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POLLACHI INSTITUTE OF ENGINEERING AND TECHNOLOGY

*(Approved by **AICTE** and Affiliated to **Anna University**)* *sky is the limit*

Department of Mechanical Engineering

Regulation 2021
III Year – IV Semester
ME3451 THERMAL ENGINERRING

ME3451 -THERMALENGINEERING

Syllabus

UNITI THERMODYNAMICCYCLES 12

AirStandardCycles–Carnot,Otto,Diesel,Dual,andBrayton–CycleAnalysis,Performanceand Comparison, Basic Rankine Cycle, modified, reheat and regenerative cycles.

UNITII STEAM NOZZLESANDINJECTOR 12

TypesandShapesofnozzles, Flowofsteamthroughnozzles, Criticalpressureratio, Variationofmass flow rate with pressure ratio. Effect of friction. Metastable flow.

UNITIII STEAMANDGASTURBINES 12

Types, Impulse and reaction principles, Velocity diagrams, Work done and efficiency – optimal operating conditions. Multi-staging, compoundingandgoverning. Gasturbinecycleanalysis –openandclosed cycle. Performance and its improvement - Regenerative, Intercooled, Reheated cycles and their combination.

UNITIV INTERNALCOMBUSTIONENGINES–FEATURESANDCOMBUSTION 12

ICengine–Classification,working,componentsandtheirfunctions.Idealandactual:Valveandporttiming diagrams,p-vdiagrams-twostroke&fourstroke,andSI&CIengines –comparison.Geometric,operating, andperformancecomparisonofSIand CIengines.Desirablepropertiesand qualitiesoffuels.Air-fuelratio calculation–leanandrichmixtures. CombustioninSI &CI Engines–Knocking–phenomenaandcontrol.

UNITV INTERNALCOMBUSTIONENGINEPERFORMANCEANDAUXILIARY SYSTEMS 12

PerformanceandEmissionTesting,Performanceparametersandcalculations.MorseandHeatBalancetests. Multipoint Fuel Injection system and Common rail direct injection systems. Ignition systems – Magneto, Batteryand Electronic. Lubrication and Cooling systems. Concepts of Supercharging and Turbocharging – Emission Norms

TOTAL:60PERIODS

TEXTBOOKS:

1. Mahesh.M.Rathore,“ThermalEngineering”,1stEdition,TataMcGrawHill,2010.

2. Ganesan.V,“InternalCombustionEngines"4thEdition,TataMcGrawHill,2012.

REFERENCES:

1. Ballaney.P,“ThermalEngineering”,25thEdition,KhannaPublishers,2017.

2. Domkundwar,Kothandaraman,&Domkundwar,“ACourseinThermal

UNIT –I

THERMODYNAMICCYCLES

1. DeriveairstandardefficiencyforanOttocyclewiththehelpofPv diagram.

TheOttocycle,whichwasfirstproposedbyaFrenchman,BeaudeRochasin1862,wasfirst used on an engine built by a German, Nicholas A. Otto, in 1876. The cycle is also called a constant volume or explosion cycle. This is the equivalent air cycle for reciprocating piston engines using spark ignition. Figures 1 and 2 show the P-V and T-s diagrams respectively

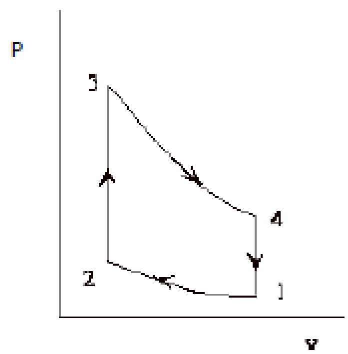


Fig.1: P-V Diagram of Otto Cycle.

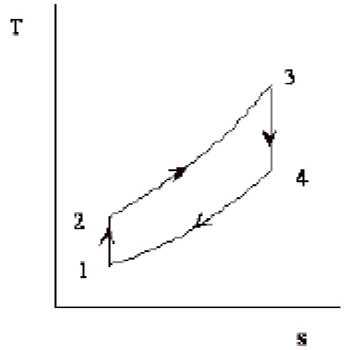


Fig.2: T-S Diagram of Otto Cycle.

At the start of the cycle, the cylinder contains a mass M of air at the pressure and volume indicated at point 1. The piston is at its lowest position. It moves upward and the gas is compressed isentropic ally to point 2. Atthis point,heatis added atconstantvolume which raises the pressure to point 3. The high pressure charge now expands isentropic ally, pushing the piston down on its expansionstroketopoint4wherethecharge rejectsheatatconstantvolumetotheinitialstate,point 1.

Theisothermalheat addition andrejection oftheCarnot cycleare replaced by theconstant volumeprocesseswhichare,theoreticallymoreplausible,althoughinpractice,eventheseprocesses are not practicable.

Theheatsupplied,Qs,perunitmassofcharge,isgiven by$c_v(T_3 – T_2)$ (1)

Theheatrejected,Qrperunitmassofchargeisgiven by

$$c_v(T_4 - T_1) \qquad (2)$$

andthethermalefficiency isgivenby

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

$$= 1 - \frac{T_1}{T_2}\left\{\frac{\left(\frac{T_4}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)}\right\} \qquad (3)$$

$$\text{Now } \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma - 1} = \left(\frac{V_3}{V_4}\right)^{\gamma - 1} = \frac{T_4}{T_3}$$

$$\text{And since } \frac{T_1}{T_2} = \frac{T_4}{T_3} \quad \textit{we have} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Hence, substitutinginEq. 3, weget, assumingthatristhecompressionratio V1/V2

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

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

$$= 1 - \frac{1}{r^{\gamma - 1}} \qquad (4)$$

2. Derivemeaneffectivepressureexpressionforan Ottocycle

ItisseenthattheairstandardefficiencyoftheOttocycledependsonlyonthecompression ratio.However,thepressures and temperatures atthevarious points inthe cycleand thenet work done, all depend uponthe initial pressure and temperature and the heat input from point 2 to point 3, besides the compression ratio.

A quantity of special interest in reciprocating engine analysis is the mean effective pressure. Mathematically, it is the net work done on the piston, W, divided by the piston displacement volume,V_1–V_2.This quantityhas the units ofpressure.Physically,it is thatconstantpressure which,if exertedonthepistonforthe wholeoutwardstroke,wouldyieldworkequaltotheworkofthecycle. Itis given by

$$mep = \frac{W}{V_1 - V_2}$$

$$= \frac{\eta Q_{2\text{-}3}}{V_1 - V_2} \qquad (5)$$

where $Q_{2\text{-}3}$ is the heat added from points 2 to 3.

