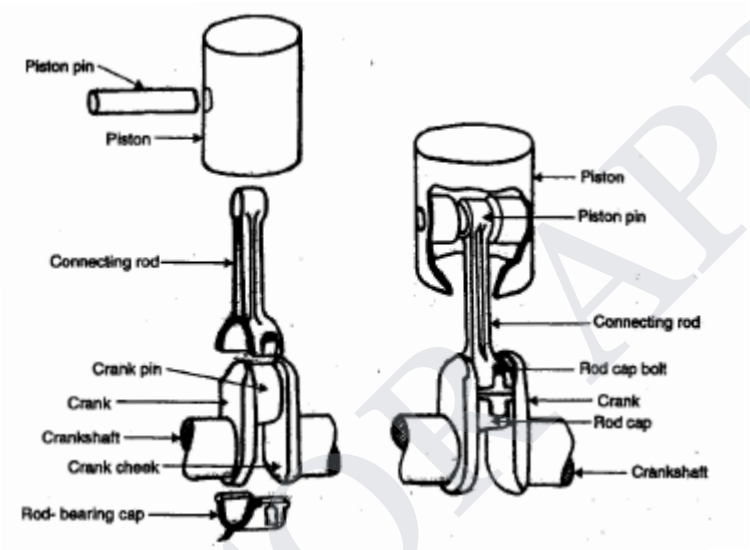
This is connected to the piston through the connecting rod and converts the linear motion of the piston into the rotational motion of the flywheel. The journals of the crankshaft are supported on main bearings, housed in the crankcase. Counter-weights and the flywheel bolted to the crankshaft help in the smooth running of the engine.

Engine Bearings:

The crankshaft and camshaft are supported on anti-friction bearings. These bearings must be capable of with standing high speed, heavy load and high temperatures. Normally, cadmium, silver or copper lead is coated on a steel back to give the above characteristics. For single cylinder vertical/horizontal engines, the present trend is to use ball bearings in place of main bearings of the thin shell type.

Valves:

To allow the air to enter into the cylinder or the exhaust, gases to escape from the cylinder, valves are provided, known as inlet and exhaust valves respectively. The valves are mounted either on the cylinder head or on the cylinder block.



Camshaft:

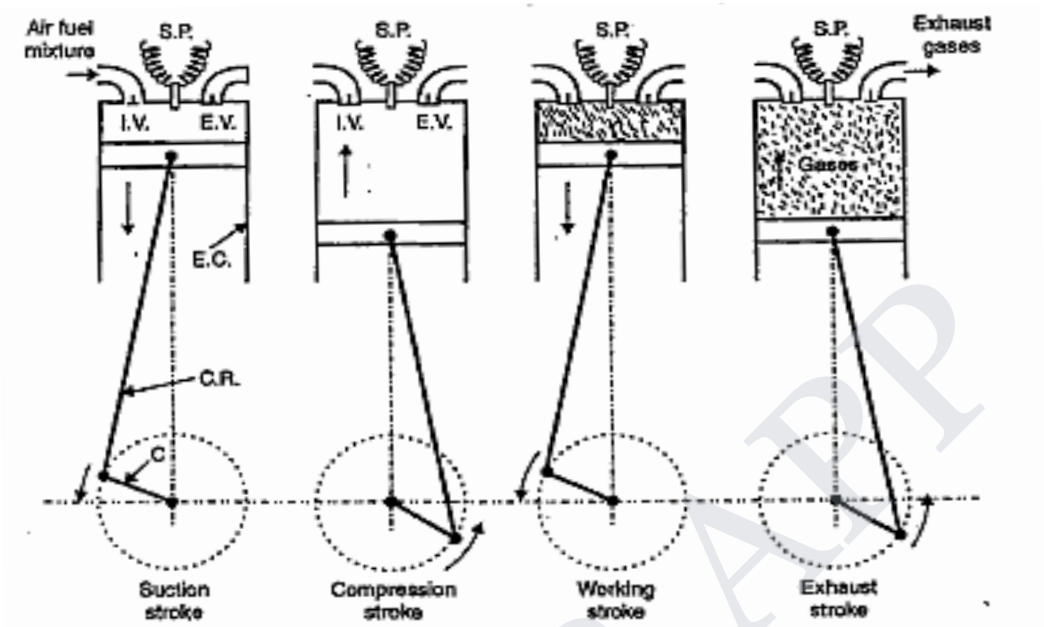
The valves are operated by the action of the camshaft, which has separate cams for the inlet, and exhaust valves. The cam lifts the valve against the pressure of the spring and as soon as it changes position the spring closes the valve. The cam gets drive through either the gear or sprocket and chain system from the crankshaft. It rotates at half the speed of the camshaft.

Flywheel

This is usually made of cast iron and its primary function is to maintain uniform engine speed by carrying the crankshaft through the intervals when it is not receiving power from a piston. The size of the flywheel varies with the number of cylinders and the type and size of the engine. It also helps in balancing rotating masses.

3. Discuss the construction, working of a 4 stroke SI engine with sketch and Theroritical and Actual P-v diagrams.

In petrol engines, gas engines, light oil engines in which the mixture of air and fuel are drawn in the engine cyünder. Since ignition in these engines is due to a spark, therefore they are also called spark ignition engines. The various strokes of a four stroke (otto) cycle engine are itetailed below.



I.V. = Inlet valve, E.V. = Exhaust valve, E.C. = Engine cylinder, C.R. = Connecting rod
C = Crank, S.P. = Spark plug.

Suction stroke:

During this stroke (also known as induction stroke) the piston mores from top dead centre (T.D.C.) to trottom dead centre (B.D.C.); the inlet valve opens and proportionate fuel air mixture is sucked in the engine cylinder. This operation is represented by the line 5-1 (Fig). The exhaust valve remains closed throughout the stroke.

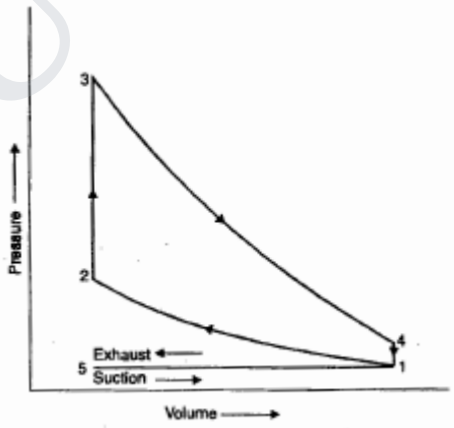


Fig. Theoretical p-V diagram of a four stroke Otto cycle engine.

Compression stroke:

In this stroke, the piston moves (1_2) towards (T.D.C.) and compresses the enclosed fuel air mixture drawn in the engine cylinder during suction.

The pressure of the mixture rises in the cylinder to a value of about 8 bar. Just before the end of this stroke the operating-plug initiates a spark which ignites the mixture and combustion takes place at constant volume (line 2--3) (Fig.). Both the inlet and exhaust valve remains closed during the stroke.

Expranslon or working stroke:

- when the mirture is ignited by the spark plug the hot geses are produced which drive or throw the piston from T.D.C. to B.D.C. and thus the work is obtained in this stroke.
- During this stroke when we get work from the engine; the other three strokes names suction, compression and exhaust being idle.
- The flywheel mounted, on the engine shaft stores energy during this stroke and supplies it during the idle strokes. The expansion of the gases is shown by 3-4. (Fig). Both the valves remain closed during the start of this stroke but when the piston just reaches the B.D.C. the exhaust valve opens.

Exhaust stroke:

- This is the last stroke of the cycle. Here the gases from which the work has been collected become useless after the completion of the expansion stroke and are made to escape through exhaust valve to the atrnosphere.
- T'his removal of gas is accomplished during this stroke. The piston moves from B.D.C. to T.D.C. and the exhaust gases are driven out of the engine cylinder, this is also called scavenging.
- This operation is represented by the line (1-5) (Fig).

Fig. shows the actual indicator diagram of four stroke Otto cycle engine.

- It may be noted that line 5-1 is below the atmospheric pressure line. This is due to the fact that owing to restricted area of the inlet passages the entering fuel air mixture cannot cope with the speed of the piston.

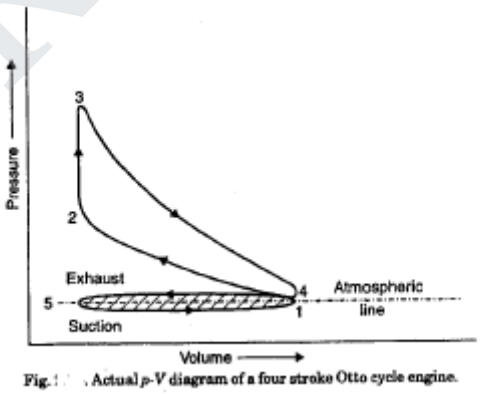
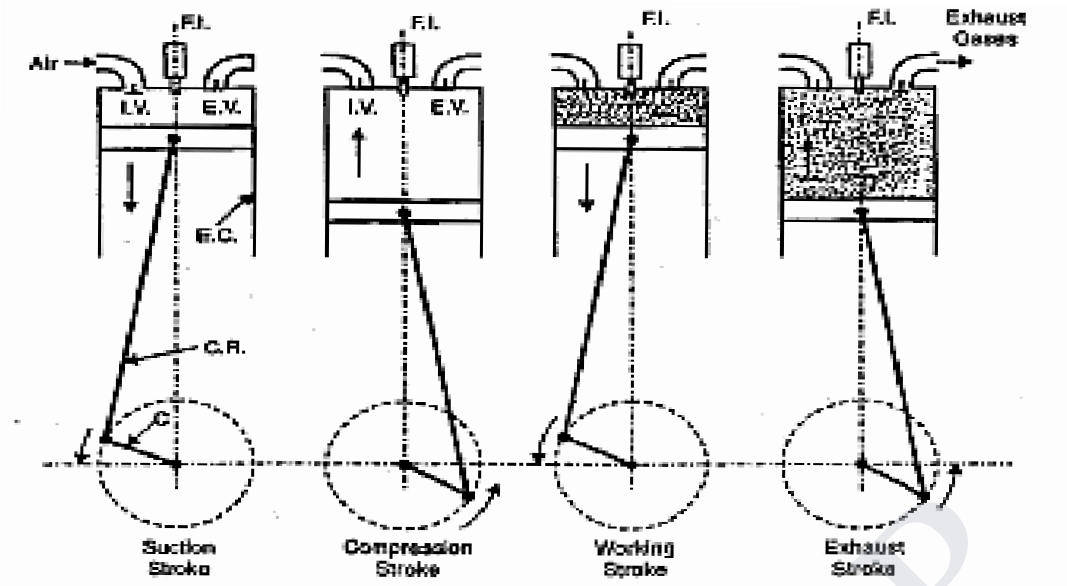


Fig. . Actual p-V diagram of a four stroke Otto cycle engine.

- The exhaust line 4-5 is slightly above the atmospheric pressure line. This is due to restricted exhaust passeges which do not allow the exhaust gases to leave the engine-cylinder quickly.
- The loop which has area 4-5-1 is called negative loop ; it gives the pumping loss due to admission of fuel air mixture and removal of exhaust gases.

The area 1-2-3-4 is the total or gross work obtained from the piston and network can be obtained by subtracting area 451 from the area t-2-3-4.

4. Discuss the construction and working of a 4 stroke CI engine with sketch.



F.I. = Fuel injector, I.V. = Inlet valve, E.V. = Exhaust valve

Fig. . . Four stroke Diesel cycle engine.

Suction stroke:

During this stroke (also known as induction stroke) the piston mores from top dead centre (T.D.C.) to trottom dead centre (B.D.C.); the inlet valve opens and proportionate fuel air mixture is sucked in the engine cylinder. This operation is represented by the line 5-1 (Fig). The exhaust valve remains closed throughout the stroke.

Compression stroke:

- Piston starts moving from BDC to TDC during this stroke.
- Both inlet valve and the exhaust valve are in closed condition.
- The air is drawn at atmospheric pressure during the suction stroke is compressed to high pressure and temperature.
- Air is compressed to 12 to 18 times inside the engine cylinder till the piston reaches TDC.
- This operation is represented by the line 1-2.

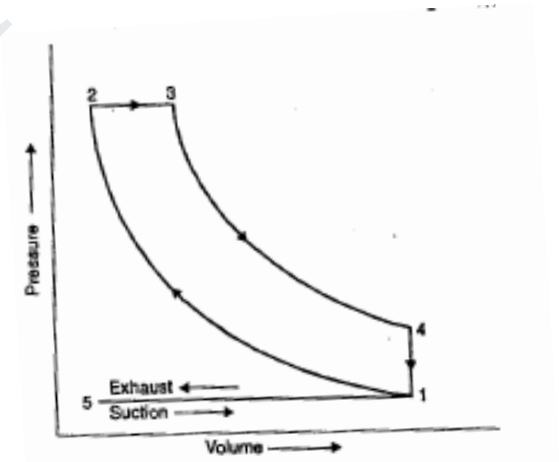


Fig. . Theoretical p-V diagram of a four stroke Diesel cycle.

Expansion or Working stroke:

- As the piston starts moving from T.D.C a matered quantity of fuel is injected into the hot compressed air in fine sprays by the fuel injector and fuel starts burning at constant pressure shown by the line 2-3.
- At the point 3 fuel supply is cut off. The fuel is injected at the end of compression stroke but in actual practice the iginition of the fuel starts before the end of the compression stroke.
- The hot gases of the cylinder expands adiabatically to point 4, thus doing work on the piston. The expansion is shown by 3-4.

Exhaust Stroke:

The piston moves from the B.D.C to T.D.C and the exhaust gases escape to the atmosphere through the exhaust valve. When the piston reaches the T.D.C. the exhaust valve closes and the cycle is completed. This stroke is represented by the line 1-5.

In fig. shows the actual indicator diagram for a four stroke diesel cycle engine.

- It may be noted that line 5-1 is below the atmospheric pressure line.
- This is due to the fact that owing to the restricted area of the intel passages the entering air cant cope with the speed of the piston.

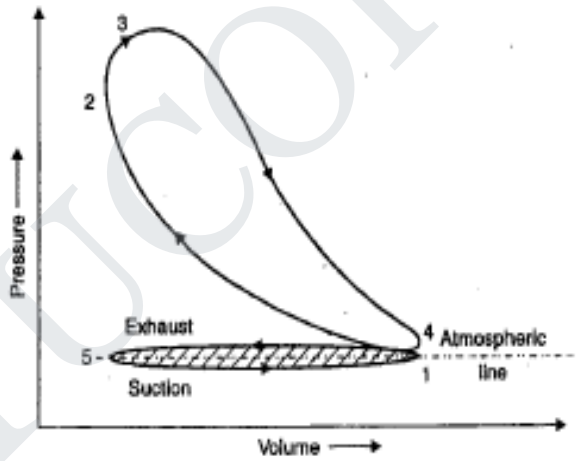


Fig. Actual p-V diagram of four stroke Diesel cycle.

- The exhust line 4-5 is slightly above the atmospheric line. This is because of the restricted exhaust passages which do not allow the exhaust gases to leave the engine cylinder quickly.
- The loop of area 4-5-1 is called negative loop, it gives the pumping loss due to admission of air and removal of exhaust gases. The area 1-2-3-4 is the total or gross work obtained from the piston the net work can be obtained by subtracting area 4-5-1 from area 1-2-3-4.

5. Draw and explain the valve timing diagram of a 4 stroke SI engine

Otto engine. Fig. shows a theoretical valve timing diagram fot four stroke "Otto cycle engines which is self explanatory.

In actual, it is difficult to open and close the valve instantaneously. So as to get better performance of the engine the valve timings are modified.

Below fig. shown Theoretical and Actual valve timing diagram.

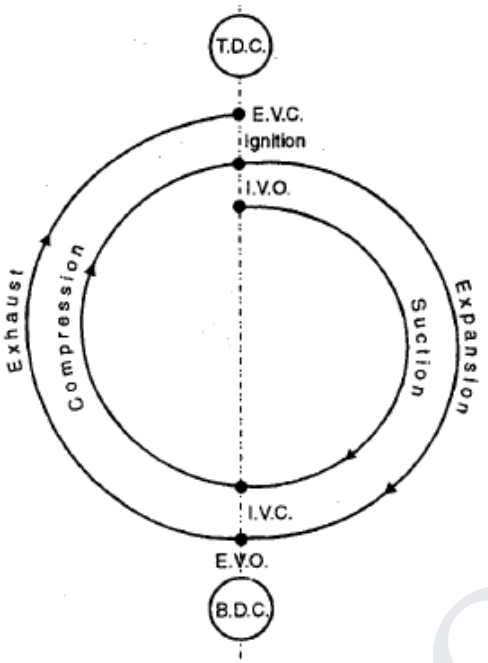
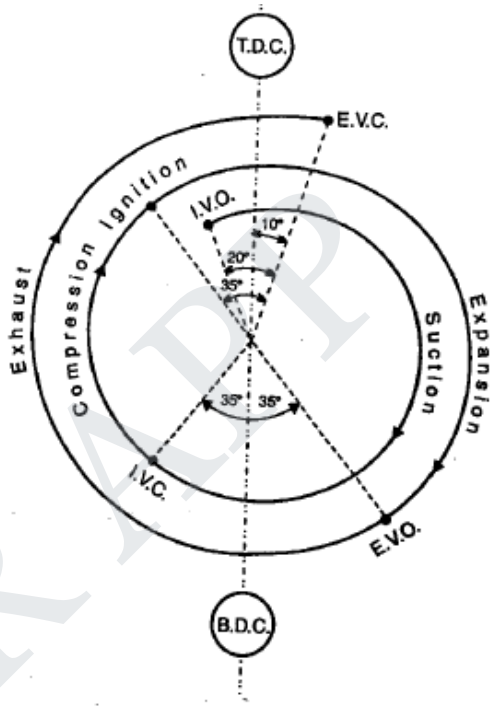


Fig. 6. Theoretical valve timing diagram (four stroke Otto cycle engine).



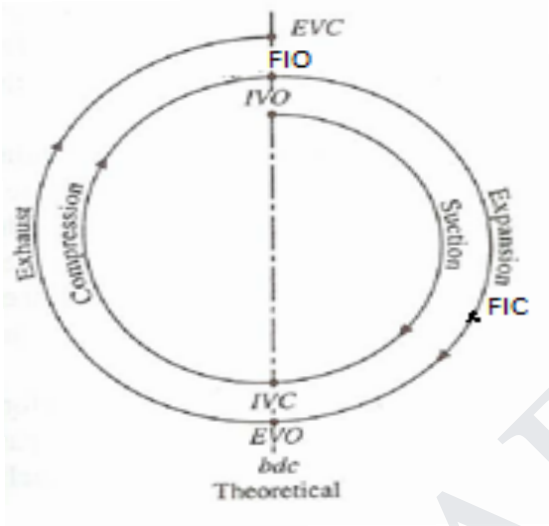
Actual valve timing diagram (four Stroke Otto cycle engines).

- The inlet valve is opened 10 to 30 in advance of the TDC position to enable the fresh charge to enter the cylinder and to help the burnt gases at the same time, to escape to the atmosphere.
- The suction of the mixture continues up to 30 to 40 or even 60 after BDC position.
- The inlet valve closes and the compression of the entrapped mixture starts.
- The spark plug produces a spark 30 to 40 before the TDC position,thus fuel gets more time to burn.
- The pressure becomes maximum nearly 10 past the TDC position.
- The exhaust valve opens 30 to 60 before the BDC position and the gases are driven out of the cylinder by piston during its upward movement.
- The exhaust valve closes when piston is nearly 10 past TDC position.

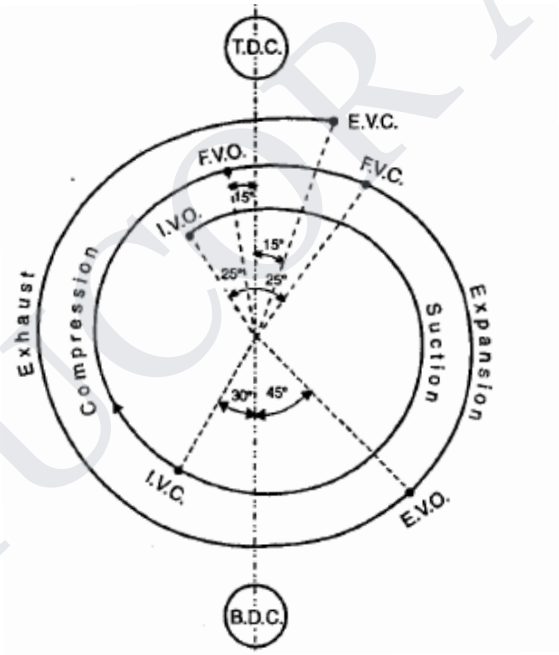
6. Draw and explain the valve timing diagram of a 4 stroke CI engine

Diesel engine. Fig. shows a theoretical valve timing diagram fot four stroke "Diesel cycle engines which is self explanatory.

In actual, it is difficult to open and close the valve instantaneously. So as to get better performance of the engine the valve timings are modified.



Below fig. shown theoretical and actual valve timing diagram:



- The inlet valve is opened 10 to 25 in advance of the TDC position to enable the air to enter the cylinder and to help the burnt gases at the same time, to escape to the atmosphere.
- The suction of the air continues up to 25 to 50 or even 60 after BDC position.
- The fuel injection takes place 5 to 10 before TDC position and continues up to 15 to 25 near TDC position.
- The pressure becomes maximum nearly 10 past the TDC position.
- The exhaust valve opens 30 to 50 before the BDC position and the gases are driven out of the cylinder by piston during its upward movement.
- The exhaust valve closes when piston is nearly 5 to 10 after TDC position.

7. Discuss the construction and working of a 2 stroke engine with sketch.

British engineer introduced a cycle which could be completed in two stroke of piston rather than four strokes as is the case with the four stroke cycle engines. The engines using this cycle were called two stroke cycle engines.

- In this engine suction and exhaust strokes are eliminated.
- Here instead of valves, ports are used.
- The exhaust gases are driven out from engine cylinder by the fresh charge of fuel entering the cylinder nearly at the end of the working stroke Fig. shows

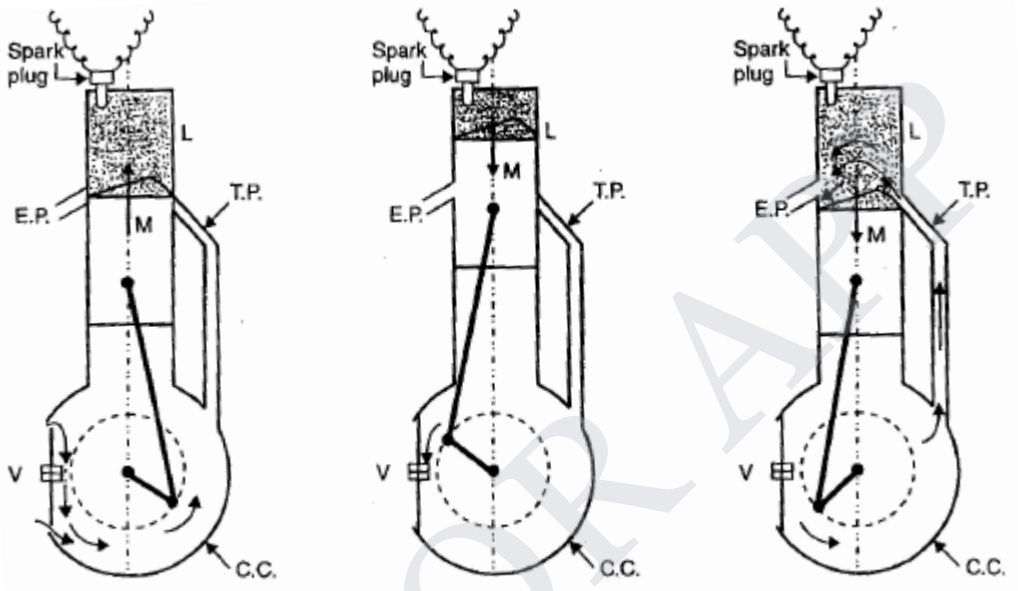


Fig. Two stroke cycle engine.

- Fig. shows a two stroke petrol engine (used in scooters, motor cycles etc.). Ttre cylinder L is connected to a closed crank chamber (C.C).
- During the upward stroke of the piston M, the gases in cylinder are compressed and at the same tirne fresh air and fuel (petrol) mixture enters the crank chamber through the inlet port V.
- When the piston rnoves downwards, inlet port closes and the mixture in the crank chamber is compressed.
- Refer Fig. (i), the piston is moving upwards and is compressing an explosive change which has previously been supplied to cylinder.
- Ignition takes place at the end of the stroke.
- The piston then travels downwards due to expansion of the gases (Fig. (ii)) and near the end of this stroke the piston uncovers the exhaust port (E.p.) and the burnt exhaust gases escape through this port.
- The transfer port (T.p.) then is uncovered inmediately, and the compressed charge from the crank chamber flows into the cylinder and is deflected upwards by the hump provided on the head of the piston.
- It may be noted that the incoming air petrol mixture helps the removal of gases from the engine-cylinder.

- If in case these exhaust gases do not leave the cylinder, the fresh charge gets diluted and efficiency of the engine will decrease.
- The piston then starts moving from BDC TO TDC and the charge gets compressed when exhaust port and transfer port are covered by the piston, thus the cycle is repeated.

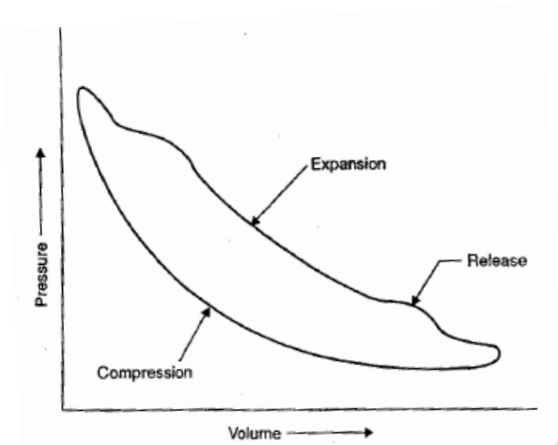


Fig. p-V diagram for a two stroke cycle engine.

Advantages of 2-Stroke Engines

- Two stroke engines do not have valves which are easy to construct and lowers their weight
- Two stroke engines fire once every revolution. Power is produced once during 2 strokes of the piston .This gives a significant power boost.
- This two stroke engines lower output in horse power
- These Engines often provide high power-to-weight ratio, usually in a narrow range of rotational speeds called the "power band".

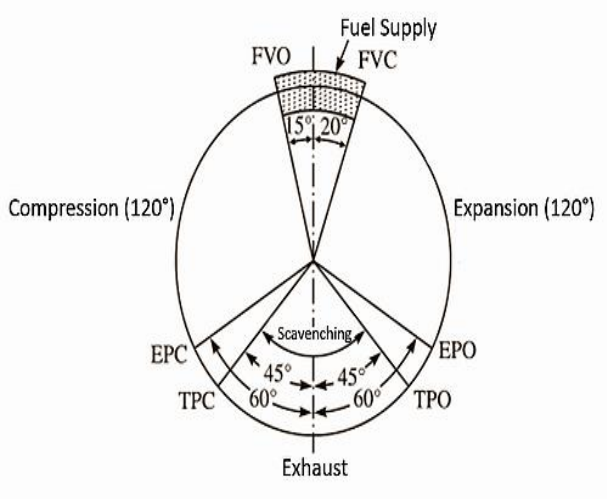
Disadvantages of 2-Stroke Engines

- Two stroke engines do not last as long as four stroke engines; there is no lubrication system in a two stroke engine so parts wear out a lot faster.
- Two stroke oil is expensive; you would burn a gallon every 1000 miles if it were in a car
- Two stroke engines are Less Efficient.
- Two stroke engines produce a lot of pollution, and the way the engine is designed that part of the air/fuel leaks out of the chamber through the exhaust port.
- The exhaust gases often get trapped inside the combustion chamber. This makes the fresh charge impure. Therefore maximum power doesn't get delivered because of improper incomplete combustion.

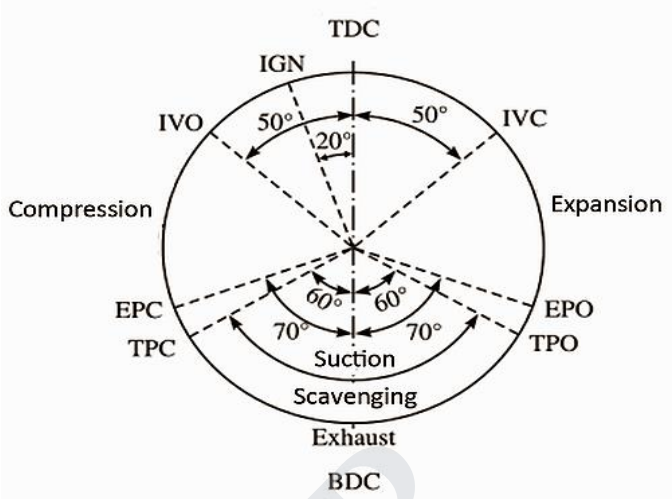
8. Draw and explain the Port timing diagram of a 2 stroke petrol engine.

- The experimental setup consists of a two stroke petrol engine which has a piston, piston rings, connecting rod, crank shaft, crank case, spark plug and flywheel.
- The piston's extreme bottom position of the cylinder is called "Inner Dead Centre" [IDC] and at the extreme top position of the cylinder is called "Outer Dead Centre" [ODC].
- The piston travel between ODC and IDC is called stroke.

Diesel engine



Petrol Engine



- In a two stroke petrol engine Compression stroke and Power stroke are performed sequentially.
- The port timing diagram (indicator diagram) shows the opening and closing of both transfer, inlet and exhaust ports with respect to crank angle.
- It also indicates the spark ignition timings. we can also find the Scavenging time from the port timing diagram.
- In practice the inlet port is opened when the piston at just after IDC and closes at just after ODC.
- Similarly the exhaust and transfer ports are opened when the piston at just before IDC and closes at just After IDC.
- Spark ignition starts when the piston reaches just before ODC.

9. Discuss difference between petrol engine and Diesel engine

S. NO	Petrol engine	Diesel engine
1.	Fuel (petrol) and air is admitted into the cylinder during suction stroke.	Air alone is admitted into the cylinder during suction stroke.
2.	Fuel admission is through carburetor.	Fuel admission is through fuel injector.
3.	Spark ignition system is used for ignition.	Compression ignition system is used for combustion.
4.	Compression ration varies from 6 to 10.	Compression ratio varies from 14 to 22.
5.	Due to light weight, they run at high speeds (2000 to 5000 rpm or above).	Due to heavy weight, they run at low speeds (400 rpm) and medium speeds (400-1200 rpm).
6.	Efficiency is less due to low compression ratio.	Efficiency is more due to higher compression ratio.
7.	Easy to start.	Difficult to start the engine since more cranking effort is required.
8.	Lighter in weight.	Heavier in weight due to high pressure.
9.	More fuel consumption.	Less fuel consumption.
10.	Fuel cost is more.	Fuel cost is less.

10. Discuss difference between 4 stroke engine and 2 stroke engine

S. No.	Four stroke cycle engine	Two stroke cycle engine
1	For every two revolutions one power stroke is produced.	For every one revolution one power stroke is produced.
2	Power produced for same size engine is small.	Power produced for same size engine is more. (Theoretically twice, actually 1.75 times).
3	For the same power more space is required.	For the same power small space is required.
4	Valves are required.	Ports are made in the cylinder walls.
5	Heavier flywheel is required because of non-uniform torque on the crank shaft.	Lighter flywheel because of uniform torque.
6	Fuel cannot escape with exhaust gases, since scavenging is better. Hence, fuel consumption is less.	Some of the fuel may escape with exhaust gases. Since, scavenging is poor. Hence, fuel consumption is more.
7	Not possible to make the engine reversible.	Possible to make the engine reversible.
8	Lesser cooling and lubrication requirements.	Greater cooling and lubrication requirements.
9	Lesser rate of wear and tear.	Greater rate of wear and tear.
10	Starting of the engine is fairly difficult.	Starting of the engine is easy.
11	It gives less noise, since exhaust gases are released in a separate stroke.	It gives more noise due to sudden release of exhaust gases.
12	Thermal efficiency is more.	Thermal efficiency is less.
13	Used in heavy vehicles like buses, lorries, trucks, cars, etc.	Used in light vehicles like motor cycles, scooters, mopeds, etc.

11. Discuss comparison between SI engine and CI engine.

S. No.	PARAMETERS	SI ENGINES	CI ENGINES
1.	Fuel	Petrol (gasoline)	Diesel oil
2.	Basic cycle	Otto cycle	Diesel cycle
3.	Intake	Fuel (petrol) and air is admitted into the cylinder during suction stroke	Air alone is admitted into the cylinder during suction stroke
4.	Fuel admission	Through carburettor	Through fuel injector
5.	Ignition system	Spark ignition system	Compression ignition system
6.	Compression ratio range	6 to 10	14 to 22
7.	Engine speed	Due to light weight, they run at high speeds (2000 to 5000 rpm or above)	Due to heavy weight, they run at low speeds (400 rpm) and medium speeds (400 to 1200 rpm)
8.	Efficiency	Efficiency is less due to low compression ratio	Efficiency is more due to higher compression ratio

9.	Starting	Easy to start	Difficult to start the engine since more cranking effort is required.
10.	Weight	Lighter	Heavier due to high pressures
11.	Fuel consumption	More	Less
12.	Fuel cost	More	Less
13.	Engine cost	Less costly	More costly
14.	Vibration and noise	Very less	More, due to high operation pressures.
15.	Engine life	Less than 60,000 km	More than 1,50,000 km.
16.	Space	For the same power output, it occupies lesser space.	For the same power output, it occupies more space.

12. Detailed explanation of properties and quality of fuels.

TYPES OF FUELS AND THEIR CHARACTERISTICS

Fuel is a substance which, when burnt, i.e. on coming in contact and reacting with oxygen or air, produces heat. Thus, the substances classified as fuel must necessarily contain one or several of the combustible elements: carbon, hydrogen, sulphur, etc.

In the process of combustion, the chemical energy of fuel is converted into heat energy. To utilize the energy of fuel in most usable form, it is required to transform the fuel from its one state to another, i.e. from solid to liquid or gaseous state, liquid to gaseous state, or from its chemical energy to some other form of energy via single or many stages. In this way, the energy of fuels can be utilized more effectively and efficiently for various purposes.

PRINCIPLES OF CLASSIFICATION OF FUELS

Fuels may broadly be classified in two ways, i.e.

(a) according to the physical state in which they exist in nature – solid, liquid and gaseous, and

(b) according to the mode of their procurement – natural and manufactured.

SOLID FUELS AND THEIR CHARACTERISTICS

Solid fuels are mainly classified into two categories, i.e. natural fuels, such as wood, coal, etc. and manufactured fuels, such as charcoal, coke, briquettes, etc.

The various advantages and disadvantages of solid fuels are given below :

Advantages

1. They are easy to transport.
2. They are convenient to store without any risk of spontaneous explosion.
3. Their cost of production is low.

4. They possess moderate ignition temperature.

Disadvantages

1. Their ash content is high.
2. Their large proportion of heat is wasted.
3. They burn with clinker formation.
4. Their combustion operation cannot be controlled easily.

LIQUID FUELS AND THEIR CHARACTERISTICS

The liquid fuels can be classified as follows:

(a) Natural or crude oil, and

(b) Artificial or manufactured oils.

The advantages and disadvantages of liquid fuels can be summarized as follows:

Advantages

(a) They possess higher calorific value per unit mass than solid fuels.

(b) They burn without dust, ash, clinkers, etc.

(c) Their firing is easier and also fire can be extinguished easily by stopping liquid fuel supply.

(d) They are easy to transport through pipes.

Disadvantages

(a) The cost of liquid fuel is relatively much higher as compared to solid fuel.

(b) Costly special storage tanks are required for storing liquid fuels.

(c) There is a greater risk of five hazards, particularly, in case of highly inflammable and volatile liquid fuels.

(d) They give bad odour.

GASEOUS FUELS

Gaseous fuels occur in nature, besides being manufactured from solid and liquid fuels. The advantages and disadvantages of gaseous fuels are given below:

Advantages

Gaseous fuels due to erase and flexibility of their applications, possess the following advantages over solid or liquid fuels:

(a) They can be lighted at ease.

(b) They have high heat contents and hence help us in having higher temperatures.

(c) They are clean in use.

(d) They do not require any special burner.

(e) They burn without any shoot, or smoke and ashes.

Disadvantages

(a) Very large storage tanks are needed.

(b) They are highly inflammable, so chances of fire hazards in their use is high.

Fuel properties

1. Viscosity of Fuel

Viscosity is the resistance offered by the fuel to its own flow. Viscosity decreases when the temperature of fuel increases and vice versa. Good fuel should have proper viscosity.

2. Pour Point of Fuel

The pour point (freezing point) of fuel must be less than the lowest climate temperature of atmosphere. In cold climate days, the fuel should be in liquid state. So its pour point should be less sufficiently.

3. Sulphur Content in the Fuel

Sulphur present in the fuel is dangerous to engine. During combustion, the sulphur in the fuel become sulfuric acid. This acid causes corrosion of engine parts. So the sulphur content in the fuel should be removed (or) sulphur content should be kept as minimum as possible.

4. Volatility

The ability to evaporate is called volatility. If the fuel evaporates in low temperature, then it has high volatility. The petrol and diesel should have adequate volatility.

5. Flash Point and Fire Point

Flash point is the minimum temperature of fuel when the fuel gives a momentary flame (or) flash. Fire point is the minimum temperature of fuel when the fuel starts continuously burning. The flash point and fire point of fuels should be adequate so that it is used in IC engine without any problem.

6. Calorific Value of Fuels:

The amount of heat liberated by burning 1 kg (or 1 m3) of fuel is known as Calorific value of fuel (or Heating value of fuel). For solid fuel, the unit for calorific value is expressed in kJ/kg. For liquid and gaseous fuel, the unit is kJ/m3 measured in S.T.P. condition (i.e., Standard Temperature and Pressure = 15° C and 760 mm of mercury). Higher Calorific Value: The amount of heat obtained by the complete combustion of 1 kg of fuel, when the products of combustion are cooled down to the temperature of the surroundings is known as Higher Calorific Value (HCV) of the fuel. Here the water vapour formed by combustion is condensed and the entire heat of steam is recovered from the products of combustion. Dulong's formula is used to determine HCV of a fuel.

$$HCV = 33800\,C + 144000\left[H_2 - \frac{O_2}{8}\right] + 9270\,S\ \frac{kJ}{kg}$$

Lower Calorific Value (LCV)

The amount of heat obtained by the combustion of 1 kg of fuel, when the product of combustion is not sufficiently cooled down to condense the steam formed during combustion is known as Lower Calorific Value (LCV) of the fuel.

So, LCV of the fuel = H.C.V – Enthalpy of evaporation of steam formed

= H.C.V. – (2466 × steam for med) kJ/kg

= H.C.V. – (2466 × 9H2)

where, 2466 kJ/kg is the specific enthalpy of evaporation of steam at 15°C.

Important characteristics of SI Engine fuel

Every engine is designed for a particular fuel according to its desired qualities. For good performance of SI engine, the fuel used must have the proper characteristics like,

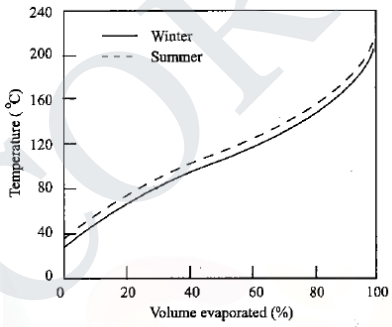
- It should readily mix with air to make an uniform mixture at inlet.
- It must be knock resistant.
- It should not pre-ignite easily.
- It should not tend to decrease the volumetric efficiency of the engine.
- Its sulphur content should be low.
- It must have adequate calorific value.
- It must have proper viscosity.

Important Qualities of engine fuel:

Gasoline which is mostly used in the present day SI engines is usually a blend of several low boiling paraffins, naphthenes and aromatics in varying proportions.

Some of the important qualities of gasoline are discussed below.

- Volatility : Volatility is one of the main characteristic properties of gasoline which determines its suitability for use in an SI engine. Since gasoline is a mixture of different hydrocarbons, volatility depends on fractional composition of the fuel.



Typical Distillation Curves of Gasoline

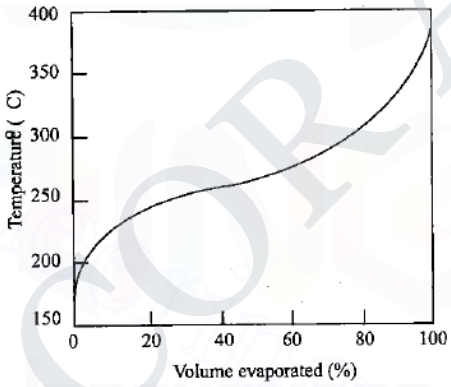
- Vapour lock characteristics: High rate of vapourization of gasoline can upset the carburetor metering or even stop the fuel flow to the engine by setting up a vapour lock in the fuel passages. This characteristic, demands the presence of relatively high boiling temperature hydrocarbons throughtout the distillation range.
- Antiknock Quality: Abnormal burning or detonation is an SI engine combustion chamber cause a very high rate of energy release, excessive temperature and pressure inside the cylinder adversely affects its thermal efficiency. The characteristics of the fuel used should be such that it resists the tendency to produce detonation and this property is called its antiknock property.
- Gum deposits: Reactive hydrocarbons and impurities in the fuel have a tendency to oxidize upon storage and form liquid and solid gummy substances.
- Sulphur content: Hydrocarbon fuels may contain free sulphur, hydrogen sulphide and other sulphur compounds which are objectionable for several reasons. The sulphur is a corrosive element of the

fuel that can corrode fuel lines, carburetors and injection pumps and it will unite with oxygen to form sulphur dioxide that in the presence of water at low temperatures, may form sulphurous acid.

- Crankcase Dilution: Liquid fuel in the cylinder causes loss of lubricating oil which deteriorates the quality of lubrication and tends to cause damage to the engine through increased friction. The liquid gasoline may also dilute the lubricating oil and weaken the oil film between rubbing surfaces.
- Starting and Warm up: A certain part of the gasoline should vapourize at the room temperature for easy starting of the engine. Hence the portion of the distillation curve between about 0 to 10% boiled off have relatively low boiling temperatures.
- Operating range performance: In order to obtain good vapourisation of the gasoline, low distillation temperature are preferable in the engine operating range.

CI Engine Fuel:

- Knock Characteristics: Knock in the CI engine occurs because of an ignition lag in the combustion of the fuel between the time of injection and the time of actual burning.
- Volatility: The fuel should be sufficiently volatile in the operating range of temperature to produce good mixing and combustion.

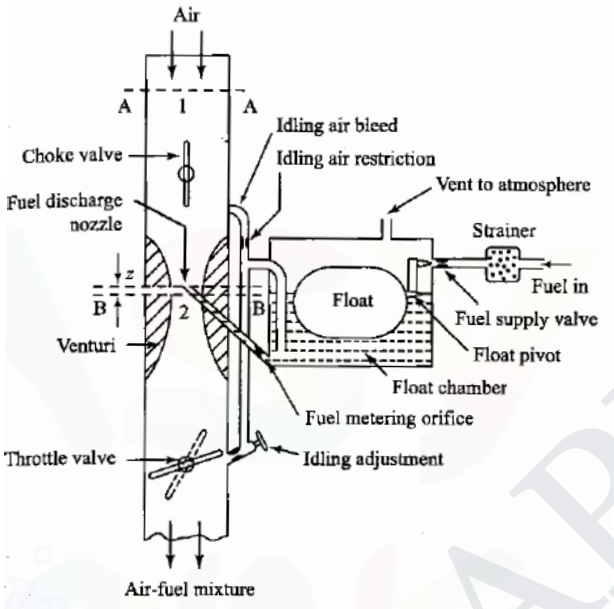


Typical Distillation Curve for Diesel

- Starting Characteristics: The fuel should help in starting the engine easily. This requirement demands high enough volatility to form a combustible mixture readily and a high cetane rating in order that the self-ignition temperature is low.
- Smoking and Odour : The fuel should not promote either smoke or odour in the engine exhaust. Generally, good volatility is the first prerequisite to ensure good mixing and therefore complete combustion.
- Viscosity: Ci engine fuels should be able to flow through the fuel system and the strainers under the lowest operating temperature to which the engine is subjected to.
- Corrosion and Wear : The fuel should not cause corrosion and wear of the engine components before or after combustion. These requirements are directly related to the presence of sulphur, Ash and residue in the fuel.
- Handling Ease: The fuel shoul be a liquid that will readily flow under all conditions that are encountered in actual use. This requirement is measured by the pour point and the viscosity of the fuel. The fuel should also have a high flash point and a high fire point.

13. Air-Fuel ratio calculation.

A simple carburetor with the tip of the fuel nozzle h meters above the fuel level in the float chamber. It may be noted that the density of air is not the same at the inlet to the carburetor (section A-A, Point 1) and the venture throat (Section B-B, point 2). The calculation of exact air mass flow involves taking this change in density or compressibility of air into account.



Applying the steady flow energy equation to section A-A and section B-B and assuming unit mass flow of air.

$$q - w = (h_2 - h_1) + \frac{1}{2}(c_2^2 - c_1^2)$$

Here q, w are the heat and work transfer from entrance to throat and C stand for enthalpy and velocity respectively.

Assuming an adiabatic flow, we get q=0, w=0 and intial velocity is negligible (C_1=0)

$$C_2 = \sqrt{2(h_1 - h_2)}$$

Assuming air to behave like ideal gas, we get $h = c_p T$, Hence the above equation can be written as

$$C_2 = \sqrt{2c_p(T_1 - T_2)}$$

As the flow process from inlet to the venturi throat can be considered to be isentropic, we have

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

From above equation:

$$T_1 - T_2 = T_1 - T_1\left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \Rightarrow T_1 - T_2 = T_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]$$

Substituting the above equation:

$$C_2 = \sqrt{2c_p(T_1 - T_2)} \Rightarrow \mathbf{C_2 = \sqrt{2c_pT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]}}$$

Now mass flow of air,

$$\dot{m}_a = \rho_1 A_1 C_1 = \rho_2 A_2 C_2$$

Where A_1 and A_2 are the cross-sectional area at the air inlet (point 1) and venture throat (point 2).