Work done per kg of air

$$W = \frac{P_3V_3 - P_4V_4}{\nu - 1} - \frac{P_2V_2 - P_1V_1}{\nu - 1} = mepV_s = P_m(V_1 - V_2)$$

$$mep = \frac{1}{(V_1 - V_2)}\left[\frac{P_3V_3 - P_4V_4}{\nu - 1} - \frac{P_2V_2 - P_1V_1}{\nu - 1}\right] \qquad (5A)$$

The pressure ratio P_3/P_2 is known as explosion ratio r_p

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\nu} = r^{\nu} \Rightarrow P_2 = P_1 r^{\nu},$$

$$P_3 = P_2 r_p = P_1 r^{\nu} r_p,$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{\nu} = P_1 r^{\nu} r_p \left(\frac{V_2}{V_1}\right)^{\nu} = P_1 r_p$$

$$\frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = r$$

$$\therefore V_s = V_c(r - 1)$$

Substituting the above values in Eq 5A

$$mep = P_1 \frac{r(r_p - 1)(r^{\gamma - 1} - 1)}{(r - 1)(\gamma - 1)}$$

, Now

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

$$= V_1\left(1 - \frac{1}{r}\right) \qquad (6)$$

Here r is the compression ratio, V_1/V_2

From the equation of state:

$$V_1 = M\frac{R_0}{m}\frac{T_1}{p_1} \qquad (7)$$

R_0 is the universal gas constant

Substituting for V_1 and for $V_1 - V_2$,

$$mep = \eta \frac{Q_{2-3}\frac{p_1 m}{MR_0 T_1}}{1 - \frac{1}{r}} \qquad (8)$$

Thequantity Q2-3/Mistheheataddedbetweenpoints2and3perunit massofair(M isthe mass of air and m is the molecular weight of air); and is denoted by Q'', thus

$$mep = \eta \frac{Q' \frac{p_1 m}{R_0 T_1}}{1 - \frac{1}{r}} \qquad (9)$$

Wecannon-dimensionalizethemepbydividingitbyp1sothatwecanobtainthefollowing equation

$$\frac{mep}{p_1} = \eta \left[\frac{1}{1 - \frac{1}{r}} \right] \left[\frac{Q' m}{R_0 T_1} \right] \qquad (10)$$

Since $\frac{R_0}{m} = c_v(\gamma - 1)$, we can substitute it in Eq. 25 to get

$$\frac{mep}{p_1} = \eta \frac{Q'}{c_v T_1} \frac{1}{\left[1 - \frac{1}{r}\right][\gamma - 1]} \qquad (11)$$

3. In an air standard Otto cyle the pressure and temperature at the beginning of the cylcle is 42^0 C and 0.1MPa.The compression ratio and maximum temperature of the cycle are 8 and 1250^0 C respectively.Find(a)thetemperatureandpressureatthecardinalpointsofthecycle,(b)heatsupplied per kg of air, (c) work doneper kg of air (d) cycle efficiency and (e) the m.e.p. of the cylcle

Givendata:

p_1=0.1MPa=100kN/m^2 T_1=42^{0}C=315K

r=8

$$r=8 \left(i.e \ \frac{V_1}{V_2} = \frac{V_4}{V_3} = 8 \right)$$

T_3=1250^{0}C=1523K

solution

Considerprocess1-2(adiabatic process)

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

$$p_2 = \left(\frac{V_1}{V_2} \right) \times P_1 = (8)^{1.4} \times 100$$

P_2=**1837.9kN/m^2**

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

$$T_2=\left(\frac{V_1}{V_2}\right)^{\gamma-1}\times T_1=(8)^{1.4-1}\times 315$$

T_2=**723.68K**

considerprocess2-3(constant volume process);

$$\frac{p_3}{p_2}=\frac{T_3}{T_2}$$

$$p_3=\frac{T_3}{T_2}\times p_2=\frac{1523}{723.68}\times 1837.9$$

p_3=**3867.89kN/m²**

Considerprocess 3-4(adiabaticprocess);

$$\frac{p_4}{p_3}=\left(\frac{V_3}{V_4}\right)^{\gamma}$$

$$p_4=\left(\frac{V_3}{V_4}\right)\times P_3=\left(\frac{1}{8}\right)^{1.4}\times 3867.89$$

P_4=**210.44kN/m²**

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

$$T4=\left(\frac{V_3}{V_4}\right)^{\gamma-1}\times T_3=\left(\frac{1}{8}\right)^{0.4}\times 1523$$

T_4=**662.9K**

Heatsupplied, $Q_s=mC_v(T_3-T_2)$

=1x0.718x(1523–723.68)

Q_s= 573.9kJ/kg

Heatrejected$Q_R=mC_v(T_4-T_1)$=1C0.718X(662.9-315) Q_R=

249.79 KJ/kg

Workdone, W=Q_S–Q_R

W= 573.9–249.79

=324.1 KjKG

Cycleefficiency,

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

$$= 1 - \frac{1}{(8)^{1.4-1}} = 0.5647$$

η=56.47%

Moreeffectivepressure,p_m

$p_1 v_1 = mRT_1$

$$v_1 = \frac{mRT_1}{P_1} = \frac{1 \times 0.287 \times 315}{100}$$

v1 =0.9m^3/kg

$$\frac{v_1}{v_2} = 8 \Rightarrow v_2 = \frac{0.9}{9} = 0.11 m^3/kg$$

$$p_m = \frac{W}{v_1 - v_2} = \frac{324.1}{0.9 - 0.11}$$

p_m=409.2kN/m^2

Alternatively,wecanusemeaneffectivepressureformula,

$$p_m = p_1 r \left[\left(\frac{k-1}{\gamma-1}\right)\left(\frac{r^{\gamma-1}-1}{r-1}\right)\right]$$

$$k = \frac{p_3}{p_2} = \frac{3867.89}{1837.9} = 2.10$$

$$p_m = 100 \times 8 \times \left[\left(\frac{2.1-1}{1.4-1}\right)\left(\frac{8^{1.4-1}-1}{8-1}\right)\right]$$

p_m=407.75Kn/m^2

4. Airentersin anairstandardOttocycleat 1barand290k.Theratioofheat rejection andheat supplied is 0.4. The maximum Temperature of the cycle is 1500k. Find efficiency, compression ratio, and network and mean effective pressure.

Given data:

p_1=1 bar T_1

= 290K

$$\frac{Q_R}{Q_S} = 0.4$$

T_3 = 1500K

Solution:

Efficiencyofthecycle,

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

=1-0.4

η= 60% Ans.

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

$$0.6 = 1 - \frac{1}{(r)^{1.4-1}} \Rightarrow r = \left(\frac{1}{0.4}\right)^{\frac{1}{0.4}}$$

r=9.88 Ans.

Compressionratio,r= 9.88

$$v_1 = \frac{RT_1}{p_1} = \frac{287 \times 290}{1 \times 10^5}$$

v_1 =0.8323m^3/kg

$$\frac{v_1}{v_2} = 9.88$$

$$v_2 = \frac{0.8323}{9.88} = 0.0842 m^3/kg$$

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

$$= 290 \times (9.88)^{1.4-1} = 725K$$

Heatsupply,Q_s=C_v(T_3–T_2)

=0.718x(1500 –725)

=556.45*kJ/kg*

Workdone, $W = \eta \times Q_s = 0.6 \times 556.45$

= 333.87*kJ/kg* **Ans.**

Meaneffectivepressure,*p_m*

$$p_m = \frac{W}{v_1 - v_2} = \frac{333.87}{0.8323 - 0.0842}$$

$Pm = 446.29 kN/m^2$ ***Ans.***

5A six cylinder petrol engine has a compression ratio of 5:1. The clearance volume of each cylinder is 110CC. It operator on the four stroke constant volume cycle and the indicated efficiency ratio referred to air standard efficiency is 0.56. At the speed of 2400 rpm. It consumer 10kg of fuel per hour. The calorific value of fuel is 44000KJ/kg. Determine the average indicated mean effective pressure.

Givendata:

$r = 5$

$V_C = 110CC$

$\eta_{relative} = 0.56$

$N = 2400rpm$

$M_f = 10kg$

$= 10/3600 kg/s$

$C_v = 44000 kJ/kg$

$Z = 6$

Solution:

Compressionratio:

$r = V_s + V_c/V_c \rightarrow 5 = V_s + 110/110 \rightarrow V_s = 440CC = 44x10^{-6} m^3$

Airstandard efficiency:

$\eta = 1 - 1/(r^{\gamma-1}) = 47.47\%$ $(\gamma = 1.4)$

Relativeefficiency:

$\eta_{relative} = \eta_{actual}/\eta_{air\text{-}standard} \rightarrow$

$0.56 = \eta_{actual}/47.47 \eta_{actual}$

$= 26.58\%$

Actualefficiency = workoutput/headinput

$0.2658 = W/m_f C_V \rightarrow W = 0.2658$

x10/3600x44000W =

32.49kw.

Thenetworkoutput:

$W = P_m x V_s x N/60 \; xZ \rightarrow 32.49x10^3 = Pmx440 \; x10^{-6} x$ 1200/60 x 6

1200/60 x 6

Pm=6.15bar

Pm = 6.15 bar

6. TheefficiencyofanOttocycleis60%andγ=1.5,whatisthecompressionratio?

Given:

Cycle efficiency(η)=0.6

γ=1.5

Required:r

Solution:

Compressionratio(r)=V_1/V_2

Weknowthat,$\eta = 1 - 1/r^{\gamma-1}$

$0.6 = 1 - 1/r^{1.5-1}$

r=6.25---Ans

7. An engine of 250 mm bore and 375 mm stroke works on Otto cycle. The clearancevolume is 0.00263 m^3. The initial pressure and temperature are 1 bar and 50^oC. If the maximum pressure is limited to 25 bar, find (i) air standard efficiency of the cycle (ii) the mean effective pressure of the cycle.

Given:

Bore(d) =0.25m

Stroke (L) = 0.375 m

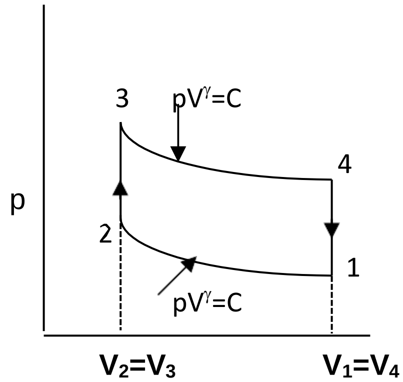
Clearancevolume(V_2)=0.00263

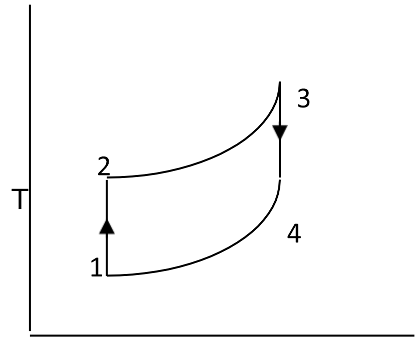
m^3Initial pressure(p_1)= 1 bar

Initialtemperature(T_1)=50^{o}C=323K

Maximumpressure(p_3)=25bar

***Required:*(i)η(ii) p_m**





Solution:

(i) Cycleefficiency(η)=1–1/$r^{\gamma-1}$

r=compressionratio=V_1/V_2

V_1=strokevolume+clearancevolume

$=(V_1-V_2)+V_2$

$=(\pi/4)D^2L+V_2$

$=(\pi/4)\text{x } 0.25^2+0.375+0.00263$

$=0.021038 m^3$

$\therefore$ **r =0.021038/0.00263=8**

$\therefore$ **$\eta=1-1/8^{1.4-1}=0.565$---Ans**

(ii) Meaneffectivepressure(p_m)

$$Pm=p_1r\left[\frac{(r^{\gamma-1}-1)(r_p-1)}{(\gamma-1)(r-1)}\right]\rightarrow \text{forOtto cycle}$$

$r_p=p_3/p_2p_2$

$V_2{}^{\gamma}=p\ V_1^{\gamma}\ _1$

$p_2=(V_1/V_2)^{\gamma}p_1=(8)^{1.4}\text{x } 1=18.38$ bar

$\therefore$ **$r_p=25/18.38 \quad =1.36$**

$$p_m=1\text{x}8\left[\frac{(8^{1.4-1}-1)(1.36-1)}{(1.4-1)(8-1)}\right]$$

p_m=1.334barAns

8Airenters anairstandardOttocyclesat100 kN/m^2and290K.Theratioofheatrejectiontoheat

suppliedis0.4.Themaximumtemperatureinthecycleis1500K.Find(a)efficiency,(b)network,(c) mep, & (d) compression ratio (r).

Given:

Initial pressure (p_1) =100kN/m^2=1

barInitial temperature (T_1)= 290 K

Heat rejection / Heat supplied = 0.4

Maximumtemperature(T_3)=1500K

Required: **(a)η(b)W_{net}(c)mep(d)**

r*Solution:*

(a) Cycleefficiency(η)=1–1/$l^{\gamma-1}$

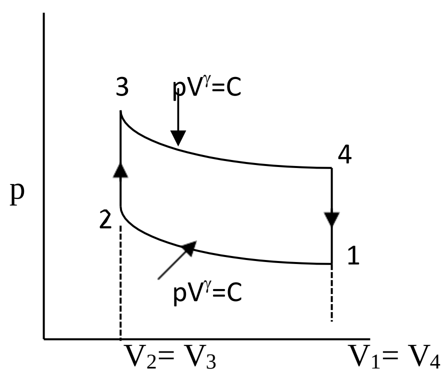
=(heatsupplied–heatrejection)/heatsupplied

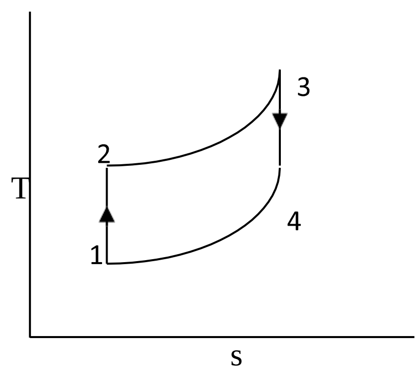
=1–(HR/HS)=1–0.4= 0.6 ----------- Ans

(b) Network(W_{net}) =Heatsupplied–Heatrejected

Heatsupplied=mC_v(T_3– T_2)