To calculate the mass flow rate of air at venture throat, we have

$$\frac{p_1}{\rho_1{}^{\gamma}} = \frac{p_2}{\rho_2{}^{\gamma}} \Rightarrow \rho_2 = \rho_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}$$

$$\dot{m}_a = \rho_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} A_2\sqrt{2c_pT_1\left[1-\left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]} \qquad \therefore \rho_1 = \frac{p_1}{RT_1}$$

$$\dot{m}_a = A_2\frac{p_1}{RT_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}\sqrt{2c_pT_1\left[1-\left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]}$$

$$\dot{m}_a = \frac{A_2p_1}{R\sqrt{T_1}}\sqrt{2c_p\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}}-\left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma+1}{\gamma}\right)}\right]}$$

Air fuel ratio:

$$\frac{A}{F}\text{ ratio} = \frac{\dot{m}_{a(actual)}}{\dot{m}_{f(actual)}} \Rightarrow \frac{A}{F} = 0.1562\frac{C_{da}}{C_{fa}}\frac{A_2}{A_F}\frac{p_1\emptyset}{\sqrt{2T_1\rho_f(p_1-p_2-gz\rho_f)}}$$

Air – Fuel ratio neglecting compressibility of air:

When air is considered as incompressible, Bernoulli's theorem is applicable to air flow

$$\frac{p_1}{\rho}+\frac{c_1^2}{2}+Z_1g = \frac{p_2}{\rho}+\frac{c_2^2}{2}+gZ_2$$

Where, $Z_1 = Z_2$, $C_1 = 0$, then

$$\frac{p_1}{\rho_a}-\frac{p_2}{\rho_a} = \frac{c_2^2}{2} \Rightarrow c_2^2 = 2\left[\frac{p_1}{\rho_a}-\frac{p_2}{\rho_a}\right] \Rightarrow C_2 = \sqrt{2\left[\frac{p_1}{\rho_a}-\frac{p_2}{\rho_a}\right]} \Rightarrow C_2 = \sqrt{2\left[\frac{p_1-p_2}{\rho_a}\right]}$$

Now mass flow of air,

$$\dot{m}_a = \rho_aA_2C_2 \Rightarrow \dot{m}_a = \rho_aA_2\sqrt{2\left[\frac{p_1-p_2}{\rho_a}\right]} \Rightarrow \dot{m}_a = C_{da}A_2\sqrt{2\,\rho_a(p_1-p_2)}$$

Air fuel ratio:

$$\frac{A}{F}\text{ ratio} = \frac{\dot{m}_a}{\dot{m}_f} \Rightarrow \frac{A}{F} = \frac{C_{da}}{C_{fa}}\frac{A_2}{A_F}\sqrt{\frac{\rho_a(p_1-p_2)}{\rho_f(p_1-p_2-gz\rho_f)}}$$

If Z=0,

$$\frac{\dot{m}_a}{\dot{m}_f} = \frac{C_{da}}{C_{fa}}\frac{A_2}{A_F}\sqrt{\frac{\rho_a}{\rho_f}}$$

1. It is clear from expression for $\dot{m}_f$ that if $(p_1 - p_2)$ is less than $gz\rho_f$ there is no fuel flow and this can happen at very low air flow.
2. The air flow increases, $(p_1 - p_2)$ increases and when $(p_1 - p_2) > gz\rho_f$, the fuel flow begins and increases with increase in the differential pressure.
3. At high air flow where $(p_1 - p_2)$ is large compared to $gz\rho_f$ the fraction $gz\rho_f/(p_1 - p_2)$ become negligible and the air-fuel ratio approaches

$$\frac{C_{da}}{C_{fa}}\frac{A_2}{A_F}\sqrt{\frac{\rho_a}{\rho_f}}$$

4. A decrease in the density of air reduces the value of air-fuel ratio (i.e., mixture become richer). It happens at (a) high air flow rates where $(p_1 - p_2)$ becomes large and ρ_2 decreases. (b) high altitudes where the density of air is low.

14. Combustion of SI engines.

COMBUSTION IN SI ENGINES

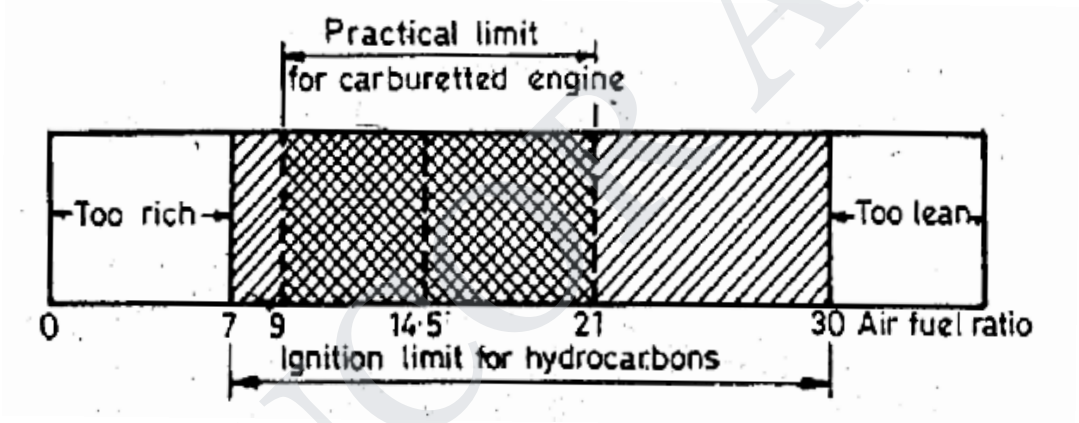
Combustion may be defined as a relatively rapid chemical combination of hydrogen and carbon in fuel with oxygen in air resulting in liberation of energy in the form of heat.

1. The presence of combustible mixture
2. Some means to initiate mixture
3. Stabilization and propagation of flame in Combustion Chamber

In S I Engines, carburetor supplies a combustible mixture of petrol and air and spark plug initiates combustion.

Ignition Limits :

- Ignition of charge is only possible within certain limits of fuel-air ratio.
- Ignition limits correspond approximately to those mixture ratios, at lean and rich ends of scale, where heat released by spark is no longer sufficient to initiate combustion in neighboring unburnt mixture.



- The ignition limits are wider at increased temperatures because of higher rates of reaction and higher thermal diffusivity coefficients of the mixture.
- The lower and upper limits of ignition of the mixture depend upon the temperature and mixture ratio.
- For hydrocarbons fuel the stoichiometric fuel air ratio is 1:15 and hence the fuel air ratio must be about 1:30 and 1:7.

Combustion in SI engine may roughly is divided into two general types:

Normal and Abnormal (knock free or knocking).

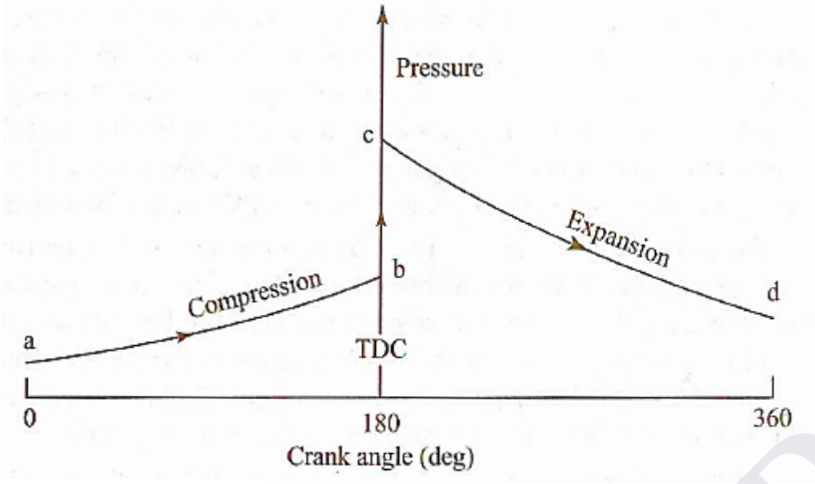
Theoretical diagram of pressure crank angle diagram is shown in figure below.

(a ⟶ b) is compression process,

(b ⟶ c) is combustion process

(c ⟶ d) is an expansion process.

In an ideal cycle it can be seen from the diagram, the entire pressure rise during combustion takes place at constant volume i.e., at TDC. However, in actual cycle this does not happen.

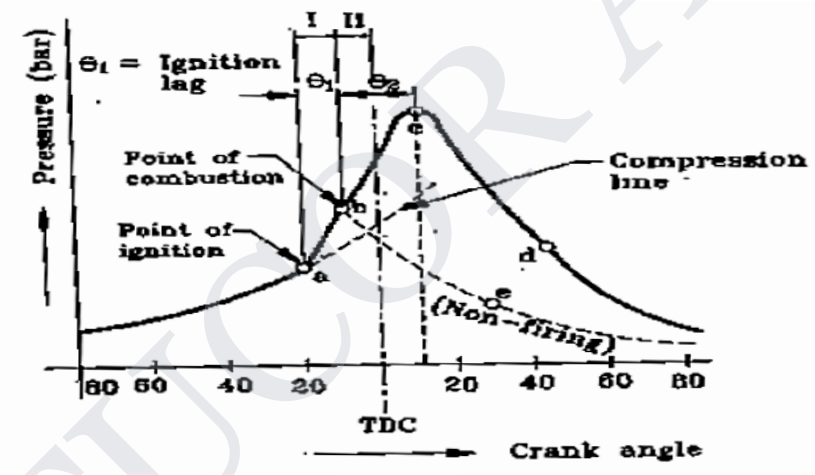


The actual combustion process of SI engines can be divided into three broad regions:

(1) Ignition and flame development,

(2) Flame propagation

(3) Flame termination.



1. Ignition lag stage:

- There is a certain time interval between instant of spark and instant where there is a noticeable rise in pressure due to combustion. This time lag is called IGNITION LAG.
- Ignition lag is the time interval in the process of chemical reaction during which molecules get heated up to self ignition temperature , get ignited and produce a self propagating nucleus of flame.
- The ignition lag is generally expressed in terms of crank angle θ_1. The period of ignition lag is shown by path (a-b).

2. Flame propagation stage:

- Once the flame is formed at "b", it should be self sustained and must be able to propagate through the mixture.
- This is possible when the rate of heat generation by burning is greater than heat lost by flame to surrounding.

- After the point "b", the flame propagation is abnormally low at the beginning as heat lost is more than heat generated.
- Therefore pressure rise is also slow as mass of mixture burned is small. Therefore, it is necessary to provide angle of advance (30-35) degrees, if the peak pressure to be attained (5-10) degrees after TDC.
- The time required for crank to rotate through an angle θ_2 is known as combustion period during which propagation of flame takes place.

3. After burning:

Combustion will not stop at point "c" but continue after attaining peak pressure and this combustion is known as after burning. This generally happens when the rich mixture is supplied to engine.

The factors that affect the flame propagations are:

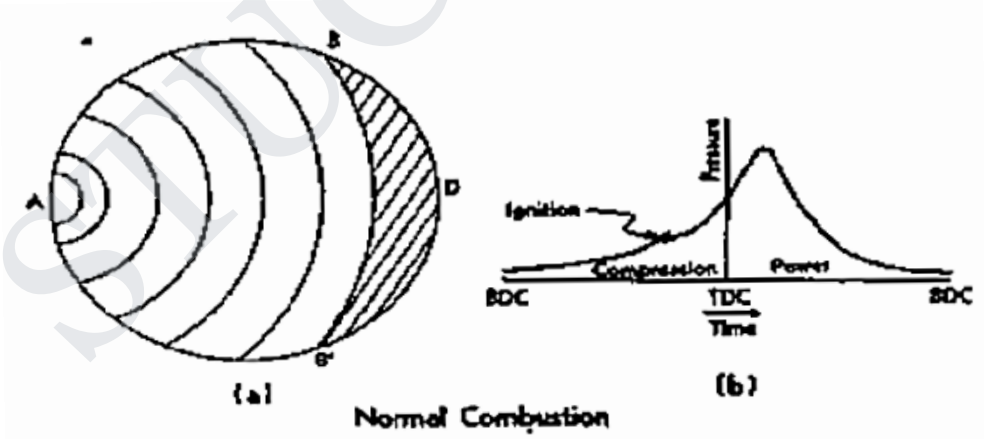
1. Air fuel ratio
2. Compression ratio
3. Load on engine
4. Turbulence and engine speed
5. Other factors

Types of combustion:

1. Normal Combustion.
2. Abnormal Combustion.

Normal Combustion:

Normal combustion rarely occurs in a real engine without some trace of autoignition appearing. After ignition, the flame front travels across the combustionchamber.



- The gas a heat of the flame front called the"end gas ".
- The end gas receives heat due to compression by expanding gases and by radiation from the advancing flame front, therefore, its temperature and density increases.
- If the temperature exceeds the self – ignition temperature and the un-burnt gas remains at or above this temperature for a period of time equal to/or greater the delay period, spontaneous ignition(or auto ignition) will occurs at variouslocations. Shortly after words an audible sound called **knock** appears.
- If the end gas does not reach its self-ignition temperature, the combustion will be normal.

Abnormal Combustion:

In Internal combustion engines, abnormal combustion is a significant phenomenon associated with the combustion processes on which the life and performance of the engine depends.

The two important abnormal combustion phenomenons are

1. KNOCK
2. SURFACE IGNITION.

These abnormal combustion phenomenons are of concern because

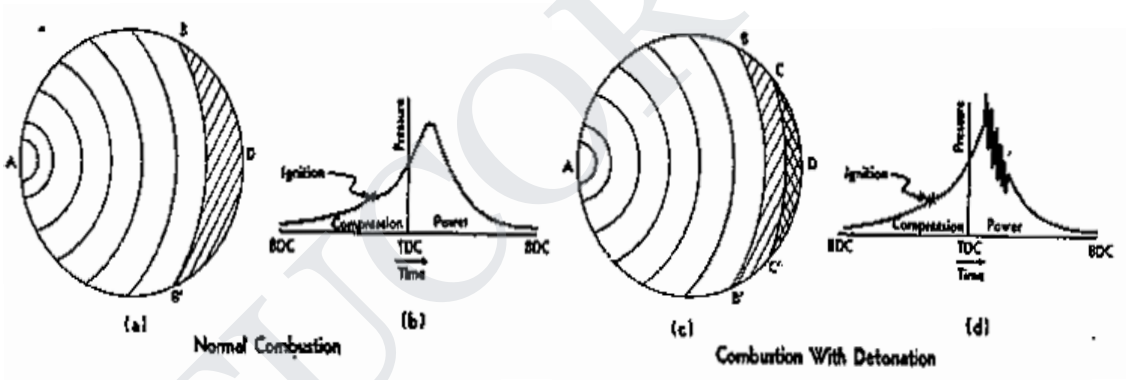
(1) when severe, they can cause major engine damage; and

(2) even if not severe, they are regarded as an objectionable source of noise by the engine.

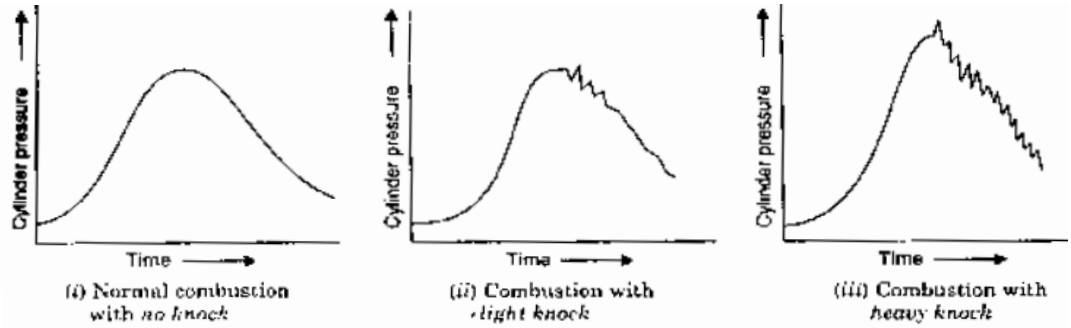
15. Phenomena of knocking in SI

PHENOMENON OF KNOCKING IN SI ENGINE

- Knocking is due to auto ignition of end portion of unburned charge in combustion chamber.
- As the normal flame proceeds across the chamber, pressure and temperature of unburned charge increase due to compression by burned portion of charge.
- This unburned compressed charge may auto ignite under certain temperature condition and release the energy at a very rapid rate compared to normal combustion process in cylinder.

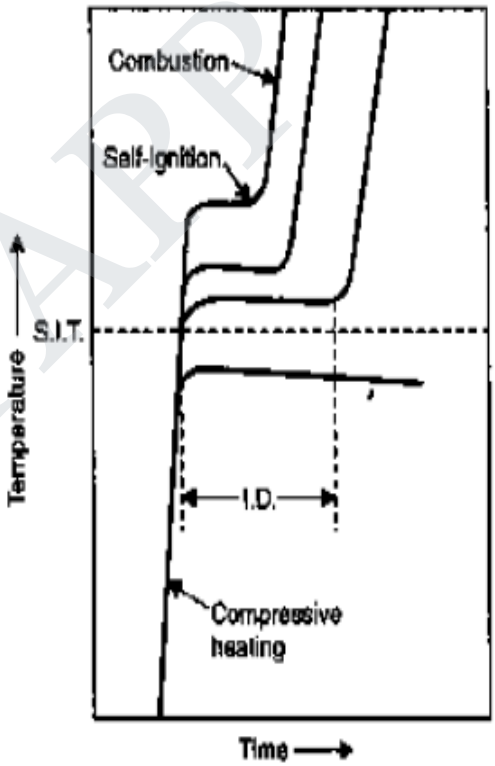


- This rapid release of energy during auto ignition causes a high pressure differential in combustion chamber and a high pressure wave is released from auto ignition region.
- The motion of high pressure compression waves inside the cylinder causes vibration of engine parts and pinging noise and it is known as knocking or detonation.
- This pressure frequency or vibration frequency in SI engine can be up to 5000 Cycles per second.



AUTO IGINITION:

- A mixture of fuel and air can react spontaneously and produce heat by chemical reaction in the absence of flame to initiate the combustion or self-ignition.
- This type of self-ignition in the absence of flame is known as Auto-Ignition.
- The temperature at which the self-ignition takes place is known as self-igniting temperature.
- The pressure and temperature abruptly increase due to auto-ignition because of sudden release of chemical energy.
- This auto-ignition leads to abnormal combustion known as detonation.
- In addition to this knocking puts a limit on the compression ratio at which an engine can be operated which directly affects the engine efficiency and output.
- Auto-ignition of the mixture does not occur instantaneously as soon as its temperature rises above the self-ignition temperature.
- Auto-ignition occurs only when the mixture stays at a temperature equal to or higher than the self-ignition temperature for a "finite time".
- This time is known as delay period or reaction time for auto-ignition.
- This delay time as a function of compression ratio is shown in adjacent figure. As the compression ratio increases, the delay period decreases and this is because of increase in initial (before combustion) pressure and temperature of the charge.
- The self-ignition temperature is a characteristic of fuel air mixture and it varies from fuel to fuel and mixture strength to mixture - strength of the same fuel.



S.I.T. → Self-ignition temperature
I.D. → Ignition delay

PRE –IGINITION:

- Pre-ignition is the ignition of the homogeneous mixture of charge as it comes in contact with hot surfaces, in the absence of spark .
- Auto ignition may overheat the spark plug and exhaust valve and it remains so hot that its temperature is sufficient to ignite the charge in next cycle during the compression stroke before spark occurs and this causes the pre-ignition of the charge.
- pre-ignition is also caused by persistent detonating pressure shockwaves scoring away the stagnant gases which normally protect the combustion chamber walls.
- The resulting increased heat flow through the walls, raises the surface temperature of any protruding poorly cooled part of the chamber, and this there fore provides a focal point for preignition.

Effects of Pre-ignition

- It increase the tendency of denotation in the engine
- It increases heat transfer to cylinder walls because high temperature gas remains in contact with for a longer time
- Pre-ignition in a single cylinder will reduce the speed and power output
- Pre-ignition may cause seizer in the multi-cylinder engines, only if only cylinders have pre-ignition

FACTORS INFLUENCE THE KNOCKING

I. Temperature Factor

Increasing the temperature of the unburned mixture increase the possibility of knock in the SI engine.

- Raising the compression ratio
- Supercharging
- Coolant temperature
- Temperature of the cylinder and combustion chamber walls

II. Density Factor

- Compression ratio (CR)
- Mass of inducted charge
- Inlet temperature of mixture
- Retarding spark timing

III. Time Factor

Time factor is the increasing the flame speed or the ignition lag will tend to reduce the tendency to knock.

- Turbulence
- Engine size
- Engine speed
- Spark plug locations

IV. Composition Factors

- Fuel-air ratio
- Octane value
- Molecular Structure
  - Paraffins
  - Olefins
  - Napthenes and Aromatics

16. Combustion of CI engines.

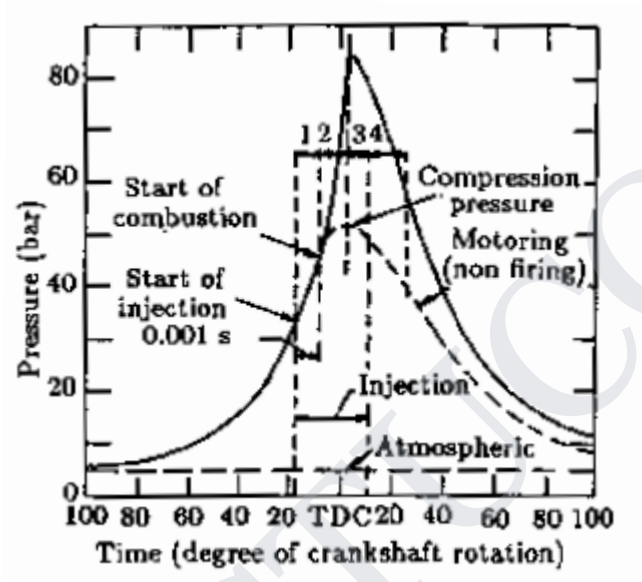
COMBUSTION IN CI ENGINES

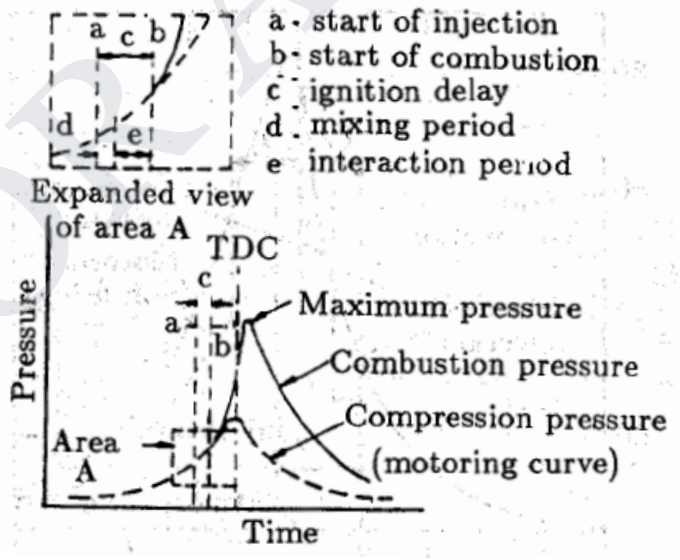
- In CI engine A: F mixture is not homogeneous and fuel remains in liquid particles, therefore quantity of air supplied is 50% to 70% more than stiochiometric mixture.
- The combustion in CI engine, the combustion takes place at number of points simultaneously and number of flames generated are also many.
- To burn the liquid fuel is more difficult as it is to be evaporated; it is to be elevated to ignition temperature and then burn.

Stages of combustion in CI engine

The combustion in CI engine is considered to be taking place in four phases:

1. Ignition Delay period /Pre-flame combustion
2. Uncontrolled combustion
3. Controlled combustion
4. After burning





1. Ignition Delay period /Pre-flame combustion

- The fuel does not ignite immediately upon injection into the combustion chamber.
- There is a definite period of inactivity between the time of injection and the actual burning this period is known as the ignition delay period.
- In Figure, the delay period is shown on pressure crank angle (or time) diagram between points a and b. Point "a" represents the time of injection and point "b" represents the time of combustion.
- The ignition delay period can be divided into two parts, the physical delay and the chemical delay.
- The delay period in the CI engine exerts a very great influence on both engine design and performance.

2. Period of Rapid Combustion / uncontrolled combustion

- The period of rapid combustion also called the uncontrolled combustion, is that phase in which the pressure rise is rapid.
- During the delay period, a considerable amount of fuel is accumulated in combustion chamber, these accumulated fuel droplets burns very rapidly causing a steep rise in pressure.
- The period of rapid combustion is counted from end of delay period or the beginning of the combustion to the point of maximum pressure on the indicator diagram.
- The rate of heat-release is maximum during this period. This is also known as uncontrolled combustion phase, because it is difficult to control the amount of burning / injection during the process of burning.

3. Period of Controlled Combustion

- The rapid combustion period is followed by the third stage, the controlled combustion.
- The temperature and pressure in the second stage are so high that fuel droplets injected burn almost as they enter and find the necessary oxygen and any further pressure rise can be controlled by injection rate.
- The period of controlled combustion is assumed to end at maximum cycle temperature.

4. Period of After-Burning

- Combustion does not stop with the completion of the injection process.
- The unburnt and partially burnt fuel particles left in the combustion chamber start burning as soon as they come into contact with the oxygen.
- This process continues for a certain duration called the after-burning period.
- This burning may continue in expansion stroke up to 70 to 80% of crank travel from TDC.

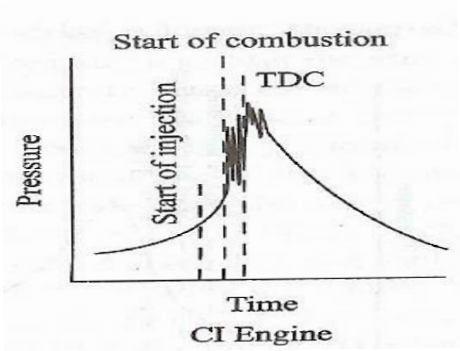
Ignition Delay or Ignition Lag

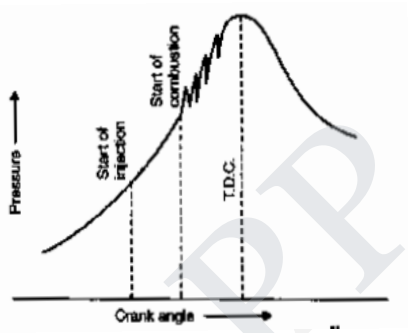
- The delay period is the time between the start of injection and start of combustion. The delay period extends for about 13 deg movement of crank.
- This delay time decreases with increase in speed. If there is no delay, the fuel would burn at injector and there would be oxygen deficiency around the injector, which results in incomplete combustion.
- If the delay period is too long, amount of fuel availability for simultaneous explosion , is too great , which results in rapid pressure rise.
- The delay period should be as short as possible since long delay period gives more rapid rise in pressure and thus causes knocking.

17. Phenomena of knocking in CI Engines.

PHENOMENON OF KNOCKING IN CI ENGINE

- Knock in compression ignition engine is related to delay period.
- When the delay period is longer, accumulation of fuel droplets causes rapid pressure rise due to ignition.
- It results jamming of forces against the piston that leads to rough engine operation.
- Knocking in CI engine occurs near the beginning of combustion.





- In C.I. engines the injection process takes place over a definite interval of time. Consequently, as the first few droplets injected are passing through the ignition lag period, additional droplets are being injected into the chamber.
- If the ignition delay is longer, the actual burning of the first few droplets is delayed and a greater quantity of fuel droplets gets accumulated in the chamber.
- When the actual burning commences, the additional fuel can cause too rapid a rate of pressure rise, as shown on pressure crank angle diagram above, resulting in Jamming of forces against the piston (as if struck by a hammer) and rough engine operation.
- If the ignition delay is quite long, so much fuel can accumulate that the rate of pressure rise is almost instantaneous.
- Such, a situation produces extreme pressure differentials and violent gas vibration known as knocking (diesel knock), and is evidenced by audible knock.
- The phenomenon is similar to that in the SI engine. However, in SI Engine knocking occurs near the end of combustion whereas in CI engine, knocking the occurs near the beginning of combustion.
- Delay period is directly related to Knocking in CI engine.

An extensive delay period can be due to following factors:

- A low compression ratio permitting only a marginal self ignition temperature to be reached.
- A low combustion pressure due to worn out piston, rings and bad valves
- Low cetane number of fuel
- Poorly atomized fuel spray preventing early combustion
- Coarse droplet formation due to malfunctioning of injector parts like spring
- Low intake temperature and pressure of air

EFFECT OF VARIOUS FACTORS ON DELAY PERIOD IN CI ENGINE

Many design and operating factors affect the delay period. The important ones are:

- Compression ratio
- Engine speed
- Output
- Injection timing
- Quality of the fuel
- Intake temperature
- Intake pressur

METHODS OF CONTROLING DIESEL KNOCK

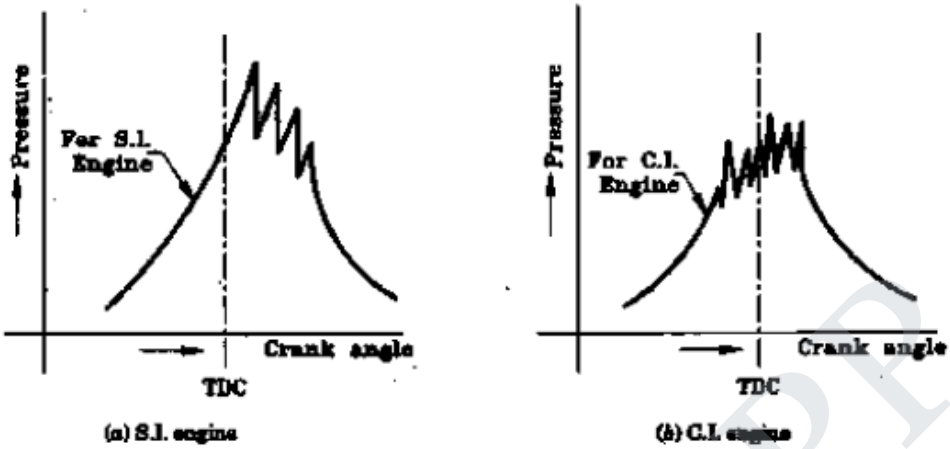
- Using a better fuel. Higher CN fuel has lower delay period and reduces knocking tendency.
- Controlling the Rate of Fuel Supply
- Knock reducing fuel injector
- By using Ignition accelerators : C N number can be increased by adding chemical called dopes.
- The two chemical dopes are used are ethyl-nitrate and amyle –nitrate.
- Increasing Swirl :

18. Combustion phenomenon in CI engine V/s combustion in SI engine.

SL NO	COMUSTION IN SI ENGINE	COMBUSTION IN CI ENGINE
1.	Homogeneous mixture of petrol vapour and air is compressed (CR 6:1 to 11:1) at the end of compression stroke and is ignited at one place by spark plug.	Air alone is compressed through large Compression ratio (12:1 to 22:1) and fuel is injected at high pressure of 110 to 200 bar using fuel injector pump.
2.	Single definite flame front progresses through air fuel mixture and entire mixture will be in combustible range	Fuel is not injected at once, but spread over a period of time. Initial droplets meet air whose temperature is above self ignition temperature and ignite after ignition delay.
3.	In SI engine physical delay is almost zero and chemical delay controls combustion	In CI engine physical delay controls combustion.
4.	In SI Engine ignition occurs at one point with a slow rise in pressure	In the CI engine, the ignition occurs at many points simultaneously with consequent rapid rise in pressure. There is no definite flame front.
5.	In SI engine , A/F ratio remains close to stoichiometric value from no load to full load	In CI engine , irrespective of load, at any speed, an approximately constant supply of air enters the cylinder. With change in load, quantity of fuel is changed to vary A/F ratio. The overall A/F can Range from 18:1 to 80:1.
6.	Delay period must be as long as possible. High octane fuel(low cetane) is required.	Delay period must be as short as possible. High cetane (low octane) fuel is required

19. COMPARISON OF KNOCK IN SI AND CI ENGINES

- It may be interesting to note that knocking in spark-ignition engines and compression ignition engines is fundamentally due to the auto ignition of the fuel-air mixture.
- In both the cases, the knocking depends on the auto ignition lag of the fuel-air mixture. But careful examination of knocking phenomenon in SI and CI engines reveals the following differences:



spark ignition engines	compression engines
Auto ignition of end gas away from the spark plug, most likely near the end of combustion causes knocking	The auto ignition of charge causing knocking is at the start of combustion
In order to avoid knocking, it is necessary to prevent auto ignition of the end gas to take place at all.	the earliest auto –ignition is necessary to avoid knocking
The knocking takes place in homogeneous mixture, therefore , the rate of pressure rise and maximum pressure is considerably high.	the mixture is not homogenous and hence the rate of pressure is lower than in SI engine.
Pre-ignition occurs	Only air is compressed, therefore there is no question of Pre-ignition in CI engines
It is lot more easy to distinguish between knocking and non-knocking condition in SI engines	no definite distinction between normal and knocking combustion
SI fuels should have long delay period to avoid knocking	CI fuels should have short delay period to avoid knocking.

The following table gives a comparative statement of various characteristics that reduce knocking in SI and CI engines

S. No	Factors affecting knock	SI engine	CI engine
1	Self ignition temperature	High	Low
2	Delay period of fuel	Long	Short
3	Compression ratio	Low	High
4	Inlet temperature	Low	High
5	Inlet pressure	Low	High
6	Speed	High	Low
7	Cylinder size	Small	Large
8	Combustion chamber wall temperature	Low	High

20. Engine emission, control and norms.

The emission exhausted into the surrounding pollute the atmosphere and cause the following problems.

i. Global warming
ii. Acid rain
iii. Smog
iv. Odours
v. Respiratory and other health hazards

Engine Emissions:

Engine emissions can be classified into two categories:

i. Exhaust emissions
ii. Non-exhaust emissions

Exhaust emissions

The emissions of concern are

i. Unburnt hydrocarbons (HC)
ii. Oxides of carbon (CO_X)
iii. Oxides of nitrogen (NO_X)
iv. Oxides of sulphur (SO_X)
v. Solid carbon particulates
vi. Soot and somke

1. Carbon monoxide (CO):

- It Is a colourless gas of about the same density as aIr.
- It is a poisonous gas which, when inhaled, replaces the oxygen in the blood stream so that the body's metabolism can not function correcfly.
- Small amounts of co concentrations, when breathed in, srow down physical and mental activity and produces headaches, while large concentration will kill.

Mechanism of formation of CO:

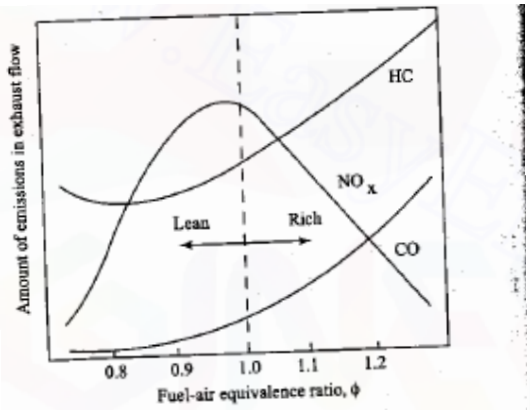
CO is generally formed when the mixture is rich in fuel. The amount of CO formed increases as the mixture becomes more and more rich in fuel. A small amount of CO will come out of the exhaust even when the mixture is slightly lean in fuel. This is due to the fact that equilibrium is not established when the products pass to the exhaust. At the high tempearature developed during the combustion, the products formed are unstable, and the following reaction take place before the equilibrium is established.

$$2H_2O + O_2 \rightarrow 2(1-y)H_2O + 2yH_2 + yO_2$$

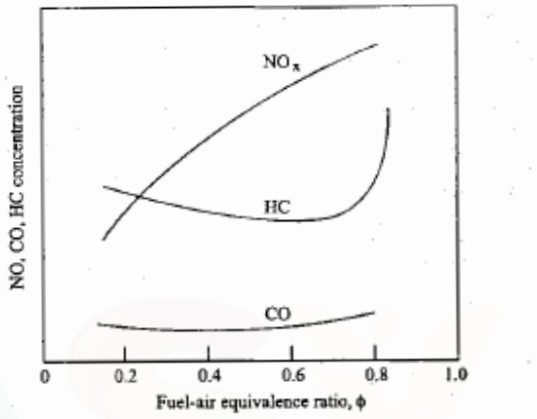
Where, y is the fraction of H_2O dissociated.

$$C + O_2 \rightarrow CO_2 \rightarrow (1-x)CO_2 + xCO_2 + \frac{x}{2}O_2$$

As the products cool down to exhaust tempearature, major part of CO reacts with oxygen to form CO_2, However a relatively small amount of CO will remain in exhaust, its concentration increasing with rich mixtures.



Emissions as a Function of Equivalence Ratio for a SI Engine



Emissions as a Function of Equivalence Ratio for a CI Engine

2. Hydrocarbons (HC) :

Hydrocarbons, derived from unburnt fuel emitted by exhausts, engine crankcase fumes and vapour escaping from the carburettor are also harmful to health,

Mechanism of fortnation of HC

- Due to existence of local very rich mixture pockets at much lower tgmperatures than the combustion chambers, unburnt hydrocarbons may appear in the exhaust.
- The hydrocarbons also appear due to flame quenching near the metallic walls.

3. Oxides of nitrogen (NOx):

Oxides of nitrogen and other obnoxious substances are produced in very small quantities and, in certain environments, can cause pollution, while prolonged exposure is dangerous to health.

Mechonism of formation of nitric oxide (NO)

At high combustion temperatures, the following chemical reactions take place behind the flame :

$$N_2 + O_2 \rightarrow 2NO$$

$$\mathrm{N_2 + 2H_2O \rightarrow 2NO + 2H_2}$$

Chemical equilibriurn calculations show that a signifrcant amount of NO will be formed at the end of combustion. The majority of NO forrned will however decompose at the low temperatures of exhaust. But due to very low reaction rate at the exhaust temperature a part of NO formed remains in exhaust. It is far in excess of the equilibrium composition at that ternperature as the formation of NO freezes at low exhaust temperatures.

Smoke or particulate:

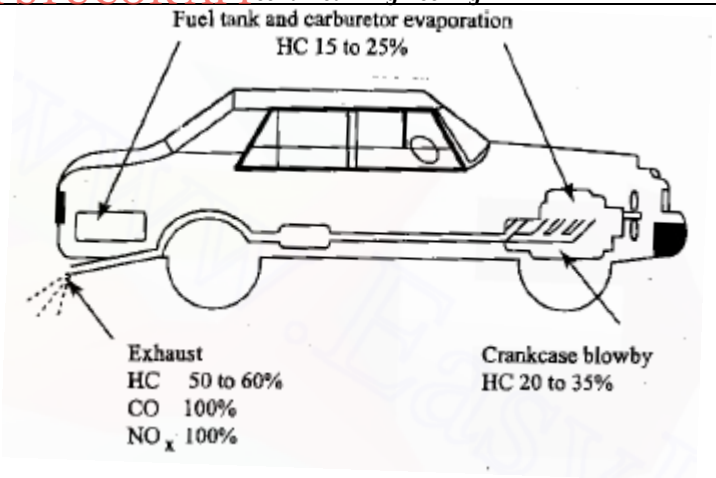
Solid particles are usually formed by dehydrogenation, polymerisation and, agglonterotion

In the combustion process of different hydrocarbons, acetylene (C_2H_2) is forrnerl as an intermediate product. These acetylene molccules after simultaneous polymerisation and dehydration produce carbon particles, which are the main constituent of the particulate.

Non-exhaust emissions

Apart from exhaust emissions there are three other sources in an automobile which emit emissions.

i. Fuel tank: The fuel tank emits fuel vapours into the atmosphere.
ii. Carburetor : The carburetor also gives out fuel vapours.
iii. Crankcase : It emits blow-by gases and fuel vapours into the atmosphere.



Emission Control:

The main methods, among various methods for SI engine emission control are :

1. Modification in the engine design aid operating parameters
2. Treatment of exhaust products of combustion
3. Modification of the fuels

Modification in the Engine Design and Operatlng Paraneters:

1. Combustion chamber configuration

- Modification of combustion chamber involves avoiding flame quenching zones where combustion might otherwise be incomplete and resulting in high HC emission This includes
- Reduced surface to volume (S/V) ratio
- Reduced squish area
- Reduced space around Piston ring
- Reduced distance of the top piston ring from the top of the piston,

2. Lower compression ratio

- A lower compression ratio reduces the quenching effect by reducing the quenching area thus reducing HC
- Lower compression ratio also reduces NO remissions due to lower maximum temperature
- Lower compression, however reduces thermal efficiency and increases fuel consumption

3. Modified induction system:

- In a multi-cylinder engine it is always difficult to supply designed A/F ratio under all conditions of load and power.
- This can be achieved by proper design of induction system or using high velocity or multi-choke carburetors.

4. Ignition timing:

- The ignition timing control is so adjusted as to provide normal required spark advance during cruising and retard the same for idle running.
- NO_x emissions are reduced due to lowering of maximum combustion temperatures.
- Also HC emissions gets reduced due to high exhaust temperatures.

5. Reduced valve overlap

- Increased overlap allows some fresh charge to escape directly and increase emission level. This can be controlled by reducing valve overlap.
- A new variable valve timing(VVT) allows for controlled scheduling of valve timing events, improves engine performance.

Treatment of exhaust products of combustion

After burner:

- An after burner is a burner where air is supplied to the exhaust gases and mixture is burnt with the help of ignition system.
- The HC and CO which are formed in the engine combustion because of inadequate O_2 and inadequate time to burn are further burnt by providing air in a separate box known as after burner.
- The after-burner is located very near to the exhaust manifold with an intension that there is no fall in the exhaust temperature.

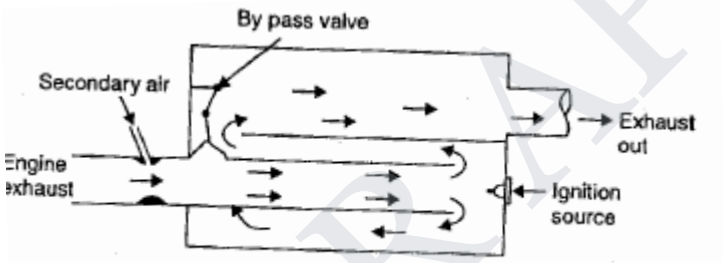
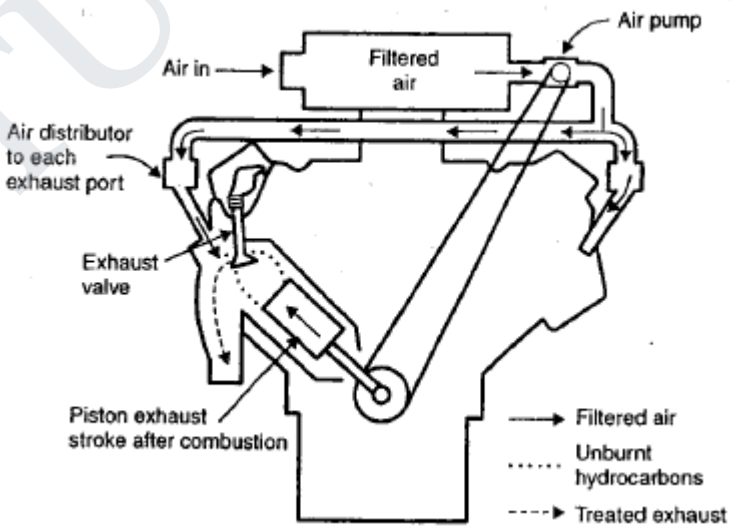


Fig. . Direct flame after-burner.

Exhaust manifold reactor:

- The design is changed so as to minimize the heat loss and to provide sufficient time for mixing of exhaust and secondary air.



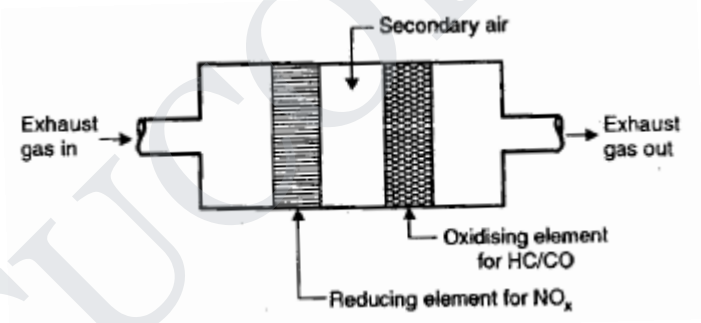
- Here a positive displacement vane pump which driven by the engine.
- Inducts air from the air cleaner or from separate air filter.
- The air passes into an internal or external distributing manifold with tubes feeding a metered amount into the exhaust port of each cylinder and close to the exhaust valve.

- Since the exhaust gases are at high temperature, the injected air reacts with HC, CO and aldehydes to reduce greatly the concentration of such emission.
- The injected air is closely metered otherwise it can decrease the temperature of the exhaust gas
- In earlier type of reactor developed by Du Pont the entry of exhaust gases was radial and the air flow peripheral.

Catalytic converter :

- A catalytic converter is a device which is placed in the vehicle exhaust system to reduce HC and CO by oxidising catalyst ond NO by reducing catalyst.
- The basic requirernents of a catalytic converter are
  - (i) High surface arca of the catalyst for better reactions.
  - (ii) Good chemical stability to prevent any deterioration in performance.
  - (iii) Low volume heat capacity to reach the operating temperatures.
  - (iv) Physical durability with attrition resistence.
  - (v) Minimum pressure drop during the flow oferhaust gases through the catalyst bed this will not increase back pressure of the engine.

Fig. shows a catalytic converter, developed by the Ford company. It consists of two separate elements, one for NO_x and the other for HC/CO emissions. The secondary air is injected ahead of the first element. The flow in the converter is axial.



Oxidatian catalytic reaction:

CO, HC and O_2 from air are catalytically converted to CO_2, and H_2O and number of catalysts are known to be effective noble metals like platinum and plutonium, copper, vanadium, iron, cobalt, nickel, chromium ets.

Reduction catalytic reaction:

The primary concept is to offer the NO molecule an activation site, say nickel or copper grids in the presence of CO but not O_2, which will cause oxidation, to from N_2, and CO_2. The NO may react with a metal molecule to form an oxide which then in turn, may react with CO to restore the metal molecule. Rhodium is best catalyst to control NO_x but A/F ratio must be within a narrow range of 14.6 : 1 to I4.7 : 1.

Major drawbacks of catalytic converter are as under :

- Owing to the exothermic reactions in the catalyst bed the exhaust systems are hotter than normal.
- Cars equipped with such converter should not use leaded fuel as lead, destroys complete catalytic activity.
- If the fuel contains surphur (as dieser oil) emission of SO_3 is increased.

Progress of emission standards for 2-and 3-wheelers: Indian Emission Standards (2 and 3 wheelers)

Standard	Reference	Date
Bharat Stage II	Euro 2	1 April 2005
Bharat Stage III	Euro 3	1 April 2010
Bharat Stage IV	Euro 4	1 April 2017
Bharat Stage VI	Euro 6	April 2020 with mandate (proposed)

In order to comply with the BSIV norms, 2- and 3-wheeler manufacturers will have to fit an evaporative emission control unit, which should lower the amount of fuel that is evaporated when the motorcycle is parked

Indian Emission Standards (4-Wheel Vehicles)

Standard	Reference	YEAR	Region
India 2000	Euro 1	2000	Nationwide
Bharat Stage II	Euro 2	2001	NCR*, Mumbai, Kolkata, Chennai
		2003	NCR*, 13 Cities
		2005	Nationwide
Bharat Stage III	Euro 3	2005	NCR*, 13 Cities
		2010	Nationwide
Bharat Stage IV	Euro 4	2010	NCR*, 13 Cities
		2017	Nationwide
Bharat Stage V	Euro 5	(to be skipped)	
Bharat Stage VI	Euro 6	2018	Delhi NCR
		2020	Nationwide
* National Capital Region (Delhi) † Mumbai, Kolkata, Chennai, Bengaluru, Hyderabad, Ahmedabad, Pune, Surat, Kanpur, Lucknow, Sholapur, Jamshedpur and Agra			

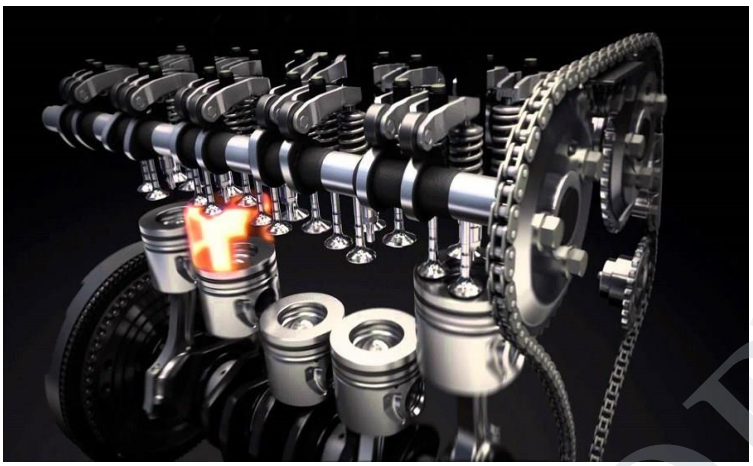
The above standards apply to all new 4-wheel vehicles sold and registered in the respective regions. In addition, the National Auto Fuel Policy introduces certain emission requirements for interstate buses with routes originating or terminating in Delhi or the other 10 cities.