Take m=1kg&C_v=0.717kJ/kgK





<u>TofindT2</u>

$$T_2/T_1=(p_2/p_1)^{(\gamma-1)/\gamma}= (V_1/V_2)^{(\gamma-1)}$$

$$=r^{\gamma-1}$$

From, $\eta=1-(1/r^{\gamma-1})$

i.e., $0.6= 1-(1/r^{1.4-1})$

$r = 9.88$

$\therefore T_2 = 290(9.88)^{1.4-1} = 725K$

$\therefore$ Heatsupplied=1x0.717x(1500–725)=555.675kJ

Heatrejected=0.4(555.675)=222.27kJ

$\therefore$ W_{net}=555.675 –222.27=333.405 kJ -------- Ans

(c) MEP

$$p_m = p_1 r\left[\frac{(r^{\gamma-1}-1)(r_p-1)}{(\gamma-1)(r-1)}\right] \rightarrow \text{forOtto cycle}$$

$r_p = p_3/p_2$

2–3→Constantvolumeprocess $p_3/$

$T_3 = p_2/T_2$

$\therefore p_3/p_2 = T_3/T_2$ To

find T_2

$T_2/T_1 = (V_1/V_2)^{\gamma-1}$

$T_2 = 290(9.88)^{1.4-1} = 724.94K$

$Rp = p_3/p_2 = 1500/724.94 = 2.069$

$$p_m = 1\text{x}9.88\left[\frac{(9.88^{1.4-1}-1)(2.069-1)}{(1.4-1)(9.88-1)}\right]$$

=4.459bar-------- Ans

Module –II

Derive thermal efficiency expression for a Diesel cycle.

This cycle, proposed by a German engineer, Dr. Rudolph Diesel to describe the processes of his engine, is also called the constant pressure cycle. This is believed to be the equivalent air cycle for the reciprocating slow speed compression ignition engine. The P-V and T-s diagrams are shown in Figs 4 and 5 respectively.

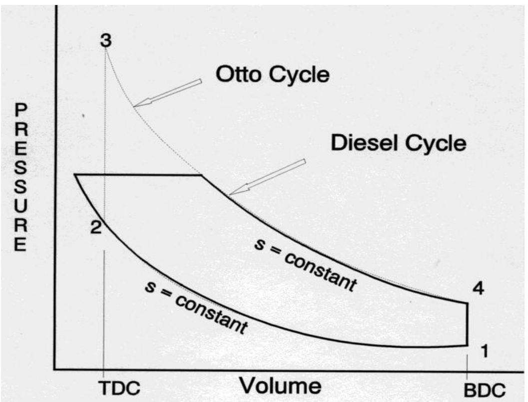


Fig.4: P-V Diagram of Diesel Cycle.

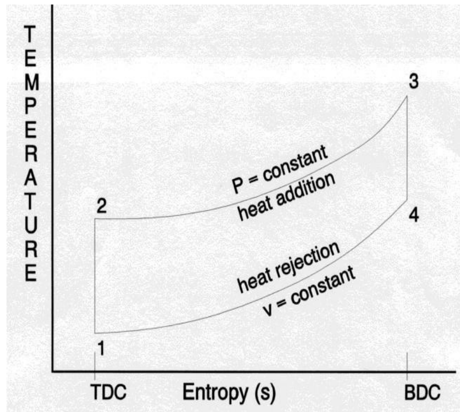


Fig.5: T-S Diagram of Diesel Cycle.

The cycle has processes which are the same as that of the Otto cycle except that the heat is added at constant pressure

The heat supplied, Qs is given by cp(T3 – T2) (22)

Whereas the heat rejected, Qr is given by cv (T4 – T1) (23) and

The thermal efficiency is given by

$$\eta_{th} = 1 - \frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \frac{1}{\gamma}\left\{\frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)}\right\} \qquad (24)$$

From the T-s diagram, Fig. 5, the difference in enthalpy between points 2 and 3 is the same as that between 4 and 1, thus

$$\Delta s_{2-3} = \Delta s_{4-1}$$

$$\therefore\ c_v \ln\left(\frac{T_4}{T_1}\right) = c_p \ln\left(\frac{T_3}{T_2}\right)$$

$$\therefore\ \ln\left(\frac{T_4}{T_1}\right) = \gamma \ln\left(\frac{T_3}{T_2}\right)$$

$$\therefore\ \frac{T_4}{T_1} = \left(\frac{T_3}{T_2}\right)^{\gamma} \text{ and } \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \frac{1}{r^{\gamma-1}}$$

Substituting in eq. 24, we get

$$\eta_{th} = 1 - \frac{1}{\gamma}\left(\frac{1}{r}\right)^{\gamma-1}\left[\frac{\left(\frac{T_3}{T_2}\right)^{\gamma} - 1}{\frac{T_3}{T_2} - 1}\right] \qquad (25)$$

$$\text{Now } \frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c = cut-off\ ratio$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_c^{\gamma} - 1}{\gamma(r_c - 1)}\right] \qquad (26)$$

When Eq. 26 is compared with Eq. 8, it is seen that the expressions are similar except for the term in the parentheses for the Diesel cycle. It can be shown that this term is always greater than unity.

Now $r_c = \frac{V_3}{V_2} = \frac{V_3}{V_4} \Big/ \frac{V_2}{V_1} = \frac{r}{r_e}$ **where r is the compression ratio and re is the expansion ratio**

Thus, the thermal efficiency of the Diesel cycle can be written as

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\left(\frac{r}{r_e}\right)^{\gamma} - 1}{\gamma\left(\frac{r}{r_e} - 1\right)}\right] \qquad (27)$$

Let $r_e = r - \Delta$ since r is greater than r_e. Here, Δ is a small quantity. We therefore have

$$\frac{r}{r_e} = \frac{r}{r-\Delta} = \frac{r}{r\left(1 - \frac{\Delta}{r}\right)} = \left(1 - \frac{\Delta}{r}\right)^{-1}$$

We can expand the last term binomially so that

$$\left(1 - \frac{\Delta}{r}\right)^{-1} = 1 + \frac{\Delta}{r} + \frac{\Delta^2}{r^2} + \frac{\Delta^3}{r^3} + \cdots$$

$$\text{Also } \left(\frac{r}{r_e}\right)^{\gamma} = \frac{r^{\gamma}}{(r-\Delta)^{\gamma}} = \frac{r^{\gamma}}{r^{\gamma}\left(1 - \frac{\Delta}{r}\right)^{\gamma}} = \left(1 - \frac{\Delta}{r}\right)^{-\gamma}$$

We can expand the last term binomially so that

$$\left(1 - \frac{\Delta}{r}\right)^{-\gamma} = 1 + \gamma\frac{\Delta}{r} + \frac{\gamma(\gamma+1)}{2!}\frac{\Delta^2}{r^2} + \frac{\gamma(\gamma+1)(\gamma+2)}{3!}\frac{\Delta^3}{r^3} + \cdots$$

Substituting in Eq. 27, we get

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\frac{\Delta}{r} + \frac{(\gamma+1)}{2!}\frac{\Delta^2}{r^2} + \frac{(\gamma+1)(\gamma+2)}{3!}\frac{\Delta^3}{r^3} + \cdots}{\frac{\Delta}{r} + \frac{\Delta^2}{r^2} + \frac{\Delta^3}{r^3} + \cdots}\right] \qquad (28)$$