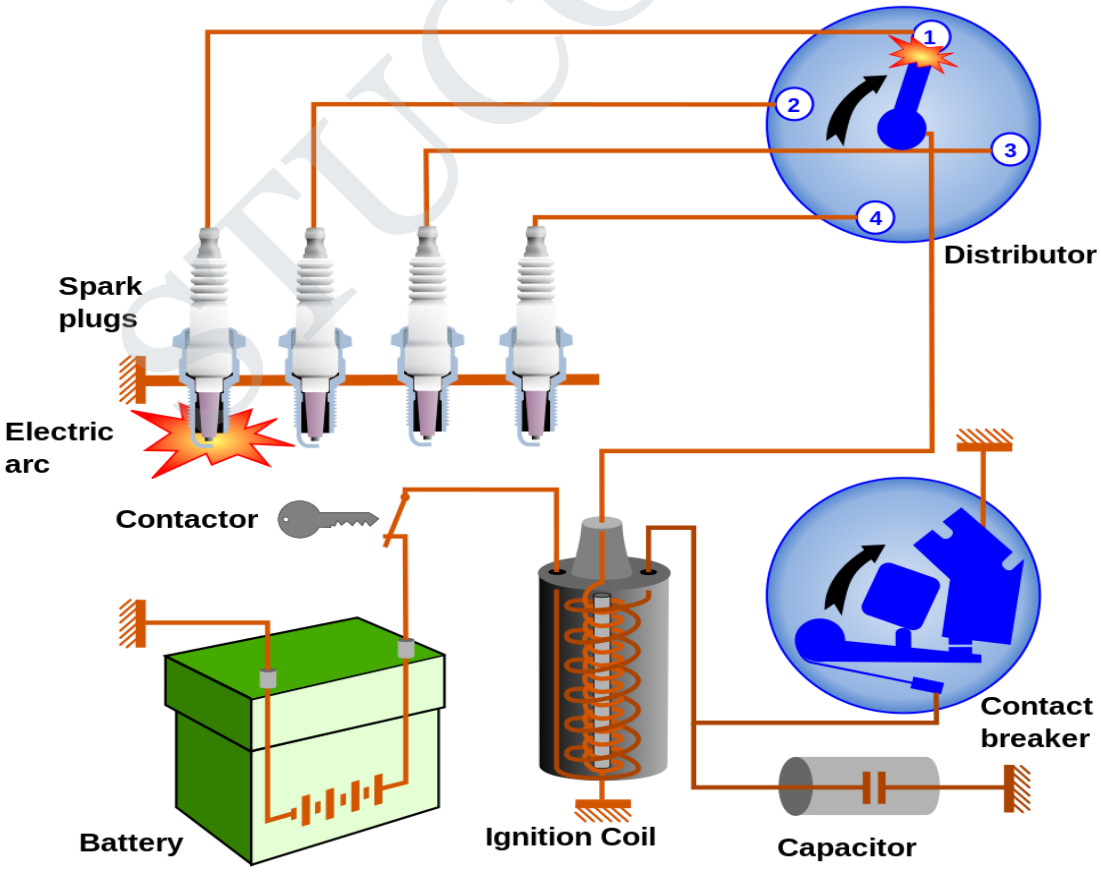
Overview of the emission norms in India

- 1991 Idle CO Limits for Petrol Vehicles and Free Acceleration Smoke for Diesel Vehicles, Mass Emission Norms for Petrol Vehicles.
- 1992 Mass Emission Norms for Diesel Vehicles.
- 1996 Revision of Mass Emission Norms for Petrol and Diesel Vehicles, mandatory fitment of Catalytic Converter for Cars in Metros on Unleaded Petrol.
- 1998 Cold Start Norms Introduced.

- 2000 India 2000 (Equivalent to Euro I) Norms, Modified IDC (Indian Driving Cycle), Bharat Stage II Norms for Delhi.
- 2001 Bharat Stage II (Equivalent to Euro II) Norms for All Metros, Emission Norms for CNG & LPG Vehicles.
- 2003 Bharat Stage II (Equivalent to Euro II) Norms for 13 major cities.
- 2005 From 1 April Bharat Stage III (Equivalent to Euro III) Norms for 13 major cities.
- 2010 Bharat Stage III Emission Norms for 2-wheelers, 3-wheelers and 4-wheelers for entire country whereas Bharat Stage – IV (Equivalent to Euro IV) for 13 major cities for only 4-wheelers. Bharat Stage IV also has norms on OBD (similar to Euro III but diluted)
- 2017 Bharat Stage IV Norms for all vehicles.
- 2018 BS-VI fuel norms from April 1, 2018 in Delhi instead of 2020
- 2020 Proposed date for country to adopt Bharat Stage VI norms for cars, skipping Bharat Stage V

UNIT IV – INTERNAL COMBUSTION ENGINE PERFORMANCE AND SYSTEMS





BRAKE POWER

Brake power is the power output of the drive shaft of the engine without the power loss caused by gears, transmission, friction etc. it is also called as useful power or true power.

SPECIFIC FUEL CONSUMPTION

The mass flow rate of fuel required to produce a unit of brake power. SFC is expressed in SI units as kilograms per hour per kilowatt (kg/kW-hr). It allows engines of all different sizes to be compared to see which is the most fuel efficient.

BRAKE THERMAL EFFICIENCY

Brake Thermal Efficiency is the ratio of brake power output to power input. It is used to evaluate how efficient an engine converts the heat from a fuel to mechanical energy. It is expressed in percentage.

INDICATED THERMAL EFFICIENCY

The ratio between the indicated power output of an engine and the rate of supply of energy from fuel.

FRICTIONAL POWER

An engine has many moving parts that produce friction. Some of these frictional forces remain constant (as long as applied load is constant); some of these frictional losses increase as engine speed increases, such as piston side forces and connecting bearing forces (due to increased inertia forces from the oscillating piston). A few frictional forces decrease at higher speed, such as the frictional force on the cam's lobes which is used to operate inlet and outlet valves (the valves' inertia at high speed tends to pull the cam follower away from the cam lobe).

TYPES OF LOADING IN THE ENGINE.

Load is when the engine is doing work. Whatever work the engine is doing places a "load" on the engine which resists the engine's turning motion and slows the engine speed down and so it requires more throttle to maintain speed. "Partial load/Part load" is when the engine is doing work that would stall the engine at idle, but does not require full throttle, just partial throttle. A good example would be a car travelling at highway speed or up a slight grade.

MOTORING TEST

It is a method of engine testing used measure the power output of the engine. The temperature of he engine's pistons and cylinder walls, together with other working parts and also the engine oil, falls bellow that of normal working temperature during the motoring tests, and with the lack of exhaust gases, etc, the frictional and pumping losses are somewhat modified.

USE OF RETARDATION TEST

It is used to find the reduction in speed with respect to time in an I C engine.

USE OF MORSE TEST

Morse test is adopted to find the indicated power of each cylinder of a high speed I C engine without using an indicator diagram.

METHODS TO IMPROVE EFFICIENCY OF A DIESEL ENGINE

(a) By increasing the injection pressure

(b) Increasing compression ratio

(c) Increasing cut-off ratio.

USE OF HEAT BALANCE TEST

Heat balance test is used to identify useful proportion of power output and various losses and thereby taking measures to minimize the losses for improving efficiency.

UNACCOUNTED LOSS

(a) Radiation loss

(b) Convection loss.

REDUCE HEAT LOSSES

(a) Optimize the cooling rate using appropriate coolant

(b) Use of exhaust gas recirculation to recover heat loss.

PURPOSES OF COOLING WATER BEING USED IN AN ENGINE

(a) To avoid engine getting overheated

(b) To prevent engine seizing.

HEAT LOSS AFFECT THE ENGINE PERFORMANCE

(a) Reduction in net power available

(b) Decrease in efficiency.

EFFECTS OF CO IN GLOBAL WARMING

(a) Rise in global temperature

(b) Rise in sea level

(c) Food shortages and hunger

(d) Climate change

METHODS TO REDUCE NO_x FROM A DIESEL ENGINE

(a) Low self ignition temperature

(b) Reduction of excess air

(c) Use of catalytic converter.

METHODS TO REDUCE HC FROM A DIESEL ENGINE

(a) Complete combustion

(b) Avoiding rapid deceleration

(c) Normal speed running.

CARBURATION

The process of vapourizing the fuel (petrol) and mixing it with air outside the cylinder in the SI Engine is known as carburation.

REQUIREMENT OF THE FUEL INJECTOR

(i) To inject the fuel into the engine cylinder by atomizing the fuel to the required degree.

(ii) To distribute the fuel such that there is a rapid and complete mixing of fuel and air.

NECESSITY OF COOLING IN IC ENGINE

(i) To avoid un even expansion of the piston in the cylinder.

(ii) To reduce the temperature of piston and cylinder.

(iii) To avoid the overheating of the cylinder.

(iv) To avoid the physical and chemical changes in the lubricating oil which may cause sticking of piston rings..

PURPOSE OF THERMOSTAT IN AN ENGINE COOLING SYSTEM

It is located in the cooling circuit,between the engine and the radiator. It opens and allows the flow of water, assisted by water pump. It is a self regulating device and it is designed at a particular temperature usually about 950C. The main purpose of thermostat is to avoid cold starting problem.

IGNITION DELAY

The period between the start of fuel injection into the combustion chamber and the start of combustion is termed as ignition delay period.

UNIT INJECTION SYSTEM

In this system each cylinder has its own individual high pressure pump and a metering unit.

OCTANE NUMBER IN I.C. ENGINES

Octane rating is a measure of a fuel's ability to resist 'knock'. The octanerequirement of an engine varies with compression ratio, geometrical and mechanical considerations and operating conditions. The higher the octane number the greater the fuel's resistance to knocking or pinging during combustion.

INDICATED POWER

$$IP = P_{IMEP} LAnk \quad kW$$

Where

N= rpm

P_{IMEP}=Indicated Mean Effective Pressure(kN/m^2)

L=length of Stroke (m)

A=Area of piston or cylinder(m^2)

n = Number of Power strokes(N/2 for four stroke)

k= Number of cylinders

BREAK POWER

$$BP = P_{BMEP} LAnk \quad kW$$

Where

N= rpm

P_{BMEP}= Break Mean Effective Pressure(kN/m^2)

L=length of Stroke (m)

A=Area of piston or cylinder(m^2)

n = Number of Power strokes(N/2 for four stroke)

k= Number of cylinders

(OR)

$$BP = \frac{2\pi NT}{60} \quad kW$$

Where

N= rpm

T=W x R (Load(kN) x Radius of drum(m))

NUMBER OF POWER STROKES

$$n = \frac{N}{60} \text{ (For two stroke Engine)}$$

$$n = \frac{N}{2 \times 60} \text{ (For Four stroke Engine)}$$

HEAT SUPPLIED

$$Q_S = \dot{m}_f \times CV \ \left(\frac{kJ}{s}\right)$$

Where

$\dot{m}_f$ = Mass of fuel $\left(\frac{kg}{s}\right)$

CV = Calorific Value of fuel $\left(\frac{kJ}{kg}\right)$

FRICTION POWER

$$FP = IP - BP \quad kW$$

INDICATED THERMAL EFFICIENCY

$$\eta_{IT} = \frac{\text{Indicated Power}}{\text{Heat Supplied}}$$

$$\eta_{IT} = \frac{IP}{\dot{m}_f \times CV}$$

BREAK THERMAL EFFICIENCY

$$\eta_{BT} = \frac{\text{Break Power}}{\text{Heat Supplied}}$$

$$\eta_{BT} = \frac{BP}{\dot{m}_f \times CV}$$

MECHANICAL EFFICIENCY

$$\eta_{Mech} = \frac{\text{Indicated Power}}{\text{Break Power}}$$

BRAKE SPECIFIC FUEL CONSUMPTION

$$BSFC = \frac{m_f}{BP} = \frac{(kg/hr)}{kW}$$

INDICATED SPECIFIC FUEL CONSUMPTION

$$ISFC = \frac{m_f}{IP} = \frac{(kg/hr)}{kW}$$

VOLUMETRIC EFFICIENCY

$$\eta_v = \frac{\text{Volume of charge (suction)}}{\text{Swept volume}} = \frac{V_a}{V_S nk}$$

HEAT BALANCE SHEET:

HEAT SUPPLIED BY THE FUEL $(Q_S) = m_f \times CV \quad \left(\frac{kJ}{hr}\right)$

HEAT SUPPLIED BY THE FUEL $(Q_{BP}) = 2\pi NT \quad \left(\frac{kJ}{hr}\right)$

HEAT LOSS DUE TO THE COOLING WATER

$$Q_w = m_w \times C_W \times (T_{w2} - T_{w1}) \quad \left(\frac{kJ}{hr}\right)$$

HEAT CARRIED AWAY BY EXHAUST GAS

$$Q_g = m_g \times C_g \times (T_{g2} - T_a) \quad \left(\frac{kJ}{hr}\right)$$

UNACCOUNTED HEAT LOSS

$$(Q_{ua}) = Q_S - Q_{BP} + Q_w + Q_g \quad \left(\frac{kJ}{hr}\right)$$

[Units is $\left(\frac{kJ}{hr}\right)$ if m_g, m_f, m_w are in $\left(\frac{kg}{hr}\right)$]

1. **Calculate the diameter and length of the stroke of a diesel engine working on four stroke constant pressure cycle from the following data. IP=18.75 kW rotation per minute=220 CR=14 fuel cut-off ratio=1/20th of stroke, index of expansion=1.3, index of compression=1.35, L/D=1.5. Assume the pressure and temperature of the air at inlet are 1 bar and 40°C respectively.**

GIVEN

IP=18.75kW, N=220rpm, CR=14, L/D=1.5 , $P_1 = 1bar$, $T_1 = 40°C$

SOLUTION:

CUTOFF RATIO:

$V_3 - V_2 = 0.05\,(V_1 - V_2) \rightarrow 0.05 = \frac{\rho-1}{r_c-1}$ ⇨ $\boldsymbol{\rho = 1.65}$

MEAN EFFECTIVE PRESSURE:

$P_m = \frac{P_1 r_c^{\gamma}\left[\gamma(\rho-1)-r_c^{\,1-\gamma}(\rho^{\gamma}-1)\right]}{(\gamma-1)(r_c-1)}$ ⇨ $P_m = \frac{1\times14^{1.3}\left[1.3(1.65-1)-14^{\,1-1.3}(1.65^{1.3}-1)\right]}{(1.3-1)(14-1)}$ ⇨ $\mathbf{P_m = 3.4bar}$

INDICATED POWER:

$IP = P_{IMEP}LAnk$ ⇨ $18.75 = 3.4\times10^2\times1.5D\times\frac{\pi\times D^2}{4}\times\frac{220}{2\times60}\times1$ ⇨ **D = 0.294m**

LENGTH OF STROKE:

$\frac{L}{D} = 1.5$ ⇨ $L = 1.5\times0.294$ ⇨ $\mathbf{L = 0.589m}$

2. **A four-cylinder, four-stroke oil engine 10 cm in diameter and 15 cm in stroke develops a torque of 185 Nm at 2000 rpm. The oil consumption is 14.5 lit/hr. The specific gravity of the oil is 0.82 and calorific value of oil is 42000 kJ/kg. If the imep taken from the indicated diagram is 6.7 bar find, (i) mechanical efficiency (ii) brake thermal efficiency (iii) Brake mean effective pressure (iv) Specific fuel consumption in litres on brake power basis.**

GIVEN:

D=10cm, L=15cm, T=185Nm, N=2000rpm, $\dot{m}_f = 14.5\,\frac{Lit}{hr}$, $\omega = 0.82$, CV=42000 kJ/kg,

$P_{Imean} = 6bar$

SOLUTION:

INDICATED POWER:

$IP = P_{IMEP}LAnk$ ⇨ $IP = 6.7\times10^2\times0.15\times\frac{\pi\times0.1^2}{4}\times\frac{2000}{2\times60}\times4$ ⇨ **IP = 52.62kW**

BRAKE POWER:

$BP = \frac{2\pi NT}{60}$ ⇨ $BP = \frac{2\times\pi\times2000\times185}{60}$ ⇨ $BP = 38746.31\ W$ ⇨ $\mathbf{BP = 38.75\ kW}$

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}}$ ⇨ $\eta_{Mech} = \frac{38.75}{52.62}$ ⇨ $\boldsymbol{\eta_{Mech} = 73.64\%}$

HEAT INPUT:

$Q_S = \dot{m}_f\times CV$ ⇨ $Q_S = (14.5\times0.82)\times42000$ ⇨ $\mathbf{Q_S = 499380\,\frac{kJ}{hr} = 138.72\,\frac{kJ}{sec}}$

BRAKE THERMAL EFFICIENCY:

$\eta_{BT} = \frac{BP}{HI}$ ⇨ $\eta_{BT} = \frac{113.09}{138.72}$ ⇨ $\boldsymbol{\eta_{BT} = 81.53\%}$

BRAKE MEAN EFFECTIVE PRESSURE

$BP = P_{BMEP}LAnk$ ⇨ $38.75 = P_{BMEP}\times0.15\times\frac{\pi\times0.1^2}{4}\times\frac{2000}{2\times60}\times4$ ⇨ $\mathbf{P_{BMEP} = 493.38\,\frac{kN}{m^2}}$

BRAKE SPECIFIC FUEL CONSUMPTION

$BSFC = \frac{m_f}{BP}$ ⇨ $BSFC = \frac{14.5\times0.82}{38.75}$ ⇨ $\mathbf{BSFC = 0.307\,\frac{kg}{kWh}}$

3. **A four stroke, four cylinder gasoline engine has a bore of 60 mm and a stroke of 100 mm. On test it develops a torque of 66.5 N m, when running at 3000 rpm. If the clearance volume in each cylinder is 60 cc, the relative efficiency with respect to brake thermal efficiency is 0.5 and the calorific value of the fuel is 42 MJ/kg, determine the fuel consumption in kg/h and the brake mean effective pressure.**

GIVEN:

D=60cm, L=100cm, T=66.5Nm, N=3000rpm, CV=42000 kJ/kg, $V_C = 60CC$, $\eta_{Rbt} = 0.5$

SOLUTION:

BRAKE POWER:

$$BP = \frac{2\pi NT}{60} \Rightarrow BP = \frac{2\times\pi\times3000\times66.5}{60} \Rightarrow BP = 20891.59\ W \Rightarrow \mathbf{BP = 20.89\ kW}$$

BRAKE MEAN EFFECTIVE PRESSURE

$$BP = P_{BMEP}LAnk \Rightarrow 20.89 = P_{BMEP} \times 0.1 \times \frac{\pi\times0.06^2}{4} \times \frac{3000}{2\times60} \times 4 \Rightarrow \mathbf{P_{BMEP} = 738.83\ \frac{kN}{m^2}}$$

FUEL CONSUMPTION:

$$\eta_{BT} = \frac{BP}{\dot{m}_f\times CV} \Rightarrow \dot{m}_f = \frac{20.89\times3600}{0.25\times42000} \Rightarrow \mathbf{\dot{m}_f = 7.16\ \frac{kg}{hr}}$$

BRAKE THERMAL EFFICIENCY:

$$\eta_{Rbt} = \frac{\eta_{bt}}{\eta_{Ideal}} \Rightarrow \eta_{bt} = \eta_{Rbt} \times \eta_{Ideal}$$

$$\Rightarrow \eta_{bt} = 0.5 \times 0.5$$

$$\Rightarrow \mathbf{\eta_{bt} = 0.25}$$

IDEAL EFFICIENCY:

$$\eta_{Ideal} = 1 - \frac{1}{r_c^{\gamma-1}} \Rightarrow \eta_{Ideal} = 1 - \frac{1}{5.742^{1.4-1}}$$

$$\Rightarrow \mathbf{\eta_{Ideal} = 0.5}$$

Where,

$$r_C = \frac{V_1}{V_2} = \frac{V_S+V_C}{V_C} \Rightarrow r_C = \frac{2.83\times10^{-3}+60\times10^{-6}}{60\times10^{-6}}$$

$$\Rightarrow \mathbf{r_C = 5.742}$$

$$V_S = \frac{\pi D^2}{4} \times L \Rightarrow V_S = \frac{\pi\times0.06^2}{4} \times 0.1$$

$$\Rightarrow \mathbf{V_S = 2.83 \times 10^{-3} m^3}$$

4. **A four cylinder, four stroke diesel engine has brake mean effective pressure of 6 bar at full load speed of 600 rpm and specific fuel consumption of 0.25 kg/kWh. The cylinder has bore of 20 cm and stroke length of 30 cm. The air fuel ratio is measured as 26 from the exhaust gas analysis. The ambient conditions are 1 bar, 27°C. Assuming the calorific value of fuel as 43 MJ/kg determine the brake thermal efficiency and the volumetric efficiency. Also find out brake power.**

GIVEN:

$P_{bmean} = 6bar$, N=600rpm, SFC=0.25 kg/kWh, D=20cm, L=30cm, $\frac{m_a}{m_f} = 26$, $P_1 = 1bar$, $T_1 = 27°C$

CV=43MJ/kg

SOLUTION:

BRAKE POWER:

$$BP = P_{BMEP}LAnk \Rightarrow BP = 6 \times 10^2 \times 0.3 \times \frac{\pi\times0.2^2}{4} \times \frac{600}{2\times60} \times 4 \Rightarrow \mathbf{BP = 113.09\ Kw}$$

FUEL CONSUMPTION:

$$BSFC = \frac{\dot{m}_f}{BP} \Rightarrow \dot{m}_f = BSFC \times BP \Rightarrow \dot{m}_f = 0.25 \times 113.09 \Rightarrow \mathbf{\dot{m}_f = 28.27\ kg/hr}$$

BRAKE THERMAL EFFICIENCY:

$$\eta_{BT} = \frac{BP}{\dot{m}_f\times CV} \Rightarrow \eta_{BT} = \frac{113.09\times3600}{28.27\times42000} \Rightarrow \mathbf{\eta_{BT} = 0.34\ or\ 34.28\%}$$

VOLUMETRIC EFFICIENCY:

$$\eta_v = \frac{V_a}{V_S \times n \times k} \Rightarrow \eta_v = \frac{0.176}{9.42\times10^{-3}\times\frac{600}{2\times60}\times4}$$

$$\Rightarrow \boldsymbol{\eta_v = 93.37\%}$$

$$\frac{\dot{m}_a}{\dot{m}_f} = 26 \Rightarrow \dot{m}_a = 26 \times \frac{28.27}{3600} \Rightarrow \boldsymbol{\dot{m}_a = 0.204\ kg/s}$$

$$P_aV_a = m_aRT_a \Rightarrow V_a = \frac{m_aRT_a}{P_a} \Rightarrow V_a = \frac{0.204\times0.287\times300}{1\times10^2}$$

$$\Rightarrow \boldsymbol{V_a = 0.176\frac{m^3}{s}}$$

$$V_s = \frac{\pi\times d^2}{4}\times L \Rightarrow V_s = \frac{\pi\times0.2^2}{4}\times0.3 \Rightarrow \boldsymbol{V_s = 9.42\times10^{-3}m^3}$$

5. **A two stroke two cylinder engine runs with speed of 3000 rpm and fuel consumption of 5 litres/hr. The fuel has specific gravity of 0.7 and air-fuel ratio is 19. The piston speed is 500 m/min and indicated mean effective pressure is 6 bar. The ambient conditions are 1.013 bar, 15°C. The volumetric efficiency is 0.7 and mechanical efficiency is 0.8. Determine brake power output considering R for gas = 0.287 kJ/kgK. (Take piston speed, m/min = 2 LN where L is stroke (m) and N is rpm)**

GIVEN:

N=3000rpm, $m_f = 5\ \frac{lit}{hr}$, $\omega = 0.7$, $\frac{m_a}{m_f} = 19$, Piston Speed=500 m/min, $P_{IMEP} = 6$ bar, $P_1 = 1.013$bar, $T_1 = 15°C$, $\eta_v = 0.7$, $\eta_{mech} = 0.8$, R = 0.287 kJ/kgK,

SOLUTION:

LENGTH OF STROKE:

$$\text{Piston Speed} = 2LN \Rightarrow 500 = 2\times L\times3000 \Rightarrow \boldsymbol{L = 0.0833m}$$

MASS FLOWRATE OF AIR:

$$\dot{m}_f = 5\ \frac{lit}{hr} \Rightarrow \dot{m}_f = 5\times0.7 \Rightarrow \boldsymbol{\dot{m}_f = 3.5\frac{kg}{hr} = 0.00972kg/s}$$

$$\frac{\dot{m}_a}{\dot{m}_f} = 19 \Rightarrow \dot{m}_a = 19\times0.00972 \Rightarrow \boldsymbol{\dot{m}_a = 0.0183\ kg/s}$$

VOLUME FLOWRATE OF AIR:

$$P_aV_a = m_aRT_a \Rightarrow V_a = \frac{m_aRT_a}{P_a} \Rightarrow V_a = \frac{0.0183\times0.287\times288}{1.013\times10^2} \Rightarrow \boldsymbol{V_a = 0.0151\frac{m^3}{s}}$$

STROKE VOLUME:

$$\eta_v = \frac{V_a}{V_S\times n\times k} \Rightarrow V_S = \frac{0.0151}{0.7\times\frac{3000}{60}\times2} \Rightarrow \boldsymbol{V_s = 2.153\times10^{-4}\ m^3}$$

DIAMETER OR BORE OF THE CYLINDER:

$$V_s = \frac{\pi\times d^2}{4}\times L \Rightarrow 2.153\times10^{-4} = \frac{\pi\times D^2}{4}\times0.0833 \Rightarrow \boldsymbol{D = 0.0574m}$$

INDICATED POWER:

$$IP = P_{IMEP}LAnk \Rightarrow IP = 6\times10^2\times0.0833\times\frac{\pi\times0.0574^2}{4}\times\frac{3000}{60}\times2 \Rightarrow \boldsymbol{IP = 12.92kW}$$

MECHANICAL EFFICIENCY:

$$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow 0.8 = \frac{\text{Brake Power}}{18.47} \Rightarrow \boldsymbol{\text{Brake Power} = 10.33kW}$$

6. **During trial of four stroke single cylinder engine the load on dynamometer is found 20 kg at radius of 50 cm. The speed of rotation is 3000 rpm. The bore and stroke are 20 cm and 30 respectively. Fuel is supplied at the rate of 0.15 kg/min. The calorific value of fuel may be taken as 43 MJ/kg. After some time the fuel supply is cut and the engine is rotated with motor which required 5 kW to maintain the same speed of rotation of engine. Determine the brake power, indicated power, mechanical efficiency, brake thermal efficiency, indicated thermal efficiency, brake mean effective pressure, indicated mean effective pressure.**

GIVEN:

W = 20kg, R=50cm, N=3000rpm, D=20cm, L=30cm, $m_f = 0.15\ \frac{kg}{min}$, CV=43MJ/kg, FP=5kW

SOLUTION:

TORQUE:

$W = \frac{T}{R\ X\ 9.81} \Rightarrow T = W \times R\ X\ 9.81 \Rightarrow T = 20 \times 0.5\ X\ 9.81 \Rightarrow \mathbf{T = 98.1Nm}$

BRAKE POWER:

$BP = \frac{2\pi NT}{60} \Rightarrow BP = \frac{2\times\pi\times3000\times98.1}{60} \Rightarrow BP = 30819.02\ W \Rightarrow \mathbf{BP = 30.82\ kW}$

INDICATED POWER:

Indicated Power = Brake Power + Friction Power ⇒ IP=30.82+5 ⇒ $\mathbf{IP = 35.82\ kW}$

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow \eta_{Mech} = \frac{30.82}{35.82} \Rightarrow \boldsymbol{\eta_{Mech} = 86.04\%}$

BRAKE SPECIFIC FUEL CONSUMPTION

$BSFC = \frac{m_f}{BP} \Rightarrow BSFC = \frac{0.15\times60}{30.82} \Rightarrow \mathbf{BSFC = 0.292\ \frac{kg}{kWhr}}$

BRAKE THERMAL EFFICIENCY:

$\eta_{BT} = \frac{BP}{\dot{m}_f\times CV} \Rightarrow \eta_{BT} = \frac{30.82\times60}{0.15\times43000} \Rightarrow \boldsymbol{\eta_{BT} = 0.2867 \text{ or } 28.67\%}$

INDICATED THERMAL EFFICIENCY:

$\eta_{IT} = \frac{IP}{\dot{m}_f\times CV} \Rightarrow \eta_{IT} = \frac{35.82\times60}{0.15\times43000} \Rightarrow \boldsymbol{\eta_{BT} = 0.3332 \text{ or } 33.32\%}$

INDICATED MEAN EFFECTIVE PRESSURE

$IP = P_{IMEP}LAnk \Rightarrow 35.82 = P_{IMEP} \times 0.3 \times \frac{\pi\times0.2^2}{4} \times \frac{3000}{2\times60} \times 1 \Rightarrow \mathbf{P_{IMEP} = 152.02\ \frac{kN}{m^2}}$

BRAKE MEAN EFFECTIVE PRESSURE

$BP = P_{BMEP}LAnk \Rightarrow 30.82 = P_{BMEP} \times 0.3 \times \frac{\pi\times0.2^2}{4} \times \frac{3000}{2\times60} \times 1 \Rightarrow \mathbf{P_{BMEP} = 130.8\ \frac{kN}{m^2}}$

7. **A four stroke four cylinder diesel engine running at 300 rpm produces 120 kW of brake power. The cylinder dimensions are 30 cm bore and 25 cm stroke. Fuel consumption rate is 1 kg/min while air fuel ratio is 10. The average indicated mean effective pressure is 0.8 MPa. Determine indicated power, mechanical efficiency, brake thermal efficiency and volumetric efficiency of engine. The calorific value of fuel is 43 MJ/kg. The ambient conditions are 1.013 bar, 27°C.**

GIVEN:

N=300rpm, BP=120kW, D=30cm, L=25cm, $m_f = 1\ \frac{kg}{min}$, $\frac{m_a}{m_f} = 10$, $P_{IMEP} = 0.8$Mpa, CV=43MJ/kg, $P_1 = 1.013$bar, $T_1 = 27°C$,

INDICATED POWER:

$IP = P_{IMEP}LAnk \Rightarrow IP = 8 \times 10^2 \times 0.25 \times \frac{\pi \times 0.3^2}{4} \times \frac{300}{2\times60} \times 4 \Rightarrow$ **IP = 141.37kW**

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow \eta_{Mech} = \frac{120}{141.37} \Rightarrow \boldsymbol{\eta_{Mech} = 84.88\%}$

BRAKE THERMAL EFFICIENCY:

$\eta_{BT} = \frac{BP}{\dot{m}_f \times CV} \Rightarrow \eta_{BT} = \frac{120\times60}{1\times43000} \Rightarrow \boldsymbol{\eta_{BT} = 0.1674 \text{ or } 16.74\%}$

VOLUMETRIC EFFICIENCY:

$\eta_v = \frac{V_a}{V_S \times n \times k} \Rightarrow \eta_v = \frac{0.142}{0.0176 \times \frac{300}{2\times60} \times 4}$

$\Rightarrow \boldsymbol{\eta_v = 80.68\%}$

$\frac{\dot{m}_a}{\dot{m}_f} = 10 \Rightarrow \dot{m}_a = 10 \times \frac{1}{60} \Rightarrow \boldsymbol{\dot{m}_a = 0.167\ kg/s}$

$P_aV_a = m_aRT_a \Rightarrow V_a = \frac{m_aRT_a}{P_a} \Rightarrow V_a = \frac{0.167\times0.287\times300}{1.013\times10^2}$

$\Rightarrow \boldsymbol{V_a = 0.142\ \frac{m^3}{s}}$

$V_s = \frac{\pi \times d^2}{4} \times L \Rightarrow V_s = \frac{\pi\times0.3^2}{4} \times 0.25 \Rightarrow \boldsymbol{V_s = 0.0176 m^3}$

8. **The following observations were taken during a test on a single cylinder 4 stroke cycle engine having a bore of 300 mm and a stroke of 450 mm.**

Ambient air temperature	**=22°C**	**Engine speed**	**= 300 rpm**
Fuel consumption	**=11 kg/h**	**CV of fuel**	**= 42000 kJ/kg.**
Mean effective pressure	**= 6 bar**	**Rope diameter**	**= 2 cm**
Net brake load	**=1.0 kN**	**Brake drum diameter**	**= 2 m**
Quantity of Jacket cooling water = 590 kg/hr		**Temperature entering cooling water = 22°C**	
Temperature of leaving cooling water =70°C		**Quantity of air as measured**	**= 225 kg/h**
Specific heat of exhaust of gases =1.005 kJ/kgK		**Exhaust gas temperature**	**= 405°C**

Determine indicated power, brake power mechanical efficiency and draw a heat balance sheet on hour basis.

SOLUTION:

INDICATED POWER:

$IP = P_{IMEP}LAnk \Rightarrow IP = 6 \times 10^2 \times 0.45 \times \frac{\pi\times0.3^2}{4} \times \frac{300}{2\times60} \times 1 \Rightarrow$ **IP = 47.71kW**

BRAKE POWER:

$BP = \frac{2\pi NT}{60} \Rightarrow BP = \frac{2\times\pi\times300\times1.01}{60}$

$\Rightarrow \boldsymbol{BP = 31.73\ kW}$

$T = W \times R \Rightarrow T = W \times \frac{d+D}{2} \Rightarrow T = 1 \times \frac{0.02+2}{2}$

$\Rightarrow \boldsymbol{T = 1.01\ kNm}$

MECHANICAL EFFICIENCY:

$$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow \eta_{Mech} = \frac{31.73}{47.71} \Rightarrow \boldsymbol{\eta_{Mech} = 66.5\%}$$

HEAT INPUT:

$$Q_S = \dot{m}_f \times CV \Rightarrow Q_S = 11 \times 42000 \Rightarrow \boldsymbol{Q_S = 462000\ \frac{kJ}{hr}}$$

HEAT LOSS DUE TO THE COOLING WATER

$$Q_w = m_w \times C_W \times (T_{w2} - T_{w1}) \Rightarrow Q_w = 590 \times 4.187 \times (70 - 22) \Rightarrow \boldsymbol{Q_w = 118575\ \frac{kJ}{hr}}$$

HEAT CARRIED AWAY BY EXHAUST GAS

$$Q_g = m_g \times C_g \times (T_{g2} - T_a) \Rightarrow Q_g = 225 \times 1.005 \times (405 - 22) \Rightarrow \boldsymbol{Q_g = 86605.87\ \frac{kJ}{hr}}$$

HEAT LOSS DUE TO BRAKE POWER:

$$P = \frac{2\pi NT}{60} \Rightarrow BP = 31.73\frac{kJ}{s} \times 3600 \Rightarrow \boldsymbol{BP = 114228.37\frac{kJ}{hr}}$$

UNACCOUNTED LOSS:

$$Q_{ua} = Q_S - Q_w + Q_g + Q_{BP} \Rightarrow Q_{ua} = 462000 - 118575 + 86605.87 + 114228.37$$

$$\boldsymbol{Q_{ua} = 142590\frac{kJ}{hr}}$$

PERCENTAGE OF HEAT LOSS:

$$\%Q_w = \frac{Q_w}{Q_S} \Rightarrow \%Q_w = \frac{118575}{462000} \Rightarrow \boldsymbol{\%Q_w = 25.6}$$

$$\%Q_g = \frac{Q_g}{Q_S} \Rightarrow \%Q_g = \frac{86605.87}{462000} \Rightarrow \boldsymbol{\%Q_g = 18.74}$$

$$\%Q_{BP} = \frac{Q_{BP}}{Q_S} \Rightarrow \%Q_{BP} = \frac{114228.37}{462000} \Rightarrow \boldsymbol{\%Q_{BP} = 24.7}$$

$$\%Q_{un} = \frac{Q_{un}}{Q_S} \Rightarrow \%Q_{un} = \frac{142590}{462000} \Rightarrow \boldsymbol{\%Q_{un} = 30.86}$$

9. During the trial of a single acting oil engine, cylinder diameter is 20 cm, stroke 28 cm, working on two stroke cycle and firing every cycle, the following observations were made:

Duration of trial	= 1 hour	Total fuel used	= 4.22 kg
Calorific value	= 44670 kJ/kg	Proportion of hydrogen in fuel	= 15%
Total number of revolutions	= 21000	Mean effective pressure	= 2.74 bar
Net brake load applied to a drum	= 600 N	Drum Diameter	= 100 cm
Total mass of cooling water circulated	= 495 kg	Cooling water enters	= 13°C
Cooling water leaves	= 38°C	Air used	=135kg
Temperature of air in test room	= 20°C	Temperature of exhaust gases	= 370°C
Cp_{gases}	= 1.005 kJ/kgK	$Cp_{\text{steam at atm}}$	= 2.093 kJ/kg K

Calculate thermal efficiency and draw up the heat balance.

SOLUTION:

INDICATED POWER:

$$IP = P_{IMEP}LAnk \Rightarrow IP = 2.74 \times 10^2 \times 0.28 \times \frac{\pi \times 0.2^2}{4} \times \frac{21000}{3600} \times 1 \Rightarrow \boldsymbol{IP = 14.06\ kW}$$

BRAKE POWER:

$BP = \frac{2\pi NT}{60} \Rightarrow BP = \frac{2\times\pi\times21000\times0.3}{3600} \Rightarrow \mathbf{BP = 10.99\ kW}$

$T = W \times R \Rightarrow T = W \times \frac{d+D}{2} \Rightarrow T = 600 \times \frac{0+1}{2} \Rightarrow \mathbf{T = 0.3\ kNm}$

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow \eta_{Mech} = \frac{10.99}{14.06} \Rightarrow \boldsymbol{\eta_{Mech} = 78.17\%}$

INDICATED THERMAL EFFICIENCY:

$\eta_{IT} = \frac{IP}{\dot{m}_f \times CV} \Rightarrow \eta_{IT} = \frac{14.06\times3600}{4.22\times44670} \Rightarrow \boldsymbol{\eta_{BT} = 0.2685 \text{ or } 26.85\%}$

HEAT INPUT:

$Q_S = \dot{m}_f \times CV \Rightarrow Q_S = 4.22 \times 44670 \Rightarrow \mathbf{Q_S = 188507.4\ \frac{kJ}{hr}}$

HEAT LOSS DUE TO THE COOLING WATER

$Q_w = m_w \times C_W \times (T_{w2} - T_{w1}) \Rightarrow Q_w = 495 \times 4.187 \times (38 - 13) \Rightarrow \mathbf{Q_w = 51814.13\ \frac{kJ}{hr}}$

HEAT CARRIED AWAY BY EXHAUST GAS:

MASS OF THE EXHAUST GAS:

$m_g = m_a + m_f \Rightarrow m_g = 135 + 4.22 \Rightarrow \mathbf{m_g = 139.22\ \frac{kg}{hr}}$

Heat carried away by exhaust gas = Heat carried away by steam in exhaust gas + Heat carried away by dry gas in exhaust gas

Mass of steam in exhaust gas $= 9 \times [0.15 \times 4.22] \Rightarrow \mathbf{m_{sg} = 5.697\ \frac{kg}{hr}}$

Mass of Dry Gas in exhaust gas $= m_g - m_{sg} \Rightarrow m_{dg} = 139.22 - 5.697 \Rightarrow \mathbf{m_{dg} = 133.523\ \frac{kg}{hr}}$

HEAT CARRIED AWAY BY STEAM IN EXHAUST GAS:

$Q_{sg} = m_{sg} \times \{\text{Sensible heat of water} + \text{Latent heat of water} + \text{Sensible heat of steam}\}$

$Q_{sg} = 5.697 \times \{[4.187 \times (100 - 20)] + 2257 + [2.093 \times (370 - 100)]\} \Rightarrow \mathbf{Q_{sg} = 17985.83\ \frac{kJ}{hr}}$

HEAT CARRIED AWAY BY DRY GAS IN EXHAUST GAS:

$Q_{dg} = m_{dg} \times C_{dg} \times (T_{g2} - T_a) \Rightarrow Q_{dg} = 133.523 \times 1.005 \times (370 - 20) \Rightarrow \mathbf{Q_{dg} = 46966.72\ \frac{kJ}{hr}}$

$Q_g = Q_{sg} + Q_{dg} \Rightarrow Q_g = 17985.83 + 46966.72 \Rightarrow \mathbf{Q_g = 64952.55\ \frac{kJ}{hr}}$

HEAT LOSS DUE TO BRAKE POWER:

$P = \frac{2\pi NT}{60} \Rightarrow BP = 10.99\ \frac{kJ}{s} \times 3600 \Rightarrow \mathbf{BP = 39564\ \frac{kJ}{hr}}$

UNACCOUNTED LOSS:

$Q_{ua} = Q_S - Q_w + Q_g + Q_{BP} \Rightarrow Q_{ua} = 188507.4 - 51814.13 + 64952.55 + 39564$

$\Rightarrow \mathbf{Q_{ua} = 32176.72\ \frac{kJ}{hr}}$

PERCENTAGE OF HEAT LOSS:

$\%Q_w = \frac{Q_w}{Q_S} \Rightarrow \%Q_w = \frac{51814.13}{188507.4} \Rightarrow \boldsymbol{\%Q_w = 27.48}$

$\%Q_g = \frac{Q_g}{Q_S} \Rightarrow \%Q_g = \frac{64952.55}{188507.4} \Rightarrow \boldsymbol{\%Q_g = 34.46}$

$\%Q_{BP} = \frac{Q_{BP}}{Q_S} \Rightarrow \%Q_{BP} = \frac{39564}{188507.4} \Rightarrow \boldsymbol{\%Q_{BP} = 20.99}$

$\%Q_{un} = \frac{Q_{un}}{Q_S} \Rightarrow \%Q_{un} = \frac{32176.72}{188507.4} \Rightarrow \boldsymbol{\%Q_{un} = 17.07}$

10. During an experiment on four stroke single cylinder engine the indicator diagram obtained has average height of 1 cm while indicator constant is 25 kN/m^2 per mm. The engine run at 300 rpm and the swept volume is 1.5×10^4 cm^3. The effective brake load upon dynamometer is 60 kg while the effective brake drum radius is 50 cm. The fuel consumption is 0.12 kg/min and the calorific value of fuel oil is 42 MJ/kg. The engine is cooled by circulating water around it at the rate of 6 kg/min. The cooling water enters at 35° C and leaves at 70°C. Exhaust gases leaving have energy of 30 kJ/s with them. Take specific heat of water as 4.18 kJ/kg K. Determine indicated power output, brake power output and mechanical efficiency. Also draw the overall energy balance in kJ/s.

GIVEN:

Indicator Diagram Height =1cm, Indicator Constant = 25 kN/m^2 per mm, N=300rpm, $V_S = 1.5 \times 10^4$ cm^3, $W_{eb} = 60$kg, $R_{eb} = 50$cm, $m_f = 0.12\frac{kg}{min}$, CV = 42 MJ/kg, $m_w = 6\frac{kg}{min}$, $T_{w1} = 35°C$, $T_{w2} = 70°C$, $Q_g = 30\frac{kJ}{s}$, $C_p = 4.18\frac{kJ}{kg.K}$

SOLUTION:

BRAKE MEAN EFFECTIVE PRESSURE:

$P_{IMEP} = \text{Ind. diagram height} \times \text{Indicator Constant} \Rightarrow P_{IMEP} = 10\text{mm} \times 25\frac{kN}{mm \times m^2} \Rightarrow \mathbf{P_{IMEP} = 250\frac{kN}{m^2}}$

INDICATED POWER:

$IP = P_{IMEP}V_S nk \Rightarrow IP = 250 \times 1.5 \times 10^4 \times 10^{-6} \times \frac{300}{2\times60} \times 1 \Rightarrow \mathbf{IP = 9.375\ kW}$

BRAKE POWER:

$BP = \frac{2\pi NT}{60} \Rightarrow BP = \frac{2\times\pi\times300\times0.1472}{60} \Rightarrow \mathbf{BP = 4.62\ kW}$

$T = 9.81 \times W \times R \Rightarrow T = 9.81 \times W \times \frac{d+D}{2} \Rightarrow T = 9.81 \times 60 \times \frac{0+0.5}{2} \Rightarrow \mathbf{T = 0.1472\ kNm}$

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow \eta_{Mech} = \frac{4.62}{9.375} \Rightarrow \mathbf{\eta_{Mech} = 49.33\%}$

HEAT INPUT:

$Q_S = \dot{m}_f \times CV \Rightarrow Q_S = 0.12 \times 42000 \Rightarrow \mathbf{Q_S = 5040\frac{kJ}{min}}$

HEAT LOSS DUE TO THE COOLING WATER

$Q_w = m_w \times C_W \times (T_{w2} - T_{w1}) \Rightarrow Q_w = 6 \times 4.18 \times (70 - 35) \Rightarrow \mathbf{Q_w = 877.8\frac{kJ}{min}}$

HEAT CARRIED AWAY BY EXHAUST GAS

$Q_g = 30\frac{kJ}{s} \times 60 \Rightarrow \mathbf{Q_g = 1800\frac{kJ}{min}}$

HEAT LOSS DUE TO BRAKE POWER:

$P = \frac{2\pi NT}{60} \Rightarrow BP = 4.62\frac{kJ}{s} \times 60 \Rightarrow \mathbf{BP = 277.2\frac{kJ}{min}}$

UNACCOUNTED LOSS:

$Q_{ua} = Q_S - Q_w + Q_g + Q_{BP} \Rightarrow Q_{ua} = 5040 - 877.8 + 1800 + 277.2 \Rightarrow \mathbf{Q_{ua} = 2085\frac{kJ}{min}}$

PERCENTAGE OF HEAT LOSS:

$\%Q_w = \frac{Q_w}{Q_S} \Rightarrow \%Q_w = \frac{877.8}{5040} \Rightarrow \mathbf{\%Q_w = 17.42}$

$\%Q_g = \frac{Q_g}{Q_S} \Rightarrow \%Q_g = \frac{1800}{5040} \Rightarrow \mathbf{\%Q_g = 35.71}$

$\%Q_{BP} = \frac{Q_{BP}}{Q_S} \Rightarrow \%Q_{BP} = \frac{277.2}{5040} \Rightarrow \mathbf{\%Q_{BP} = 5.5}$

$\%Q_{un} = \frac{Q_{un}}{Q_S} \Rightarrow \%Q_{un} = \frac{2085}{5040} \Rightarrow \mathbf{\%Q_{un} = 41.37}$

11. During 15 minutes trial of an internal combustion engine of 2-stroke single cylinder type the total 4 kg fuel is consumed while the engine is run at 1500 rpm. Engine is cooled employing water being circulated at 15 kg/min with its inlet and exit temperatures as 27°C and 50°C. The total air consumed is 150 kg and the exhaust temperature is 400°C. The atmospheric temperature is 27°C. The mean specific heat of exhaust gases may be taken as 1.25 kJ/kg K. The mechanical efficiency is 0.9. Determine the brake power, brake specific fuel consumption and indicated thermal efficiency. Also draw energy balance on per minute basis. Brake torque is 300 Nm and the fuel calorific value is 42 MJ/kg.

GIVEN:

Trail=15min, $m_f = 4kg = \frac{4}{15}\frac{kg}{min}$, N=1500rpm, $m_w = 15\frac{kg}{min}$, $T_{w1} = 27°C$, $T_{w2} = 50°C$, $m_g = 150kg$, $T_{g1} = 27°C$, $T_{g2} = 400°C$, $C_g = 1.25\frac{kJ}{kg.K}$, $\eta_{mech} = 0.9$, T=300Nm

SOLUTION:

BRAKE POWER:

$BP = \frac{2\pi NT}{60} \Rightarrow BP = \frac{2\times\pi\times1500\times300}{60} \Rightarrow BP = 47123.89\ W \Rightarrow \mathbf{BP = 47.124\ kW}$

BRAKE SPECIFIC FUEL CONSUMPTION

$BSFC = \frac{m_f}{BP} \Rightarrow BSFC = \frac{4\times60}{15\times47.124} \Rightarrow \mathbf{BSFC = 0.339\ \frac{kg}{kWhr}}$

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow 0.9 = \frac{47.124}{IP} \Rightarrow \mathbf{IP = 52.36\ kW}$

INDICATED THERMAL EFFICIENCY:

$\eta_{IT} = \frac{IP}{\dot{m}_f\times CV} \Rightarrow \eta_{IT} = \frac{52.36\times60\times15}{4\times42000} \Rightarrow \mathbf{\eta_{BT} = 0.2805\ Or\ 28.05\%}$

HEAT INPUT:

$Q_S = \dot{m}_f \times CV \Rightarrow Q_S = \frac{4}{15}\times42000 \Rightarrow \mathbf{Q_S = 11200\frac{kJ}{min}}$

HEAT LOSS DUE TO THE COOLING WATER

$Q_w = m_w \times C_W \times (T_{w2} - T_{w1}) \Rightarrow Q_w = 15\times4.18 \times(50-27) \Rightarrow \mathbf{Q_w = 1444.5\ \frac{kJ}{min}}$

HEAT CARRIED AWAY BY EXHAUST GAS

$Q_g = (m_a + m_f)\times C_g \times (T_{g2} - T_a) \Rightarrow Q_g = \frac{150+4}{15}\times1.25 \times(400-27) \Rightarrow \mathbf{Q_g = 4786.83\ \frac{kJ}{min}}$

HEAT LOSS DUE TO BRAKE POWER:

$P = \frac{2\pi NT}{60} \Rightarrow BP = 47.124\frac{kJ}{s}\times60 \Rightarrow \mathbf{BP = 2827.44\frac{kJ}{min}}$

UNACCOUNTED LOSS:

$Q_{ua} = Q_S - Q_w + Q_g + Q_{BP} \Rightarrow Q_{ua} = 11200 - 1444.5 + 4786.83 + 2827.44$

$\Rightarrow \mathbf{Q_{ua} = 2141.23\frac{kJ}{min}}$

PERCENTAGE OF HEAT LOSS:

$\%Q_w = \frac{Q_w}{Q_S} \Rightarrow \%Q_w = \frac{1444.5}{11200} \Rightarrow \mathbf{\%Q_w = 12.89}$

$\%Q_g = \frac{Q_g}{Q_S} \Rightarrow \%Q_g = \frac{4786.83}{11200} \Rightarrow \mathbf{\%Q_g = 42.74}$

$\%Q_{BP} = \frac{Q_{BP}}{Q_S} \Rightarrow \%Q_{BP} = \frac{2827.44}{11200} \Rightarrow \mathbf{\%Q_{BP} = 25.24}$

$\%Q_{un} = \frac{Q_{un}}{Q_S} \Rightarrow \%Q_{un} = \frac{2141.23}{11200} \Rightarrow \mathbf{\%Q_{un} = 19.14}$

12. **During trial of a four cylinder four stroke petrol engine running at full load it has speed of 1500 rpm and brake load of 250 N when all cylinders are working. After some time each cylinder is cut one by one and then again brought back to same speed of engine. The brake readings are measured as 175 N, 180 N, 182 N and 170 N. The brake drum radius is 50 cm. The fuel consumption rate is 0.189 kg/min with the fuel whose calorific value is 43 MJ/kg and A/F ratio of 12. Exhaust gas temperature is found to be 600°C. The cooling water flows at 18 kg/min and enters at 27°C and leaves at 50°C. The atmospheric air temperature is 27°C. Take specific heat of exhaust gas as 1.02 kJ/kg K. Determine the brake power output of engine, its indicated power and mechanical efficiency. Also draw a heat balance on per minute basis.**

GIVEN:

N=1500rpm, $W_{bl} = 250N$, 175 N, 180 N, 182 N and 170 N, R=50cm, $m_f = 0.189\ \frac{kg}{min}$, CV=43MJ/kg, $\frac{m_a}{m_f} = 12$, $T_{g2} = 600°C$, $m_w = 18\frac{kg}{min}$, $T_{w1} = 27°C$, $T_{w2} = 50°C$, , $T_{g1} = 27°C$, , $C_g = 1.02\frac{kJ}{kg.K}$,

SOLUTION:

BRAKE POWER WHEN ALL CYLINDERS ARE WORKING:

TORQUE:

$W_B = \frac{T_B}{R}$ ⇨ $T_B = W_B \times R$ ⇨ $T_B = 250 \times 0.5$ ⇨ $\mathbf{T_B = 125Nm}$

BRAKE POWER:

$BP = \frac{2\pi NT}{60\times1000}$ ⇨ $BP = \frac{2\times\pi\times1500\times125}{60\times1000}$ ⇨ $\mathbf{BP = 19.63\ kW}$

BRAKE POWER WHEN CYLINDER ONE CUT:

TORQUE 1 :

$W_{B1} = \frac{T_{B1}}{R}$ ⇨ $T_{B1} = W_{B1} \times R$ ⇨ $T_{B1} = 175 \times 0.5$ ⇨ $\mathbf{T_{B1} = 87.5Nm}$

BRAKE POWER1:

$BP_1 = \frac{2\pi NT_{B1}}{60\times1000}$ ⇨ $BP_1 = \frac{2\times\pi\times1500\times87.5}{60\times1000}$ ⇨ $\mathbf{BP_1 = 13.74\ kW}$

INDICATED POWER1:

Indicated Power1 $= BP - BP_1$ ⇨ $IP_1 = 19.63 - 13.74$ ⇨ $\mathbf{IP_1 = 5.89kW}$

BRAKE POWER WHEN CYLINDER TWO CUT:

TORQUE2:

TORQUE2:

$W_{B2} = \frac{T_{B2}}{R}$ ⇨ $T_{B2} = W_{B2} \times R$ ⇨ $T_{B2} = 180 \times 0.5$ ⇨ $\mathbf{T_{B2} = 90Nm}$

BRAKE POWER2:

$BP_2 = \frac{2\pi NT_{B2}}{60\times1000}$ ⇨ $BP_2 = \frac{2\times\pi\times1500\times90}{60\times1000}$ ⇨ $\mathbf{BP_2 = 14.14\ kW}$

INDICATED POWER2:

Indicated Power2 $= BP - BP_2$ ⇨ $IP_2 = 19.63 - 14.14$ ⇨ $\mathbf{IP_2 = 5.49\ kW}$

BRAKE POWER WHEN CYLINDER THREE CUT:

TORQUE 3 :

$W_{B3} = \frac{T_{B3}}{R}$ ⇨ $T_{B3} = W_{B3} \times R$ ⇨ $T_{B3} = 182 \times 0.5$ ⇨ $\mathbf{T_{B3} = 91Nm}$

BRAKE POWER3:

$BP_3 = \frac{2\pi NT_{B3}}{60\times1000}$ ⇨ $BP_3 = \frac{2\times\pi\times1500\times91}{60\times1000}$ ⇨ $\mathbf{BP_3 = 14.29\ kW}$

INDICATED POWER3:

Indicated Power3 $= BP - BP_3$ ⇨ $IP_3 = 19.63 - 14.29$ ⇨ $\mathbf{IP_3 = 5.34\ kW}$

BRAKE POWER WHEN CYLINDER FOUR CUT:

TORQUE 4 :

$W_{B4} = \frac{T_{B4}}{R}$ ⇨ $T_{B4} = W_{B4} \times R$ ⇨ $T_{B4} = 170 \times 0.5$ ⇨ $\mathbf{T_{B4} = 85Nm}$

BRAKE POWER4:

$BP_4 = \frac{2\pi NT_{B4}}{60}$ ⇨ $BP_4 = \frac{2\times\pi\times1500\times85}{60}$ ⇨ $\mathbf{BP_4 = 13.35\ kW}$

INDICATED POWER4:

Indicated Power4 $= BP - BP_4$ ⇨ $IP_4 = 19.63 - 13.35$ ⇨ $\mathbf{IP_4 = 6.28\ kW}$

TOTAL INDICATED POWER:

$IP = IP_1 + IP_2 + IP_3 + IP_4$ ⇨ $IP = 5.89 + 5.49 + 5.34 + 6.28$ ⇨ $\mathbf{IP = 23\ kW}$

MECHANICAL EFFICIENCY:

$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}}$ ⇨ $\eta_{Mech} = \frac{19.63}{23}$ ⇨ $\mathbf{\eta_{Mech} = 85.35\%}$

HEAT INPUT:

$Q_S = \dot{m}_f \times CV$ ⇨ $Q_S = 0.189 \times 43000$ ⇨ $\mathbf{Q_S = 8127\ \frac{kJ}{min}}$

HEAT LOSS DUE TO THE COOLING WATER

$Q_w = m_w \times C_W \times (T_{w2} - T_{w1})$ ⇨ $Q_w = 18 \times 4.18 \times (50 - 27)$ ⇨ $\mathbf{Q_w = 1730.52\ \frac{kJ}{min}}$

HEAT CARRIED AWAY BY EXHAUST GAS

$Q_g = (m_a + m_f) \times C_g \times (T_{g2} - T_a$ ⇨ $Q_g = (0.189 + 2.27) \times 1.02 \times (600 - 27)$

⇨ $\mathbf{Q_g = 1437.19\ \frac{kJ}{min}}$

HEAT LOSS DUE TO BRAKE POWER:

$P = \frac{2\pi NT}{60}$ ⇨ $BP = 19.63\frac{kJ}{s} \times 60$ ⇨ $\mathbf{BP = 1177.8\ \frac{kJ}{min}}$

UNACCOUNTED LOSS:

$Q_{ua} = Q_S - Q_w + Q_g + Q_{BP}$ ⇨ $Q_{ua} = 8127 - 1730.52 + 1437.19 + 1177.8$

⇨ $\mathbf{Q_{ua} = 3781.49\ \frac{kJ}{min}}$

PERCENTAGE OF HEAT LOSS:

$$\%Q_w = \frac{Q_w}{Q_S} \Rightarrow \%Q_w = \frac{1730.52}{8127} \Rightarrow \%\mathbf{Q_w = 21.29}$$

$$\%Q_g = \frac{Q_g}{Q_S} \Rightarrow \%Q_g = \frac{1437.19}{8127} \Rightarrow \%\mathbf{Q_g = 17.68}$$

$$\%Q_{BP} = \frac{Q_{BP}}{Q_S} \Rightarrow \%Q_{BP} = \frac{1177.8}{8127} \Rightarrow \%\mathbf{Q_{BP} = 14.49}$$

$$\%Q_{un} = \frac{Q_{un}}{Q_S} \Rightarrow \%Q_{un} = \frac{3781.49}{8127} \Rightarrow \%\mathbf{Q_{un} = 46.53}$$

13. During Morse Test experiment on a six cylinder petrol engine the brake power output was found 50 kW when all cylinders run at full load. When one by one each cylinder is cut and load is reduced to bring engine back to original speed, the measured brake power outputs are as under. Determine the indicated power of engine and mechanical efficiency of engine.

No. of cylinders	1	2	3	4	5	6
Brake power (kW)	40.1	39.5	39.1	39.6	39.8	40

INDICATED POWER WHEN CYLINDER ONE CUT:

$$\text{Indicated Power1} = BP - BP_1 \Rightarrow IP_1 = 50 - 40.1 \Rightarrow \mathbf{IP_1 = 9.9kW}$$

INDICATED POWER WHEN CYLINDER TWO CUT:

$$\text{Indicated Power2} = BP - BP_2 \Rightarrow IP_2 = 50 - 39.5 \Rightarrow \mathbf{IP_2 = 10.5\ kW}$$

INDICATED POWER WHEN CYLINDER THREE CUT:

$$\text{Indicated Power3} = BP - BP_3 \Rightarrow IP_3 = 50 - 39.1 \Rightarrow \mathbf{IP_3 = 10.9\ kW}$$

INDICATED POWER WHEN CYLINDER FOUR CUT:

$$\text{Indicated Power4} = BP - BP_4 \Rightarrow IP_4 = 50 - 39.6 \Rightarrow \mathbf{IP_4 = 10.4\ kW}$$

INDICATED POWER WHEN CYLINDER FIVE CUT:

$$\text{Indicated Power5} = BP - BP_5 \Rightarrow IP_5 = 50 - 39.8 \Rightarrow \mathbf{IP_5 = 10.2\ kW}$$

INDICATED POWER WHEN CYLINDER SIX CUT:

$$\text{Indicated Power6} = BP - BP_6 \Rightarrow IP_6 = 50 - 40 \Rightarrow \mathbf{IP_6 = 10\ kW}$$

TOTAL INDICATED POWER:

$$IP = IP_1 + IP_2 + IP_3 + IP_4 + IP_3 + IP_4 \Rightarrow IP = 9.9 + 10.5 + 10.9 + 10.4 + 10.2 + 10 \Rightarrow \mathbf{IP = 61.9kW}$$

MECHANICAL EFFICIENCY:

$$\eta_{Mech} = \frac{\text{Brake Power}}{\text{Indicated Power}} \Rightarrow \eta_{Mech} = \frac{50}{61.9} \Rightarrow \boldsymbol{\eta_{Mech} = 80.78\%}$$

PART - B (THEORY)

1. Working of simple carburettor with neat sketch

A device for atomizing, vapourising the fuel (petrol) and mixing it with air in correct proportion.

- Air fuel mixture obtained is called the combustible mixture.
- Process of mixing the petrol with air is called carburetion.

Functions of carburettor

- To atomize , vapourise the fuel and mix it homogeneously with air.
- To supply the require quantity of air- fuel mixture at correct proportion according to the load and speed of engine
- To maintain a small reserve of petrol at constant head
- To provide easy starting of the engine in cold conditions.

Components of carburettor

1. Float, Float Chamber And Needle Valve Assembly

- A float is placed inside the float chamber.
- Petrol level in the float chamber is maintained constant and slightly below the top of the jet by the float and needle valve arrangement.
- The float chamber is vented to the atmosphere.

2. Fuel Jet

- Fuel jet is connected to the float chamber.
- If petrol in the float chamber falls below the required level, the float lowers.
- It is possible if the supply from the fuel tank to the float chamber is less than the fuel supplied by the jet. This happens when the load on the engine increases.
- The needle valve is opened for increased fuel supply from the fuel tank.
- When the required level is reached , the float closes the needle valve.
- As the level of fuel in the float chamber rises, the float also rises.
- It is possible if the supply from the fuel tank to the float chamber is more than the fuel supplied by the jet. This happens when the load on the engine decreases.
- The needle valve closes the fuel inlet and a constant level is maintained.

3. Venturi and Venturi Throat

Venturi is a tube of decreasing cross-section which reaches a minimum at the throat , called “venture throat”.

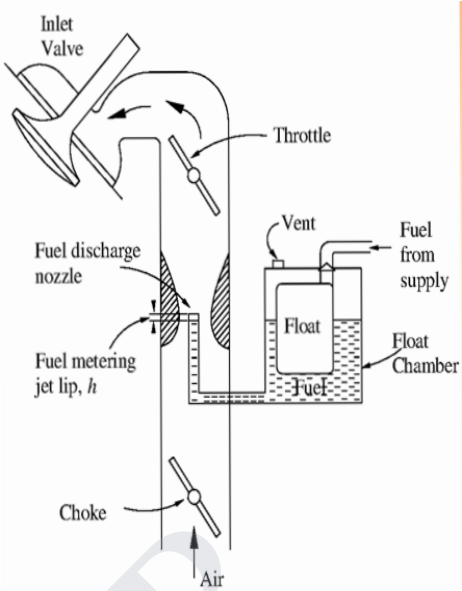
4. Throttle Valve

Control of the throttle valve is normally with the accelerator pedal.

5. Choke valve

It is provided in the air passage before the venturi.

Working :



- During suction stroke , air flows through the choke valve and passes through the venturi; its velocity increases and pressure in the venture throat decreases.
- Now, because the pressure at the float chamber is atmospheric and that at the tip of the jet (i.e. throat) below atmospheric, a pressure differential called "carburetor depression" exists between them.
- This causes the flow of fuel from the float chamber through the jet in fine spray.
- Fuel gets vaporized and an uniform air-fuel mixture is supplied to the cylinder.
- At normal speed, the carburetor delivers a normal mixture of air- fuel ratio 15:1
- The quantity of mixture supplied to the cylinder is controlled by the throttle valve.
- As the throttle is closed, less air flows through the venturi and less is the quantity of air-fuel mixture delivered to the cylinder and hence less is the power developed.
- As the throttle is opened, more quantity of air-fuel mixture is delivered to the cylinder.

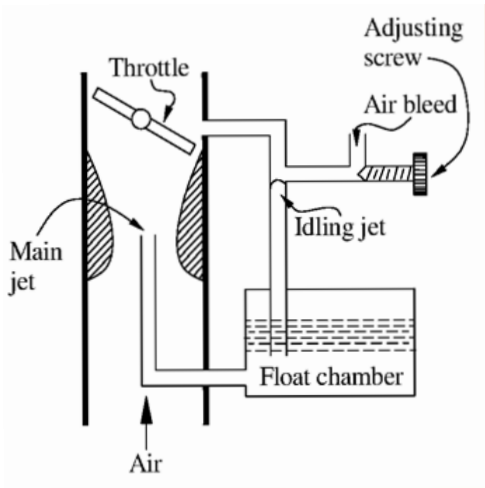
2. Complete carburetor: (i) main metering system (ii) idling system (iii) economizer system (iv) acceleration pump system (v) choke

A simple carburetor is capable to supply a correct air-fuel mixture to the engine only at a particular load and speed. In order to meet the engine demand at various operating conditions, the following additional systems are added to the simple carburetor.

- Idling system
- Auxiliary port system
- Power enrichment by economizer system
- Accelerating pump system
- Chock

Idling System:

During starting or idling, engine runs without load and the throttle valve remains in closed position. Engine produces power only to covercome friction between the parts and a rich mixture is to be fed to the engine to sustain combustion.

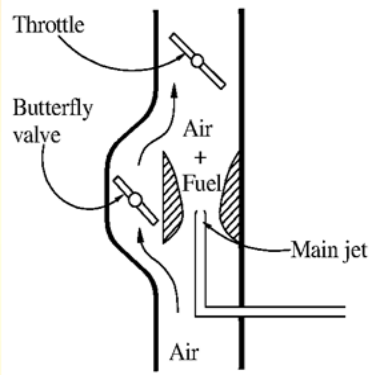


Choke:

During cold starting period, at low cranking speed and before the engine gets warmed up, a rich mixture has to be supplied. The most common method of obtaining this rich mixture is to use a choke valve between the entry to the carburetor and the venture throat.

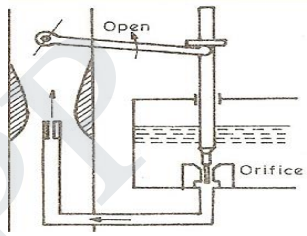
Auxiliary Port System:

During normal power or crusing operation, where the engine runs for most of the period, the fuel economy has to be maintained. Thus, it is necessary to have lower fuel consumption for maximum economy. One such arrangement used is the auxiliary port carburetor as shown, where opening of butterfly valve allows additional air to be admitted and at the same time depression at the venture throat gets reduced, therby decreasing the fuel flow rate



Powerenrichment System:

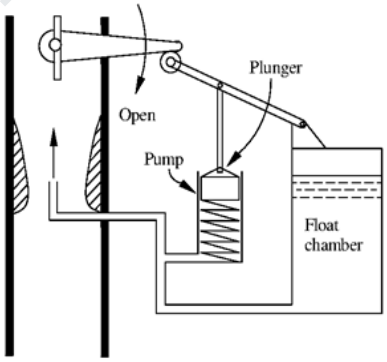
In order to obtain maximum power, the carburetor must supply a rich mixture. This additional fuel required is supplied by a power enrichment system that contains a meter rod economizer that provides a larger orifice opening to the main jet as the throttle is opened beyond a certain point.



Accelerating Pump System:

During sudden acceleration of an engine(e.g., overtaking a vehicle), an extra amount of fuel is momentarily required to supply a rich mixture. This is obtained by an accelerating pump system. It consists of a spring-loaded plunger and the necessary linkage mechanism.

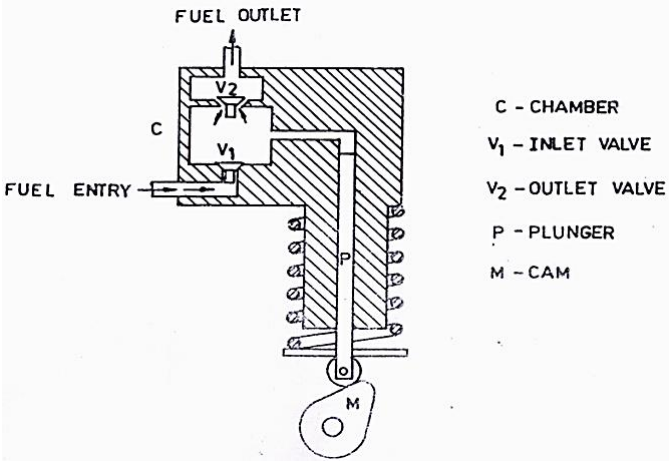
The rapid opening of the throttle moves the plunger into the cylinder, and an additional amount of fuel is forced into the venturi



3. Working of diesel pump & diesel injector with neat sketch

FUEL PUMP

- In the diesel engine , diesel is to be injected in the compressed air at the high pressure.
- So it is necessary to increase the fuel pressure using fuel pump. The fuel pump raises the pressure of the fuel very high .
- Fuel is injected into the cylinder in the atomized from, i.e. in the form of fine spray by means of an injector.
- There are two methods of fuel injection
    - Air injection system - requires an air compressor
    - Airless injection system - does not require air compressor



Diesel Fuel Pump

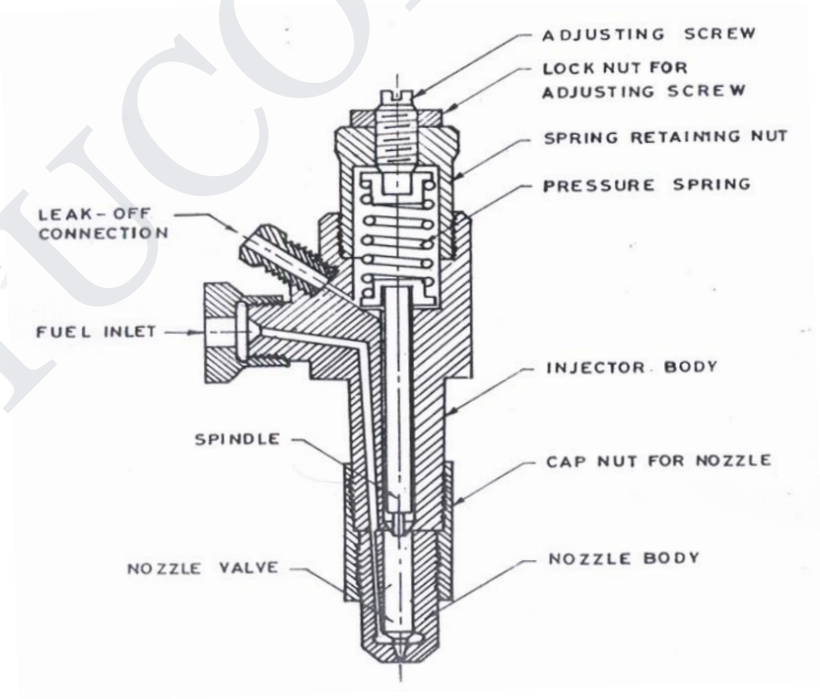
- It is used for airless injection.
- The injection fuel pump consists of a chamber, plunger, spring, inlet and the outlet valves.
- It is operated by a cam fitted over a cam shaft. It is fitted at the head of the cylinder.

Working Principle

- When the plunger moves in the downward direction due to the action of the spring, the fuel comes into the chamber through the inlet valve from the fuel tank.
- When the cam rotates, the plunger moves in the upward direction against the compression of the spring.
- The fuel is forced through the outlet valve from the chamber to the injector.
- The injector forces the fuel into the cylinder of the engine in an atomized form at the end of the compression stroke.

FUEL INJECTOR

- Atomization of diesel is secure through the fuel injector.
- Function of the injector is to split up the fuel into a spray and inject it directly into the engine cylinder such that it is completely consumed, without smoke in the exhaust.
- The fuel pump delivers an accurately metered quantity of fuel under high pressure at the correct moment to the injector.



- The high pressure fuel from the fuel pump is injected into the cylinder with the help of the fuel injector.
- The functions of the fuel injector are:
  - To inject and atomise the fuel to the required degree;
  - To distribute the fuel in such a way that there is complete mixing of fuel and air.

Working Principle:

A typical spring loaded Bosch fuel injector or fuel atomiser consists of

1) Nozzle valve
2) Nozzle body
3) Cap nut for nozzle.
4) Spindle
5) Injector body
6) Pressure spring
7) Spring retainer nut
8) Adjusting screw
9) Lock nut for adjusting screw
10) Fuel inlet
11) Leak-off connection

- The high pressure fuel from the fuel pump enters into the fuel injector through the fuel inlet.
- It acts on the nozzle valve from the bottom.
- The valve is lifted up due to the pressure of the fuel against the spring.
- Fuel is now injected into the combustion chamber of the engine cylinder in the form of fine spray.
- As the pressure of the fuel falls, the spring passes the spindle and the valve is automatically closed by the spring force.
- The amount of fuel injected is regulated by the duration of the open period of the valve.
- The pressure of the spring can be adjusted by means of the spring adjusting screw.
- Due to the upward motion of the valve, a communication between the fuel inlet and the leak- off connection is established.
- Any leakage or overflow of fuel in the valve is taken out through the leak- off connection as shown.

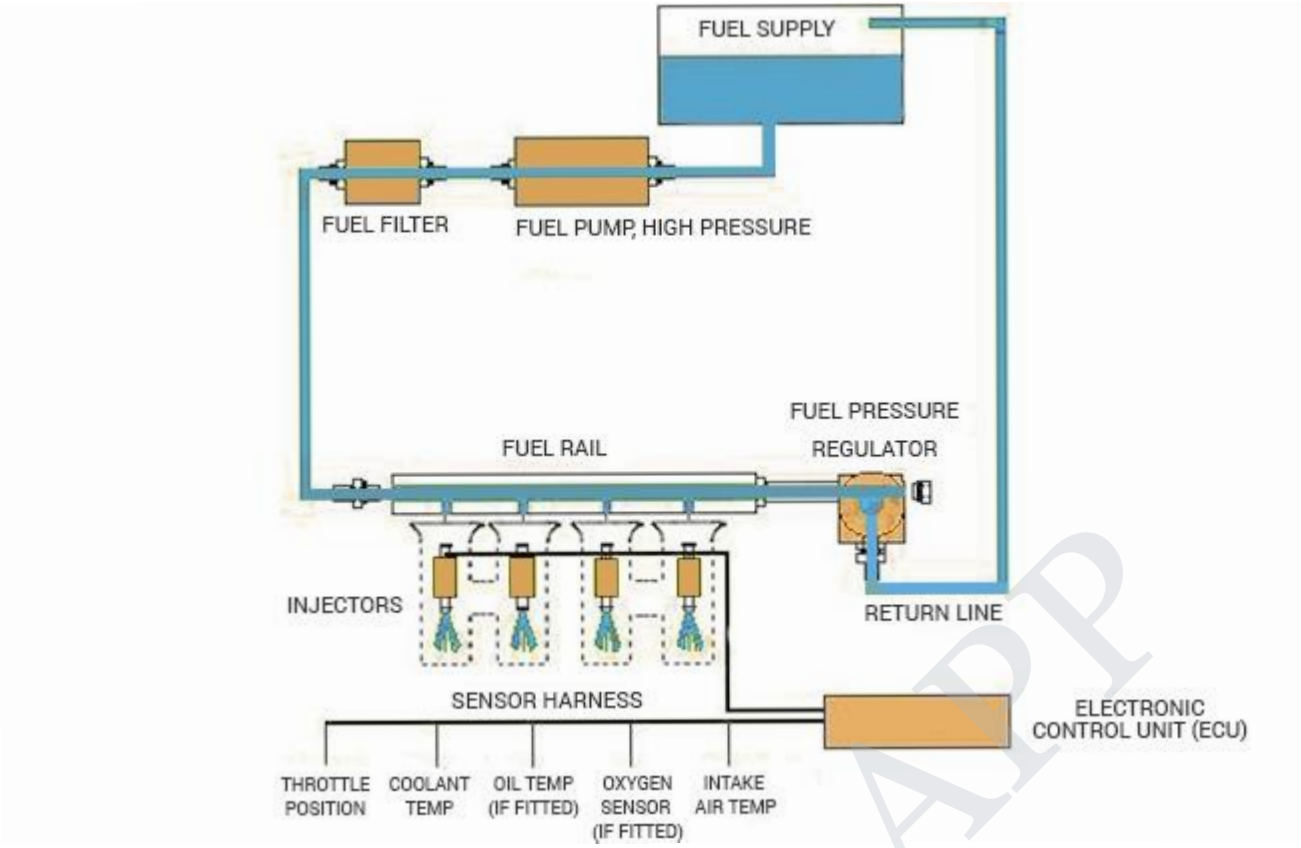
4. Working of Multi point fuel injector:

Principle MPFI:

- Multi point fuel injection system is an electronic system in petrol engine which aims to have efficient combustion with reduction in emmisions.
- Electronic system senses the parameters of engine like speed,load,temperature, rpm to calculte the amount of fuel which is to be injected and the pressure at which the air fuel mixture is to be injected.
- The timing of injection is also taken in account by sensing the crank angle.

Working of MPFI engine:

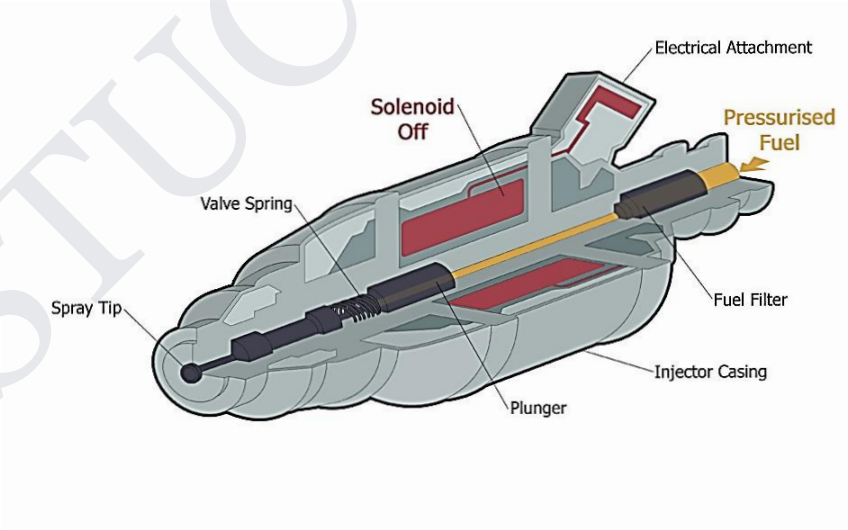
- When you step on the gas pedal, the throttle valve opens up more, letting in more air. The engine control unit (ECU, the computer that controls all of the electronic components on your engine) "sees" the throttle valve open (with the help of Mass airflow sensor) and increases the fuel rate in anticipation of more air entering the engine.
- It is important to increase the fuel rate as soon as the throttle valve opens; otherwise, when the gas pedal is first pressed, there may be a hesitation as some air reaches the cylinders without enough fuel in it.



- Sensors monitor the mass of air entering the engine, as well as the amount of oxygen in the exhaust. The ECU uses this information to fine-tune the fuel delivery so that the air-to-fuel ratio is just right.

KEY PARTS OF MPFI

Fuel Injector



- A fuel injector is nothing but an electronically controlled valve. It is supplied with pressurized fuel by the fuel pump in your car, and it is capable of opening and closing many times per second

Engine Sensors

- In order to provide the correct amount of fuel for every operating condition, **the engine control unit** (ECU) has to monitor a huge number of input sensors. Here are just a few-
- **Mass airflow sensor -** Tells the ECU the mass of air entering the engine.

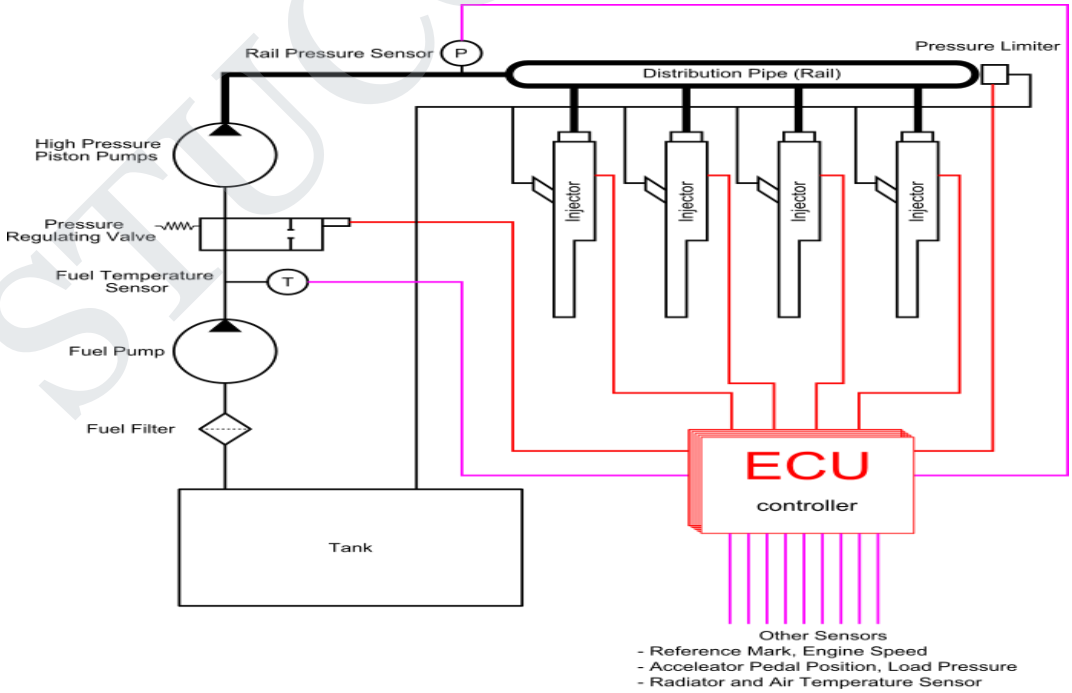
- **Oxygen sensor(s)** - Monitors the amount of oxygen in the exhaust so the ECU can determine how rich or lean the fuel mixture is and make adjustments accordingly
- **Throttle position sensor** - Monitors the throttle valve position (which determines how much air goes into the engine) so the ECU can respond quickly to changes, increasing or decreasing the fuel rate as necessary.
- **Coolant temperature sensor** - Allows the ECU to determine when the engine has reached its proper operating temperature Voltage sensor - Monitors the system voltage in the car so the ECU can raise the idle speed if voltage is dropping (which would indicate a high electrical load).
- **Engine speed sensor** - Monitors engine speed, which is one of the factors used to calculate the pulse width

Advantage of Electronic Fuel injection over carburettor:-

- Better atomization of fuel
- Lower emission of pollutant
- Better flow due to elimination of venture
- Rapid response time with respect to the changes
- Improved fuel efficiency

5. Detailed working of common rail fuel Injection system.

Solenoid or piezoelectric valves make possible fine electronic control over the fuel injection time and quantity, and the higher pressure that the common rail technology makes available provides better fuel atomisation.



❖ To lower engine noise, the engine's electronic control unit can inject a small amount of diesel just before the main injection event ("pilot" injection), thus reducing its explosiveness and vibration, as well as optimising injection timing and quantity for variations in fuel quality, cold starting and so on.

- Some advanced common rail fuel systems perform as many as five injections per stroke.
- Common rail engines require a very short to no heating-up time, depending on the ambient temperature, and produce lower engine noise and emissions than older systems.
- Two common types include the unit injection system and the distributor/inline pump systems. While these older systems provide accurate fuel quantity and injection timing control, they are limited by several factors:
  - They are cam driven, and injection pressure is proportional to engine speed.
  - They are limited in the number and timing of injection events that can be commanded during a single combustion event. While multiple injection events are possible with these older systems, it is much more difficult and costly to achieve.
  - For the typical distributor/inline system, the start of injection occurs at a pre-determined pressure and ends at a pre-determined pressure. This characteristic results from "dumb" injectors in the cylinder head which open and close at pressures determined by the spring preload applied to the plunger in the injector. Once the pressure in the injector reaches a pre-determined level, the plunger lifts and injection starts.
- In common rail systems, a high-pressure pump stores a reservoir of fuel at high pressure — up to and above 2,000 bars (200 MPa; 29,000 psi).
- The term "common rail" refers to the fact that all of the fuel injectors are supplied by a common fuel rail which is nothing more than a pressure accumulator where the fuel is stored at high pressure.
- This accumulator supplies multiple fuel injectors with high-pressure fuel. This simplifies the purpose of the high-pressure pump in that it only needs to maintain a target pressure (either mechanically or electronically controlled).
- The fuel injectors are typically ECU-controlled. When the fuel injectors are electrically activated, a hydraulic valve (consisting of a nozzle and plunger) is mechanically or hydraulically opened and fuel is sprayed into the cylinders at the desired pressure.
- Since the fuel pressure energy is stored remotely and the injectors are electrically actuated, the injection pressure at the start and end of injection is very near the pressure in the accumulator (rail), thus producing a square injection rate.
- If the accumulator, pump and plumbing are sized properly, the injection pressure and rate will be the same for each of the multiple injection events.

Advantages:

1. Reduced noise and vibration.
2. Reduced smoke , particulates and exhaust.
3. Increased fuel economy.
4. Higher power output even at lower rpm.

Disadvantages:

1. High cost due to high pressure pump and ECU.
2. Technology cannot be employed in present engines.

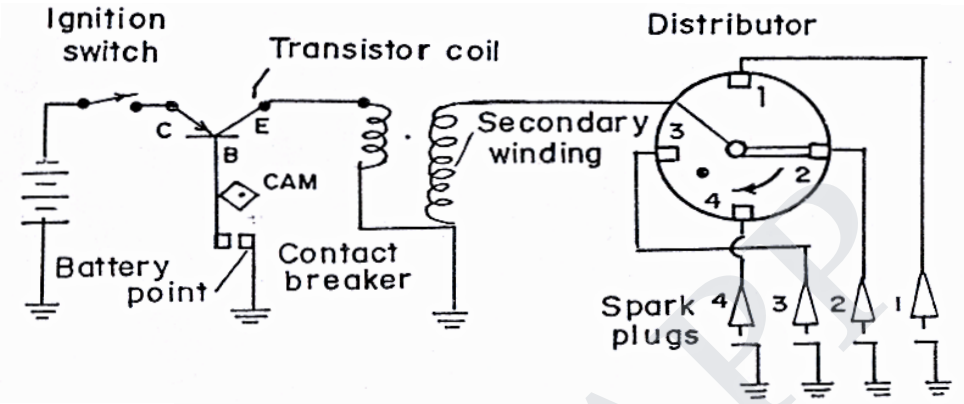
6. Detailed working of battery ignition system and magneto ignition system with neat sketch.

COIL/BATTERY IGNITION SYSTEM

Functions of Ignition System

1. To produce spark at the end of compression stroke to ignite the fuel .
2. To fire the air fuel mixture at correct time by giving electrical power to the spark plug.

Modern SI engines use the battery or coil ignition system. Energy required for establishing the arc is obtained from a 6V or 12V battery.



Construction :

Battery

Electrical power source.

Provides a voltage of 6 or 12 volts.

Ignition coil

Purpose of the ignition coil is to step up the battery voltage (i.e 6 or 12 volts to 20,000 or 30,000 volts) required to produce spark for ignition in the spark plug.

Ignition switch

It acts as a locking switch. Whenever it is in 'ON' position, the current will flow through the circuit.

Contact breaker

It is the mechanical device for making and breaking the primary ignition circuit.

Distributor

It distributes the high voltage to the respective spark plug at regular intervals in the correct sequence.

Condenser

It is connected across the contact breaker.

It is used to avoid excess sparking at contact breaker points and to induce a high voltage in the secondary circuit by causing more rapid contact break of the primary circuit.

Spark plug

It is fitted on the cylinder head of the engine.

Working Principle:

The system has two circuits.

Primary circuit - Battery, Primary coil, Condensor & Contact breaker

Secondary circuit - Secondary coil , Distributor & Spark plug.

- When the primary circuit is closed by the contact breaker, current begins to flow through the primary coil .

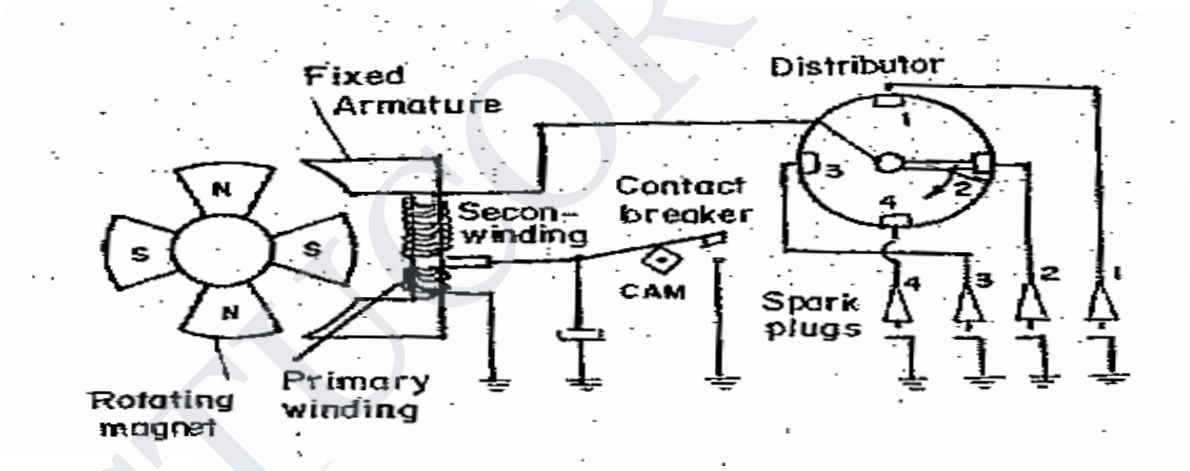
- EMF induced in the secondary coil is proportional to the rate at which the magnetic flux increases due to primary current flow.
- This EMF produced in the secondary coil is not sufficient to produce a spark at the spark plug. When primary circuit is opened by the contact breaker, the magnetic field collapses.
- A very high voltage in the range of 10,000 to 20,000 volts is induced in the secondary coil due to the sudden collapse of magnetic field in the coil.
- This high voltage is supplied through the high tension distribution to the corresponding spark plug thereby producing a spark.
- Condenser in the circuit helps to achieve very rapid collapse of the magnetic field for inducing a very high voltage.
- Without a condenser, the magnetic field collapse would be slow, producing secondary voltage.

MAGNETO IGNITION SYSTEM :

Description:

Main difference between coil (battery) and magneto ignition system is that the battery is replaced by a rotating magnet.

- It consist of a switch, magneto, condenser, contact breaker, distributor and spark plugs.
- The magneto consist of a rotating magnet assembly and a fixed armature.
- The armature contains the primary windings and secondary windings.
- The magnet is located on the outer rim of the flywheel



Working Principle:

- Engine is cranked by switching on the ignition switch.
- The engine rotate the magnetic assembly.
- As the magnet rotates, the EMF is induced in the primary circuit and the current flows through the contact breaker point.
- When the primary circuit is opened by the contact breaker, the magnetic field collapses.
- A very high voltage is induced in the secondary coil due to the sudden collapse of magnetic field in the coil.
- High voltage is supplied through the high tension distribution to the corresponding spark plug thereby producing a spark.
- Condenser in the circuit helps achieve very rapid collapse of the magnetic field for inducing a very high voltage.
- Condenser also avoids sparking at the contact break points.

S. No.	Coil/Battery ignition system	Magneto ignition system
1	The source of energy is battery	The source of energy is an electro magnet
2	Easy engine starting	Difficult starting
3	Sparking is good even at low speed	Poor sparking at low speed
4	If is the battery is discharged the engine cannot be started.	No such difficult.
5	Application: Used in cars, buses, trucks	Application: Used in motor cycles, scooters

7. Detailed working of electronic ignition system.

TYPES OF ELECTRONIC IGNITION SYSTEM

Electronic Ignition System is as follow :

(a) Capacitance Discharge Ignition system

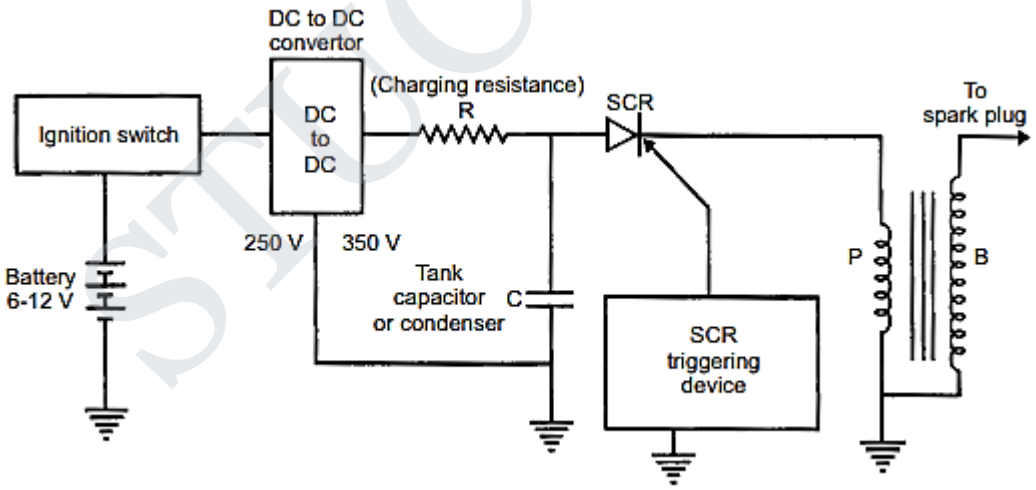
(b) Transistorized system

(c) Piezo-electric Ignition system

(d) The Texaco Ignition system

Capacitance Discharge Ignition System

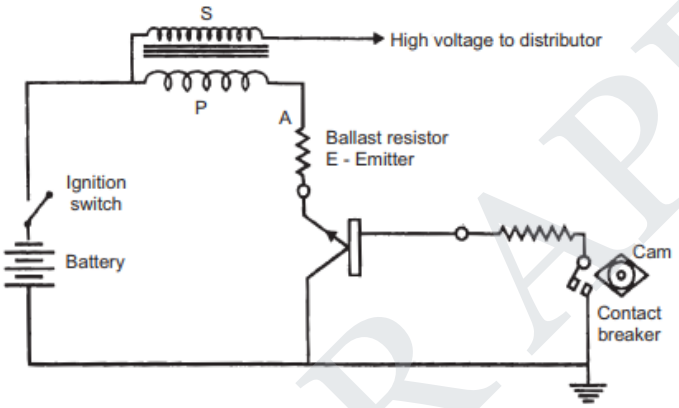
It mainly consists of 6-12 V battery, ignition switch, DC to DC convertor, charging resistance, tank capacitor, Silicon Controlled Rectifier (SCR), SCR-triggering device, step up transformer, spark plugs. A 6-12 volt battery is connected to DC to DC converter i.e. power circuit through the ignition switch, which is designed to give or increase the voltage to 250-350 volts. This high voltage is used to charge the tank capacitor (or condenser) to this voltage through the charging resistance. The charging resistance is also so designed that it controls the required current in the SCR.



Depending upon the engine firing order, whenever the SCR triggering device, sends a pulse, then the current flowing through the primary winding is stopped. And the magnetic field begins to collapse. This collapsing magnetic field will induce or step up high voltage current in the secondary, which while jumping the spark plug gap produces the spark, and the charge of air fuel mixture is ignited.

Transistorized Assisted Contact (TAC) Ignition System

This system incorporates a normal mechanical breakers, which drives a transistor to control the current in the primary circuit. Since a very small breaker current is used, erosion of the contacts is eliminated so that good coil output is maintained. Also it provides accurate spark timing for a much longer period. When a low inductive coil and ballast resistor are used with this system, excessive contact arcing produced by the high primary current is also eliminated. The basic principle of a breaker-triggered, inductive, semiconductor ignition system is illustrated in Fig. where a transistor works as of the contact breaker, by acting as a power switch to make and break the primary circuit. The transistor performs as a relay, which is operated by the current supplied by a cam-operated control switch and thereby called as breaker-triggered.