6. Derive the mean effective pressure expression for a diesel cycle

$$mep = \frac{1}{V_s}\left[P_2(V_3 - V_2) + \frac{P_3V_3 - P_4V_4}{\nu - 1} - \frac{P_2V_2 - P_1V_1}{\nu - 1}\right] \qquad (29)$$

The pressure ratio P3/P2 is known as explosion ratio rp

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\nu} = r^{\nu} \Rightarrow P_2 = P_1 r^{\nu},$$

$$P_3 = P_2 = P_1 r^{\nu}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{\nu} = P_1 r^{\nu}\left(\frac{V_2}{V_1}\right)^{\nu} = P_1 r_c^{\ \nu}$$

$$V_4 = V_1, V_2 = V_c,$$

$$\frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = r$$

$$\therefore V_s = V_c(r-1)$$

Substituting the above values in Eq 29 to get Eq (29A)

In terms of the cut-off ratio, we can obtain another expression for mep/p1 as follows

$$mep = P_1 \frac{\gamma r^{\gamma}(r_c - 1) - r(r_c^{\gamma} - 1)}{(r-1)(\gamma - 1)} \qquad (29A)$$

We can obtain a value of rc for a Diesel cycle in terms of Q" as follows:

$$r_c = \frac{Q'}{c_p T_1 r^{\gamma-1}} + 1 \qquad (30)$$

We can substitute the value of η from Eq. 38 in Eq. 26, reproduced below and obtain the value of mep/p1 for the Diesel cycle.

$$\frac{mep}{p_1} = \eta \frac{Q'}{c_v T_1} \frac{1}{\left[1 - \frac{1}{r}\right][\gamma - 1]}$$

For the Diesel cycle, the expression for mep/p3 is as follows:

$$\frac{mep}{p_3} = \frac{mep}{p_1}\left(\frac{1}{r^{\gamma}}\right) \qquad (31)$$

Modern high speed diesel engines do not follow the Diesel cycle. The process of heat addition is partly at constant volume and partly at constant pressure. This brings us to the dual cycle.

7. A diesel engine working on air standard cycle takes in air at 1bar and 25^{0}C. The specify volume of air at inlet is 0.8^3/kg. The compression ratio is 14 and heat is added at constant pressure is 840kJ/kg. Find cut – off ratio and air standard efficiency.

Given data:

p_1 = 1bar

$T_1 = 25^0\ C = 25 + 273 = 293\ K$

$v_1 = 0.8 m^3/kg$

r = 14

Q_s = 840kJ/kg

Solution:

$$\frac{v_1}{v_2} = 14$$

$$v_2 = \frac{0.8}{14} = 0.05714\, m^3 / kg$$

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

$$T_2 = 298 \times (14)^{1.4-1}$$

$$T_2 = 856.38K$$

We know that, $Q_s = m \times C_v (T_3 - T_2)$

$$T_3 = \frac{Q_s}{C_v} + T_2 = \frac{840}{0.718} + 856.38$$

$$T_3 = 2026.3K$$

Consider process 2-3:

$$\frac{v_2}{T_2} = \frac{v_3}{T_3}$$

$$v_3 = \frac{v_2}{T_2} \times T_3 = \frac{0.05714}{856.38} \times 2026.3$$

$$= 0.1352 m^3/kg$$

Cut-off ratio, $\rho = \frac{v_3}{v_2} = \frac{0.1352}{0.05714}$

$= 2.37$ Ans.

$\therefore$ **Air Standard efficiency,** $\eta = 1 - \frac{1}{\gamma\,(\,)^{\gamma-1}}\left(\frac{\rho^{\gamma}-1}{\rho-1}\right)$

$$= 1 - \frac{1}{1.4(14)^{1.4-1}}\left[\frac{(2.37)^{1.4}-1}{(2.37-1)}\right]$$

= 57.42 % **Ans.**

8. An engine working on an ideal air standard diesel cycle has the compression ratio 15 and heat transfer 1400kJ/kg. Find the pressure and temperature at the end of the each process if the inlet conditions are 280K and 1.1 bar. Find also the air standard efficiency and mean effective pressure.

Given data:

r=15

Q_s = 1400 k J/kg

T_1 = 280 K

P_1=1.1bar = 110 k N/m^2

Solution :

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

$$T_2 = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \times T_1 = (15)^{1.4-1} \times 280$$

$$T_2 = 827.17K$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \times p_1 = \left(\frac{827.17}{280}\right)^{\frac{1.4}{1.4-1}} \times 110$$

p_3 =p_2 = 4874.39kN/m^2

Consider the process 2-3 (Constant pressure process);

$$Qs = m \times (T_3 - T_2) = 1400\ kJ/kg$$

1400 = 1 × 1.005 **× (T_3 -827.17)**

T_3 = 2220.2K

$$\frac{V_2}{T_2} = \frac{V_3}{T_3} \Rightarrow \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{2220.2}{827.17} = 2.684$$

Cut-off ratio, $\rho = \dfrac{V_3}{V_2} = 2.684$

Consider the process 3-4 (adiabatic expansion);

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

$$T_4 = \left(\frac{V_3}{V_4}\right)^{\gamma-1} \times T_3 \qquad (\because V_4 = V_1)$$

$$= \left(\frac{V_3}{V_2} \times \frac{V_2}{V_1}\right)^{\gamma-1} \times T_3 = \left(\frac{2.684}{15}\right)^{1.4-1} \times 2220.2$$

***T_4* = 1115.5K** **Ans.**

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

$$p_4 = \left(\frac{T_4}{T_3}\right)^{\frac{\gamma}{\gamma-1}} \times p_3$$

$$= \left(\frac{1115.5}{2220.2}\right)^{\frac{1.4}{0.4}} \times 4874.39$$

$p_4 = 438.24 kN/m^3$ **Ans.**

Air standard efficiency,

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

$$= 1 - \frac{1}{1.4(15)^{1.4-1}}\left(\frac{(2.684)^{1.4}-1}{2.684-1}\right)$$

= 57.16% **Ans**

Mean effective pressure,

$$P_m = \frac{p_1 r^{\gamma}\left[\gamma(\rho-1) - r^{1-\gamma}\left(\rho^{\gamma}-1\right)\right]}{(\gamma-1)(r-1)}$$

$$= \frac{100 \times (15)^{1.4}\left[1.4(2.684-1) - (15)^{1-1.4}\left[(2.684)^{1.4}-1\right]\right.}{(1.4-1)(15-1)}$$

***p_m* = 1066.33kN/m²** **Ans.**

9. In an engine working on diesel cycle, inlet pressure and temperature are l bar and 17^0 C respectively. Pressure at the end of adiabatic compression is 35 bar. The ratio of expansion i.e., after constant pressure heat addition is 5. Calculate the hear addition, heat rejection the efficiency of the cycle, and mean effective pressure.

Assume =1.4, C_p = 1.004kJ/kgK and C_v = 0.717kJ/kgK.