A small control current passes through the base-emitter of the transistor when the contact breaker is in closed condition. This switches-on the collector-emitter circuit of the transistor and allows full current to flow through the primary circuit to energize the coil. The flow of current, at this stage, in the control circuit and transistor base is governed by the total and relative values of the resistors R\ and R2. These resistance values are chosen to provide a control current of about 0.3 A, which is sufficient to provide a self-cleaning action of the contact surfaces without overloading the breaker. When the spark is required, the cam opens the contact to interrupt the base circuit, which causes the transistor to switch-off. With sudden opening of the primary circuit a high voltage is induced into the secondary, which produces a spark at the plug.

Advantages

(a) The low breaker-current ensures longer life.

(b) The smaller gap and lighter point assembly increase dwell time minimize contact bouncing and improve repeatability of secondary voltage.

(c) The low primary inductance reduces primary inductance reduces primary current drop-off at high speeds.

Disadvantages

(a) As in the conventional system, mechanical breaker points are necessary for timing the spark.

(b) The cost of the ignition system is increased.

(c) The voltage rise-time at the spark plug is about the same as before.

Piezo-electric Ignition System

The development of synthetic piezo-electric materials producing about 22 kV by mechanical loading of a small crystal resulted in some ignition systems for single cylinder engines. But due to difficulties of high mechanical loading need of the order of 500 kg timely control and ability to produce sufficient voltage, these systems have not been able to come up.

The Texaco Ignition System

Due to the increased emphasis on exhaust emission control, there has been a sudden interest in exhaust gas recirculation systems and lean fuel-air mixtures. To avoid the problems of burning of lean mixtures, the Texaco Ignition system has been developed. It provides a spark of controlled duration which means that the spark duration in crank angle degrees can be made constant at all engine speeds. It is a AC system. This system consists of three basic units, a power unit, a control unit and a distributor sensor. This system can give stable ignition up to A/F ratios as high as 24 : 1.

Following are the advantages of electronic ignition system :

(a) Moving parts are absent-so no maintenance.

(b) Contact breaker points are absent-so no arcing.

(c) Spark plug life increases by 50% and they can be used for about 60000 km without any problem.

(d) Better combustion in combustion chamber, about 90-95% of air fuel mixture is burnt compared with 70-75% with conventional ignition system.

(e) More power output.

(f) More fuel efficiency.

8. Different methods of lubricating IC engine

Lubricating System of IC Engine:

Moving parts rub against each other causing frictional force because of which heat is generated and the engine parts wear easily.

- Power is lost due to friction.
- To reduce the wear and tear and also the power loss of the moving parts, lubricant is used in between the rubbing surfaces.

Parts to be Lubricated

1. Main crankshaft bearings
2. Big end bearings
3. Gudgeon pin bearings
4. Piston rings and cylinder walls
5. Timing gears
6. Cam shaft and its bearings
7. Valve mechanism
8. Electrical equipment
9. Crank pin
10. Piston pin bushes

Functions of Lubrication System

1. Reduces friction between the rubbing surfaces.
2. Reduces wear and tear of the rubbibg parts.
3. Reduces the temperature of the working parts.
4. Reduces the noise.

5. To keep the rubbing parts clean by removing worn out materials and carbon dust.
6. To act as a sealing between the cylinder and piston and it prevents the leakage of gases.
7. To serve as a cushion against the shocks of the engine.
8. To reduce the power loss due to friction.

Types of Lubricating System:

1.Mist lubrication system or Petroil lubricating system

2. Wet sump lubrication system

(a) Splash lubrication system

(b) Pressure or forced lubrication system

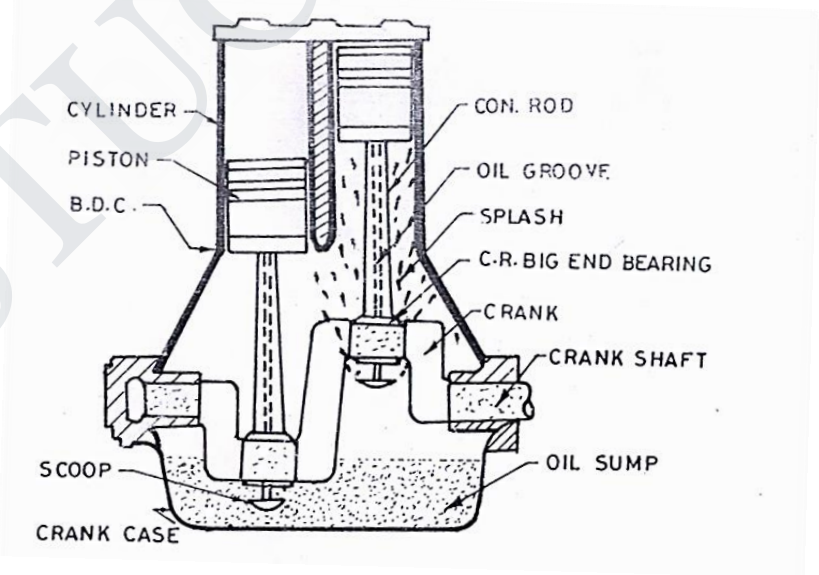
(c) Gravity lubrication system

(d) Semi-pressure lubrication system

3. Dry-sump lubrication system

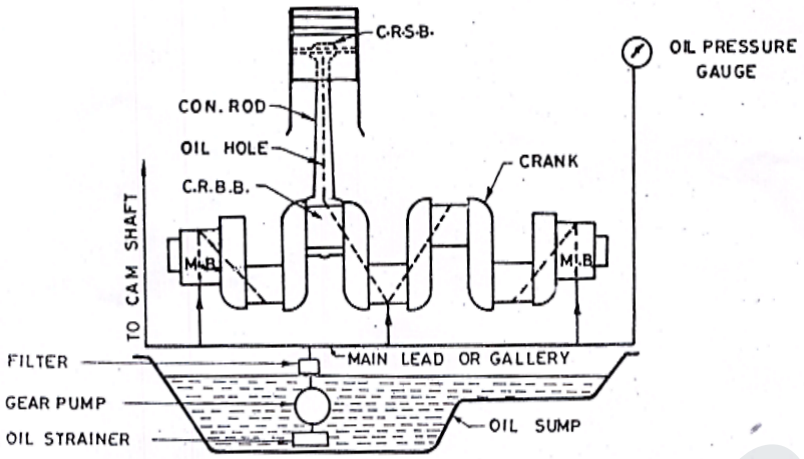
SPLASH LUBRICATION

- An oil sump or reservoir is fixed to the bottom of the crank case.
- During each revolution of the crank, when the piston reaches BDC, the protruding portion of the connecting rod, namely, scoop is dipped in the sump.
- Due to the centrifugal force of the revolving crank, the oil is splashed to the piston, cylinder walls, connecting rod, crank and crank-shaft.
- Grooves are provided in the connecting rod to distribute the oil to the portions of the engine not splashed directly.
- Grooves also facilitate flow of oil back to the sump for reuse.



- To maintain proper oil level in the sump to ensure effective splashing, an oil gauge in the form of a sight glass can be installed in the sump.
- Since the oil is splashed by the centrifugal force of the revolving crank, speed of the engine using this system should not be below 200 rpm.
- It is used in small output stationary IC engines

PRESSURE OR FORCED LUBRICATION SYSTEM



- Oil is stored in oil sump. oil is pumped by the gear pump from the sump through an oil stainer, at a pressure of about
- 2 kgf/m^2.
- High pressure oil is filtered in the filter and is flown to the main lead or gallery.
- From the main lead oil is supplied to various smaller leads.
- Some of the oil from the main lead is supplied at high pressure to the crank shaft main bearing from where the oil flows to the connecting rod big end bearing through the diagonal holes drilled
- Oil from the small end bearing flows to the pistons and cylinder walls at a relative lower pressure.
- Camshaft is also lubricated from one of the smaller leads of the main leads.
- After lubrication, oil falls into the sump.
- An oil pressure gauge is connected to the main gallery.
- The pressure lubrication system is used for engines which are exposed to the high engine loads.
- Supply of the oil under prescribed pressure and maintenance of correct sump are of vital important for the safety as well as efficient engine operation.

MIST (OR) PETROIL LUBRICATING SYSTEM

- It is used for two stroke cycles engines (scooters and motor cycles).
- 3 to 6% lubricating oil is added with petrol in petrol tank.
- Oil and the fuel is injected through the carburetor.
- Petrol is vaporized. Oil in the form of mist, goes into cylinder via crankcase.
- Bearings, piston, connecting rod, cylinder walls are lubricated by this oil mist.
- If insufficient quantity of oil is mixed with petrol, the lubrication is inadequate and under severe conditions may even result in the seizure of the engine.
- On the other hand, the larger proportions would give raise to exhaust smoke and deposition of carbon in the cylinder head and exhaust port.

Advatages

- System is very simple
- Low cost

Disavantages

- Suitable only for small engines.
- If the oil added is less, there will not be sufficient lubrication. It may results in the seizure of the engine.
- If the oil added is more, it will lead to exhaust smoke and carbon deposits in the cylinder, exhaust ports and plugs.

DRY SUMP LUBRICATION SYSTEM

- It is similar to a pressure lubrication system.
- But lubrication oil is not kept in the oil sump, hence it is called as dry sump system.
- In this system, the oil is kept in a separate oil tank.
- oil is fed to the engine by using pressure pump.
- The oil which falls into the sump of the engine during working is sent back to the tank by means of a separate pump (scavenging pump)
- This system is applicable in aircrafts.

9. Different methods of Cooling IC engine

AIR COOLING SYSTEM

- Air is made to contact with the cylinder block and head and takes away the excessive heat.
- To increase the air contact area on the cylinder blocks and cylinder head fins or ribs are made.
- Air cooling is mostly used in small cars as well as engines of motor cycles and scooters.
- Also aircooling is used for heavy diesel engines as used in Eicher tractor and Kirloskar generator set engines.
- Motor cycles – flow of air is achieved by forward motion.
- Cars – air is thrown over the engine by a fan or blower built into the engine fly wheel.

Advantages

- Engine weight is less (no radiator, water, water pump)
- Engine requires less space.
- Design of air-cooled system is simple and less costly.
- No danger of leakage of the coolant.
- No water freezing hazard in air-cooling system.
- No maintenance of water pump fan, fan-belt, hoses etc.
- Installation is easy - it doesn't require radiator, headers & piping connections.

Disadvantages

- Air cooling engine produces more sound (no water jacket for damping sound).
- Volumetric efficiency is less.
- Air cooling system is not suitable for multi-cylinder engines.

WATER COOLING SYSTEM

- Water is used to take away the heat.
- Water is circulated through water jackets around combustion chambers, cylinders, valve seat and valve stem.
- Water takes heat of the combustion and it is cooled by the radiator fan for water recirculation.

Main parts in the water cooling system are:

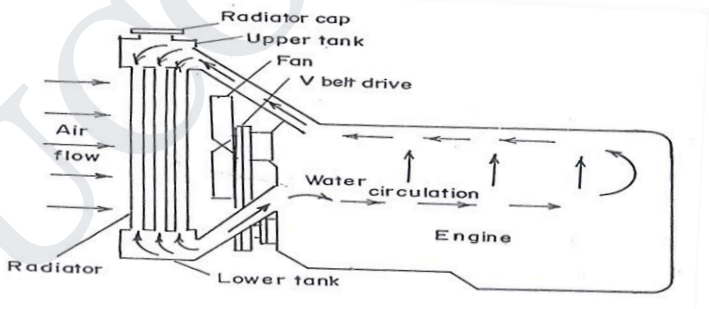
1. Radiator
2. Pressure cap
3. Fan and fan belt
4. Water jackets in cylinder block and head
5. Thermostat
6. Hoses
7. Water
8. Water pump

a) THERMO-SYPHON/NATURAL COOLING SYSTEM

Principle:Water circulates by virtue of pressure difference arising from the difference in density between the hot water in the jackets and the cold water in the radiator.

Working

- Force required to circulate the water through the system is the difference in pressure head due to hot and cold water.
- Density difference between the hot and cold water is the basic working principle of thermo-syphon system. Density of hot water is lesser than the cold water.



- Water when heated up becomes hot and light. Hence it moves up.
- At the same time, the water at the top being cool and heavy ; it moves down.
- In this way circulation starts by itself is known as thermo-syphon cooling system.

LIMITATIONS

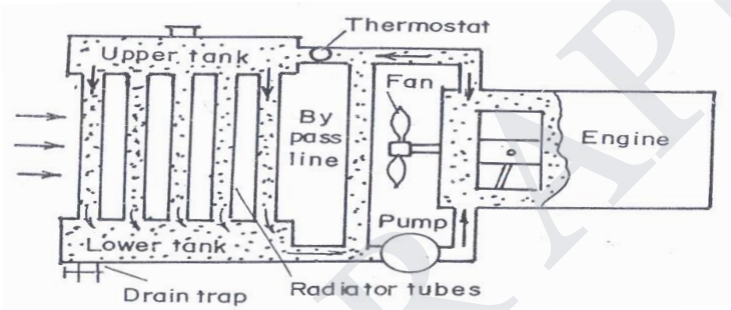
1. More quantity of water is required.
2. Complete system will stop working in case of damage or choked hose pipe.
3. Dead weight of vehicle increases for carrying more water and bigger radiator.
4. The use of this system is recommended for small capacity engines
5. Engine should be placed as low as possible in relation to the radiator since the force causing the flow is limited by the temperature difference of hot and cold water.
6. The water level in the system should not fall below the level of the delivery pipe, otherwise the circulation of water in the system will stop.

b) FORCED WATER COOLING SYSTEM

- Most commonly used systems for cooling the engine.
- A water pump is used to pump the water from the lower tank of radiator to cylinder and cylinder heads to water jackets.
- Water is circulated with the help of pump. The pump is driven by means of a belt which is driven by the crank shaft.

ADVANTAGES

- As water is in direct touch with cylinder walls and cylinder head, it takes away heat quickly.
- For multi-cylinder engines, it is more suitable than air cooling.
- This system is cheap because water is cheap and easily available.
- Engine temperature can be controlled properly due to use of thermostat.



- Water jackets reduce the engines noise.
- Engine with water cooled system can be fitted in any position in the vehicle.
- Uniform cooling of calve, cylinder head and cylinder is possible with water cooling.

DISADVANTAGES

- Weight of radiator with water pump, etc., increases dead weight of the vehicles.
- Water freezes at zero degree temperature.
- Water boils and evaporates at 100^0C.
- Water corrodes the metal parts in the system.
- Pump requires considerable power compared with the power required for the fan.
- Initial and maintenance are higher.
- It is dependent on supply of water.
- Serious damage may be caused in the case of failure of the cooling system.

ANTI-FREEZE SOLUTIONS

- To prevent the cooling water from freezing down, some chemicals known as anti-freeze solutions are mixed up with water.
- If water freezes in the engine, the resulting expanding force is sufficient to crack cylinder block and radiator.
- Anti-freeze solutions are added and mixed with cooling water to prevent its freezing.

AIR COOLING SYSTEM Vs WATER COOLING SYSTEM

AIR COOLING SYSTEM	WATER COOLING SYSTEM
Weight of the engine is less	Weight is more due to water in the radiator
No problem of leakage or freezing of water	Both the problem is exist
More noise	Less noise as water dampens the vibration
Maintenance of the cooling system is easier	Maintenance is difficult
Control of temperature is different	It can be easily controlled by fitting a thermostat
The heat transfer rate in the system is less	The heat transfer rate in the system is more
Air cooling system is mostly used in small vehicles like scooters, mopeds, motor cycles etc…	Water cooling system is used in medium and high capacity engines like car, bus, trucks etc…

10. Working Spark Plug and Crank Compression Ignition

SPARK PLUG

- Spark plug transforms the required voltage generated by the ignition system into a spark within the combustion chamber of the cylinder.
- It consists of a casing which is screwed in the combustion chamber.
- An insulated electrode is sealed within the casing to prevent leakage to the spark plug gap.
- A second electrode is fastened to the grounded part of the casing.
- At the end of compression stroke, the charge is ready for ignition.
- The spark that is formed, jumps the gap between ends of the two electrodes.
- The gap between them varies from 0.45 to 0.65mm

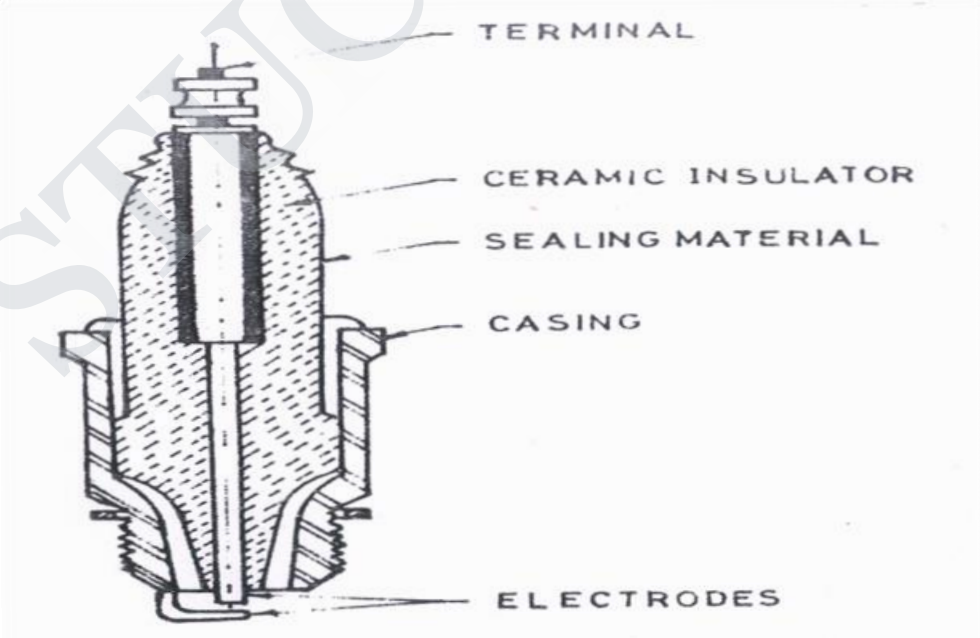


Fig. 4.15 Spark Plug

COMPRESSION IGNITION SYSTEM

- No special arrangement is necessary for ignition, since the temperature
- of the compressed air is higher than the ignition temperature of diesel.
- Diesel is injected into the cylinder before the end of the compression stroke.
- As diesel particles come in contact with the hot compressed air, they vaporize
- and ignite.

11. Detailed explanation of supercharger and turbocharger.

The power output of an engine depends upon the amount of air inducted per unit time and the degree of utilization of the air, and the thermal efficiency of the engine

Three possible methods utilized to increase the air consumption of an engine are as follows.

- Increasing the piston displacement: This increases the size and weight of the engine, and introduces additional cooling problems
- Running the engine at higher speeds: This results in increased mechanical friction losses and imposes greater inertia stresses on engine parts.
- Increasing the density of the charge: This allows a greater mass of the charge to be inducted into the same volume.

Supercharging:

The method of increasing the air capacity of an engine is known as supercharging. The device used to increase the air density is known as supercharger. Supercharger is merely a blower or a compressor that provides a denser charge to the engine.

SI Engine

In an IC engine, Supercharging in SI engine is employed only in aircraft and racing car engines. Apart from supercharging is the process of improving the volumetric efficiency of the engine by using the power of engine, supercharging results in an increase in the intake temperature of the engine. This reduces the ignition delay and increases the flame speed. Both these effects result in a greater tendency to knock or pre-ignite. For this reason, the supercharged petrol engines employ lower compression ratios.

CI Engines

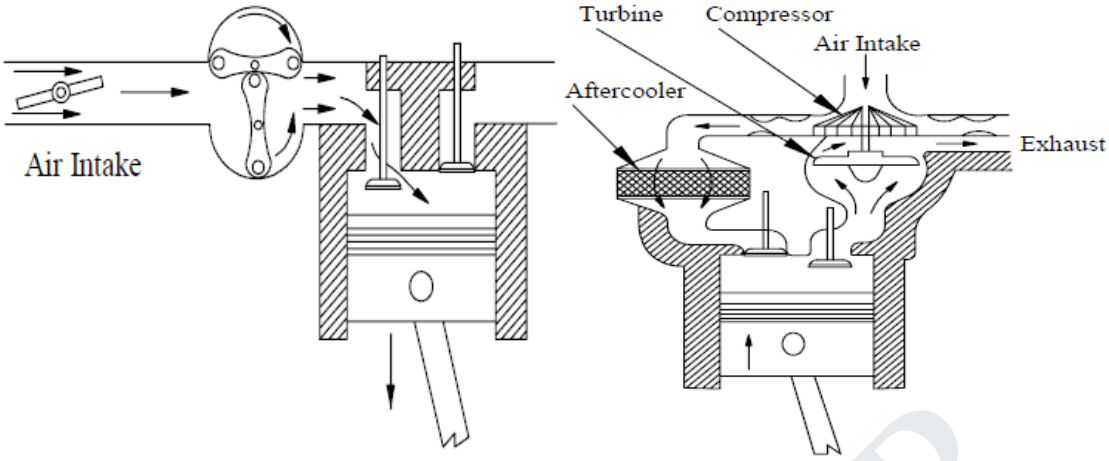
In case of CI engines, supercharging does not result in any combustion problem, rather it improves combustion

Increase of pressure and temperature of the inducted air reduces ignition delay and hence the rate of pressure rise results in a better, quieter and smoother combustion.

Mechanical Supercharger:

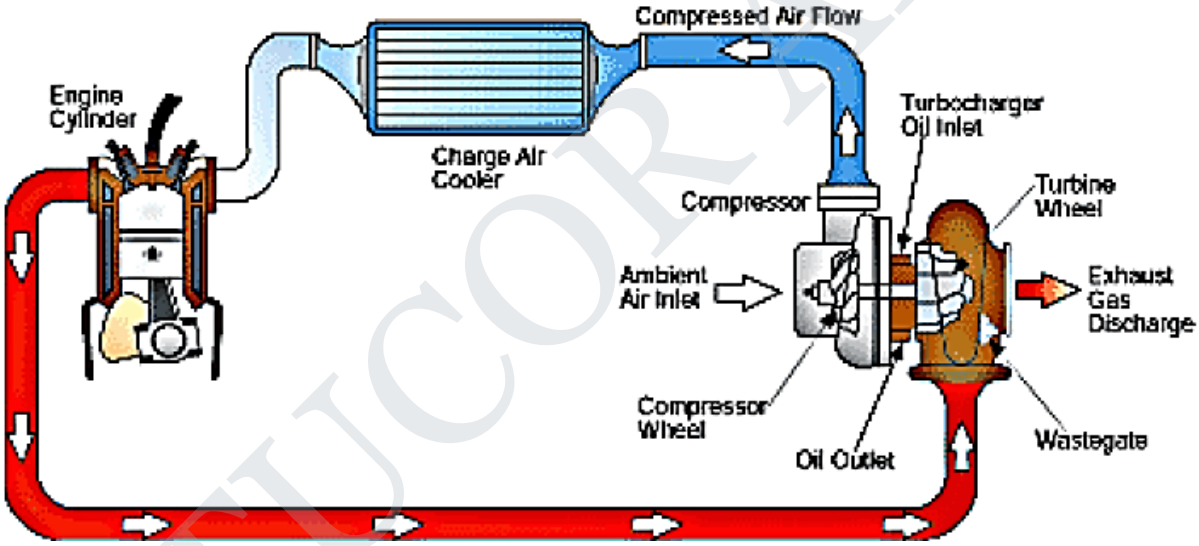
A supercharged engine gives better power for the same amount of fuel that a non-supercharged engine does. A turbine which acts as a compressor is connected to the crankshaft through a belt. When the

combustion begins and the crankshaft rotates the belt connecting the compressor rotates sucking in more air and improving the air fuel mixture. Thus improving the power output for the same fuel quantity.



Turbocharger:

The blower/compressor and the turbine are mounted on the same shaft. The compressor is run by the turbine, and the turbine in turn is run by exhaust gaese.



The difference between supercharger and turbocharger is the turbocharger get power from exhaust gas to run the compressor through turbine. In petrol engine combustion pressure is low so the energy in exhaust gas is also low compared to diesel engine so turbocharger for petrol engine not so efficient.

The advantage of supercharging is better combustion and so the total power produced by engine is also increased. This advantage is due to availability of higher oxygen during the combustion which ensures better combustion.

Supercharger and turbocharger are provided for the same function to increase volumetric efficiency. But the supercharger commonly used in petrol engine and turbocharger used in diesel engine.

The difference between the two devices is their source of energy.

Turbochargers are powered by the mass-flow of exhaust gases driving a turbine.

Superchargers are powered mechanically by belt- or chain-drive from the engine's crankshaft.

Superchargers increase intake by compressing air above atmospheric pressure, without creating a vacuum. This forces more air into the engine, providing a "boost." With the additional air in the boost, more fuel can be added to the charge, and the power of the engine is increased. Supercharging adds an average of 46 percent more horsepower. In high-altitude situations, where engine performance deteriorates because the air has low density and pressure, a supercharger delivers higher-pressure air to the engine so it can operate optimally. To pressurize the air, a supercharger must spin rapidly -- more rapidly than the engine itself. Making the drive gear larger than the compressor gear causes the compressor to spin faster. Superchargers can spin at speeds as high as 50,000 to 65,000 rotations per minute (RPM). As the air is compressed, it gets hotter, which means that it loses its density and can not expand as much during the explosion. This means that it can't create as much power when it's ignited by the spark plug. For a supercharger to work at peak efficiency, the compressed air exiting the discharge unit must be cooled before it enters the intake manifold. The intercooler is responsible for this cooling process. Intercoolers come in two basic designs: air-to-air intercoolers and air-to-water intercoolers. Both work just like radiator, with cooler air or water sent through a system of pipes or tubes. As the hot air exiting the supercharger encounters the cooler pipes, it also cools down. The reduction in air temperature increases the density of the air, which makes for a denser charge entering the combustion chamber.

WORKED EXAMPLES

Example 11.1. *A four cylinder four-stroke engine having diameter and length of stroke as 100 m and 120 mm respectively is running at 1800 r.p.m. Its carburettor venturi has a 28 mm throat. Assuming co-efficient of air flow 0.8, density of air 1.2 kg/m³ and volumetric efficiency of the engine as 75 per cent, determine the suction at the throat.*

Solution. *Given* : $D = 100$ mm $= 0.1$ m ; $L = 120$ mm $= 0.12$ m ; $N = 1800$ r.p.m. ;
Throat diameter, $d_2 = 28$ mm $= 0.028$ m ; $C_{da} = 0.8$; $\rho_a = 1.2$ kg/m³ ; $\eta_{vol} = 75\%$.

Suction at the throat Δp_a :

Stroke volume $= \frac{\pi}{4} \times (0.1)^2 \times 0.12 \times 4 = 0.00377$ m³

Actual volume per strokes $= \eta_{vol} \times 0.00377 = 0.75 \times 0.00377 = 0.00283$ m³

∴ Actual volume sucked per second

$$= 0.00283 \times \frac{1800}{2} \times \frac{1}{60} = 0.04245 \text{ m}^3/\text{s}$$

$$\dot{m}_a = 0.04245 \times 1.2 = 0.05094 \text{ kg/s}$$

As the initial temperature and pressure are not given, the problem is solved by approximate method i.e., neglecting compressibility of the air.

$$\dot{m}_a = C_{da} \times A_2 = \sqrt{2\rho_a \Delta p_a} \qquad \text{...[Eqn. 11.4]}$$

$$0.05094 = 0.8 \times \frac{\pi}{4}(0.028)^2 \sqrt{2 \times 1.2 \times \Delta p_a}$$

$$= 7.63 \times 10^{-4} \sqrt{\Delta p_a}$$

$$\therefore \quad \Delta p_a = \left(\frac{0.05094}{7.63 \times 10^{-4}}\right)^2 = 4457 \text{ N/m}^2 = 0.04457 \text{ bar.} \quad \textbf{(Ans.)}$$

Example 11.2. *A spark ignition engine on test consumes 5 kg/h of petrol when running on an air-fuel ratio of 16 : 1. The engine uses a single-jet carburettor having a fuel orifice area of 2 sq mm and the tip of the jet is 5 mm above the level of petrol in the float chamber, when the engine is not running. Calculate the depression in the venturi throat to maintain the required fuel flow rate through the carburettor. Assume specific gravity of petrol as 0.75 and the coefficient of discharge of the fuel orifice as 0.8. What area of venturi throat will be required to maintain the desired flow rate ? Density of air is 1.20 kg/m³ and the coefficient of discharge for venturi throat is 0.8. Neglect compressibility of air.* **(Roorkee University, AMIE, S-2000)**

Solution. *Given* : $\dot{m}_f = \frac{5}{3600} = 0.001389$ kg/s ; A/F ratio = 16 : 1 ;
Fuel orifice area, $A_f = 2$ mm² $= 2 \times 10^{-6}$ m² ; $z = 5$ mm $= 0.005$ m ;
Sp. gr. of petrol = 0.75 ; $C_{df} = 0.8$, $\rho_a = 1.2$ kg/m³ ; $C_{da} = 0.8$.

Depression in venturi throat, Δp_a :

The actual fuel flow rate is given by,

$$\dot{m}_f = C_{df} \cdot A_f \sqrt{2\rho_f(\Delta p_a - gz\rho_f)} \qquad \text{...[Eqn. (11.7)]}$$

where Δp_a is in N/m².

or $$0.001389 = 0.8 \times (2 \times 10^{-6}) \sqrt{2 \times (0.75 \times 1000)(\Delta p_a - 9.81 \times 0.005 \times 0.75 \times 1000)}$$

or $$\frac{0.001389}{0.8 \times 2 \times 10^{-6}} = 38.73\sqrt{\Delta p_a - 36.79}$$

or $$\Delta p_a - 36.79 = \left(\frac{0.001389}{0.8 \times 2 \times 10^{-6} \times 38.73}\right)^2$$

or $$\Delta p_a = \mathbf{539.2\ N/m^2.}\quad \textbf{(Ans.)}$$

Throat area, A_t :

Air flow rate, $$\dot{m}_a = \frac{5}{3600} \times 16 = 0.02222\ \text{kg/s}$$

Also, $$\dot{m}_a = C_{da} \times A_t \sqrt{2\rho_a \Delta p_a} \quad \text{...(Eqn. (11.4)}$$

(Here $A_2 = A_t$)

$$0.02222 = 0.8 \times A_t \sqrt{2 \times 1.2 \times 539.21}$$

$$\therefore \quad A_t = \frac{0.02222}{0.8\sqrt{2 \times 1.2 \times 539.21}} = 7.7209 \times 10^{-4}\ \text{m}^2$$

$$= \mathbf{7.7209\ cm^2.}\quad \textbf{(Ans.)}$$

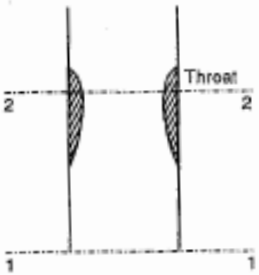


Fig. 11.23

Example 11.3. *The following data relate to a petrol engine :*

Petrol consumed per hour	*= 7.2 kg*
The specific gravity of the fuel	*= 0.75*
The temperature of air	*= 27°C*
The air fuel ratio	*= 1 : 15*
The diameter of the choke tube	*= 24 mm*

The height of top of the jet above the petrol level = 4.2 mm = 0.0042 m in the float chamber

The co-efficient of discharge for air	*= 0.8*
The co-efficient of discharge for fuel	*= 0.7*
Atmospheric pressure	*= 1.013 bar*

Calculate the diameter of the fuel jet of a simple carburettor.

Solution. *Given :* $\dot{m}_f = \frac{7.2}{3600}$ kg/s ; $\rho_f = 0.75 \times 1000 = 750$ kg/m³ ; $T_1 = 27 + 273 = 300$ K ;
A / F ratio = 1 : 15 ; $d_2 = 24$ mm = 0.024 m ; $z = 4.2$ mm = 0.0042 m ;
$C_{da} = 0.8$; $C_{df} = 0.7$; $p_1 = 1.013$ bar.

Diameter of the fuel jet, d_f :

We know that, $$\rho_a = \frac{p_1}{RT_1} = \frac{1.013 \times 10^5}{(0.287 \times 1000) \times 300} = 1.176\ \text{kg/m}^3$$

Air flow rate, $$\dot{m}_a = A_2\, C_{da} \sqrt{2\rho_a \Delta p_a}$$

$$\frac{15 \times 7.2}{3600} = \frac{\pi}{4} \times (0.024)^2 \times 0.8\sqrt{2 \times 1.176 \times \Delta p_a}$$

$$= 5.55 \times 10^{-4}\sqrt{\Delta p_a}$$

$$\therefore \quad \Delta p_a = \left(\frac{15 \times 7.2}{3600 \times 5.55 \times 10^{-4}}\right)^2 = 2922\ \text{N/m}^2$$

Fuel flow rate, $$\dot{m}_f = A_f C_{df} \sqrt{2\rho_f(\Delta p_a - gz\rho_f)} \quad \text{...[Eqn. 11.7]}$$

$$\frac{7.2}{3600} = \frac{\pi}{4}(d_f)^2 \times 0.7\sqrt{2 \times 750(2922 - 9.81 \times 0.0042 \times 750)}$$

$$= 1144.89\,(d_f)^2$$

$$\therefore \quad \mathbf{d_f} = \left(\frac{7.2}{3600 \times 1144.89}\right)^{1/2} = 1.32 \times 10^{-3}\ \text{m} \quad \text{or} \quad \mathbf{1.32\ mm.}\quad \textbf{(Ans.)}$$

Example 11.4. *A simple carburettor under a certain condition delivers 5.45 kg/h of petrol with an air-fuel ratio of 15. The fuel jet area is 2 mm² with a coefficient of discharge of 0.75. If the tip of the fuel jet is 0.635 cm above the level of petrol in the float chamber and the venturi throat coefficient of discharge is assumed to be 0.80, calculate :*

(i) The venturi depression in cm of H_2O necessary to cause air and fuel flow at the desired rate.

(ii) The venturi throat diameter.

(iii) The velocity of air across the venturi throat.

You may take density of air = 1.29 kg/m³ and specific gravity of period = 0.72.

(Madras University)

Solution. *Given :* $\dot{m}_f = \frac{5.45}{3600} = 0.001514$ kg/s ; A/F ratio = 15 ; $A_f = 2$ mm²
$= 2 \times 10^{-6}$ m² ; $C_{df} = 0.75$; $z = 0.635$ cm = 0.00635 m ; $C_{da} = 0.8$;
$\rho_a = 1.29$ kg/m³ ; Sp. gr. of petrol = 0.72.

(i) **Venturi depression, Δp_a :**

$$\dot{m}_f = C_{df} \cdot A_f \sqrt{2\rho_f(\Delta p_a - gz\rho_f)} \quad \text{...[Eqn. (11.7)]}$$

where Δp_a is in N/m².

$$0.001514 = 0.75 \times 2 \times 10^{-6}\sqrt{2 \times (0.72 \times 1000)(\Delta p_a - 9.81 \times 0.00635 \times 0.72 \times 1000)}$$

$$\frac{0.001514}{0.75 \times 2 \times 10^{-6}} = 37.95\sqrt{(\Delta p_a - 44.85)}$$

or $$\Delta p_a - 44.85 = \left(\frac{0.001514}{0.75 \times 10^{-6} \times 37.95}\right)^2 = 707.37$$

$$\therefore \quad \Delta p_a = 752.22\ \text{N/m}^2 = \frac{752.22}{9810}\ \text{m of water} = \mathbf{7.67\ cm\ of\ H_2O.}\quad \textbf{(Ans.)}$$

(ii) **Venturi throat diameter, D_t :**

Air flow rate $= \frac{5.45}{3600} \times 15 = 0.02271$ kg/s

Also, $$\dot{m}_a = C_{da} A_t \sqrt{2\rho_a \Delta p_a}$$

$$\therefore \quad 0.02271 = 0.8 \times A_t \sqrt{2 \times 1.29 \times 752.22}$$

or $$A_t = 6.444 \times 10^{-4}\ \text{m}^2 = \frac{\pi}{4} D_t^2$$

$$\therefore \quad D_t = \left[\frac{6.444 \times 10^{-4} \times 4}{\pi}\right]^{1/2} = 0.0286\ \text{m} = \mathbf{2.86\ cm.}\quad \textbf{(Ans.)}$$

(iii) **Velocity of air across the venturi throat C_t :**

$$C_t \text{ (or } C_2) = \sqrt{\frac{2(gz\rho_f)}{\rho_a}} \qquad \text{...(Eqn. 11.18)}$$

$$= \sqrt{\frac{2 \times 9.81 \times 0.00635 \times (0.72 \times 1000)}{1.29}} = \mathbf{8.34\ m/s.} \quad \textbf{(Ans.)}$$

Example 11.5. *A carburettor ; tested in the laboratory has its float chamber vented to atmosphere. The main metering system is adjusted to give an air-fuel ratio of 15 : 1 at sea level conditions. The pressure at the venturi throat is 0.8 bar. The atmospheric pressure is 1 bar. The same carburettor is tested again when an air cleaner is fitted at the inlet to the carburettor. The pressure drop to air cleaner is found to be 30 mm of Hg when air flow at sea level condition is 240 kg/h. Assuming zero tip and constant coefficient of flow, calculate (i) the throat pressure when the air cleaner is fitted and (ii) air-fuel ratio when the air cleaner is fitted.* **(Bombay University)**

Solution. *Given* : A / F ratio = 15 : 1 at sea level conditions ;

$p_1 = 1$ bar ; $p_2 = 0.8$ bar ;

(i) **The throat pressure when the air cleaner is fitted :**

Quantity of air flowing is same in both the cases.

$$(\dot{m}_a)_{actual} = C_{da} A_t \sqrt{2\rho_a (p_1 - p_2)}$$

When there is *no air cleaner*,

$$\Delta p_a = p_1 - p_2 = 1 - 0.8 = 0.2 \text{ bar}$$

When the *air cleaner is fitted*, let p_t be the throat pressure, then

$$\Delta p_a' = \left[1 - \left(1000 \times 13.6 \times 9.81 \times \frac{30}{1000} \times 10^{-5}\right) - p_t\right] \text{bar}$$

$$= (0.96 - p_t) \text{ bar} \qquad (\because \ 1 \text{ bar} = 10^5 \text{ N/m}^2)$$

For the same air flow and constant coefficients,

$$\Delta p_a = \Delta p_a'$$

$$\therefore \quad 0.2 = 0.96 - p_t$$

or $\quad \mathbf{p_t = 0.76\ bar.}$ **(Ans.)**

(ii) **Air-fuel ratio when the air cleaner is fitted :**

Without air cleaner, $\Delta p_f = \Delta p_a = 0.2$ bar

With air cleaner fitted (with float-chamber still vented to atmosphere),

$$\Delta p_f = 1 - 0.76 = 0.24 \text{ bar}$$

As Δp_f has increased more fuel will flow making the mixture richer.

$$\text{New A / F ratio} = \text{A / F ratio when air cleaner is not fitted} \times \sqrt{\frac{\Delta p_f \text{ without air cleaner}}{\Delta p_f \text{ with air cleaner}}}$$

$$= 15 \times \sqrt{\frac{0.2}{0.24}} = \mathbf{13.69.} \quad \textbf{(Ans.)}$$

Example 11.6. *A simple jet carburettor is required to supply 4.6 kg of air per minute. The pressure and temperature of air are 1.013 bar and 25°C respectively. Assuming flow to be isentropic and compressible and velocity coefficient as 0.8, calculate the throat diameter of the choke for air flow velocity of 80 m/s.*

Solution. *Given* : $\dot{m}_a = \frac{4.6}{60} = 0.0767$ kg/s ; $p_1 = 1.013$ bar, $T_1 = 25 + 273 = 298$ K ;

$C_1 = 0$; $C_2 = 80$ m/s ; $C_v = 0.8$.

Throat diameter d_2 :

Applying S.F.E.E. at sections 1-1 and 2-2, we have

$$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W$$

or $$h_1 = h_2 + \frac{C_2^2}{2} \qquad \left(\because \ Q = 0 ; W = 0 ; C_1 = 0\right)$$

or $$C_2 = \sqrt{2(h_1 - h_2)}$$

$$= \sqrt{2c_p(T_1 - T_2)}$$

$$= \sqrt{2c_pT_1\left(1 - \frac{T_2}{T_1}\right)}$$

$$= \sqrt{2c_pT_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

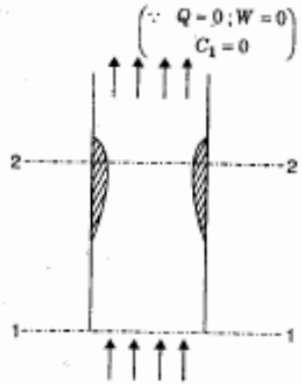


Fig. 11.24

$$\therefore \quad (C_2)_{actual} = C_v \sqrt{2c_pT_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

$$c_p = R\left(\frac{\gamma}{\gamma - 1}\right) = 0.287 \times \frac{1.4}{4.1 - 1} = 1.005 \text{ kJ/kg K}$$

$$80 = 0.8 \sqrt{2 \times 1.005 \times 1000 \times 298\left[1 - \left(\frac{p_2}{1.013}\right)^{\frac{1.4-1}{1.4}}\right]} = 619.15 \sqrt{1 - \left(\frac{p_2}{1.013}\right)^{0.2857}}$$

or $$1 - \left(\frac{p_2}{1.013}\right)^{0.2857} = \left(\frac{80}{619.15}\right)^2 = 0.01669$$

or Throat pressure, $$p_2 = \left[(1 - 0.01669)^{\frac{1}{0.2857}}\right] \times 1.013 = 0.955 \text{ bar}$$

$$\rho_1 = \frac{p_1}{RT_1} = \frac{1.013 \times 10^5}{(0.287 \times 1000) \times 298} = 1.1844 \text{ kg/m}^3$$

Now $$p v^\gamma = \text{constant}$$

or $$\frac{p}{\rho^\gamma} = \text{constant}$$

or $$\frac{p_1}{\rho_1^\gamma} = \frac{p_2}{\rho_2^\gamma}$$

or $$\rho_2 = \rho_1\left(\frac{p_2}{p_1}\right)^{1/\gamma} = 1.1844\left(\frac{0.955}{1.013}\right)^{\frac{1}{1.4}} = 1.1356 \text{ kg/m}^3$$

We know that, $\dot{m}_a (= \rho AC) = \rho_2 A_2 C_2$

$$0.0767 = 1.1356 \times A_2 \times 80 \quad (\text{where } A_2 = \text{throat area})$$

or $$A_2 = 8.443 \times 10^{-4}\ \text{m}^2$$

or $$A_2 = 8.443 \times 10^{-4} = \frac{\pi}{4} d_2^2$$

$\therefore$ *Throat diameter,* $$\mathbf{d_2} = \left(\frac{8.443 \times 10^{-4} \times 4}{\pi}\right)^{1/2} = 0.0328\ \text{m} \quad \text{or} \quad \mathbf{3.28\ cm.}\ \textbf{(Ans.)}$$

Example 11.7. *A simple jet carburettor is required to supply 6 kg of air per minute and 0.45 kg of fuel of density 740 kg/m³. The air is initially at 1.013 bar and 27°C.*

(i) Calculate the throat diameter of the choke for a flow velocity of 92 m/s. Velocity coefficient = 0.8.

(ii) If the pressure drop across the fuel metering orifice is 0.75 of that at the choke, calculate the orifice diameter assuming $C_d = 0.60$. **(AMIE, S-2001 ; Nagpur University)**

Solution. *Given :* $\dot{m}_a = \frac{6}{60} = 0.1$ kg/s ; $\dot{m}_f = \frac{0.45}{60} = 0.0075$ kg/s ; $\rho_f = 740$ kg/m³ ;

$p_1 = 1.013$ bar ; $T_1 = 27 + 273 = 300$ K ; $C_2 = 92$ m/s ; $C_{da} = 0.8$;

$C_{df} = 0.60$.

(i) **Throat diameter, D_2 :**

Velocity of air at venturi throat,

$$C_2 = C_{da}\sqrt{2c_pT_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

$$92 = 0.8\sqrt{2 \times 1.005 \times 1000 \times 300\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{1.4-1}{1.4}}\right]}$$

or $$1-\left(\frac{p_2}{p_1}\right)^{\frac{0.4}{1.4}} = \left(\frac{92}{0.8}\right)^2 \times \frac{1}{2 \times 1.005 \times 1000 \times 300} = 0.021932$$

$\therefore$ $$\left(\frac{p_2}{p_1}\right)^{0.2857} = 0.97807, \quad \therefore \quad \frac{p_2}{p_1} = 0.925$$

or $$p_2 = 1.013 \times 0.925 = 0.937\ \text{bar}$$

Now, $$p_1v_1^\gamma = p_2v_2^\gamma$$

$$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{1/\gamma}$$

$$= \frac{RT_1}{p_1}\left(\frac{1}{0.925}\right)^{\frac{1}{1.4}}$$

$$= \frac{287 \times 300}{1.013 \times 10^5}\left(\frac{1}{0.925}\right)^{0.7143} = 0.898\ \text{m}^3/\text{kg}$$

Now, $$\dot{m}_a = \frac{A_2C_2}{v_2}$$

$\therefore$ Throat area, $$A_2 = \frac{\dot{m}_a \times v_2}{C_2} = \frac{0.1 \times 0.898}{92} = 9.76 \times 10^{-4}\ \text{m}^2 = 9.76\ \text{cm}^2$$

But $$A_2 = \frac{\pi}{4}D_2^2 = 9.26$$

$\therefore$ $$\mathbf{D_2}\ (\text{or } D_t) = \sqrt{\frac{9.76 \times 4}{\pi}} = \mathbf{3.525\ cm.}\ \textbf{(Ans.)}$$

(ii) **Orifice diameter, d_f :**

Pressure drop at venturi = 1.013 – 0.937 = 0.076 bar

Pressure drop at jet = 0.75 × 0.076 = 0.057 bar

Now, $$\dot{m}_f = A_f C_{df}\sqrt{2\rho_f \Delta p}$$

$\therefore$ $$0.0075 = A_f \times 0.6\sqrt{2 \times 740 \times 0.057 \times 10^5} = 1742.68$$

$$A_f = 4.304 \times 10^{-6}\ \text{m}^2 \quad \text{or} \quad 4.304\ \text{mm}^2$$

But, $$A_f = \frac{\pi}{4}d_f^2 = 4.304$$

$\therefore$ $$\mathbf{d_f} = \sqrt{\frac{4 \times 4.304}{\pi}} = \mathbf{2.34\ mm.}\ \textbf{(Ans.)}$$

Example 11.8. *The following data relate to a 4-stroke petrol engine of Hindustan Ambassador :*

Capacity of the petrol engine	*= 1489 c.c.*
Speed at which maximum power is developed	*= 4200 r.p.m.*
The volumetric efficiency (at the above speed)	*= 75 percent*
The air-fuel ratio	*= 13 : 1*
Theoretical air speed at choke (at peak power)	*= 85 m/s*
The co-efficient of discharge for venturi	*= 0.82*
The co-efficient of discharge of the main petrol jet	*= 0.65*
The specific gravity of petrol	*= 0.74*
Level of petrol surface below the choke	*= 6 mm*
Atmospheric pressure and temperature	*= 1.013 bar, 20°C respectively*

An allowance should be made for the emulsion tube, the diameter of which can be taken as 40 percent of the choke diameter.

Calculate the sizes of a suitable choke and main jet.

Solution. *Given :* $V_s = 1489$ c.c. $= 1489 \times 10^{-6}$ m³ $= 0.001489$ m³ ; $N = 4200$ r.p.m ;

$\eta_{vol.} = 75\%$; A/F ratio = 13 : 1 ; $C_t (= C_2) = 85$ m/s ; $C_{da} = 0.82$; $C_{df} = 0.65$;

$\rho_f = 0.74 \times 1000 = 740$ kg/m³ ; $p_1 (= p_a) = 1.013$ bar ; $p_2 (= p_t) = ?$; $T_1 (= T_a) = 20 + 273 = 293$ K ;

$z = 6$ mm $= 0.006$ m ; $d = 0.4\,D$.

Volume of air induced $= \eta_{vol.} \times V_s$

$$= \frac{0.75 \times 0.001489 \times 4200}{2 \times 60} = 0.03909\ \text{m}^3/\text{s}$$

∴ Mass flow of air, $\dot{m}_a = \frac{p_1 v}{RT_1} = \frac{1.013 \times 10^5 \times 0.03909}{0.287 \times 10^3 \times 293} = 0.04709$ kg/s

For compressible flow, velocity at throat,

$$C_t = \sqrt{\left[2T_a c_p \left\{1 - \left(\frac{p_t}{p_a}\right)^{\frac{\gamma-1}{\gamma}}\right\}\right]} \quad \text{...[Eqn. (11.13)]}$$

$$85 = \sqrt{\left[2 \times 293 \times (1.005 \times 10^3)\left\{1 - \frac{p_t}{p_a}\right\}^{\frac{1.4-1}{1.4}}\right]}$$

$$85 = 767.4 \sqrt{\left\{1 - \left(\frac{p_t}{p_a}\right)^{0.2857}\right\}}$$

∴ $$\frac{p_t}{p_a} = \left[1 - \left(\frac{85}{767.4}\right)^2\right]^{\frac{1}{0.2857}} = 0.9576$$

or $$p_t = 1.013 \times 0.9576 = 0.97 \text{ bar}$$

Volume flow of air at choke,

$$v_t = 0.03909 \times \left(\frac{p_a}{p_t}\right)^{\frac{1}{\gamma}}$$

$$= 0.03909 \times \left(\frac{1.013}{0.97}\right)^{\frac{1}{1.4}} = 0.04032 \text{ m}^3/\text{s}$$

∴ $$A_t = \frac{v_t}{C_t \times C_{da}} = \frac{0.04032}{85 \times 0.82} = 0.0005785 \text{ m}^2 = 578.5 \text{ mm}^2$$

Now, $$\frac{\pi}{4}(D^2 - d^2) = 578.5$$

or $$\frac{\pi}{4}[D^2 - (0.4\,D)^2] = 578.5$$

or $$\frac{\pi}{4} \times 0.84\,D^2 = 578.5$$

or Choke dia., **D = 29.61 mm. (Ans.)**

Mass flow of fuel, $\dot{m}_f = \frac{\dot{m}_a}{13} = \frac{0.04709}{13} = 0.003622$ kg/s

$$\dot{m}_f = C_{df} \cdot A_j \sqrt{2\rho_f(\Delta p_a - gz\rho_f)} \quad \text{...[Eqn. (11.7)]}$$

$$0.003622 = 0.65 \times A_j \sqrt{2 \times 740\,[(1.013 - 0.97) \times 10^5 - 9.81 \times 0.006 \times 740]}$$
$$= 1639.75\,A_j$$

∴ $$A_j = 2.209 \times 10^{-6} \text{ m}^2 \quad \text{or} \quad 2.209 \text{ mm}^2$$

$$A_j = \frac{\pi}{4} D_{jet}^2 = 2.209$$

∴ **D_{jet} = 1.68 mm. (Ans.)**

Example 11.9. *The following data refer to a simple carburettor :*

Throat diameter	= 18 mm
Diameter of fuel orifice	= 1.2 mm
Co-efficient of air flow	= 0.82
Co-efficient of fuel flow	= 0.65
Level of petrol surface below the throat	= 6 mm
Density of air	= 1.2 kg/m³
Density of fuel	= 750 kg/m³.

Calculate :

(i) The A/F ratio for a pressure drop of 0.065 bar when the nozzle lip is neglected ;

(ii) The A/F ratio when the nozzle lip is taken into account ;

(iii) The minimum velocity of air or critical air velocity required to start the fuel flow when nozzle lip is provided.

Solution. *Given :* d_a = 18 mm = 0.018 m ;
d_f = 1.2 mm = 0.0012 m ; C_{da} = 0.82 ; C_{df} = 0.65 ; z = 6 mm = 0.006 m ;
ρ_a = 1.2 kg/m³ ; ρ_f = 750 kg/m³

(i) A / F ratio when the nozzle lip is neglected :

Air flow, $$\dot{m}_a = C_{da} A_a \sqrt{2\rho_a \Delta p_a} \quad \text{...[Eqn. (11.4)]}$$

Fuel flow, $$\dot{m}_f = C_{df} A_f \sqrt{2\rho_f \Delta p_a}$$

∴ $$\text{A / F ratio} = \frac{C_{da}}{C_{df}} \cdot \frac{A_a}{A_f} \sqrt{\frac{\rho_a}{\rho_f}}$$

$$= \frac{0.82}{0.65} \times \left(\frac{0.018}{0.0012}\right)^2 \sqrt{\frac{1.2}{750}} = \mathbf{11.35.} \quad \textbf{(Ans.)}$$

$$\left[\because \frac{A_a}{A_f} = \frac{\frac{\pi}{4} d_a^2}{\frac{\pi}{4} d_f^2} = \left(\frac{d_a}{d_f}\right)^2\right]$$

(ii) A / F ratio when the nozzle lip is taken into account :

The *air flow will remain same.* The *fuel flow* will become,

$$\dot{m}_f = C_{df} \times A_f \sqrt{2\rho_f(\Delta p_a - gz\rho_f)} \quad \text{...[Eqn. (11.7)]}$$

∴ $$\text{A / F ratio} = \frac{C_{da}}{C_{df}} \cdot \frac{A_a}{A_f} \sqrt{\frac{\rho_a}{\rho_f}} \sqrt{\frac{\Delta p_a}{\Delta p_a - gz\rho_f}}$$

$$= \frac{0.82}{0.65} \times \left(\frac{0.018}{0.0012}\right)^2 \sqrt{\frac{1.2}{750}} \sqrt{\frac{0.065}{0.065 - (9.81 \times 0.006 \times 750/10^5)}}$$

$$= 11.35 \times \sqrt{\frac{0.065}{0.065 - 0.00044145}} = \mathbf{11.39.} \quad \textbf{(Ans.)}$$

(iii) **Minimum velocity of air, C_2 :**

The *flow of fuel when lip is provided will start only when the minimum velocity of air required to create requisite pressure difference for flow of fuel to overcome nozzle lip exists.*

∴ Pressure difference Δp_a must be *equal to* $gz\rho_f$. Assuming velocity at entrance of venturi, $C_1 = 0$,

$$\frac{p_1}{\rho_a} = \frac{p_2}{\rho_a} + \frac{C_2^2}{2}$$

or
$$\frac{\Delta p_a}{\rho_a} = \frac{C_2^2}{2}$$

or
$$\frac{C_2^2}{2} = \frac{gz\rho_f}{\rho_a}$$

or
$$C_2 = \sqrt{\frac{2gz\rho_f}{\rho_a}} = \sqrt{\frac{2 \times 9.81 \times 0.006 \times 750}{1.2}} = \mathbf{8.58\ m/s.} \quad \textbf{(Ans.)}$$

Note. The **nozzle lip** is *the height of fuel nozzle in the throat above petrol surface in the carburettor. It is provided so that there is no spilling of fuel due to vibration or slight non-horizontal position of carburettor. This would avoid wastage of fuel and fire hazard.*

Example 11.10. *The following data refer to an eight-cylinder four-stroke petrol engine :*

Bore	*= 110 mm*
Stroke	*= 110 mm*
Composition of the fuel used	*= C = 84% ; H_2 = 16%*
Throat diameter of the choke tube	*= 42 mm*
Volumetric efficiency at 300 r.p.m.	*= 75% (referred to 0°C and 1.013 bar)*
The pressure depression	*= 0.12 bar*
The temperature at the throat	*= 15°C*
Characteristic gas constant : For air	*= 287 J/kg K*
For fuel vapour	*= 97 J/kg K*

If chemically correct air-fuel ratio is supplied for combustion, determine :

(i) Fuel consumption in kg/h ;

(ii) The air velocity through the tube.

Solution. *Given :* $D = 110$ mm $= 0.11$ m ; $L = 110$ mm $= 0.11$ m ; $d_a = 42$ mm $= 0.042$ m ; $\eta_{vol.} = 75\%$, $N = 3000$ r.p.m. ; $R_a = 287$ J/kg K ; $R_o = 97$ J/kg K ; $\Delta p_a = 0.12$ bar.

(i) **Fuel consumption in kg/h :**

The volume of mixture supplied at 0°C and 1.013 bar per minute

$$= \frac{\pi}{4} D^2 L \times 8 \times \frac{N}{2} \times \eta_{vol.}$$

$$= \frac{\pi}{4} \times (0.11)^2 \times 0.11 \times 8 \times \frac{3000}{2} \times 0.75 = 9.408\ m^3/\text{min.}$$

Also

$$C + O_2 = CO_2$$
$$12 \quad 32 \quad 44$$

nd

$$2H_2 + O_2 = 2H_2O$$
$$4 \quad 32 \quad 36$$
$$1 \quad 8 \quad 9$$

Thus, air required for combustion of 1 kg of fuel

$$= \left(0.84 \times \frac{32}{12} + 0.16 \times 8\right) \times \frac{100}{23} = 15.3\ \text{kg}$$

Thus A / F ratio $= \frac{m_a}{m_f} = 15.3$

The volume of one kg of air at 0°C and 1.013 bar,

$$v_a = \frac{R_a T}{p} = \frac{287 \times 273}{1.013 \times 10^5} = 0.773\ m^3/\text{kg}$$

Similarly volume of 1 kg of fuel vapour at 0°C and 1.013 bar,

$$v_f = \frac{R_f T}{p} = \frac{97 \times 273}{1.013 \times 10^5} = 0.2614\ m^3/\text{kg}$$

Thus $m_a/\text{min} \times v_a + m_f/\text{min} \times v_f = 9.408$

$15.3\ m_f/\text{min} \times v_a + m_f/\text{min} \times v_f = 9.408$

Thus,
$$m_f/\text{min} = \frac{9.408}{15.3\ v_a + v_f} = \frac{9.408}{15.3 \times 0.773 + 0.2614} = 0.778\ \text{kg/min}$$

∴ Fuel consumption $= 0.778 \times 60 =$ **46.68 kg/h.** **(Ans.)**

(ii) **The air velocity through the tube, C_2 (= C_a)**

Density of air at the throat,

$$\rho_a = \frac{p_2}{R_a T_2}$$

$$= \frac{p_1 - \Delta p_a}{R_a T_2} \qquad (\because\ p_1 - p_2 = \Delta p_a)$$

$$= \frac{(1.013 - 0.12) \times 10^5}{287 \times (15 + 273)} = 1.08\ \text{kg/m}^3.$$

∴ Velocity at the throat in m/s,

$$C_a = \frac{m_a}{A_a \rho_a} = \frac{15.3\ m_f}{\frac{\pi}{4} \times (0.42)^2 \times 1.08}$$

$$= \frac{15.3 \times (0.778/60)}{\frac{\pi}{4} \times (0.042)^2 \times 1.08} = \mathbf{132.59\ m/s.} \quad \textbf{(Ans.)}$$

Example 11.11. *Determine the air-fuel ratio supplied at 4500 m altitude by a carburettor which is adjusted to give an air-fuel ratio of 14 : 1 at sea level where air temperature is 25°C and pressure 1.013 bar.*

The temperature of air decreases with altitude as given by the expression,

$$t = t_s - 0.0064\ h$$

where h is the height in metres and t_s is sea level temperature in °C.

The pressure of air decreases with altitude as per relation :

$$h = 19300 \log_{10}\left(\frac{1.013}{p}\right)$$

where p is expressed in bar at altitude.

Solution.

$$t = t_s - 0.0064\,h$$
$$= 25 - 0.0064 \times 4500 = -3.8°C$$

Now

$$h = 19300 \log_{10}\left(\frac{1.013}{p}\right)$$

$$4500 = 19300 \log_{10}\left(\frac{1.013}{p}\right)$$

$$\therefore \quad \log_{10}\left(\frac{1.013}{p}\right) = \frac{4500}{19300} = 0.2332$$

or

$$\frac{1.013}{p} = 1.711$$

or

$$p = \frac{1.013}{1.711} = 0.592 \text{ bar}$$

Now,

$$\frac{\text{A / F ratio at altitude}}{\text{A / F ratio at sea level}} = \sqrt{\frac{\rho_{\text{altitude}}}{\rho_{\text{sea level}}}}$$

$$\therefore \quad \textbf{A/F ratio at altitude} = 14\sqrt{\frac{p_{\text{alt.}}/RT_{\text{alt.}}}{p_{\text{sea}}/RT_{\text{sea}}}}$$

$$= 14\sqrt{\frac{p_{\text{alt.}} \times T_{\text{sea}}}{p_{\text{sea}} \times T_{\text{alt.}}}}$$

$$= 14\sqrt{\frac{0.592 \times (25 + 273)}{1.013 \times (-3.8 + 273)}}$$

$$= \mathbf{11.26.} \quad \textbf{(Ans.)}$$

WORKED EXAMPLES

Example 12.1. *A six-cylinder, four-stroke diesel engine develops 125 kW at 3000 r.p.m. Its brake specific fuel consumption is 200 g/kWh. Calculate the quantity of fuel to be injected per cycle per cylinder. Specific gravity of the fuel may be taken as 0.85.*

Solution. *Given :* $n = 6$; B.P. = 125 kW ; $N = 3000$ r.p.m. ; b.s.f.c. = 2.00 g/kWh ;
Sp. gr. of fuel = 0.85.

Fuel consumption per hour = b.s.f.c. × B.P.

$$= \frac{200}{1000} \times 125 = 25 \text{ kg}$$

$\therefore$ Fuel consumption per cylinder $= \frac{25}{n} = \frac{25}{6} = 4.167$ kg/h

Fuel consumption per cycle $= \frac{\text{Fuel consumption per cylinder per min.}}{\text{No. of cycles per min.}}$

$$= \frac{(4.167/60)}{(3000/2)} = 4.63 \times 10^{-5} \text{ kg} = 0.0463 \text{ g}$$

$\therefore$ Volume of fuel injected per cycle

$$\therefore \quad = \frac{\text{Fuel consumption per cycle}}{\text{Specific gravity of fuel}}$$

$$= \frac{0.0463}{0.85} = \mathbf{0.05447 \text{ c.c.}} \quad \textbf{(Ans.)}$$

Example 12.2. *A 6-cylinder 4-stroke C.I. engine develops 220 kW at 1500 r.pm. with brake specific fuel consumption of 0.273 kg/kWh. Determine the size of the single hole injector nozzle if the injection pressure is 160 bar and the pressure in the combustion chamber is 40 bar. The period of injection is 30° of crank angle. Specific gravity of fuel = 0.85 and orifice discharge coefficient = 0.9.*

Solution. *Given :* $n = n_0 = 6$; $N = 1500$ r.p.m. ; B.P. = 220 kW, b.s.f.c. = 0.273 kg/kWh
$\theta = 30°$, Sp. gr. of oil = 0.85, $C_f = 0.9$, $\Delta p = p_1 - p_2 = 160 - 40 = 120$ bar.

Diameter of the nozzle orifice, d_0 :

We know that, *actual* fuel velocity of injection,

$$V_f = C_f \sqrt{\frac{2(p_1 - p_2)}{\rho_f}} = C_f \sqrt{\frac{2\Delta p}{\rho_f}} \quad \text{...(Eqn. 12.4)}$$

$$= 0.9 \times \sqrt{\frac{2 \times 120 \times 10^5}{(0.85 \times 1000)}} = 151.23 \text{ m/s}$$

Volume of fuel injected per second,

$$Q_f = \frac{0.273 \times 220}{(0.85 \times 1000) \times 3600} = 1.963 \times 10^{-5} \text{ m}^3/\text{s}$$

Also, volume of fuel injected per second,

$$Q_f = \left[\frac{\pi}{4} d_0^2 \times n_0\right] \times V_f \times \left[\frac{\theta}{360} \times \frac{60}{N}\right] \times \frac{N_i}{60} \quad \text{...(Eqn. (12.5)}$$

(where N_i = No. of injection/min. $= \frac{1500}{2} = 750$)

$$1.963 \times 10^{-5} = \left[\frac{\pi}{4} d_0^2 \times 6\right] \times 151.23 \times \left[\frac{30}{360} \times \frac{60}{1500}\right] \times \frac{750}{60} = 29.694\, d_0^2$$

$$\therefore \quad \mathbf{d_0} = \left(\frac{1.963 \times 10^{-5}}{29.694}\right)^{1/2} = 8.13 \times 10^{-4} \text{m} \quad \text{or} \quad \mathbf{0.813 \text{ mm}.} \quad \textbf{(Ans.)}$$

Example 12.3. *Fuel injection in a single cylinder, 4-stroke cycle C.I. engine running at 650 r.p.m. takes place through a single orifice nozzle and occupies 28° of crank travel. The fuel consumption of the engine is 2.2 kg/hour and the fuel used has a specific gravity of 0.875. If injection pressure is 150 bar and the combustion chamber pressure is 32 bar estimate the volume of fuel injected per cycle and the diameter of the orifice. Take coefficient of discharge of orifice = 0.88.*

Solution. *Given :* $n = 1$, $N = 650$ r.p.m. ; $\theta = 28°$ of crank travel ;
Fuel consumption = 2.2 kg/h ; Sp. gr. = 0.875 ;
$\Delta p = p_1 - p_2 = 150 - 32 = 118$ bar ; $C_d = 0.88$

Volume of fuel injected per cycle :

Fuel to be injected per cycle $= \frac{\text{Fuel consumption per cylinder}}{\text{No. of cycles per min.}}$

$$= \frac{(2.2/60)}{(650/2)} = 1.128 \times 10^{-4} \text{ kg}$$

Volume of fuel injected per cycle $= \frac{\text{Mass of fuel injected per cycle}}{\text{Density of fuel } (\rho_f)}$

$$= \frac{1.128 \times 10^{-4}}{0.875 \times 1000} = 1.289 \times 10^{-7}$$

$$= \mathbf{0.1289 \text{ cm}^3}. \quad \textbf{(Ans.)}$$

Diameter of the orifice, d_0 :

Time for fuel injection per cycle $= \left(\frac{\theta}{360} \times \frac{60}{N}\right)$ sec.

$$= \frac{28}{360} \times \frac{60}{650} = 0.00718 \text{s}$$

Mass of fuel injected per second,

$$m_f = \frac{\text{Fuel injected per cycle}}{\text{Time for fuel injection}} = \frac{1.128 \times 10^{-4}}{0.00718} = 0.0157 \text{ kg/s}$$

Actual velocity of per cycle injection,

$$V_f = C_f \sqrt{\frac{2\Delta p}{\rho_f}} = 0.88 \sqrt{\frac{2 \times 118 \times 10^5}{(0.875 \times 1000)}} = 144.5 \text{ m/s}$$

Now, $m_f = A_0 \times V_f \times \rho_f$

or $$0.0157 = \left(\frac{\pi}{4} d_0^2\right) \times 144.5 \times (0.875 \times 1000)$$

or $$d_0 = \left[\frac{0.0157 \times 4}{\pi \times 144.5 \times (0.875 \times 1000)}\right]^{1/2} = 3.976 \times 10^{-4} \text{ m} = \mathbf{0.4 \text{ mm}}. \quad \textbf{(Ans.)}$$

Example 12.4. *A four-stroke engine using 0.272 kg/kWh fuel of 32° API develops 15 kW per cylinder at 2000 r.p.m. The fuel injection pressure is 120 bar and the combustion chamber pressure is 30 bar. If the duration of injection is 30° of crank travel and velocity coefficient is 0.9 determine the diameter of the fuel orifice.*

$$\text{Table Sp. gr.} = \frac{141.5}{131.5 + {}^\circ API}$$ **(Madras University)**

Solution. *Given* : s.f.c. = 0.272 kg/kWh, Power developed = 15 kW ;

$\Delta p = p_1 - p_2 = 120 - 30 = 90$ bar ; Duration of injection = 30° of crank angle ; $C_f = 0.9$

Diameter of the orifice, d_0 :

$$\text{Sp. gr.} = \frac{141.5}{131.5 + 32} = 0.8654$$

$$\text{Fuel consumption/cycle} = \frac{\text{s.f.c.} \times \text{kW}}{\text{cycle / hour}}$$

$$= \frac{0.272 \times 15}{\left(\frac{2000}{2}\right) \times 60} = 6.8 \times 10^{-5} \text{ kg}$$

$$\text{Duration of injection} = \frac{\theta}{360} \times \frac{60}{N} = \frac{30}{360} \times \frac{60}{2000} = 0.0025\text{s}$$

$$\therefore \quad m_f = \frac{6.8 \times 10^5}{0.0025} = 0.0272 \text{ kg/s}$$

Actual fuel velocity of injection,

$$V_f = C_f \sqrt{\frac{2\Delta p}{\rho_f}} = 0.9 \times \sqrt{\frac{2 \times 90 \times 10^5}{(0.8654 \times 1000)}} = 129.8 \text{ m/s}$$

Now,

$$m_f = A_f \times V_f \times \rho_f$$

$$= \frac{\pi}{4} d_0^2 \times V_f \times \rho_f$$

or

$$0.0272 = \frac{\pi}{4} d_0^2 \times 129.8 \times (0.8654 \times 1000)$$

$$\therefore \quad d_0 = \left[\frac{0.0272 \times 4}{\pi \times 129.8 \times (0.8654 \times 1000)}\right]^{1/2} = 5.55 \times 10^{-4} \text{ m}$$

$$= 5.55 \times 10^{-4} \text{ m} \quad \text{or} \quad \mathbf{0.555\ mm.} \quad \textbf{(Ans.)}$$

Example 12.5. *A 4-stroke cycle C.I. engine develops 11 kW per cylinder while running at 1800 r.p.m. and using fuel oil of 32° API. Fuel injection occupies 32° of crank travel and takes place through a fuel injection orifice 0.47 mm diameter with flow coefficient of 0.9. Fuel is injected at a pressure of 118.2 bar into combustion chamber where the pressure is 31.38 bar.*

Estimate the quantity of fuel injected in kg/kWh. Specific gravity of fuel oil is given by :

$$\frac{141.5}{131.5 + {}^\circ API}$$

Solution. *Given* : Power developed = 11 kW ; N = 1800 r.p.m., θ = 32° of crank travel ; d_0 = 0.47 mm ; C_f = 0.9 ; $\Delta p = p_1 - p_2 = 118.2 - 31.38 = 86.82$ bar

$$\text{Sp. gr.} = \frac{141.5}{131.5 + {}^\circ\text{API}} = \frac{141.5}{131.5 + 32} = 0.8654$$

Actual fuel velocity of injection,

$$V_f = C_f \sqrt{\frac{2\Delta p}{\rho_f}} = 0.9 \sqrt{\frac{2 \times 86.82 \times 10^5}{(0.8654 \times 1000)}} = 127.48 \text{ m/s}$$

Now,

$$m_f = A_0 \times V_f \times \rho_f$$

$$= \frac{\pi}{4} d_0^2 \times V_f \times \rho_f$$

$$= \frac{\pi}{4} \times \left(\frac{0.47}{1000}\right)^2 \times 127.48 \times (0.8654 \times 1000) = 0.01914 \text{ kg/s}$$

$$\text{Time for fuel injection per cycle} = \frac{\theta}{360} \times \frac{60}{N} = \frac{32}{360} \times \frac{60}{1800} = 2.963 \times 10^{-3}\text{s}$$

Mass of fuel injected per cycle = $0.01914 \times 2.963 \times 10^{-3} = 5.671 \times 10^{-5}$ kg/cycle

$$\text{Total number of cycles per hour} = \frac{1800}{2} \times 60 = 54000$$

$\therefore$ **Fuel consumption in kg/kWh**

$$= 5.671 \times 10^{-5} \times 54000 \times \frac{1}{11} = \mathbf{0.278\ kg/kWh.} \quad \textbf{(Ans)}$$

Example 12.6. *An high-cylinder, four-stroke diesel engine has a power output of 386.4 kW at 800 r.p.m. The fuel consumption is 0.25 kg/kWh. The pressure in the cylinder at the beginning of injection is 32 bar and the maximum cylinder pressure is 55 bar. The injector is expected to be set at 207 bar and the maximum pressure at the injector is set to be about 595 bar. Calculate the orifice area required per injector if the injection takes place over 12° crank angle.*

Assume the following :

Specific gravity of fuel = 0.85 ; coefficient of discharge for the injector = 0.6 ; atmospheric pressure = 1.013 bar ; The effective pressure difference is the average pressure difference over the injection period. **(Madras University)**

Solution. *Given* : n = 8 ; Power output = 386.4 kW, N = 800 r.p.m.,

Fuel consumption = 0.25 kg / kWh, θ = 12° crank angle ; Sp. gr. = 0.85 ;

$C_f = 0.6$; $p_{atm.} = 1.013$ bar.

Orifice area reqd. per injection, A_0 :

$$\text{kW per cylinder} = \frac{386.4}{8} = 48.3$$

$\therefore$ Fuel consumption per cylinder = 48.3 × 0.25 = 12.075 kg / h or 0.2012 kg/ min.

$$\text{Fuel to be injected per cycle} = \frac{0.2012}{(800/2)} = 5.03 \times 10^{-4} \text{ kg}$$

$$\text{Time for fuel injection per cycle} = \left(\frac{\theta}{360} \times \frac{60}{N}\right) = \frac{12}{360} \times \frac{60}{800} = 0.0025 \text{ s}$$

$\therefore$ Mass of fuel injected per second,

$$m_f = \frac{5.03 \times 10^{-4}}{0.0025} = 0.2012 \text{ kg/s}$$

Pressure difference at beginning = 207 − 32 = 175 bar

Pressure difference at end = 595 − 55 = 540 bar

$$\text{Average pressure difference} = \frac{175 + 540}{2} = 357.5 \text{ bar}$$

Now,
$$m_f = A_0 \times V_f \times \rho_f$$
$$= A_0 \times \left[C_f\sqrt{\frac{2\Delta p}{\rho_f}}\right] \times \rho_f = A_0 \times C_f\sqrt{2\Delta p.\rho_f}$$

or
$$0.2012 = A_0 \times 0.6\sqrt{2 \times (357.5 \times 10^5) \times (0.85 \times 1000)}$$
$$= 147915.5\, A_0$$

$\therefore$ $\mathbf{A_0 = 1.36 \times 10^{-6}\ m^2}$ or **0.0136 cm².** **(Ans.)**

Example 12.7. *A six-cylinder, four-stroke oil engine operates on A/F ratio = 20. The diameter and stroke of the cylinder are 100 mm and 140 mm respectively. The volumetric efficiency is 80 per-cent. The condition of air at the beginning of compression are 1 bar, 27° C.*

(i) Determine the maximum amount of fuel that can be injected in each cylinder per second.

(ii) If the speed of the engine is 1500 r.p.m., injection pressure is 150 bar, air pressure during fuel injection is 40 bar and fuel injection is carried out for 20° crank angle, determine the diameter of the fuel orifice assuming only one orifice is used.

Take, $\rho_f = 860\ kg/m^3$; $C_f = 0.67$. **(Roorkee University)**

Solution. *Given* : $n = n_0 = 6$; A / F ratio = 20 ; $d = 100$ mm = 0.1 m, $l = 140$ mm = 0.14 m ; $\eta_{vol.} = 80\%$; $p_a = 1$ bar, $T_a = 27 + 273 = 300$ K, $N = 1500$ r.p.m. ; $\theta = 20°$ crank angle ; $\rho_f = 960$ kg/m³ ; $C_f = 0.67$, $\Delta_p = 150 - 40 = 110$ bar.

(i) Amount of fuel injected into each cylinder per cycle :

Volume of air supplied per cylinder per cycle
$$= \text{Stroke volume} \times \eta_{vol.}$$
$$= \frac{\pi}{4}d^2 \times l \times \eta_{vol.} = \frac{\pi}{4} \times (0.1)^2 \times 0.14 \times 0.8 = 8.8 \times 10^{-4} m^3$$

Mass of this air at suction conditions,
$$m_a = \frac{p_a V_a}{RT_a} = \frac{1 \times 10^5 (8.8 \times 10^{-4})}{287 \times 300} = 1.022 \times 10^{-3}\ \text{kg/cycle}$$
$$\text{A / F ratio} = \frac{m_a}{m_f} = \frac{20}{1}$$

$\therefore$
$$m_f = \frac{m_a}{20} = \frac{1.022 \times 10^{-3}}{20} = 5.11 \times 10^{-5}\ \text{kg/cycle}$$

Time taken for fuel injection per cycle
$$= \left(\frac{\theta}{360} \times \frac{60}{N}\right) = \frac{20}{360} \times \frac{60}{1500} = 0.00222\text{s}$$

$\therefore$ *Amount/mass of fuel injected into each cylinder per second,*
$$\dot{m}_f = \frac{5.11 \times 10^{-5}}{0.00222} = \mathbf{0.023\ kg/s.}\ \textbf{(Ans.)}$$

(ii) Diameter of the fuel orifice, d_0 :

The mass of fuel injected into each cylinder per second,
$$\dot{m}_f = A_0 \times V_f \times \rho_f$$

or
$$0.023 = \frac{\pi}{4}d_0^2 \times C_f\sqrt{\frac{2\Delta p}{\rho_f}} \times \rho_f$$

$$= \frac{\pi}{4}d_0^2 \times C_f\sqrt{2\Delta p \times \rho_f}$$
$$= \frac{\pi}{4}d_0^2 \times 0.67\ \sqrt{2 \times 110 \times 10^5 \times 960} = 7238.11\, d_0^2$$

or $d_0 = 5.637 \times 10^{-4}$ m or **0.5637 mm.** **(Ans.)**

Example 12.8. *In a diesel fuel injection pump, the volume of fuel in the pump barrel before commencement of the effective stroke is 7 c.c. The diameter of the fuel line from pump to injector is 3 mm and is 700 mm long. The fuel in the injection valve is 2 c.c.*

(i) To deliver 0.10 c.c. of fuel at a pressure of 150 bar, how much displacement the plunger undergoes ? Assume a pump inlet pressure of 1 bar ;

(ii) What is the effective stroke of the plunger if its diameter is 7 mm.

Assume coefficient of compressibility of oil as 78.8 × 10⁻⁶ per bar at atmospheric pressure.

Solution. *Given* : The volume of fuel in the pump barrel before commencement of the effective stroke = 7 c.c.

The diameter and length of the fuel line from pump to injector = 3 mm, 700 mm,

Volume of fuel in the injection valve = 2 c.c.

Volume of fuel to be delivered = 0.10 c.c.

The pressure at which fuel to be delivered, p_1 = 150 bar

Atmospheric pressure, p_2 = 1 bar

Coefficient of compressibility, $C_c = 78.8 \times 10^{-6}$ per bar at atmospheric pressure

Diameter of plunger, d_p = 7 mm

(i) Displacement of plunger :

Coefficient of compressibility of oil,
$$C_c = \frac{\text{Change in volume per unit volume}}{\text{Difference in pressure causing compression}}$$
$$= \frac{(V_1 - V_2)}{V_1(p_1 - p_2)}$$

Total initial fuel volume,

V_1 = Volume of fuel in barrel + volume of fuel in the delivery line + volume of fuel in the injection valve
$$= 7 + \frac{\pi}{4}(0.3)^2 \times 70 + 2 = 13.95\ \text{c.c.}$$

No pressure is built up till the pump plunger closes the inlet port. Further advance of plunger will compress the fuel oil and raise the pressure to a required value. Once the delivery pressure is attained, further movement of plunger results in delivery of fuel oil at constant pressure.

Change in volume due to compression $= C_c(p_1 - p_2) \times V_1$

or
$$(V_1 - V_2) = 78.8 \times 10^{-6} \times (150 - 1) \times 13.95$$
$$= 0.16379\ \text{c.c.}$$

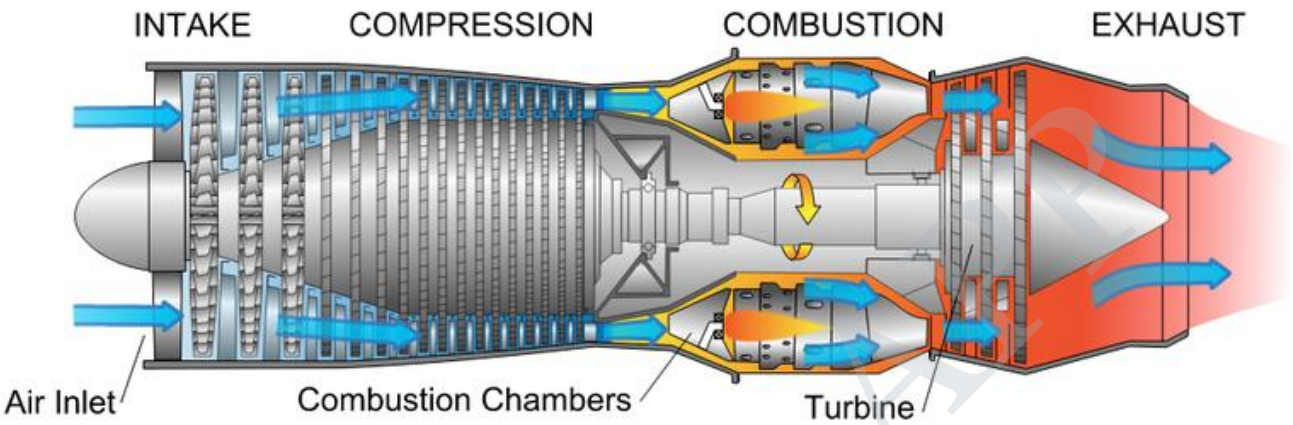
Total displacement of plunger
$$= (V_1 - V_2) + 0.1 = 0.16379 + 0.1 = \mathbf{0.26379\ c.c.}\ \textbf{(Ans.)}$$

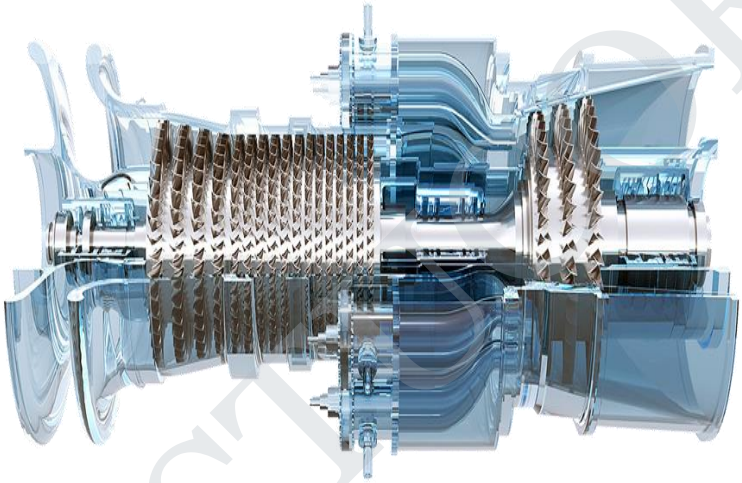
(ii) Effective stroke of the plunger, l_p :
$$\frac{\pi}{4}d_p^2 \times l_p = 0.26379 \quad \text{or} \quad \frac{\pi}{4} \times (0.7)^2 \times l_p = 0.26379$$

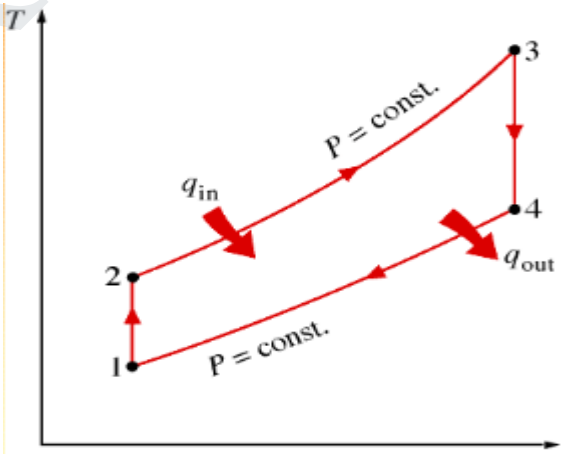
$\therefore$
$$l_p = \frac{0.26379 \times 4}{\pi \times (0.7)^2} = 0.6854\ \text{cm or } \mathbf{6.854\ mm.}\ \textbf{(Ans.)}$$

THERMAL ENGINEERING 1

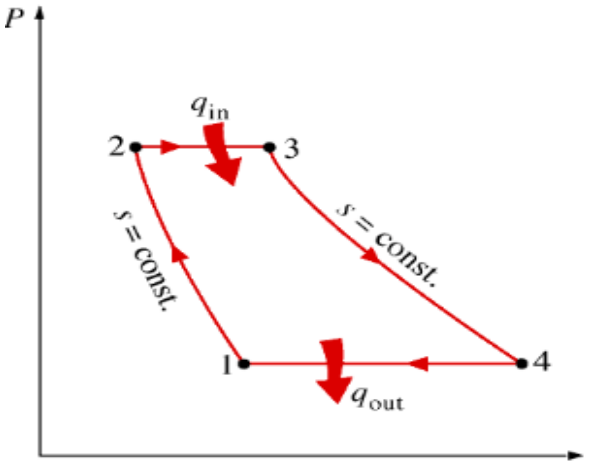
UNIT V - GAS TURBINE







(a) *T-s* diagram



(b) *P*- υdiagram

CLASSIFICATION OF GAS TURBINES

Though the gas turbines maybe classified in many ways,yet the following are important from the subject point of view:

I. According to path of the working substance

(a) Closed cycle gas turbines

(b) Open cycle gas turbines

(c) Semi-closed gas turbines

(d) Continuous-combustion (constant pressure) gas turbine may work with open or closed cycle

(e) Constant volume gas turbine works with open cycle

(f) Closed cycle gas turbine is an external combustion engine while open cycle gas turbine is an internal combustion engine

2. According to process of heat obsorption.

(a) Continuous-combustion or Constant pressure gas turbines

(b) Explosive type combustion or Constant volume gas turbines.

3. Thermodynamic (Gas Power) Cycle

(a) Brayton or Joule cycle (for constant volume gas turbines),

(b) Atkinson cycle (for constant volume gas turbines)

(c) Ericsson cycle (for constant pressure gas turbine with large number of intercooling and reheating.)

4. Arrangement of shafts.

(a) Single shaft gas turbines (compressor is run by power turbine)

(b) Multi-shaft gas turbines (separate compressor turbine and power turbine),

(c) Series Flow Gas Turbines

(d) Parallel flow gas turbines

5. F u e l

(a) Liquid Fuel

(b) Gaseous Fuel

(c) Solid Fuel Gas Turbine

6: Application

(a) Stationary

(b) Automotive

(c) Locomotive

(d) Marine

(e) Air-craft Gas Turbine

INTRODUCTION

- In a gas turbine, the air is obtained from the atmosphere and compressed in an air compressor.
- The compressed air is then passed into the combustion chamber, where it is heated considerably.
- The hot air is then made to flow over the moving blades of the gas turbine, which imparts rotational motion to the runner.
- During this process, the air gets expanded and finally it is exhausted into the atmosphere.
- A major part of the power developed by the turbine is consumed for driving th; compressor (which supplies compressed air to the combustion chamber). The remaining power is utilised for doing some external work.

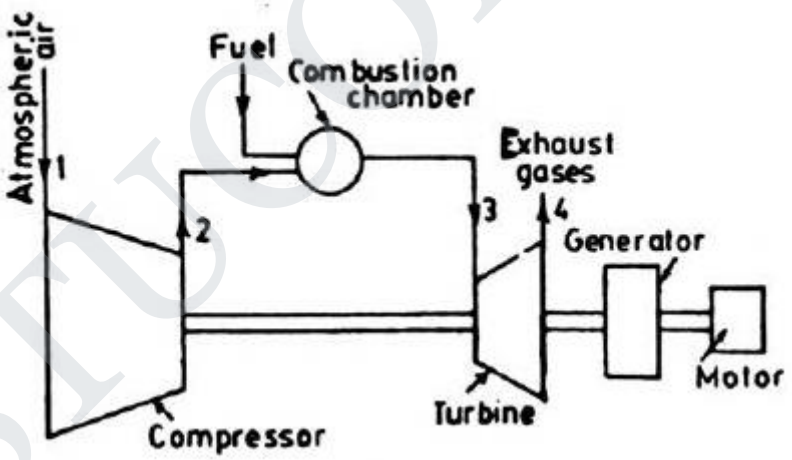
COMPARISON OF GAS TURBINES AND I.C. ENGINES

S. No	Gas turbines	I.C engines
1.	The mass of gas turbine per kW developed is less	The mass of an IC. engine per kW deveoped is more
2.	The installation and running cost is less.	The installation and running cost is more.
3.	Its efficiency is higher.	Its efficiency is less.
4.	The balancing of a gas turbine is perfect.	The balancing of an IC. engine is not perfect.
5.	The torque produced is uniform. Thus no flywheel is required.	The torque produced is not uniform. Thus flywheel is necessary.
6.	The lubrication and ignition systems are simple.	The lubrication and ignition systems are difficult.
7.	It can be driven at a very high speed.	It can not be driven at a very high speed.
8.	The pressures used are very low (about 5 bar)	The pressures used are high (above 60 bar).
9.	The exhaust of a gas turbine is free from smok and less polluting.	The exhaust of an I.C. engine is more polluting.
10.	They are very suitable for air crafts.	They are less suitable for air crafts.
11.	The s tarting of a gas turbine is not simple.	The starting ofan IC. engine is simple

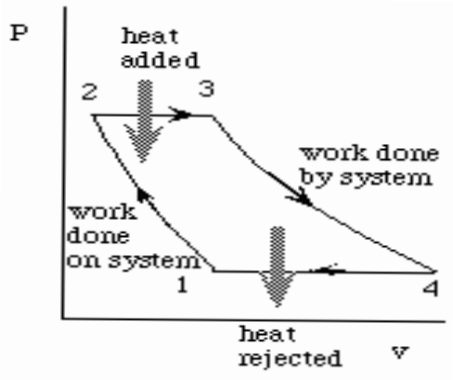
COMPARISON OF GAS TURBINES AND STEAM TURBINES

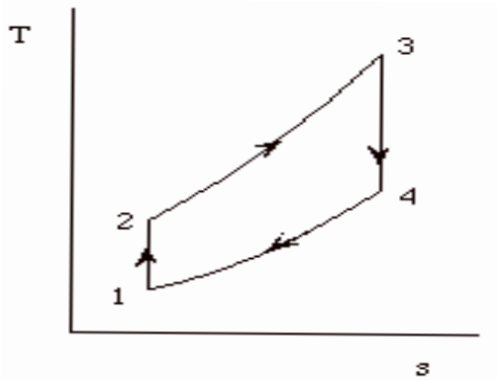
S. No	Gas turbines	Steam turbines
1.	The important components are compressor and combustion chamber.	The important components are steam boiler and accessories.
2.	The mass of gas turbine per kW developed is less,	The mass of steam turbine per kW developed is more.
3.	It requires less space for installation.	It requires more space for idstallation.
4.	The instaitation and running cost is less.	The installation and running cost is more
5.	The starting ofgas turbine is very easy and quick	The starting of steam turbine is difficult and takes long time.
6.	Its control, with the changing load conditions, is easy.	Its control, with the changing load conditions, is difficult.
7.	A gas turbine does not depend on water supply	A steam turbine depends on water supply.
8.	Its efficiency is less.	Its efficiency Is hi&her.

OPEN CYCLE GAS TURBINE



- In this type of gas turbine liquid (or) gaseous fuels are used for power generation. The basic components are shown in figure above.
- Initially, atmospheric air is allowed to pass through rotary compressor in which its Pressure and temperature is increased, isentropically.
- Then this compressed air is passed through combustion chamber in which fuel is injected for combustion purpose. After combustion of fuel in combustion chamber the heat is added under constant pressure condition the temperature of compressed air is further increased.
- Now high pressure and temperature gases are expanded in gas turbine which is helpful to run the gas turbine or blades (generally of reaction type)





- This gas turbine is directly connected to electric generator to produce electricity and finally exhausted into the atmosphere.
- This type of gas turbine works on open cycle because here working fluid is used only once. After single use it is thrown into atmosphere.
- Here inlet and outlet both the ends are open to atmosphere hence termed as open cycle gas turbine. It is also called as continuous combustion gas turbine

Advantages:

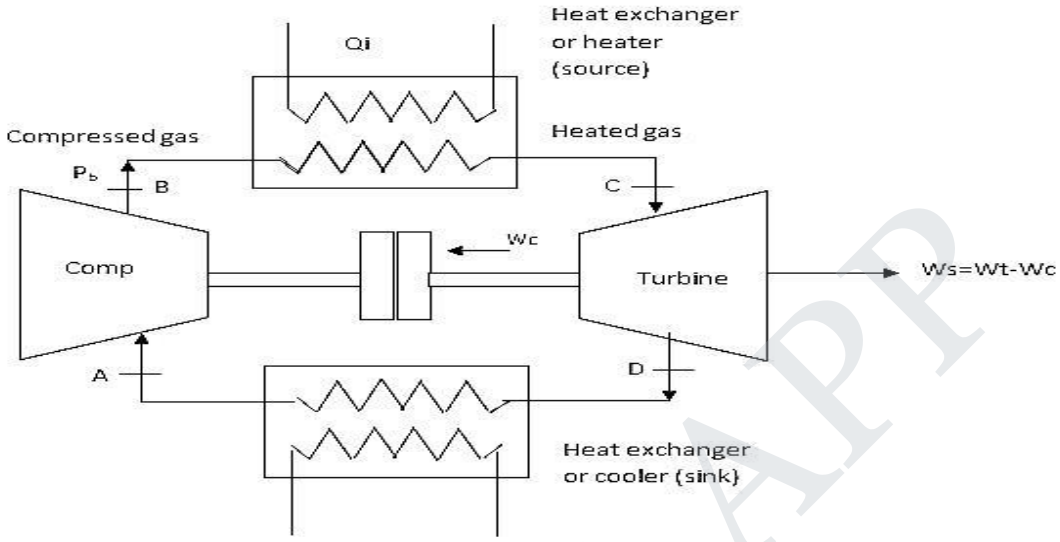
1. **Warm-up time:** Once the turbine is brought up to the rated speed by the starting motor and the fuel is ignited, the gas turbine will be accelerated from cold start to full load without warm-up time.
2. **Low weight and size:** The weight in kg per kW developed is less.
3. Fuels**:** Almost any hydrocarbon fuel from high-octane gasoline to heavy diesel oils can be used in the combustion chamber.
4. Open cycle plants occupies less space compared to close cycle plants.
5. The stipulation of a quick start and take-up of load frequently are the points in favor of open cycle plant when the plant is used as peak load plant.
6. Component or auxiliary refinements can usually be varied in open cycle gas turbine plant to improve the thermal efficiency and can give the most economical overall cost for the plant load factors and other operating conditions envisaged.
7. Open cycle gas turbine power plant, except those having an intercooler, does not need cooling water. Therefore, the plant is independent of cooling medium and becomes self-contained.

Disadvantages:

1. The part load efficiency of the open cycle gas turbine plant decreases rapidly as the considerable percentage of power developed by the turbine is used for driving the compressor.
2. The system is sensitive to the component efficiency; particularly that of compressor. The open cycle gas turbine plant is sensitive to changes in the atmospheric air temperature, pressure and humidity.
3. The open cycle plant has high air rate compared to the closed cycle plants, therefore, it results in increased loss of heat in the exhaust gases and large diameter duct work is needed.
4. It is essential that the dust should be prevented from entering into the compressor to decrease erosion and depositions on the blades and passages of the compressor and turbine. So damages their profile. The deposition of the carbon and ash content on the turbine blades is not at all desirable as it reduces the overall efficiency of the open cycle gas turbine plant.