Given data

Diesel cycle

p_1 = 1 bar = 100 kN/m^2

T_1 = 17^0C = 290K

p_2 =35bar = 3500 kN/m^2

$$\frac{V_4}{V_3} = \frac{V_1}{V_3} = 5$$

$$\gamma = 1.4; C_P = 1.004 \text{ kJ/kgK};$$

C_v=0.717 kJ/kgK

Solution

Consider process 1-2 ;

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

$$T_2 = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \times T_1 = \left(\frac{3500}{100}\right)^{\frac{1.4-1}{1.4}} \times 290 = 800.87K$$

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma} \Rightarrow r = \frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}$$

Compression ratio, $r = \left(\frac{3500}{100}\right)^{\frac{1}{1.4}} = 12.67$

$$\frac{V_1}{V3} = \frac{V_1}{V_2} \times \frac{V_2}{V_3}$$

$$5 = 12.67 \times \frac{V_2}{V_3}$$

Cut – off ratio, $\rho = \frac{V_3}{V_2} = \frac{12.67}{5} = 2.534$

Consider process: Constant pressure process;

$$\frac{V_2}{T_2} = \frac{V_3}{T_3} \Rightarrow T_3 = \frac{V_3}{V_2} \times T_2 = 2.534 \times 800.87$$

T_3 = 2090.4K

Heat supplied during 2 – 3

$Q_s = C_p (T_3 - T_2)$

$= 1.004(2029.4 - 800.87)$

Q_s = 1233.45kJ/kg **Ans.**

Consider process 3-4: Adiabatic process;

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

$$T_4 = \left(\frac{V_3}{V_4}\right)^{\gamma - 1} \times T_3 = \left(\frac{1}{5}\right)^{1.4-1} \times 2029.4$$

T_4 = 1066K

Heat rejection during constant volume process 4-1

$Q_R = C_V (T_4 - T_1) = 0.717(1066-290)$

Q_R = 556.43kJ/kg **Ans.**

Efficiency, $\eta = 1 - \frac{Q_R}{Q_S} = 1 - \frac{556.43}{1233.45}$

= 54.88% **Ans.**

Mean effective pressure, p_m

$$P_m = \frac{p_1 r^{\gamma}\left[\gamma(\rho - 1) - r^{1-\gamma}\left(\rho^{\gamma} - 1\right)\right]}{(\gamma - 1)(r - 1)}$$

$$= \frac{1 \times (12.67)^{1.4}\left[1.4(2.534 - 1) - (12.67)^{1-1.4}\left((2.534)^{1.4} - 1\right)\right]}{(1.4 - 1)(12.67 - 1)}$$

P_m = 8.83bar **Ans.**

10. In an air standard diesel cycle, the pressure and temperatures of air at the beginning of Cycle are 1bar and 40°C. The temperatures before and after the heat supplied are 400^0C and 1500^0C. Find the air standard efficiency and mean effective pressure of the cycle. What is the Power output if it makes 100cycles / min?

Given data:

$p_1 = 1bar = 100kN/m^2$

$T_1 = 40^0C = 40 + 273 = 313K$

$T_2 = 400^0C = 673K$

$T_3 = 1500^0C = 1773K$

Solution:

Consider the process 1-2 (Isentropic compression)

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

Compression ratio, $r = \frac{v_1}{v_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma-1}} = \left(\frac{673}{313}\right)^{\frac{1}{1.4-1}}$

$= 6.779$

Consider the process 2-3 (Constant pressure heating)

$$\frac{v_2}{T_2} = \frac{v_3}{T_3}$$

Cut off ratio, $\rho = \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{1773}{673} = 2.634$

Efficiency, $\eta = 1 - \frac{1}{\gamma (r)^{\gamma-1}} \left(\frac{\rho^\gamma - 1}{\rho - 1}\right)$

$$= 1 - \frac{1}{1.4(6.779)^{1.4-1}} \left(\frac{2.634^{1.4} - 1}{2.634 - 1}\right)$$

= 0.4142 = 41.42 % **Ans.**

Mean effective pressure,

$$\rho_m = \frac{p_1 r^\gamma \left(\gamma(\rho - 1) - r^{1-\gamma}\left(\rho^\gamma - 1\right)\right)}{(\gamma - 1)(r - 1)}$$

$$\frac{100\times(6.779)^{1.4}\left(1.4(2.634-1)-(6.779)^{1-1.4}\left((2.634)^{1.4}-1\right)\right)}{(1.4-1)(6.779-1)}$$

$\rho_m = 597.77kN/m^2$ **Ans.**

Heat supplied = m x C_p (T_3-T_2)

= 1 x 1.005 (1773 – 673)

Q_S = 1105.5*kJ/kg*

Work done $= \eta \times Q_S = 0.4142 \times 1105.5$

=457.89*kJ/kg*

$$\left[\because \eta = 1 - \frac{Q_R}{Q_S} = \frac{W}{Q_S}\right]$$

Power = *W* x cycles/min = 457.89 x 100

= 45 x 10^{-3}*kJ/kg-min* = 763.16*kJ/kg-sec*

= *763.16kW/kg* **Ans.**

11) Find the air standard efficiency of a diesel cycle when the compression ratio andcut–off ratio are 15 & 1.84 respectively. Assume γ = 1.4.

Given:

Compression ratio (r) =

15 Cut – off ratio(ρ) = 1.84

γ = 1. 4

Required: η

***Solution*:**

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

$$= 1 - \frac{1}{1.4(15)^{1.4 - 1}}\left[\frac{1.84^{1.4 - 1} - 1}{1.84 - 1}\right]$$

η= 0.612---- Ans

12)In a diesel engine the pressure at the beginning of compression is 1 bar. Compression ratio is 14 : 1 and cut–off takes place at 10% of the stroke. Calculatethe air standard efficiency and ideal mep of the cycle (γ = 1. 4 for air).

Given:

Initial pressure (p_1) = 1 bar.Compression ratio (r) = 14.

Cut – off takes place at 10 % of stroke,i.e., $V_3 - V_2 = 0.1 (V_1 - V_2)$

$\gamma = 1.4$

***Required* : η & p_m**

Solution:

$$\eta = 1 - \frac{1}{\gamma\, r^{\gamma - 1}} \left[\frac{\rho^{\gamma - 1} - 1}{\rho - 1}\right]$$

ρ = cut – off ratio = V_3 / V_2

r = V_1 / V_2 = 14

$\therefore V_1 = 14 V_2$

$\therefore V_3 - V_2 = 0.1 (14 V_2 - V_2)$

$V_3 = 2.3 V_2$.