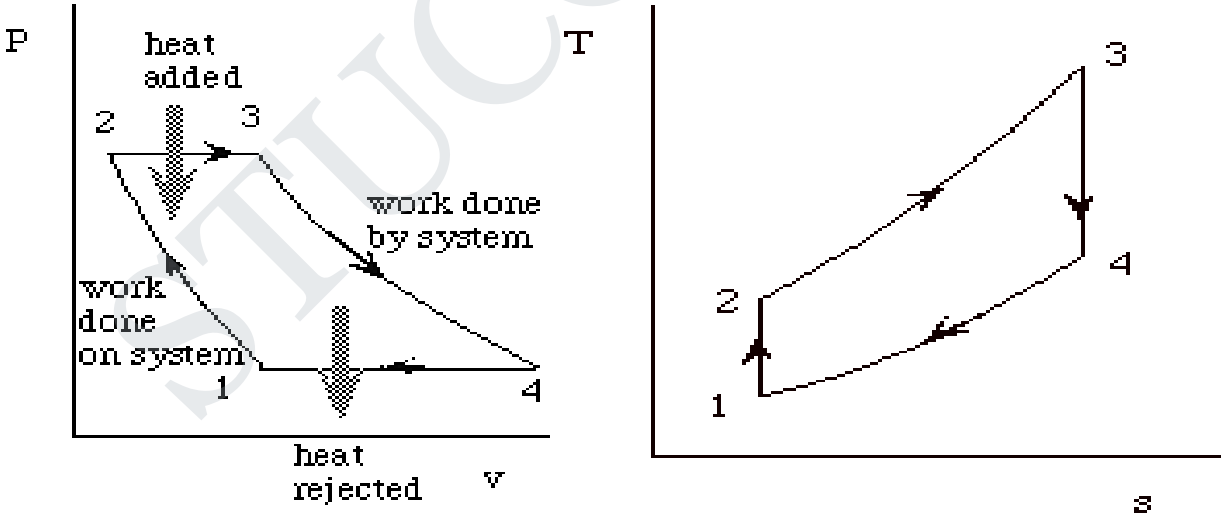
CLOSED CYCLE GAS TURBINES

In the closed cycle gas turbine, compressed air leaves the compressor and passes via the heat exchanger through the air heater. In the air heater there are tubes (not shown) through which the compressed air passes. The air is therefore further heated in the heater. This hot high pressure air then passes through the blade rings. Whilst passing over the rotor blades, the air is continuously expanding, its pressure energy being converted into kinetic energy, which in turn, is absorbed by the turbine motor.



The hot air on leaving the turbine passes through the heat exchanger. As the air is still at a high temperature, it is cooled in a pre-cooler before entering the compressor.

Part of the power developed by the turbine is used to drive he compressor and the remainder in driving the alternator. The turbine is started by an electric motor.



Advantages:

- Use of higher pressure throughout the cycle which is useful for reduce size of plant.
- No outside air is used for compressing so there is no problem of dust and dirt.
- Also there is no need of filtration of incoming air.
- Any type of fuel can be used for combustion purpose.

- It is not necessary that air is used as working fluid any other fluid having better thermodynamic property can be used.
- Working fluid circulated continuously.
- It avoids erosion of turbine blade due to contaminated gases.
- The exhaust gas from the turbine is passed into cooling chamber.
- Mass of installation per KW is more.
- Maintenance cost is low
- Longer life.

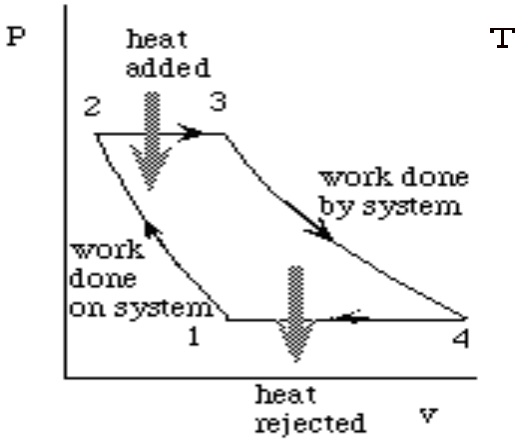
Disadvantages :

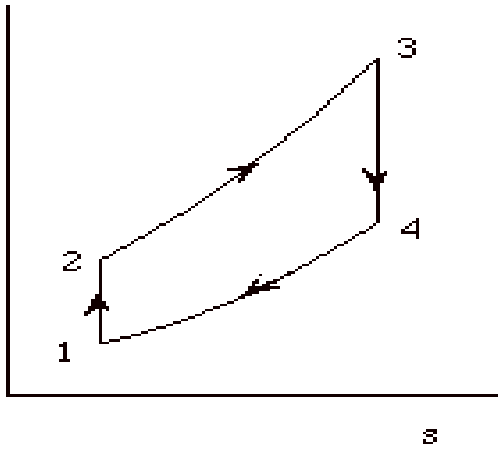
- Weight of system is high compared to open cycle.
- Large amount of water is required for cooling in cooler.
- System should be air tight when working substance other than air is used.
- If load on system increases then performance of system is poor.

DIFFERENCE BETWEEN OPEN AND CLOSED GAS TURBINE:

S. No	Closed cycle gas turbine	Open cycle gas turbine
1.	Combustion of fuel is external	Combustion of fuel is internal.
2.	Gas from turbine is passed into cooling chamber.	Gas from turbine is exhausted to atmosphere.
3.	Any type of fluid is used.	Only air can be used.
4.	Turbine blades cannot be contaminated.	Turbine blades get contaminated.
5.	Working fluid circulated continuously.	Working fluid replaced continuously.
6.	Mass of installation per KW is more.	Mass of installation per KW is less.
7.	Heat exchanger is used.	Heat exchanger is not used.
8.	This system required more space.	This system required less space.
9.	Since exhaust is cooled by circulating water, it is best suited fo stationary installation, marine use.	Since turbine exhaust is discharged into atmosphere, it is best suited for moving Vehicle like Aircraft.
10.	Maintenance cost is high.	Maintenance cost is low.

PERFORMANCE CALCULATION OF OPEN AND CLOSED GAS TURBINE CYCLE:





1-2 Process: Adiabatic compression process

Compressor Work $(W_C) = mc_p(T_2 - T_1)$ **kJ**

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

2-3 Process: Constant pressure heat addition

$$Q_S = mc_p(T_3 - T_2) \text{ kJ}$$

3-4 Process: Adiabatic Expansion process

Turbine Work $(W_T) = mc_p(T_3 - T_4)$ **kJ**

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}}$$

4-1 Process: Constant pressure Heat Rejection

$$Q_R = mc_p(T_4 - T_1) \text{ kJ}$$

Net Work done

$$W_{net} = Q_s - Q_R \text{ kJ}$$

Thermal Efficiency

$$\eta = \frac{\text{Work done}}{\text{Heat Supplied}} = \frac{W_{net}}{Q_S} = \frac{mc_p(T_3 - T_2) - mc_p(T_3 - T_4)}{mc_p(T_3 - T_2)} = 1 - \frac{T_3 - T_4}{T_3 - T_2}$$

Efficiency

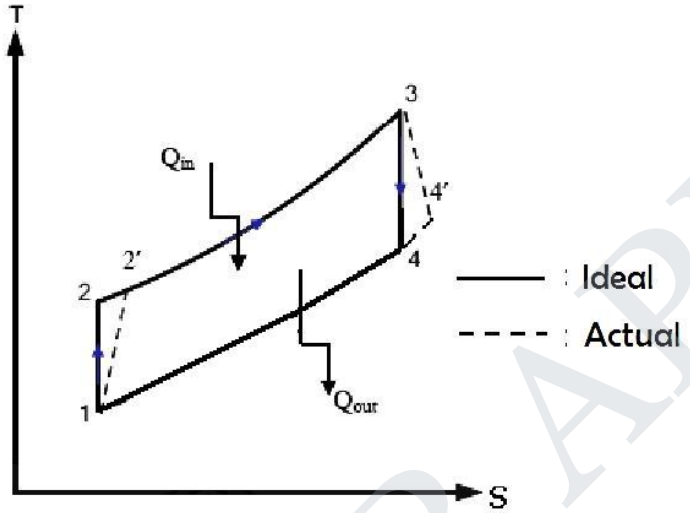
$$\eta = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

Turbine Work Ratio

$$W_R = \frac{W_T}{W_{net}} = 1 - \frac{T_1}{T_3}\left(r_c^{\gamma-1}\right)$$

Causes of Departure of Actual Cycle from Ideal Cycle

In practice, losses due to friction, heat transfer, shock, etc. occur in both the compressor and turbine components (i.e. the compression in the compressor and expansion in the turbine are not isentropic) so that actual power absorbed by the compressor increases and actual output of the turbine decreases compared with isentropic operation. Thus, in practice the compression is polytropic (1-2) and not isentropic (1-2'). Similarly, expansion is polytropic (3-4) and not isentropic (3-4'). If η_c and η_T are isentropic efficiencies of compressor and turbine respectively,



Ideal Gas turbine cycle: 1-2-3-4

Actual Gas turbine cycle: $1' - 2' - 3' - 4'$

Actual compression work: $pV^n = C$

Actual Compressor Work $(W_C) = c_p(T_{2'} - T_1)$

Actual Compressor Work $(W_C) = \frac{\text{Ideal compressor work}}{\text{Compressor efficiency}} \Rightarrow W_C = \frac{c_p(T_2 - T_1)}{\eta_c} \Rightarrow W_C = \frac{c_p T_1\left(\frac{T_2}{T_1} - 1\right)}{\eta_c}$

Where,

$$\textbf{Actual Compressor Work } (W_C) = \frac{c_p T_1\left((r_c)^{\frac{\gamma-1}{\gamma}} - 1\right)}{\eta_c} \qquad \therefore \frac{T_2}{T_1} = (r_c)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

The value of η_c depends upon the type of air compressor, its pressure ratio, and the weight of air passing through it. For modern high-speed centrifugal compressors a value of 0-75 appears to be reasonable and for axial-flow compressors value varies from 0-85 to 0-90.

It should be noted that owing to the pressure loss in the combustion chamber, the expansion ratio for the turbine is smaller than pressure ratio for compressor. Further because of the injection of fuel in the combustion chamber, the mass flow of gases is greater in the turbine than air flow through compressor.

Actual Turbine Work: $pV^n = C$

$$\text{Actual Turbine Work } (W_T) = c_p(T_3 - T_{4'})$$

$$\text{Actual Turbine Work } (W_T) = \text{Ideal Turbine Work} \times \text{Turbine Efficiency} \Rightarrow W_T = c_p(T_3 - T_4) \times \eta_T$$

$$\Rightarrow W_T = c_p\eta_T\left(1 - \frac{T_4}{T_3}\right) \Rightarrow \mathbf{W_T = c_p\eta_T\left(1 - \frac{1}{(r_c)^{\frac{\gamma-1}{\gamma}}}\right)} \qquad \therefore \frac{T_4}{T_3} = (r_c)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

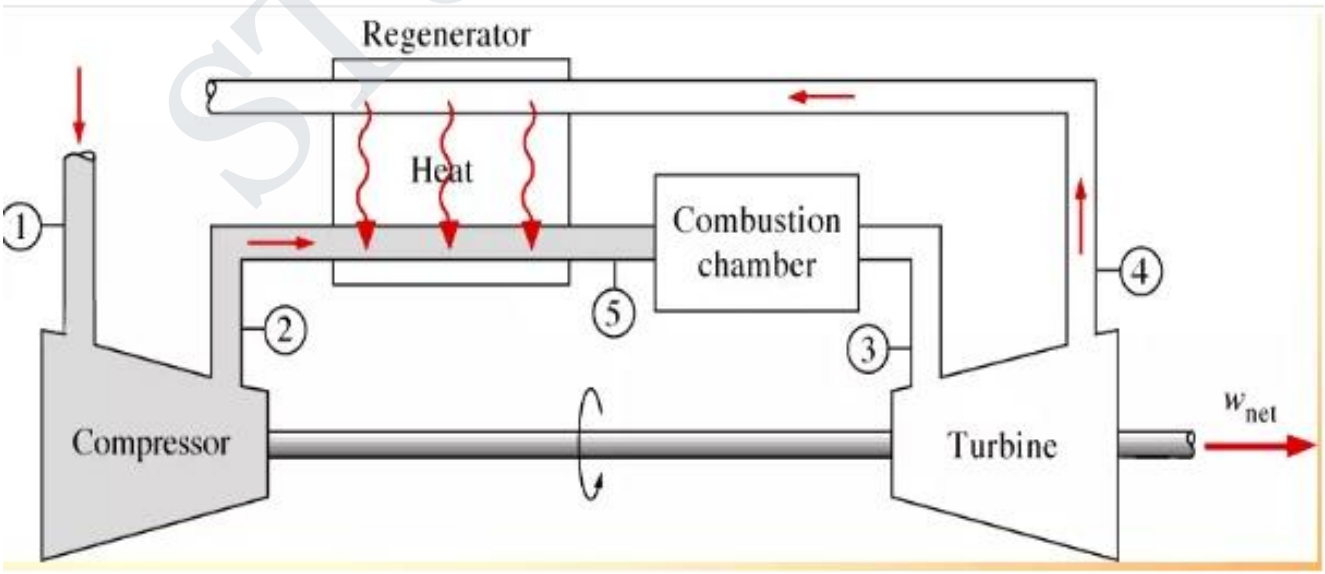
The isentropic efficiency of the turbine is affected by its size and number of stages. For single-stage impulse turbine, values vary from 0-8 to 0-85, depending upon the care taken to reduce the blading losses, nozzle friction, leakage, etc.

In addition to factors responsible for lowering isentropic efficiencies of compressor and turbine as mentioned above, we may add briefly other factors responsible for departure of actual cycle from ideal cycle. These factors are:

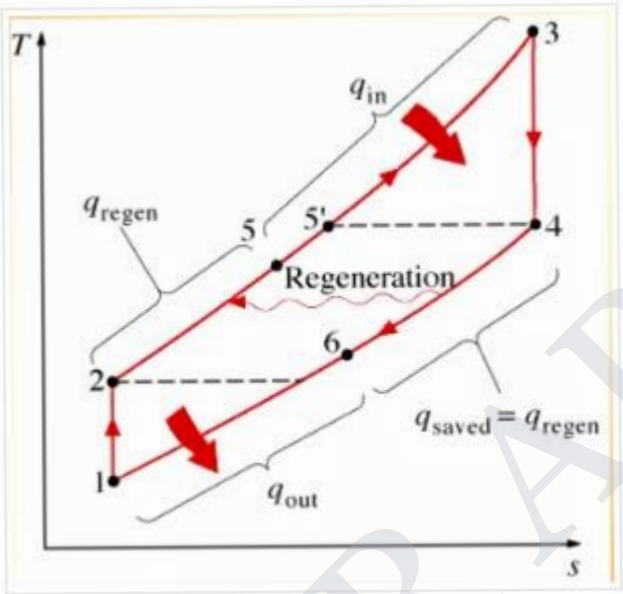
- Pressure losses in the pipes connecting the various components, combustion chamber and heat exchanger,
- Mechanical losses at compressor and turbine bearings (2 to 4%),
- Variation of specific heats of working fluid with temperature,
- Variation of mass flow of the working fluid, and
- Heat exchange in the heat exchanger (if included) being incomplete.

REGENERATIVE GAS TURBINE CYCLE

In this method, a regenerator (heat exchanger) is used for utilising heat of exhaust gases from turbine, in pre-heating the compressed air before it enters the combustion chamber. The preheating of the compressed air reduces the fuel consumption and consequently improves the thermal efficiency. Regeneration is shown in fig. As a result of regeneration, compressed air is preheated and exhaust gases are cooled.



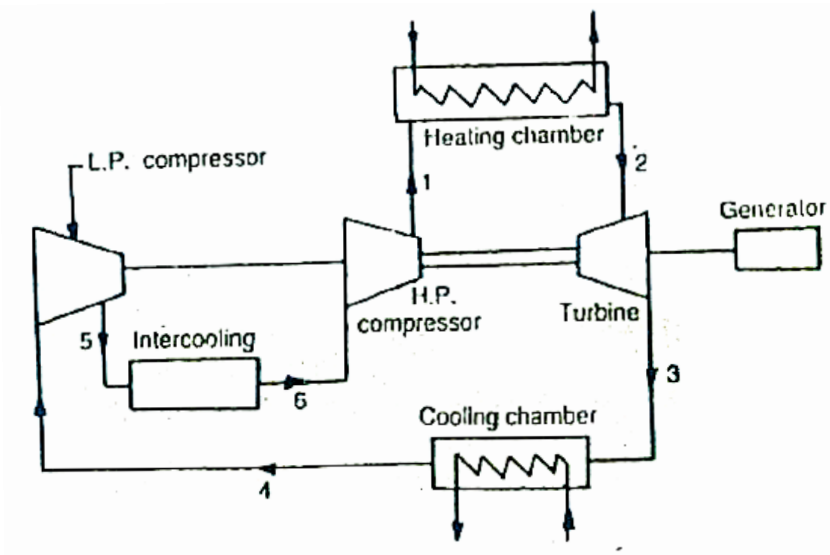
- Temperature of the exhaust gas leaving the turbine is higher than the temperature of the air leaving the compressor.
- The air leaving the compressor can be heated by the hot exhaust gases in a counter-flow heat exchanger (a regenerator or recuperator) – a process called regeneration
- The thermal efficiency of the Brayton cycle increases due to regeneration since less fuel is used for the same work output.



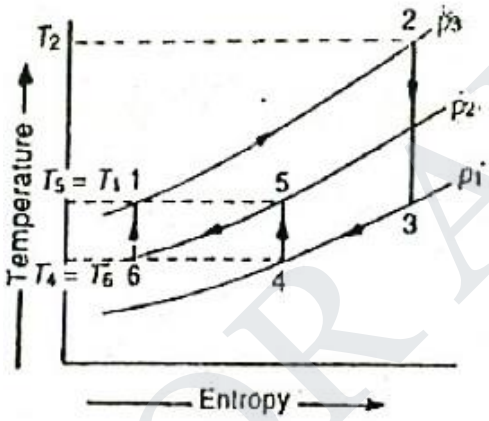
Regeneration process involves the installation of a heat exchanger in the gas turbine cycle. The heat-exchanger is also known as the recuperator. This heat exchanger is used to extract the heat from the exhaust gas . This exhaust gas is used to heat the compressed air. This compressed and pre-heated air then enters the combustors. When the heat exchanger is well designed , the effectiveness is high and pressure drops are minimal. And when these heat exchangers are used an improvement in the efficiency is noticed. Regenerated Gas turbines can improve the efficiency more than 5 % . Regenerated Gas Turbine work even more effectively in the improved part load applications.

GAS TURBINE WITH INTERCOOLING:

We have already discussed that a major portion of the power developed by the gas turbine is utilised by the compressor. It can be reduced by compressing the air in two stages with an intercooler between the two. This improves the efficiency of the gas turbine. The schematic arrangement of a closed cycle gas turbine with an intercooler is shown.



- In this arrangement, first of all, the air is compressed in the first compressor, known as low pressure (L.P.) compressor. We know that as a result of this compression, the pressure and temperature of the air is increased.
- Now the air is passed to an intercooler which reduces the temperature of the compressed air to its original temperature, but keeping the pressure constant.
- After that, the compressed air is once again compressed in the second compressor known as high pressure (H.P.) compressor.
- Now the compressed air is passed through the heating chamber and then through the turbine. Finally, the air is cooled in the cooling chamber and again passed into the low pressure compressor as shown.
- The process of intercooling the air in two stages of compression is shown on T-s diagram in Fig.



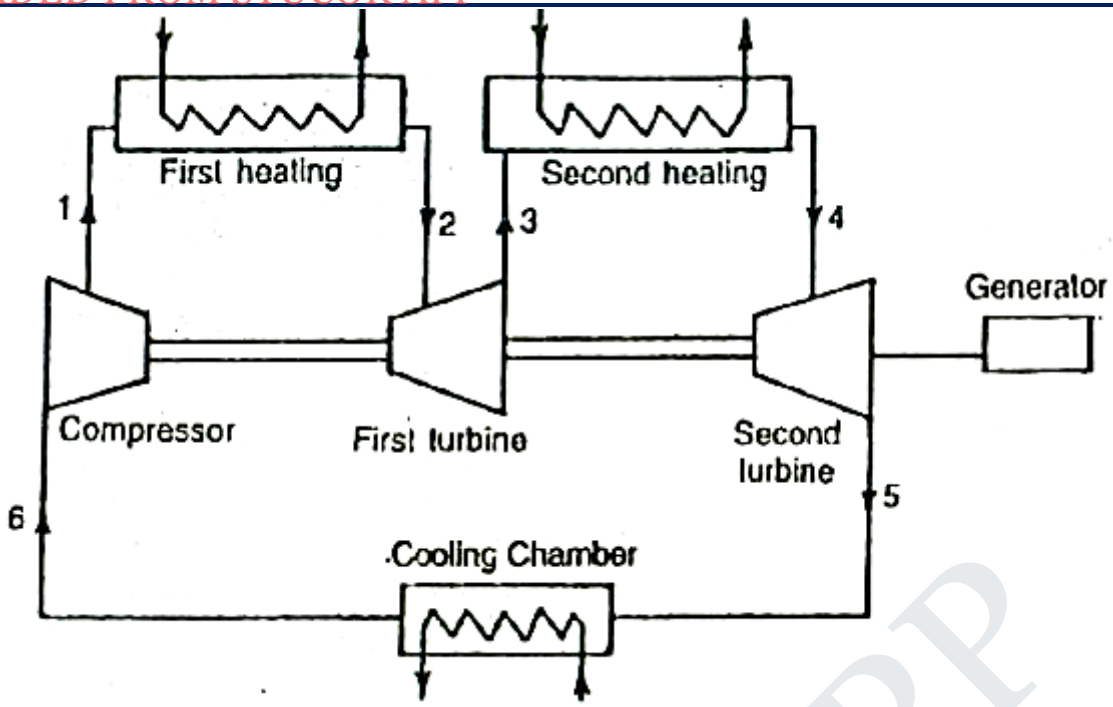
- The process 1-2 shows heating of the air in heating chamber at constant pressure.
- The process 2-3 shows isentropic expansion of air in the turbine.
- The process 3-4 shows cooling of the air in the cooling chamber at constant pressure.
- The process 4-5 shows compression of air in the L.P. compressor.
- The process 5-6 shows cooling of the air in the intercooler at constant pressure.
- Finally, the process 6-I shows compression of air in the H.P. compressor

The work required to compress air depends upon its temperature during compression. The efficiency of gas turbjne is improved by adopting multi-stage compression with intercooling in between two stages as it reduces the work required to compress the air.

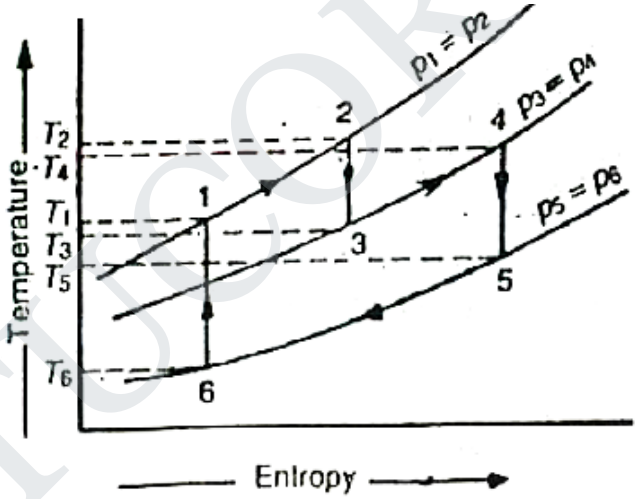
GAS TURBINE WITH REHEATING:

The expressions for thermal efficiency, for Brayton and Ericsson cycles, suggest that higher inlet temperature to the turbine result in improved efficiencies. Thus, to improve the efficiency, the temperature of the gases after partial expansion in the turbine is increased by reheating the gases, before gases start further stage of expansion.

The output of a gas turbine can be considerably improved by exánding the hot air in two stages with a reheater between the two. The schematic arrangement of a closed cycle gas turbine with reheating is shown



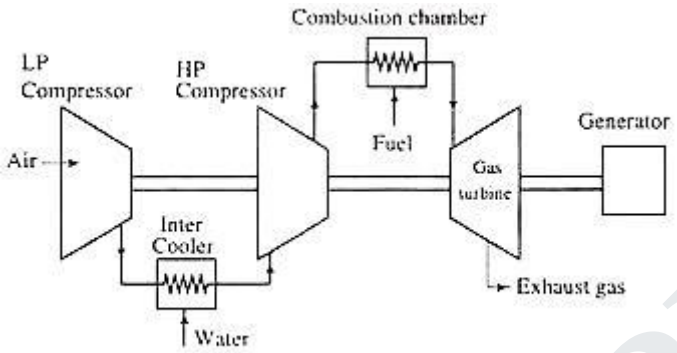
In this arrangement, the air is first compressed in the compressor, passed into the heating chamber, and then to the first turbine. The air is once again passed onto another heating chamber and then to the second turbine. Finally, the air is cooled in the cooling chamber and again passed into the compressor as shown in Fig.



- The process of reheating in the two turbines is shown on T-s diagram in Fig.
- The process 1-2 shows heating of the air in the first heating chamber at constant pressure.
- The process 2-3 shows isentropic expansion of air in the first turbine.
- The process 3-4 shows heating of the air in the second heating chamber at constant pressure.
- The process 4-5 shows isentropic expansiot of air in the second turbine.
- The process 5-6 shows cooling of the air in the intercooler at Constant pressure.
- Finally, the process 6-I shows compression of air in the compressor.

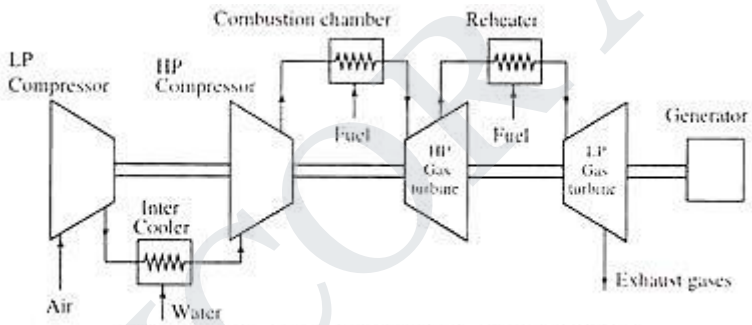
GAS TURBINE WITH INTERCOOLING AND REHEATING:

Fig. shows the layout of the plant in which the compression of air takes place in two stages. An intercooler is employed when high pressure ratios are involved to cool the air from the low pressure compressor before entry into the high pressure compressor, thus reducing the overall power required for compression. Though the intercooler reduces the compression work, there is some pressure loss. But even then it contributes towards overall economy.



Gas turbine with intercooling

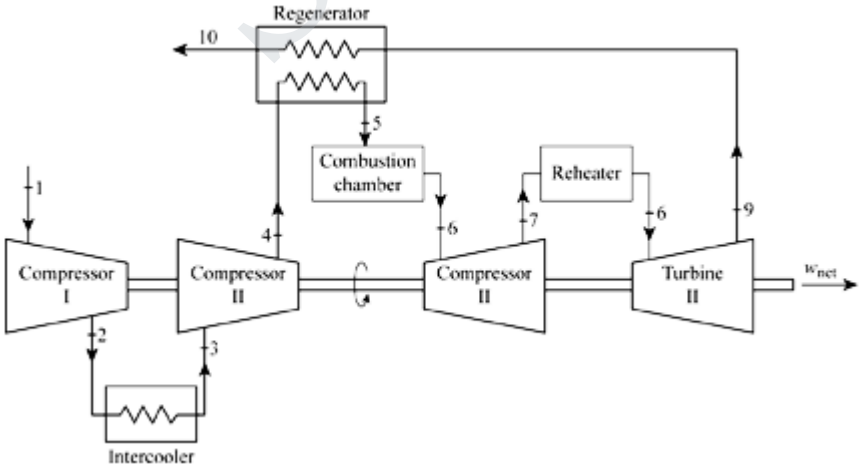
Fig. shows the gas turbine with expansion in two stages as in HP and LP turbine and the gases exhausted from HP turbine are reheated to same temperature before expansion or less and then sent to the LP turbine.



Gas turbine with intercooling and reheating

The air fuel ratio in a gas turbine is quite high. Therefore, the products of combustion are exhausted after expansion in the high pressure turbine which are still rich in oxygen and are subjected to combustion once again in the second combustion chamber just before the air.

After reheating is same as that before entry into the high pressure turbine. The compressors may be driven by power from the high pressure turbine or low pressure turbine.



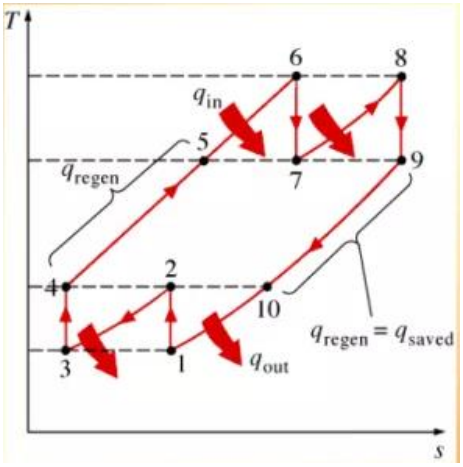
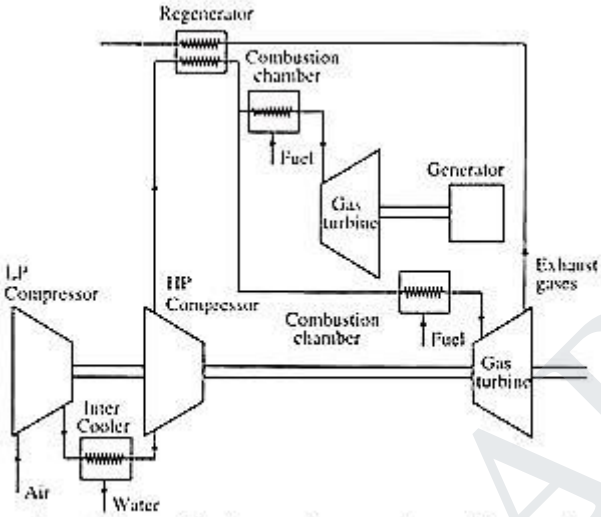


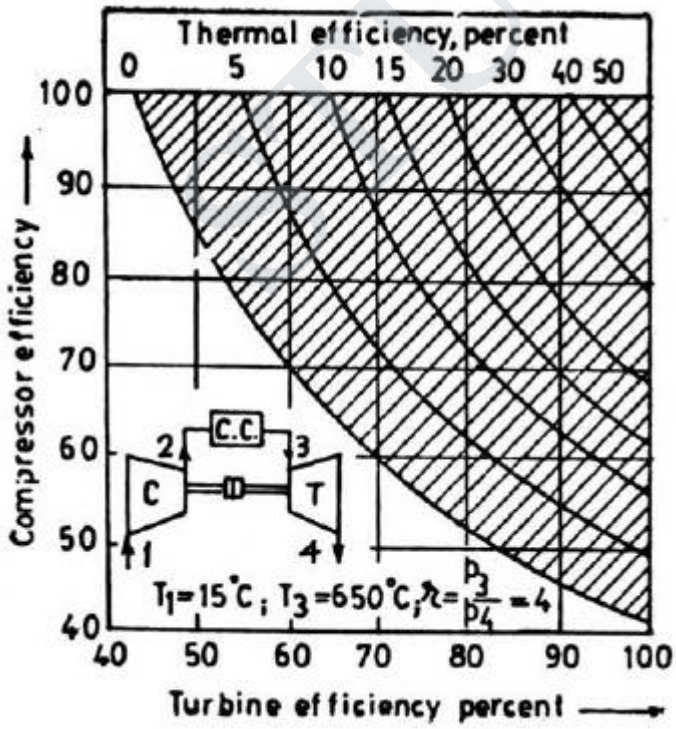
Fig. shows the gas turbine power plant with intercooler, reheater and heat exchanger for heat recovery. It should be noted that though by employing heat exchanger, intercooler and reheaters, there is some gain in the overall efficiency of the gas turbine plant this gain is at the expense of the increased weight and cost. Sometimes two turbines are connected in tandem (fig. 15-29) in which the compressed air after passing through the heat exchanger is split into two parts each part passing through a separate combustion chamber immediately before entry into each of the two turbines.

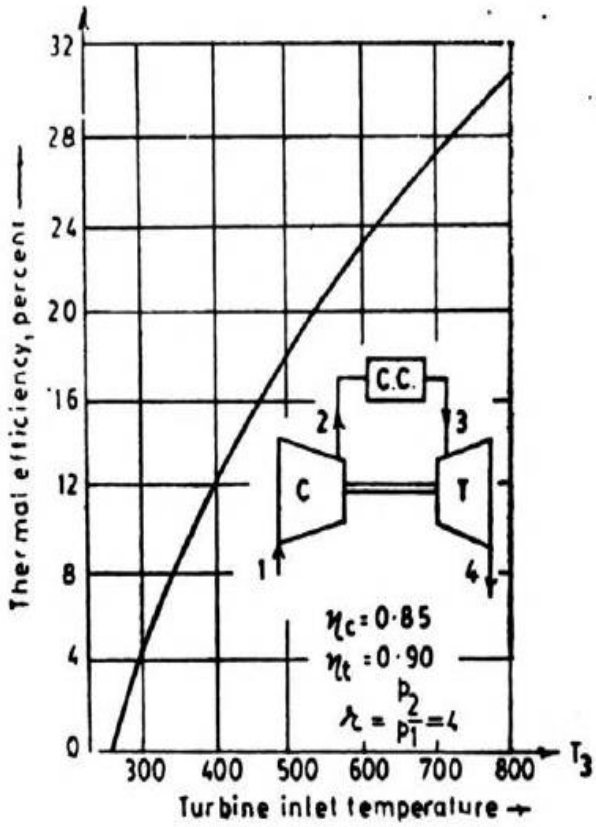


Gas turbine with intercooling, tandam turbine and regeneration with compressor driven by LP turbine

METHODS OF IMPROVING THERMAL EFFICIENCY OF SIMPLE CYCLE

A clear understanding of the thermodynamic cycle is necessary in order to appreciate the efficient operation of the gas turbine i.e. production of largest mechanical energy with least fuel consumed. To achieve this end in a gas turbine, following steps in design and operation are necessary:





Turbine and compressor efficiencies : The turbine and compressor should be designed to give highest efficiency. Efficiencies obtained in present day designed compressors and turbines are of the order of 85 to 90% and future progress in this direction will be slow. Axial flow compressors are efficient than centrifugal compressors. Figure illustrates the critical effect of compressor and turbine efficiencies on the thermal efficiency of the simple open gas turbine plant

Effect of compressor intake temperature : The intake temperature of air, affects the temperature at the end of compression. The compressor work for a fixed pressure ratio is proportional to the absolute temperature at the inlet to compression, that is, T |. Consequently, if the intake temperature is reduced and all other variables remain unchanged, the net power output is increased and the efficiency of the cycle is raised.

Effect of Turbine Inlet Temperature : The turbine efficiency is greatly increased by increasing turbine inlet temperature (fig. 4-5). A practical limitation to increasing the turbine inlet temperature, however, is the ability of materials available for the turbine blading to withstand the high rotative and thermal stresses. For a turbine inlet temperature range of 650° to 750°C, a simple gas turbine may realize an efficiency between 18 to 26 percent, depending upon design. Considerable effort is being made to find new materials, coatings and techniques, to increase the permissible turbine inlet temperature.

TURBINE MATERIALS:

- A key limiting factor in early jet engines was the performance of the materials available for the hot section (combustor and turbine) of the engine.
- The need for better materials spurred much research in the field of alloys and manufacturing techniques, and that research resulted in a long list of new materials and methods that make modern gas turbines possible.
- One of the earliest of these was Nimonic, used in the British Whittle engines.
- The development of superalloys in the 1940s and new processing methods such as vacuum induction melting in the 1950s greatly increased the temperature capability of turbine blades.
- Further processing methods like hot isostatic pressing improved the alloys used for turbine blades and increased turbine blade performance. Modern turbine blades often use nickel-based superalloys that incorporate chromium, cobalt, and rhenium.
- Aside from alloy improvements, a major breakthrough was the development of directional solidification (DS) and single crystal (SC) production methods. These methods help greatly increase strength against fatigue and creep by aligning grain boundaries in one direction (DS) or by eliminating grain boundaries altogether (SC).
- Ceramic matrix composites (CMC), where fibers are embedded in a ceramic matrix, are being developed for use in turbine blades. The main advantage of CMCs over conventional superalloys is their light weight and high temperature capability.
- SiC/SiC composites consisting of silicon matrix reinforced by silicon carbide fibers have been shown to withstand operating temperatures 200°-300 °F higher than nickel superalloys.
- GE Aviation successfully demonstrated the use of such SiC/SiC composite blades for the low-pressure turbine of its F414 jet engine.

PART - B

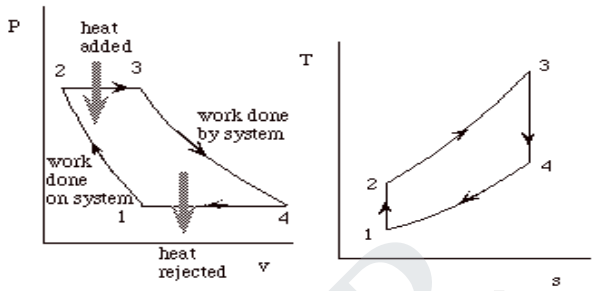
GAS TURBINE PLANT OR BRAYTON CYCLE

1. A gas turbine on works air standard brayton cycle. The intial condition of air is 25 °C and 1 bar the maximum pressure and temperature are limited to 3bar and 650°C. Determine i) Cycle efficiency ii)Heat supplied and rejected per kg of air iii)Work output iv) Exhaust temperature.

GIVEN:

$p_1 = 1\text{bar},\ T_1 = 25°C + 273$

$p_3 = 3\text{bar}, T_3 = 650°C + 273$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 3^{\frac{1.4-1}{1.4}} \times 298 \Rightarrow \mathbf{T_2 = 407.88\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(407.88 - 298) \Rightarrow \mathbf{W_C = 110.43\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(923 - 407.88) \Rightarrow \mathbf{Q_S = 517.7\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{923}{3^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 674.34\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(923 - 674.34) \Rightarrow \mathbf{W_T = 249.9\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(674.34 - 298) \Rightarrow \mathbf{Q_R = 378.22\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 517.7 - 378.22 \Rightarrow \mathbf{W_{net} = 139.48\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{139.48}{517.7} \Rightarrow \boldsymbol{\eta = 26.94\%}$$

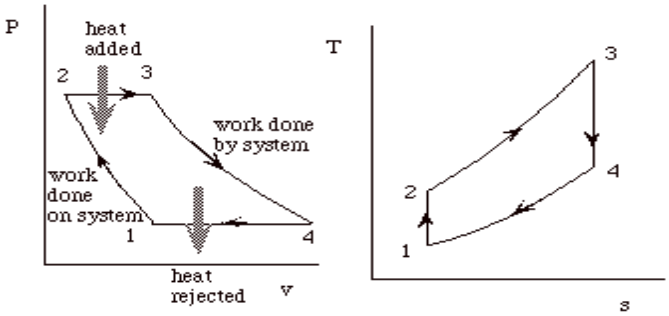
2. Air enters the compressor of a gas turbine plant operating on brayton cycle at 1 bar, 27 °C. The pressure ratio in the cycle is 6. If Wt = 2.5 Wc where Wt and Wc are the turbine and compressor work respectively, calculate the maximum temperature and the cycle efficiency.

GIVEN:

$p_1 = \mathbf{1bar},\ T_1 = \mathbf{27^\circ C + 273}$

$r_p = \mathbf{6bar},$

$W_t = 2.5\ W_c$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 6^{\frac{1.4-1}{1.4}} \times 300 \Rightarrow \mathbf{T_2 = 500.55\ K}$$

$$W_t = 2.5\ W_c \Rightarrow c_p(T_3 - T_4) = 2.5c_p(T_2 - T_1) \Rightarrow \left(T_3 - \frac{T_3}{1.67}\right) = 501.38 \Rightarrow \mathbf{T_3 = 1249.7\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(500.55 - 300) \Rightarrow \mathbf{W_C = 201.55\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(1249.7 - 500.55) \Rightarrow \mathbf{Q_S = 752.89\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{1249.7}{6^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 748.32\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(1249.7 - 748.32) \Rightarrow \mathbf{W_T = 503.89\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(748.32 - 300) \Rightarrow \mathbf{Q_R = 450.56\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 752.89 - 450.56 \Rightarrow \mathbf{W_{net} = 301.99\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{301.99}{752.89} \Rightarrow \boldsymbol{\eta = 40.11\%}$$

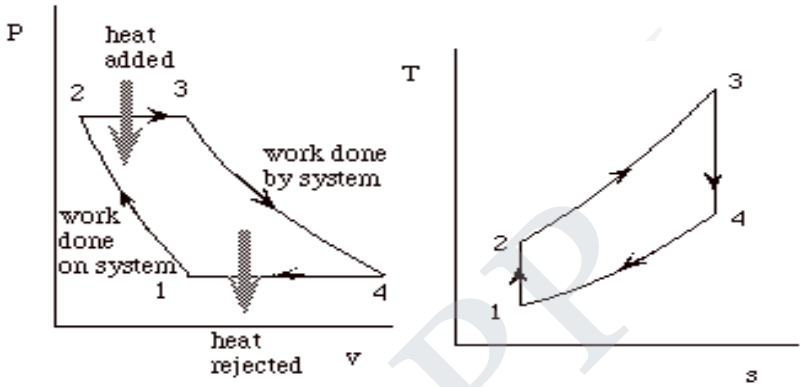
3. Consider an air standard cycle in which the air enters the compressor at 1.0 bar and 20°C. The pressure of air leaving the compressor is 3.5 bar and the temperature at turbine inlet is 600°C. Determine per kg of air, (i)Efficiency of the cycle, (ii) Heat supplied to air, (iii) Work available at the shaft,(iv) Heat rejected in the cooler and (v) Temperature of air leaving the turbine. For air γ = 1.4 and Cp = 1.005 kJ/kg K.

GIVEN:

$p_1 = \mathbf{1bar},\ T_1 = \mathbf{20^\circ C + 273}$

$p_3 = \mathbf{3.5bar},$

$T_3 = \mathbf{600^\circ C + 273}$



1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\ \frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\ \frac{\gamma-1}{\gamma}} \times T_1 = 3.5^{\frac{1.4-1}{1.4}} \times 293 \Rightarrow \mathbf{T_2 = 419.09\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(419.09 - 293) \Rightarrow \mathbf{W_C = 126.72\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(873 - 419.09) \Rightarrow \mathbf{Q_S = 456.18\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\ \frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\ \frac{\gamma-1}{\gamma}}} = \frac{873}{3.5^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 610.33\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(873 - 610.33) \Rightarrow \mathbf{W_T = 263.98\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(610.33 - 293) \Rightarrow \mathbf{Q_R = 318.92\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 456.18 - 318.92 \Rightarrow \mathbf{W_{net} = 137.26\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{137.26}{456.18} \Rightarrow \boldsymbol{\eta = 30.08\%}$$

4. A closed cycle ideal gas turbine plant operates between temperature limits of 800°C and 30°C and produces a power of 100 kW. The plant is designed such that there is no need for a regenerator. A fuel of calorific value 45000 kJ/kg is used. Calculate the mass flow rate of air through the plant and rate of fuel consumption. Assume Cp = 1 kJ/kg K and γ= 1.4.

GIVEN:

$T_1 = 30^\circ C + 273$

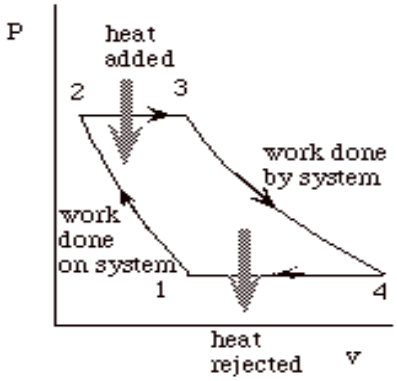
$T_3 = 800^\circ C + 273$

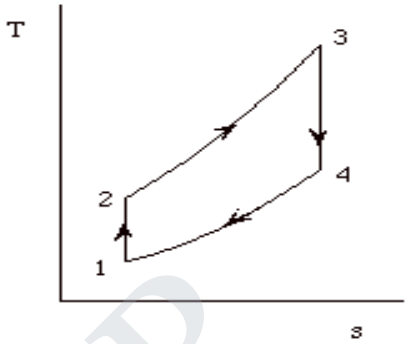
Assume $r_p = 6$

Power=100kW

CV=45000kJ/kg

$C_p = 1$ kJ/kg K, $\gamma = 1.4$





1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 6^{\frac{1.4-1}{1.4}} \times 303 \Rightarrow \mathbf{T_2 = 505.56\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(505.56 - 303) \Rightarrow \mathbf{W_C = 203.57\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(1073 - 505.56) \Rightarrow \mathbf{Q_S = 570.28\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{1073}{6^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 643.09\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(1073 - 643.09) \Rightarrow \mathbf{W_T = 432.06\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(643.09 - 303) \Rightarrow \mathbf{Q_R = 341.79\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 570.28 - 341.79 \Rightarrow \mathbf{W_{net} = 228.49\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{Work\ done}{Heat\ Supplied} = \frac{W_{net}}{Q_S} = \frac{228.49}{570.28} \Rightarrow \boldsymbol{\eta = 40.06\%}$$

mass flow rate

$$\dot{m} = \frac{\text{Power}}{\text{Work done}} = \frac{100}{228.49} \Rightarrow \dot{m} = 0.438\ \text{kg/s}$$

5. **A simple closed cycle gas turbine plant receives air at 1 bar and 15° C, and compresses it to 5 bar and then heats it to 800 °C in the heating chamber. The hot air expands in a turbine back to I bar. Calculate the power developed per kg of air supplied per second. Take Cp for air as I kJ/kg K.**

Solution. Given : $p_3 = p_4 = 1$ bar ; $T_4 = 15°\,C = 15 + 273 = 288$ K ; $p_1 = p_2 = 5$ bar ; $T_2 = 800°\,C = 800 + 273 = 1073$ K ; $c_p = 1$ kJ/kg K

The p-v and T-s diagram for the closed cycle gas turbine is shown in Fig. 32.3.

Let T_1 and T_3 = Temperature of air after isentropic compression and expansion (*i.e.* at points 1 and 3 respectively).

We know that for isentropic expansion 2-3,

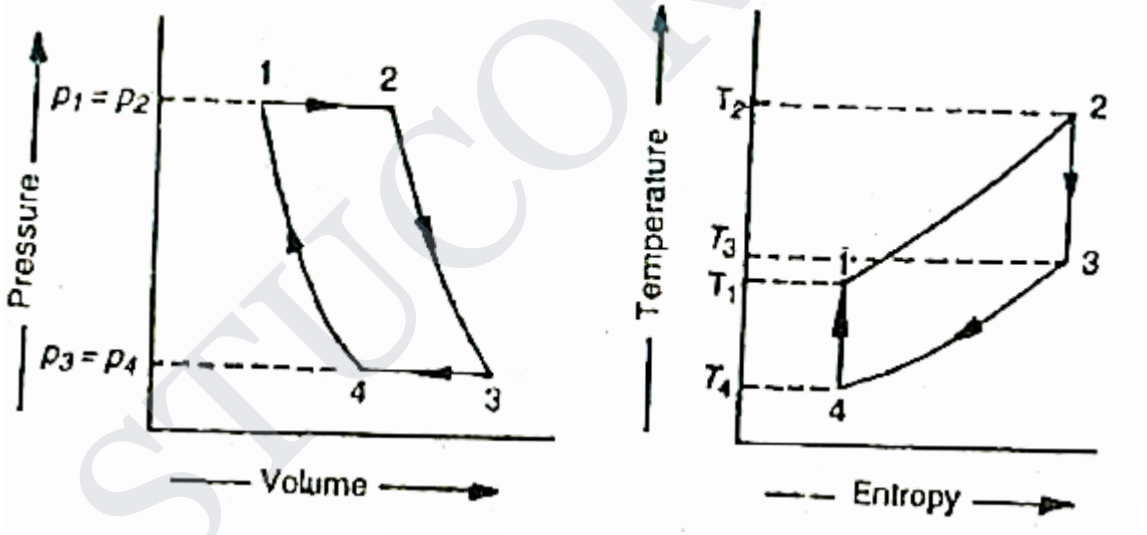
$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{5}\right)^{\frac{1.4-1}{1.4}} = (0.2)^{0.286} = 0.631$$

$$\therefore \quad T_3 = T_2 \times 0.631 = 1073 \times 0.631 = 677 \text{ K}$$

Similarly, for isentropic compression 4-1,

$$\frac{T_4}{T_1} = \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{5}\right)^{\frac{1.4-1}{1.4}} = 0.631$$

$$\therefore \quad T_1 = T_4 / 0.631 = 288 / 0.631 = 456 \text{ K}$$



We know that work developed by the turbine,

$$W_T = c_p (T_2 - T_3) = 1\,(1073 - 677) = 396 \text{ kJ/s}$$

and work required by the compressor,

$$W_C = c_p (T_1 - T_4) = 1\,(456 - 288) = 168 \text{ kJ/s}$$

∴ Net work done by the turbine,

$$W = W_T - W_C = 396 - 168 = 228 \text{ kJ/s}$$

and power developed, $P = 228$ kW **Ans.** ...(∵ 1 kJ/s = 1 kW)

6. **In an oil-gas turbine installation, it is taken at pressure of 1 bar and 22° C ana compressed to a pressure of 4 bar. The oil with a calorific value of 42 000 kJ/kg is burnt in the combustion chamber to raise the temperature of air to 550°C. If the air flows at the rate of 1.2 kg/s find the net power of the installation. Also-find air fuel ratio. Take Cp = 1.05 kJ/kg K.**

Solution. Given : $p_3 = p_4 = 1$ bar ; $T_4 = 27°\,C = 27 + 273 = 300$ K ; $p_1 = p_4 = 4$ bar ; $C = 42\,000$ kJ/kg ; $T_2 = 550°\,C = 550 + 273 = 823$ K ; $m = 1.2$ kg/s ; $c_p = 1.05$ kJ/kg K

Net power of the Installation

Let T_1 and T_3 = Temperature of air after isentropic compression and expansion respectively.

We know that for isentropic expansion 2-3 (Refer Fig. 32.3),

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = (0.25)^{0.286} = 0.673$$

$$\therefore \quad T_3 = T_2 \times 0.673 = 823 \times 0.673 = 553.9 \text{ K}$$

Similarly, for isentropic compression 4-1,

$$\frac{T_4}{T_1} = \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 0.673$$

$$\therefore \quad T_1 = T_4 / 0.673 = 300 / 0.673 = 445.8 \text{ K}$$

We know that work done by the turbine,

$$W_T = m\,c_p\,(T_2 - T_3) = 1.2 \times 1.05\,(823 - 553.9) = 339.1 \text{ kJ/s}$$

and work done by the compressor,

$$W_C = m\,c_p\,(T_1 - T_4)$$
$$= 1.2 \times 1.05\,(445.8 - 300) = 183.7 \text{ kJ/s}$$

∴ Net power of the installation,

$$= 339.1 - 183.7 = 154.4 \text{ kJ/s} = 154.4 \text{ kW} \quad \textbf{Ans.}$$

Air-fuel ratio

We know that heat supplied by the oil

$$= m\,c_p\,(T_2 - T_1)$$
$$= 1.2 \times 1.05(823 - 445.8) = 475.3 \text{ kJ/s}$$

∴ Mass of fuel burnt per second

$$= \frac{\text{Heat supplied}}{\text{Calorific value}} = \frac{475.3}{42\,000} = 0.011 \text{ kg}$$

and air-fuel ratio $= \dfrac{\text{Mass of air}}{\text{Mass of fuel}} = \dfrac{1.2}{0.011} = 109.1$ **Ans.**

7. A gas turbine plant consists of two stage compressor with perfect intercooler and a single stage turbine, if the plant works between the temperature limits of 300 K and 1000 K and 1 bar and 16 bar ;find the net power of the plant per kg of air. Take specific hear constant pressure as I kJ/kg K.

Solution. Given : $T_4 = 300$ K ; $T_2 = 1000$ K ; $p_3 = p_4 = 1$ bar ; $p_1 = p_2 = 16$ bar ; $c_p = 1$ kJ/kg K

The T-s diagram is shown in Fig. 32.6.

Let T_1, T_3, T_5 and T_6 = Temperature of air at corresponding points.

We know that for perfect intercooling, the intermediate pressure,

$$p_5 = p_6 = \sqrt{p_1 \times p_4} = \sqrt{16 \times 1} = 4 \text{ bar}$$

Now for the isentropic process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{1}{16}\right)^{\frac{1.4-1}{1.4}} = 0.453$$

$$\therefore \quad T_3 = T_2 \times 0.453$$

$$= 1000 \times 0.453 = 453 \text{ K}$$

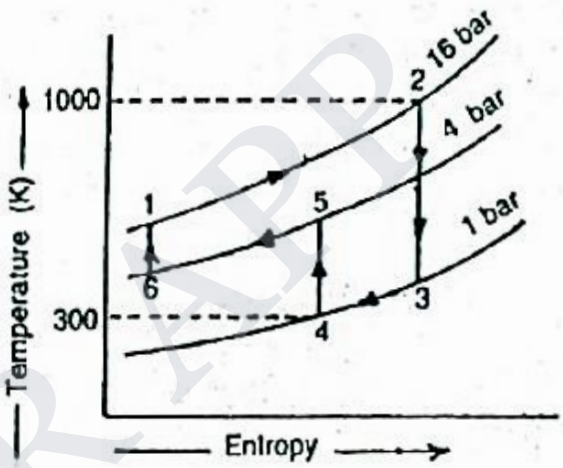


Fig. 32.6

Similarly for the isentropic process 4-5,

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 0.673$$

$$\therefore \quad T_5 = T_4 / 0.673 = 300 / 0.673 = 446 \text{ K}$$

We know that for perfect inter cooling,

$$T_1 = T_5 = 446 \text{ K}$$

and for isentropic process 6-1, $\frac{T_6}{T_1} = \left(\frac{p_6}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{16}\right)^{\frac{1.4-1}{1.4}} = 0.673$

$$\therefore \quad T_6 = T_1 \times 0.673 = 446 \times 0.673 = 300 \text{ K}$$

Now work done by the turbine per kg of air,

$$W_T = c_p (T_2 - T_3) = 1\,(1000 - 453) = 547 \text{ kJ/s}$$

and work absorbed by the compressor per kg of air,

$$W_C = c_p [(T_1 - T_6) + (T_5 - T_4)]$$

$$= 1 [(446 - 300) + (446 - 300)] = 292 \text{ kJ/s}$$

We know that work done by the plant per kg of air,

$$W = W_T - W_C = 547 - 292 = 255 \text{ kJ/s}$$

∴ Net power of the plant, $P = 255$ kW Ans.

8. In a gas turbine plans, the air is compressed in a single stage compressor from 1 bar to 9 bar and from an initial temperature of 300 K. The same air is then heated to a temperature of 800 K and then expanded in the turbine. The air is then reheated to a temperature of 800 K and then expanded in the second turbine. Find the maximum power that can be obtained frøn the installation, if the mass of air circulated per second is 2 kg. Take Cp = 1 kJ/kg K.

Solution. Given : $p_6 = p_5 = 1$ bar ; $p_1 = p_2 = 9$ bar ; $T_6 = 300$ K ; $T_2 = T_4 = 800$ K ; $m = 2$ kg/s ; $c_p = 1$ kJ/kg K

The T-s diagram of the reheat cycle is shown in Fig.

Let T_1, T_3, T_5 = Temperature of air at the corresponding points.

We know that for maximum power (or work), the intermediate pressure,

$$p_3 = p_4 = \sqrt{p_1 \times p_6}$$

$$= \sqrt{9 \times 1} = 3 \text{ bar}$$

We also know that for isentropic compression of air in the compressor (process 6-1),

$$\frac{T_1}{T_6} = \left(\frac{p_1}{p_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$= (9)^{0.286} = 1.873$$

∴ $T_1 = T_6 \times 1.873$

$$= 300 \times 1.873 = 562 \text{ K}$$

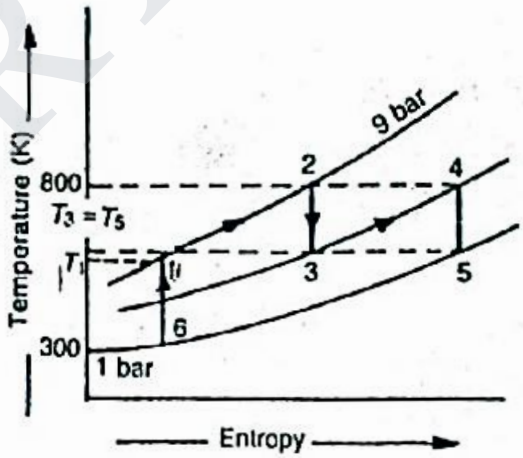


Fig. 32.9

For isentropic expansion of air in the first turbine (process 2-3),

$$\frac{T_2}{T_3} = \left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9}{3}\right)^{\frac{1.4-1}{1.4}} = (3)^{0.286} = 1.369$$

∴ $T_3 = T_2 / 1.369 = 800 / 1.369 = 584$ K

Similarly, for isentropic expansion of air in the second turbine (process 4-5),

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3}{1}\right)^{\frac{1.4-1}{1.4}} = (3)^{0.286} = 1.369$$

∴ $T_5 = T_4 / 1.369 = 800 / 1.369 = 584$ K

We know that work done by the turbine,

$$W_T = m c_p [(T_2 - T_3) + (T_4 - T_5)]$$

$$= 2 \times 1 [(800 - 584) + (800 - 584)] = 864 \text{ kJ/s}$$

and work absorbed by the compressor,

$$W_C = m c_p (T_1 - T_6) = 2 \times 1 (562 - 300) = 524 \text{ kJ/s}$$

We also know that net work available,

$$W = W_T - W_C = 864 - 524 = 340 \text{ kJ/s}$$

∴ Power that can be obtained from the installation,

$$P = 340 \text{ kJ/s} = 340 \text{ kW} \quad \textbf{Ans.}$$

9. A closed cycle gas turbine consists of a two stage compressor with perfect intercooler and a two stage turbine, with a reheater. All the components are mounted on the same shaft. The pressure and temperature at the inlet of the low pressure compressor are 2 bar and 300 K. The maximum pressure and temperature are limited to 8 bar and 1000 K. The gases are heated in the reheater to 1000 K Calculate mass of fluid circulated in the turbine, if the net power developed by the turbine is 370kW Also find the amount of heat supplied per second from the external source.

Solution. Given : $p_6 = p_5 = 2$ bar ; $T_6 = 300$ K ; $p_1 = p_2 = 8$ bar ; $T_2 = 1000$ K ; $T_4 = 1000$ K ; $P = 370$ kW

The T-s diagram of the reheat cycle is shown in Fig. 32.10.

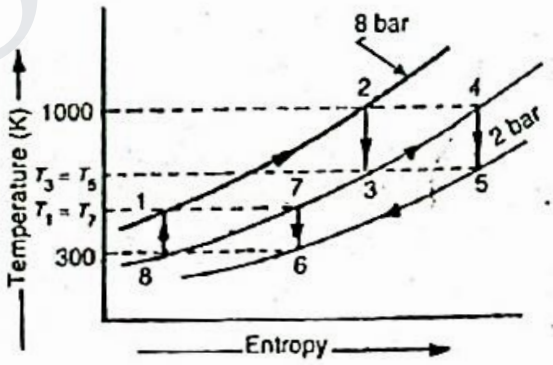


Fig. 32.10

Mass of fluid circulated in the turbine

Let m = Mass of air circulated in the turbine,

T_1, T_3, T_5, T_7, T_8 = Temperature of air at the corresponding points.

We know that for perfect cooling, the intermediate pressure,

$$p_8 = p_7 = p_3 = p_4 = \sqrt{p_1 \times p_6} = \sqrt{8 \times 2} = 4 \text{ bar}$$

Now for the isentropic process 6-7,

$$\frac{T_6}{T_7} = \left(\frac{p_6}{p_7}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{2}{4}\right)^{\frac{1.4-1}{1.4}} = (0.5)^{0.286} = 0.82$$

$$\therefore \quad T_7 = T_6 / 0.82 = 300 / 0.82 = 366 \text{ K}$$

We know that for perfect cooling,

$$T_1 = T_7 = 366 \text{ K}$$

Again, for the isentropic process 8-1,

$$\frac{T_8}{T_1} = \left(\frac{p_8}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{8}\right)^{\frac{1.4-1}{1.4}} = (0.5)^{0.286} = 0.82$$

$$\therefore \quad T_8 = T_1 \times 0.82 = 366 \times 0.82 = 300 \text{ K}$$

and for the isentropic process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{8}\right)^{\frac{1.4-1}{1.4}} = 0.82$$

$$\therefore \quad T_3 = T_2 \times 0.82 = 1000 \times 0.82 = 820 \text{ K}$$

Similarly, for the isentropic process 4-5,

$$\frac{T_5}{T_4} = \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{2}{4}\right)^{\frac{1.4-1}{1.4}} = 0.82$$

$$\therefore \quad T_5 = T_4 \times 0.82 = 1000 \times 0.82 = 820 \text{ K}$$

We know that work done by the turbine,

$$W_T = m\,c_p\,[(T_2 - T_3) + (T_4 - T_5)]$$

$$= m \times 1\,[(1000 - 820) + (1000 - 820)] = 360\,m \text{ kJ/s}$$

and work absorbed by the compressor,

$$W_C = m\,c_p\,[(T_1 - T_8) + (T_7 - T_6)]$$

$$= m \times 1\,[(366 - 300) + (366 - 300)] = 132\,m \text{ kJ/s}$$

∴ Net work done by the turbine,

$$W = W_T - W_C = 360\,m - 132\,m = 228\,m \text{ kJ/s}$$

We also know that power developed by the turbine (P),

$$370 = 228\,m \quad \text{or} \quad m = 1.62 \text{ kg/s} \quad \textbf{Ans.}$$

Heat supplied from the external source

We know that heat supplied from the external source

$$= m\,c_p\,[(T_2 - T_1) + (T_4 - T_3)]$$

$$= 1.62 \times 1\,[(1000 - 366) + (1000 - 820)] \text{ kJ/s}$$

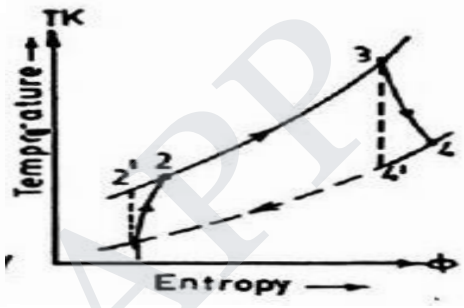
$$= 1318.7 \text{ kJ/s} \quad \textbf{Ans.}$$

10. A simple constant pressure gas turbine plant draws in air at 30°C and compresses it through pressure ratio of 6. The air passes to the combustion chamber and after combustion of fuel, gases enter the turbine at a temperature of 787°C and expand to the initial low pressure. Assuming isentropic efficiencies of both the compressor and turbine a s 89%, calculate: (a) the thermal efficiency of the plant, and (b) the percentage increase in the thermal efficiency if the air temperature at compressor inlet is 300° C and other parameters remain the same. Take γ = 1.4 for air and gases.

(a) Referring to fig. , $T_1 = 30 + 273 = 303$ K; $T_3 = 787 + 273 = 1{,}060$ K;

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = r_p = 6; \quad \eta_c = \eta_t = 89\%; \quad \gamma = 1{\cdot}4$$

For isentropic compression,



$$\frac{T_2'}{T_1} = \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{0{\cdot}4}{1{\cdot}4}} = 1{\cdot}668$$

$$\therefore\ T_2' = T_1 \times 1{\cdot}668 = 303 \times 1{\cdot}668 = 505{\cdot}4\ K$$

Further, $\eta_c = \dfrac{T_2' - T_1}{T_2 - T_1}$

$$\therefore\ T_2 - T_1 = \frac{T_2' - T_1}{\eta_c} = \frac{505{\cdot}4 - 303}{0{\cdot}89} = \frac{202{\cdot}4}{0{\cdot}89} = 227{\cdot}4$$

$\therefore\ T_2 = 227{\cdot}4 + 303 = 530{\cdot}4\ K$ (actual temp. of air after compression)

Compression work, $W_c = k_p(T_2 - T_1) = k_p \times 227{\cdot}4$ kJ/kg

Heat supplied, $Q_s = k_p \times (T_3 - T_2)$

$= k_p(1{,}060 - 530{\cdot}4) = k_p \times 529{\cdot}6$ kJ/kg

Now, for isentropic expansion in turbine,

$$\frac{T_3}{T_4'} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{0{\cdot}4}{1{\cdot}4}} = 1{\cdot}668$$

$$\therefore\ T_4' = \frac{T_3}{1{\cdot}668} = \frac{1{,}060}{1{\cdot}668} = 635{\cdot}5\ K$$

Now, $\eta_t = \dfrac{T_3 - T_4}{T_3 - T_4'}$

$$\therefore\ T_3 - T_4 = \eta_t(T_3 - T_4') = 0{\cdot}89(1{,}060 - 635{\cdot}5) = 377{\cdot}8$$

Turbine work. $W_t = k_p(T_3 - T_4) = k_p \times 377{\cdot}8$ kJ/kg

Thermal efficiency, $\eta_T = \dfrac{W_t - W_c}{Q_s} = \dfrac{k_p \times 377{\cdot}8 - k_p \times 227{\cdot}4}{K_p \times 529{\cdot}6}$

$$= \frac{377{\cdot}8 - 227{\cdot}4}{529{\cdot}6} = \frac{150{\cdot}4}{529{\cdot}6} = 0{\cdot}284 \text{ or } \mathbf{28{\cdot}4\%}$$

(b) Now, $T_1 = -30 + 273 = 243$ K; $T_3 = 787 + 273 = 1{,}060$ K;

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = r_p = 6; \quad \gamma = 1{\cdot}4; \quad \eta_c = \eta_t = 89\%$$

$$\frac{T_2'}{T_1} = \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{0.4}{1.4}} = 1.668$$

$$\therefore\ T_2' = T_1 \times 1.668 = 243 \times 1.668 = 405.3\ K$$

$$\text{But, } \eta_c = \frac{T_2' - T_1}{T_2 - T_1} \text{ i.e. } 0.89 = \frac{405.3 - 243}{T_2 - T_1}$$

$$\therefore\ T_2 - T_1 = \frac{405.3 - 243}{0.89} = \frac{162.3}{0.89} = 182.36$$

$$\therefore\ T_2 = 182.36 + 243 = 425.36\ K$$

Compression work per kg, $W_c = k_p(T_2 - T_1) = k_p \times 182.36$ kJ

Turbine work per kg, $W_t = k_p\ (T_3 - T_4)$

$= k_p \times 377.8$ kJ[same as in part (a)]

Heat supplied per kg, $Q_s = k_p(T_3 - T_2)$

$= k_p(1{,}060 - 425.36) = k_p \times 634.64$ kJ

$$\text{Thermal efficiency, } \eta_T = \frac{W_t - W_c}{Q_s} = \frac{k_p \times 377.8 - k_p \times 182.36}{k_p \times 634.64}$$

$$= \frac{377.8 - 182.36}{634.64} = \frac{195.44}{634.64} = 0.3079 \text{ or } 30.79\%$$

Thus, % increase in thermal efficiency,

$$\eta_T = \frac{30.79 - 28.4}{28.4} \times 100 = 8.42$$

11. A simple constant pressure open cycle gas turbine plant draws air at 100 Kpa (1 bar) and 17°C and compresses it through a pressure ratio of 4. The air then passes to the combustion chamber and after combustion of fuel, the gases enter the turbine at a temperature of 650°C and expand to 100 kPa. Assuming the isentropic efficiency of both the compressor and the turbine as 85 per cent, calculate: (a) the power required to drive the compressor if it has to handle 2 kg of air per second, (b) the power developed by the turbine, (c) the net plant work output per kg of air, (d) the thermal efficiency of the plant, and (e) the work ratio of the plant. Assume kp = 1026 KJ/kg K and 7 = 14 for both air and gases. Neglect the mass of fuel burnt and the loss of pressure in the combustion chamber.

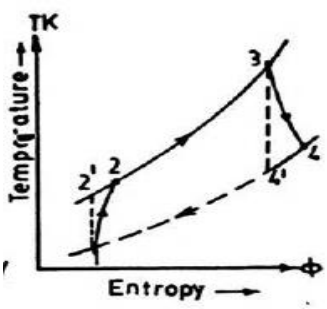
(a) Referring to fig. , the following data is available:

$T_1 = 17 + 273 = 290$ K; $T_3 = 650 + 273 = 923$ K; $p_1 = 100$ kPa

$\frac{p_2}{p_1} = \frac{p_3}{p_4} = 4$; $k_p = 1.026$ kJ/kg K; $\gamma = 1.4$

Now, for isentropic compression, $\frac{T_2'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$

$$\therefore\ T_2' = T_1\left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} = 290 \times \{4\}^{\frac{0.4}{1.4}} = 290 \times 1.485 = 431\ \text{K}$$



$\therefore$ Isentropic temperature rise $= T_2' - T_1 = 431 - 290 = 141$,

Now, $\eta_c = \dfrac{\text{Isentropic temperature rise}}{\text{Actual temperature rise}} = \dfrac{T_2' - T_1}{T_2 - T_1} = 0{\cdot}85$

$\therefore\ T_2 - T_1 = \dfrac{T_2' - T_1}{0{\cdot}85} = \dfrac{141}{0{\cdot}85} = 165{\cdot}5$

$\therefore\ T_2 = 165{\cdot}5 + T_1 = 165{\cdot}5 + 290 = 455{\cdot}5$ K

Again for isentropic expansion,

$$\frac{T_3}{T_4'} = \left\{\frac{p_3}{p_4}\right\}^{\frac{\gamma-1}{\gamma}} = \{4\}^{\frac{0{\cdot}4}{1{\cdot}4}} = 1{\cdot}485$$

$\therefore\ T_4' = \dfrac{T_3}{1{\cdot}485} = \dfrac{923}{1{\cdot}485} = 621\ K$

$\therefore$ Isentropic temperature drop $= T_3 - T_4' = 923 - 621 = 302$

$\therefore\ \eta_t = \dfrac{\text{Actual temp. drop}}{\text{Isentropic temp. drop}} = \dfrac{T_3 - T_4}{T_3 - T_4'} = 0{\cdot}85$

$\therefore\ T_3 - T_4 = (T_3 - T_4') \times 0{\cdot}85 = 302 \times 0{\cdot}85 = 257$

Compression work, $W_c = k_p(T_2 - T_1)$

$= 1{\cdot}026 \times 165{\cdot}5 = 169{\cdot}75$ kJ/kg of air

$\therefore$ Power required to drive the compressor

$= 169{\cdot}75 \times 2 = 339{\cdot}5$ kJ/sec. = **339·5 kW**

(b) Turbine work output, $W_c = k_p(T_3 - T_4)$

$= 1{\cdot}026 \times 257 = 264{\cdot}7$ kJ/kg of air

$\therefore$ Turbine power $= 264{\cdot}7 \times 2 = 529{\cdot}4$ kJ/sec. or **529·4 kW.**

(c) Net plant work output $= W_t - W_c = 264{\cdot}7 - 169{\cdot}75 =$ **94·95 kJ/kg of air**

(d) Heat supplied $= k_p\ (T_3 - T_2) = 1{\cdot}026\ (923 - 455{\cdot}5) =$ **479·7 kJ/kg of air**

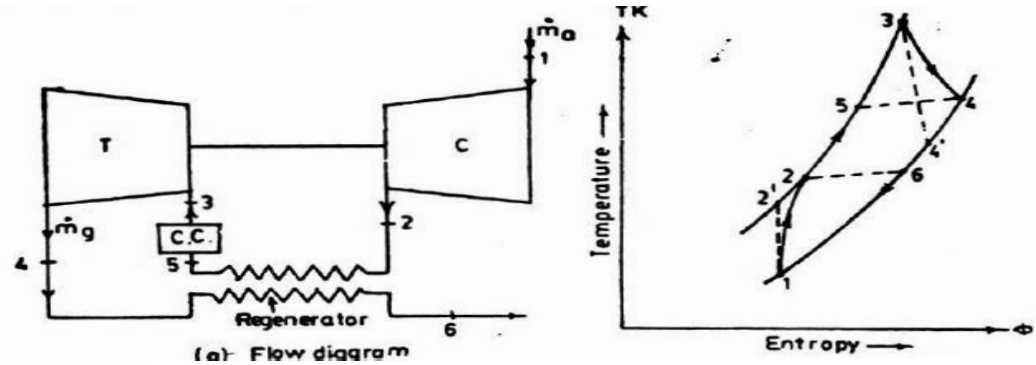
$\therefore$ Thermal efficiency of the plant,

$$\eta_T = \frac{\text{Turbine work–Compressor work}}{\text{Heat supplied}}$$

$$= \frac{264{\cdot}7 - 169{\cdot}75}{479{\cdot}7} = \frac{94{\cdot}95}{479{\cdot}7} = 0{\cdot}1979 \text{ or } 19{\cdot}79\%$$

(e) Work ratio $= \dfrac{\text{Net plant work output per kg}}{\text{Turbine work per kg}} = \dfrac{94{\cdot}95}{264{\cdot}7} = 0{\cdot}3587$

12. In a gas turbine installation the compressor takes in air at a temperature of 20°C and compresses it to four times the initial pressure with an isentropic efficiency of 84%. The air is then passed through a heat exchanger and heated by the turbine exhaust before reaching the combustion chamber. In the heat exchanger,80% of the available heat is given to compressed air. The maximum temperature after constant pressure combustion is 580°C and the isentropic efficiency of the turbine is 75%. Determine the overall efficiency of the plant. Take γ= 1.4 and kp=1.005 kJ/kg K for air and gases.



(a) Flow diagram

$T_1 = 20 + 273 = 293$ K; $T_3 = 580 + 273 = 853$ K; $\frac{p_2}{p_1} = 4 = \frac{p_3}{p_4}$

Now, $\frac{T_2'}{T_1} = \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}}$

$\therefore T_2' = T_1 \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} = 293 \times (4)^{\frac{0.4}{1.4}} = 293 \times 1.486 = 435.4\ K$

But, $\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$ *i.e.* $0.84 = \frac{435.4 - 293}{T_2 - T_1}$ $\therefore T_2 - T_1 = \frac{142.4}{0.84} = 169.5\ K$

$\therefore T_2 = 293 + 169.5 = 462.5K$

Now, $\frac{T_3}{T_4'} = \left\{\frac{p_3}{p_4}\right\}^{\frac{\gamma-1}{\gamma}} = (4)^{\frac{0.4}{1.4}} = 1.486$ i.e. $T_4' = \frac{T_3}{1.486} = \frac{853}{1.486} = 574$ K

$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'}$ i.e. $0.75 = \frac{T_3 - T_4}{853 - 574}$

$\therefore T_3 - T_4 = 0.75 \times 279 = 209.25$ K

$\therefore T_4 = T_3 - 209.25 = 853 - 209.25 = 643.75$ K

Effectiveness of heat exchanger $= \frac{T_5 - T_2}{T_4 - T_2}$

i.e., $0.8 = \frac{T_5 - 462.5}{643.75 - 462.5}$ $\therefore T_5 = 607.5$ K

Heat supplied/kg $= k_p(T_3 - T_5) = 1.005 \times (853 - 607.5) = 246.73$ kJ

Compressor work/kg $= k_p(T_2 - T_1) = 1.005(462.5 - 293) = 170.35$ kJ

Turbine work/kg $= k_p(T_3 - T_4) = 1.005(853 - 643.75) = 210.3$ kJ

Net work output/kg = Turbine work/kg − Compressor work/kg

$= 210.3 - 170.35 = 39.95$ kJ/kg

Overall thermal efficiency $= \frac{\text{Net work output per kg}}{\text{Heat supplied per kg}}$

$= \frac{39.95}{246.73} = 0.1619$ or **16.19%**

7. **A simple open cycle gas turbine takes in air at 1 bar (100 kpa) and 15°C and compresses it to 5 times the initial pressure, the isentropic efficiency of compressor being 85 per cent. The air passes to the combustion chamber, and after combustion, the gases enter the turbine at a temperature of 557°C and expand to 1 bar, with an isentropic efficiency of 82%. Estimate the mass flow of air and gases in kg/min for a net power output of 1500 kW, making the following assumptions. Fall of pressure through the combustion system= 0.09 bar, kp = 1025 kJ/kg K and y = 14, far both air and combustion gases. Assume that mass flows through the turbine and compressor are equal.**

$$\frac{T_2'}{T_1} = \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}}$$

$$\therefore T_2' = T_1 \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} = 288\,(5)^{\frac{0.4}{1.4}} = 288 \times 1.584 = 456.2\ K$$

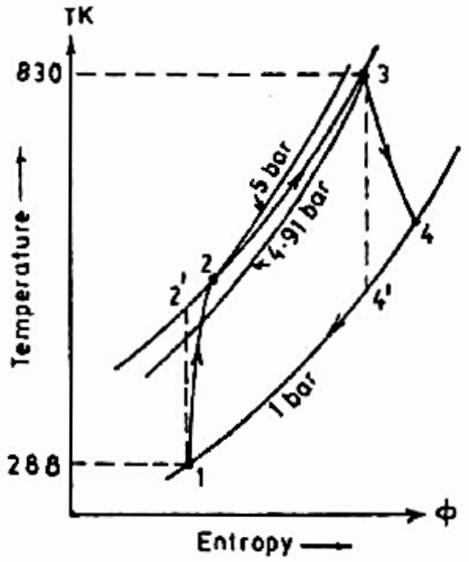


Fig. 4-11 Cycle on $T-\phi$ diagram.

Isentropic temperature rise

$$= T_2' - T_1 = 456{\cdot}2 - 288 = 168{\cdot}2$$

$$\eta_c = \frac{\text{Isentropic temp. rise}}{\text{Actual temp.rise}}$$

$$= \frac{T_2' - T_1}{T_2 - T_1} = 0{\cdot}85$$

$$\therefore\ T_2 - T_1 = \frac{T_2' - T_1}{0{\cdot}85} = \frac{168{\cdot}2}{0{\cdot}85} = 197{\cdot}88$$

$$\therefore\ T_2 = 197{\cdot}88 + 288 = 485{\cdot}88\ K$$

Compressor work, $W_c = k_p(T_2 - T_1)$

$= 1{\cdot}025 \times 197{\cdot}88 = 202{\cdot}83$ kJ/kg of air.

Since, there is a fall of pressure of 0·09 bar through the combustion system, the pressure ratio across the turbine now is $\frac{5 - 0{\cdot}09}{1} = 4{\cdot}91$.

$$\text{Now, } \frac{T_3}{T_4'} = \left\{\frac{p_3}{p_4}\right\}^{\frac{\gamma-1}{\gamma}}$$

$$\therefore\ T_4' = \frac{T}{\left\{\frac{p_3}{p_4}\right\}^{\frac{\gamma-1}{\gamma}}} = \frac{830}{(4{\cdot}91)^{\frac{0{\cdot}4}{1{\cdot}4}}} = \frac{830}{1{\cdot}575} = 527\ K$$

Isentropic temp. drop $= T_3 - T_4' = 830 - 527 = 303\ K$

Actual temperature drop $= T_3 - T_4 = 0{\cdot}82 \times 303 = 248{\cdot}86\ K$

$\therefore$ Turbine work output, $W_t = k_p(T_3 - T_4)$

$= 1{\cdot}025\,(248{\cdot}46) = 256{\cdot}67$ kJ/kg of air.

Of this, 202·83 kJ/kg of air are absorbed in driving the compressor.

$\therefore$ Net work output of the turbine plant per kg of air

= Turbine output − Compressor work = 256·67 − 202·83 = 53·84 kJ

$\therefore$ Net output power in kW = mass flow of air in kg per second × net work output in kJ/kg of air.

i.e., $1{,}500 = m \times 53{\cdot}84$

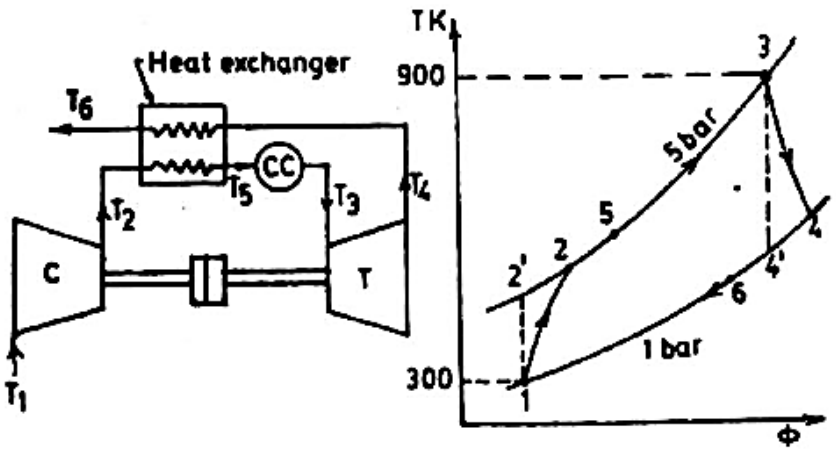
$$\therefore\ m = \frac{1{,}500}{53{\cdot}84} = 27{\cdot}86 \text{ kg/sec.}$$

$\therefore$ Mass flow of air and gases in kg/min. = 27·86 × 60 = **1,671·6**

8. In an open cycle constant pressure gas turbine plant, air enters the compressor at a pressure of 1 bar (100 kPa) and at a temperature of 27'C and leaves it at a pressure of 5 bar (500 kPa). The gases enter the turbine at a temperature of 627°C. The gases are expanded in the turbine to the initial pressure of 1 bar (100 kpa). The isentropic efficiency of the turbine is 84 per cent and that of the compressor is 86 per cent. Determine the thermal efficiency of the gas turbine plant: (a) When a regenerator (heat exchanger) with 70 per cent effectiveness (efficiency) is used to preheat the compressed air before it enters the combustion chamber and (b) if no heat exchanger is used.

Assume no pressure lo sses in the connecting pipes, combustion chamber and heat exchanger. Take kp = 1-005 kJ/kg K and y = 14 both for air and gases.

$T_1 = 27 + 273 = 300$ K; $T_3 = 627 + 273 = 900$ K; $p_1 = 1$ bar;
$p_2 = 5$ bar; $\gamma = 1{\cdot}$; $k_p = 1{\cdot}005$ kJ/kg K.



(a) Flow diagram (b) T-Φ diagram

Fig. 4-12

(a) With Heat Exchanger :

For isentropic compression, $\dfrac{T_2'}{T_1} = \left\{\dfrac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}}$

∴ Temp. of air after insentropic compression,

$$T_2' = T_1 = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300\left(\frac{5}{1}\right)^{\frac{0.4}{1.4}} = 300 \times 1{\cdot}584 = \mathbf{475{\cdot}2\ K}$$

Isentropic temp. rise $= T_2' - T_1 = 475{\cdot}2 - 300 = 175{\cdot}2$ K

Isentropic efficiency of compressor, $\eta_c = \dfrac{T_2' - T_1}{T_2 - T_1}$

∴ Actual temperatures rise, $T_2 - T_1 = \dfrac{T_2' - T_1}{\eta_c} = \dfrac{175{\cdot}2}{0{\cdot}86} = 203{\cdot}7$

∴ Actual temp. after compression, $T_2 = 203{\cdot}7 + 300 = 503{\cdot}7\ K$

Again for isentropic expansion in the turbine,

$$\frac{T_3}{T_4'} = \left\{\frac{p_3}{p_4}\right\}^{\frac{\gamma-1}{\gamma}} = (5)^{\frac{0.4}{1.4}} = 1{\cdot}584$$

∴ Final temp. after expansion in the turbine, if the expansion were isentropic,

$$T_4' = \frac{T_3}{1{\cdot}584} = \frac{900}{1{\cdot}584} = 568{\cdot}2\ \text{K}$$

Isentropic temp. fall $= T_3 - T_4' = 900 - 568{\cdot}2 = 331{\cdot}8$

Isentropic eff. of turbine, $\eta_t = \dfrac{\text{Actual temp. fall}}{\text{Isentropic temp. fall}} = \dfrac{T_3 - T_4}{T_3 - T_4'}$

∴ $T_3 - T_4 = \eta_t(T_3 - T_4') = 0{\cdot}84 \times 331{\cdot}8 = 278{\cdot}8$

∴ Actual temp. of gases after expansion in the turbine,

$T_4 = 900 - 278{\cdot}7 = 621{\cdot}3$ K

With heat exchanger, the air after compression is preheated from T_2 to T_3 and the fuel is supplied to raise the temperature of compressed air from T_5 to T_3, *i.e.*, the heat supplied per kg of air = $k_p\ (T_3 - T_5)$. The actual temperature after compression, T_2 is 503·7 K, so that the temperature drop $(T_4 - T_2)$ = 621·3 – 503·7 = 117·6°C is available for heating the compressed air by the exhaust heat exchanger. Since only 70% of the available heat is given to the compressed air, the actual temperature drop available for heating the compressed air will be

$$0{\cdot}7\,(T_4 - T_2) = 0{\cdot}7\,(621{\cdot}3 - 503{\cdot}7) = 82{\cdot}3$$

In passing through the heat exchanger, the compressed air will be heated from T_2 to T_5. T_5 can be found from heat balance equation for the heat exchanger,

i.e., $T_5 - T_2 = 0{\cdot}7\,(T_4 - T_2)$, where 0·7 is the effectiveness of the heat exchanger.

$\therefore\ T_5 = 0{\cdot}7\,(621{\cdot}3 - 503{\cdot}7) + 503{\cdot}7 = 586\ K$

Heat supplied per kg of air = $k_p\,(T_3 - T_5) = k_p\,(900 - 586) = 314\ k_p$ kJ

Work required to drive the compressor per kg of air

$= k_p\,(T_2 - T_1) = k_p\,(503{\cdot}7 - 300) = 203{\cdot}7\ k_p$ kJ

Turbine work output per kg = $k_p\,(T_3 - T_4) = k_p\,(900 - 621{\cdot}3) = 278{\cdot}7\ k_p$ kJ

$\therefore$ Net work output per kg = $278{\cdot}7\ k_p - 203{\cdot}7\ k_p = 75\ k_p$ kJ

$\therefore$ Thermal efficiency of the plant

$$= \frac{\text{Net work output of the plant per kg}}{\text{Heat supplied per kg}} = \frac{75\ k_p}{314\ k_p} = 0{\cdot}2388 \text{ or } 23{\cdot}88\%$$

(b) Without heat exchanger :

Work required to drive the compressor per kg

$= k_p\,(T_2 - T_1) = k_p\,(503{\cdot}7 - 300) = 203{\cdot}7\ k_p$ kJ

Turbine work output per kg

$= k_p\,(T_3 - T_4) = k_p\,(900 - 621{\cdot}3) = 278{\cdot}7\ k_p$ kJ

$\therefore$ Net work output of the plant per kg

$= 278{\cdot}7\ k_p - 203{\cdot}7\ k_p = 75\ k_p$ kJ

Heat supplied per kg = $k_p\,(T_3 - T_2) = k_p\,(900 - 503{\cdot}7) = 396{\cdot}3\ k_p$ kJ

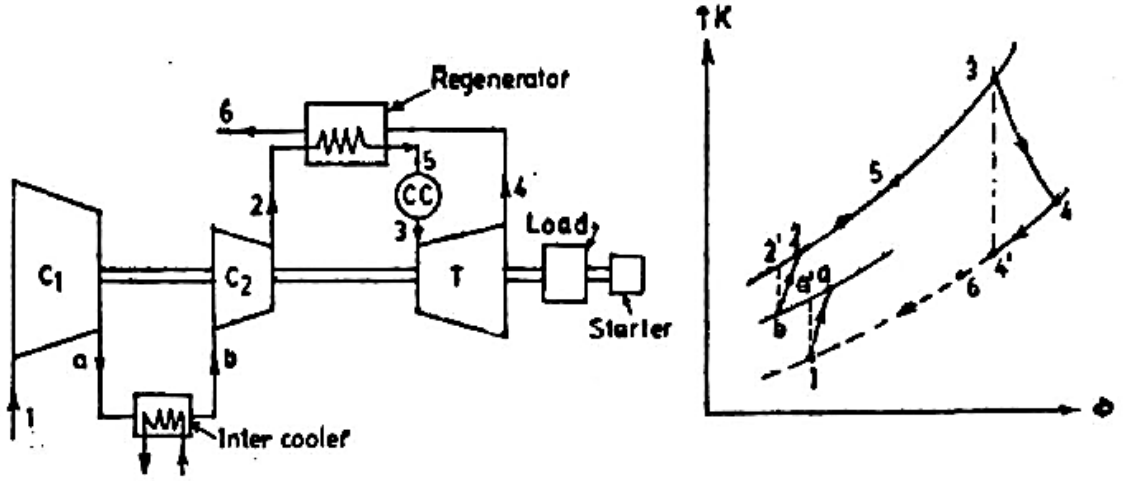
$\therefore$ Thermal efficiency of the plant without heat exchanger,

$$\eta_T = \frac{\text{Net work output of the plant per kg}}{\text{Heat supplied per kg}}$$

$$= \frac{75 k_p}{396{\cdot}3 k_p} = 0{\cdot}1925 \text{ or } 19{\cdot}25\%$$

It should be noted that improvement in the thermal efficiency of plant due to introduction of the regenerator (heat exchanger) is achieved at the expense of increased weight and cost of the plant.

9. In a single-shaft constant pressure open cycle gas turbine plant, a two-stage compression with intercooling and regeneration is employed. The inlet pressure and temperature are 1 bar and 20°C. The pressure ratio in each stage is 2 5 and isentropic efficiency of the compressor is 85 per cent. The effectiveness of the intercooler is 75 per cent. The gases enter the turbine at a temperature of 700* C. The gases are expanded in a turbine to the initial pressure of 1 bar. The isentropic efficiency of the turbine is 82 per cent. The effectiveness (thermal ratio) of the regenerator (heat exchanger) is 75 per cent. Assuming air to be working medium throughout the cycle with kp = 1005 kJ/kg K and γ = 1-4, find the thermal efficiency of gas turbine plant.



(a) Flow diagram (b) T-φ diagram

Fig 4-13

Referring to fig. 4–13,

$T_1 = 20 + 273 = 293\ K$; $T_3 = 700 + 273 = 973\ K$; $p_1 = 1$ bar; $\frac{p_a}{p_1} = 2{\cdot}5$;

$\frac{p_2}{p_b} = 2{\cdot}5$; $\frac{p_2}{p_1} = \frac{p_2}{p_b} \times \frac{p_a}{p_1} = 2{\cdot}5 \times 2{\cdot}5 = 6{\cdot}25$; $\frac{p_3}{p_4} = 6{\cdot}25$;

$\eta_{c1} = \eta_{c2} = 85\%$; $\eta_t = 82\%$; Effectiveness of intercooler = 75%;

Effectiveness of heat exchanger = 75% ; $k_p = 1{\cdot}005$ kJ/kg K, and $\gamma = 1{\cdot}4$.

Now, $\frac{T_a'}{T_1} = \left\{\frac{p_a}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} = (2{\cdot}5)^{\frac{0{\cdot}4}{1{\cdot}4}} = (2{\cdot}5)^{0{\cdot}286} = 1{\cdot}3$

$\therefore\ T_a' = T_1 \times 1{\cdot}3 = 293 \times 1{\cdot}3 = 381$ K

Now, $\eta_{c1} = \frac{\text{Isentropic increase in temperature}}{\text{actual increase in temperature}} = \frac{T_a' - T_1}{T_a - T_1}$

$\therefore\ T_a - T_1 = \frac{T_a' - T_1}{\eta_{c1}} = \frac{381 - 293}{0{\cdot}85} = 103{\cdot}5$

$\therefore\ T_a = 103{\cdot}5 + T_1 = 103{\cdot}5 + 293 = 396{\cdot}5$ K

Effectiveness of intercooler $= \frac{T_a - T_b}{T_a - T_1} = 0{\cdot}75$

$\therefore\ T_a - T_b = (T_a - T_1) \times 0{\cdot}75 = (396{\cdot}5 - 293) \times 0{\cdot}75 = 77{\cdot}63$

$\therefore\ T_b = T_a - 77{\cdot}63 = 396{\cdot}5 - 77{\cdot}63 = 318{\cdot}87$ K

Now, considering second stage compression,

$\frac{T_2'}{T_b} = \left\{\frac{p_2}{p_b}\right\}^{\frac{\gamma-1}{\gamma}} = (2{\cdot}5)^{\frac{0{\cdot}4}{1{\cdot}4}} = (2{\cdot}5)^{0{\cdot}286} = 1{\cdot}3$

$\therefore\ T_2' = T_b \times 1{\cdot}3 = 318{\cdot}87 \times 1{\cdot}3 = 414{\cdot}5$ K

Now, $\eta_{c2} = \frac{T_2' - T_b}{T_2 - T_b} = 0{\cdot}85$

$\therefore\ T_2 - T_b = \frac{T_2' - T_b}{0{\cdot}85} = \frac{414{\cdot}5 - 318{\cdot}87}{0{\cdot}85} = 112{\cdot}5$

$\therefore\ T_2 = 112{\cdot}5 + T_b = 112{\cdot}5 + 318{\cdot}87 = 431{\cdot}37$ K

Compressor work, $W_c = W_{c1} + W_{c2}$

$$= k_p(T_a - T_1) + k_p(T_2 - T_b)$$

$$= 1{\cdot}005\ (396{\cdot}5 - 293) + 1{\cdot}005\ (431{\cdot}37 - 318{\cdot}87)$$

$$= 1{\cdot}005 \times 216{\cdot}0 = 217{\cdot}08 \text{ kJ/kg}$$

Now, considering turbine expansion,

$$\frac{T_3}{T_4'} = \left\{\frac{p_3}{p_4}\right\}^{\frac{\gamma-1}{\gamma}} = (6{\cdot}25)^{\frac{0{\cdot}4}{1{\cdot}4}} = (6{\cdot}25)^{0{\cdot}286} = 1{\cdot}69$$

$$\therefore\ T_4' = \frac{T_3}{1{\cdot}69} = \frac{973}{1{\cdot}69} = 575{\cdot}74 \text{ K}$$

$$\text{Now, } \eta_t = \frac{\text{Actual temperature drop}}{\text{Isentropic temperature drop}} = \frac{T_3 - T_4}{T_3 - T_4'} = 0{\cdot}82$$

$$\therefore\ T_3 - T_4 = (T_3 - T_4') \times 0{\cdot}82 = (973 - 575{\cdot}74) \times 0{\cdot}82 = 325{\cdot}75$$

$$\therefore\ T_4 = T_3 - 325{\cdot}75 = 973 - 325{\cdot}75 = 647{\cdot}25 \text{ K}$$

Turbine work, $W_t = k_p\ (T_3 - T_4) = 1{\cdot}005(973 - 647{\cdot}25) = 327{\cdot}37$ kJ/kg

$$\text{Effectiveness of heat exchanger} = 0{\cdot}75 = \frac{T_5 - T_2}{T_4 - T_2}$$

$$i.e.\ 0{\cdot}75 = \frac{T_5 - 431{\cdot}37}{647{\cdot}25 - 431{\cdot}37}$$

$$\therefore\ T_5 = 0{\cdot}75\,(647{\cdot}25 - 431{\cdot}37) + 431{\cdot}37 = 593{\cdot}28 \text{ K}$$

Heat supplied $= k_p(T_3 - T_5) = 1{\cdot}005\ (973 - 593{\cdot}28) = 381{\cdot}61$ kJ/kg

Thermal eff. of the gas turbine plant,

$$\eta_T = \frac{\text{Turbine work} - \text{Compressor work}}{\text{Heat supplied}}$$

$$= \frac{327{\cdot}37 - 217{\cdot}08}{381{\cdot}61} = \frac{110{\cdot}29}{381{\cdot}61} = 0{\cdot}289 \text{ or } 28{\cdot}9\%$$