$V_3 / V_2 = \rho = 2.3$

$$\therefore \quad \eta = 1 - \frac{1}{1.4\ (14)^{1.4 - 1}} \left[\frac{2.3^{1.4 - 1} - 1}{2.3 - 1}\right]$$

= 0.577 ------ Ans

Mean effective pressure (p_m)

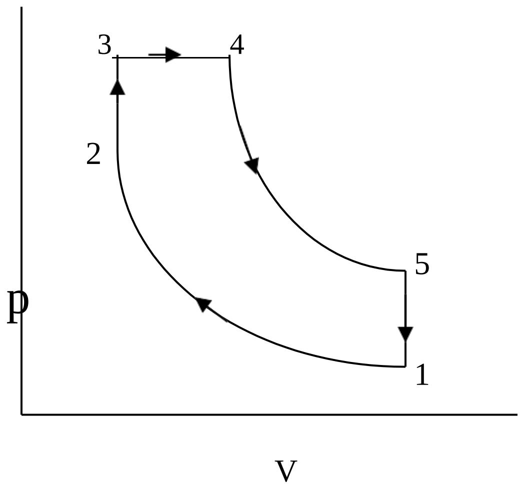
$$p_m = p_1 r^{\gamma} \left[\frac{\gamma (\rho - 1) - r^{1 - \gamma} (\rho^{\gamma} - 1)}{(\gamma - 1)(r - 1)}\right] \rightarrow \text{for Diesel cycle}$$

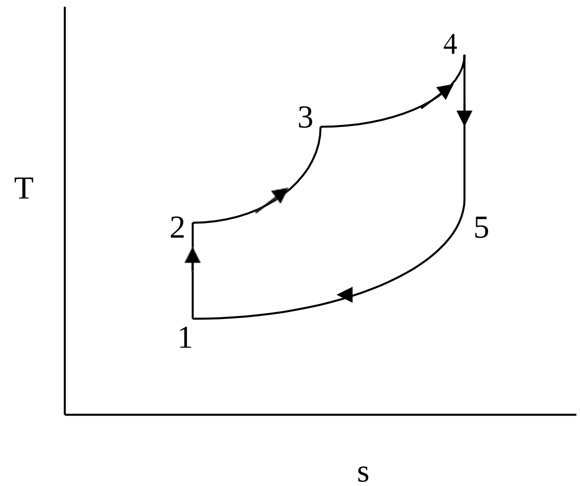
$$= 1 \times 14^{1.4} \left[\frac{1.4\ (2.3 - 1) - 14^{1 - 1.4} (2.3^{1.4} - 1)}{(1.4 - 1)\ (14 - 1)}\right] = = 8.133 \text{ bar} \text{---------- Ans}$$

MODULE III

Dual Cycle (Limited Pressure or Mixed Cycle)

This cycle is a combination of Otto and Diesel cycles. In this cycle the heat is added partially at constant volume and partially at constant pressure. The advantage of this cycle is increased time to fuel for injection.





1 – 2 → Isentropic compression

2 – 3 → Constant volume heat addition

3 – 4 → Constant pressure heat addition

4 – 5 → isentropic expansion

5 – 1 → Constant volume heat rejection

ρ = Cut-off ratio = V_4/V_3

r_p = Explosion ratio or Pressure ratio = p_3/p_2

The cycle is the equivalent air cycle for reciprocating high speed compression ignition engines. The P-V and T-s diagrams are shown in Figs.6 and 7. In the cycle, compression and expansion processes are isentropic; heat addition is partly at constant volume and partly at constant pressure while heat rejection is at constant volume as in the case of the Otto and Diesel cycles.

The heat supplied, Qs per unit mass of charge is given by

$$c_v(T_3 - T_2) + c_p(T_{3'} - T_2) \qquad (32)$$

Whereas the heat rejected, Qr per unit mass of charge is given by

cv (T4 – T1) and

The thermal efficiency is given by

$$\eta_{th} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2) + c_p(T_{3'} - T_2)} \qquad (33A)$$

$$= 1 - \left\{ \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right) + \gamma T_3\left(\frac{T_{3'}}{T_3} - 1\right)} \right\} \qquad (33B)$$

$$= 1 - \frac{\frac{T_4}{T_1} - 1}{\frac{T_2}{T_1}\left(\frac{T_3}{T_2} - 1\right) + \frac{\gamma T_3}{T_2}\frac{T_2}{T_1}\left(\frac{T_{3'}}{T_3} - 1\right)} \qquad (33C)$$

From thermodynamics

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = r_p \qquad (34)$$

the explosion or pressure ratio and

$$\frac{T_{3'}}{T_3} = \frac{V_{3'}}{V_3} = r_c \qquad (35)$$

the cut-off ratio.

$$\text{Now, } \frac{T_4}{T_1} = \frac{p_4}{p_1} = \frac{p_4}{p_{3'}}\frac{p_{3'}}{p_3}\frac{p_3}{p_2}\frac{p_2}{p_1}$$

$$\text{Also } \frac{p_4}{p_{3'}} = \left(\frac{V_{3'}}{V_4}\right)^\gamma = \left(\frac{V_{3'}}{V_3}\frac{V_3}{V_4}\right)^\gamma = \left(r_c\frac{1}{r}\right)^\gamma$$

$$\text{And } \frac{p_2}{p_1} = r^\gamma$$

$$\text{Thus } \frac{T_4}{T_1} = r_p r_c^\gamma$$

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

Therefore, the thermal efficiency of the dual cycle is

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_p r_c^\gamma - 1}{(r_p - 1) + \gamma r_p (r_c - 1)}\right] \qquad (36)$$

1. An air standard dual cycle has a compression ratio of 16 and compression begins at 1bar and 50^0 C. The maximum pressure is 70bar. The heat transferred to air at constant Pressure is equal to heat transferred at constant volume. Find the temperature at all cardinal Points, cycle efficiency and mean effective pressure. Take C_p = 1.005kJ/kgK; C_V=0.718kJ/kgK.

Given data:

$r = 16$

$P_1 = 1bar$

$T_1 = 50^0C = 323K$

$P_3 = 70bar$

$Q_{S_1} = Q_{S_2}$

$C_p = 1.005kJ/kgK$

$C_V = 0.718kJ/kgK$

Solution:

Specific volume, $v_1 = \frac{RT_1}{P_1} = \frac{287 \times 323}{1 \times 10^5}$

= 0.92701m3/*kg*

$V_2 = 0.05794m^3/kg$

1-2 ⇒ Isentropic compression process

$$p_2 = (r)^{\gamma} \times p_1 = (16)^{1.4} \times 1$$

= 48.5bar **Ans.**

$$T_2 = (r)^{\gamma - 1} \times T_1 = (16)^{1.4-1} \times 323$$

= 979K **Ans.**

2-3 ⇒ Constant volume heat addition process

$$T_3 = \left(\frac{p_3}{p_2}\right) \times T_2 = \frac{70}{48.5} \times 979$$

= 1413 K **Ans.**

$$Q_{S_1} = C_V(T_3 - T_2)$$

= 0.718 (1413 – 979)

= 311.612kJ/kg

3-4 $\Rightarrow$ ***Constant pressure heat addition***

$$Q_{S_1} = Q_{S_2} = C_P(T_4 - T_3)$$

311.612 = 1.005 (T_4 – 1413)

T_4 = 1723K **Ans.**

$$v_4 = \frac{T_4}{T_3} \times v_3 = \frac{1723}{1413} \times 0.05794 \qquad (\because v_2 = v_3)$$

= 0.070652*m³/kg*

Expansion ratio, $r_e = \frac{v_4}{v_1} = \frac{0.070652}{0.92701}$

r_e = 0.076215

4-5 $\Rightarrow$ ***isentropic expansion process***

$$p_5 = (r_e)^{\gamma} \times p_4 = (0.076215)^{1.4} \times 70$$

= 1.9063bar **Ans.**

$\therefore$ **Cut off ratio,** $\rho = \frac{v_4}{v_3} = \frac{0.070652}{0.05744}$

= 1.2194

$\therefore$ **Pressure ratio** $(k) = \left(\frac{p_3}{p_2}\right) = \left(\frac{70}{48.5}\right)$

= 1.4433

The cycle efficiency,

$$\eta = 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{(k\rho^{\gamma} - 1)}{(k-1) + k \times \gamma \times (\rho - 1)}\right]$$

$$=1-\frac{1}{(16)^{1.4-1}}\left[\frac{\left(1.4433\times(1.2194)^{1.4}-1\right)}{(1.4433-1)+1.4\times1.4433\times(1.2194-1)}\right]$$

= 66.34% **Ans.**

Net heat supplied to the cycle,

$$Q_S = Q_{S_1} + Q_{S_2}$$

= 311.612 + 311.612
= 623.224*kJ/kg*

Net work done of the cycle,

$$W = Q_S \times \eta$$

= 623.224 x 0.6634

= 413.45*kJ/kg*

The mean effective pressure,

$$\rho_m = \frac{W}{v_1 - v_2} = \frac{413.45}{0.92701 - 0.05794}$$

= 4.75*bar* **Ans.**

2. **In engine working on dual cycle, the temperature and pressure at the beginning of the Cycle of the are 90^0C and 1bar. The compression ratio is 9. The maximum pressure is limited to 68bar and total heat supplied per kg of air is 1750kJ. Determine air standard efficiency and mean effective pressure.**

Given data:

$p_1 = 1$ bar

$T_1 = 90^0C = 363K$

$p_3 = p_4 = 68bar$

$r=9$

$Q_S = 1750kJ/kg$

Solution:

1-2 $\Rightarrow$ isentropic comp. process

$$p_2 = (r)^{\gamma} \times p_1 = (9)^{1.4} \times 1$$

= 21.67bar

$$T_2 = (r)^{\gamma-1} \times T_1 = (9)^{0.4} \times 363$$

= 874K

2-3 $\Rightarrow$ Constant volume heat addition process

$$T_3 = \left(\frac{p_3}{p_2}\right) \times T_2 = \left(\frac{68}{21.67}\right) \times 874 = 2743\,K$$

3-4 $\Rightarrow$ Constant pressure heat addition process

$$Q_S = C_V(T_3 - T_2) + C_p(T_4 - T_3)$$

1750 = 0.718 (2743 – 874) + 1.005 (T_4 – 2743)

T_4 = 3149K

$$v_1 = \frac{RT_1}{p_1} = \frac{287 \times 363}{1 \times 105} = 1.04181\, m3/kg$$

$$v_3 = v_2 = \frac{v_1}{r} = \frac{1.04181}{9}$$

=0.11576m^3/kg

$$v_4 = \left(\frac{T_4}{T_3}\right) \times v_3 = \left(\frac{3149}{2743}\right) \times 0.11576$$

= 0.132894m3/kg

Cut off ration, $\rho = \frac{v_4}{v_3} = \frac{0.13289}{0.11576}$

= 1.148

Pressure ratio, $k = \frac{p_3}{p_2} = \frac{68}{21.67}$

= 3.138

Efficiency of the cycle,

$$\eta = 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{k\rho^{\gamma} - 1}{(k-1) + k \times \gamma(\rho - 1)}\right]$$

$$1 - \frac{1}{(9)^{1.4-1}}\left[\frac{3.138 \times 1.148^{1.4} - 1}{(3.13-1) + 3.138 \times 1.4(1.148)}\right]$$

= 58.19 % **Ans.**

Net work done of the cycle,

$$W_{net} = \eta \times Q_S$$

= 0.5819 x 1750

= 1018.33*kJ/kg*

Mean effective pressure,

$$p_m = \frac{W_{net}}{v_1 - v_2}$$

$$= \frac{1018.33}{1.04181 - 0.11576} = 10.98 bar$$ **Ans.**

3. An air standard dual cycle has a compression ratio of 16 and compression begins at 1 bar and 50ºC. The maximum pressure is 70 bar. The heat transformed to air at constant pressure is equal to heat transferred at constant volume. Find the temperature at all cardial points, cycle efficiency and mean effective pressure take Cp= 1.005KJ/kgK, Cv = 0.718KJ/kgK.

Given data:
r = 16
P1 = 1 bar
T1 =50ºC = 323K
P3 = 70 bar Qs1 = Qs2
Cp = 1.005kJ/kgk
Cv = 0.718kJ/kgk

Solution:
Specific volume,

V1RT1/P1 = 287 x 323/1 x 105 V1= 0.92701m3/kg
V_2 = 0.05794m^3/kg

1-2 isentropic compression process:

P_2 = (r) x P_1 = (16) $^{1.4}$ x 1 =

48.5 barT_2 = (r) $^{-1}$ x T_1 = (16)

$^{1.4-1}$ x 323

T_2 = 979K

2-3 constant volume heat addition

process: $T_3 = (P_3/P_2) \times T_2$

$= 70/48.5 \times 979$ $T_3 = 1413K$

$Q_{s1} = Cv (T_3 - T_2)$; $0.718(1413 – 979)$

$Q_{s1} = 311.612 KJ/kg$

3-4 constant pressure heat addition: $Q_{s1} = Q_{s2} = C_p (T_4 – T_3)$ $311.62 = 1.005 (T_4 – 1413)$

$T_4 = 1723K$

$V_4 = T_4/T_3 \times V_3 = 1723/1413$ X

$0.05794 V_4 = 0.070652 m^3/kg$

Expansion ratio:

$r_e = V_4/V_1 = 0.70652/0.92701 = 0.06215$

4-5 isentropic expansion process:

$P_5 = (r) \times P_4 = (0.076215)^{1.4}$

x 70 $P_5 = 1.9063$ bar

$T_5 = (r)^{□-1} \times T_4$

$= (0.076215)^{1.4-1} \times 1723$

$= 567$ K

Cut off ratio,

ρ= V4/ V3

=0.00652/0.05744

ρ = 1.2194

Pressure ratio (K) = (P3/P_2) =

(70/ 48.5) K = 1.4433

The cycle efficiency:

η= 1- 1/(r)$^{□-1}$ [(kp□-1)/ (k-1 + K□(p-1)]

= 66.34%

Net heat supplied to

the cycle: Q_s =

$Q_{s1} + Q_{s2}$

$= 311.612 + 311.612$

$= 623.224$ KJ/kg

The mean effective pressure:

$P_m = W/ V_1 – V_2 = 413.45/ (0.92701 -0.05794)$

$P_m = 4.75$ bar

4. In a air standard dual cycle, the compression ratio is 12 and the maximum pressure and temperature of the cycle are 1 bar and 300K. haet added during constant pressure process is upto 3% of the stroke. taking diameter as 25cm and stroke as 30cm, determine.

1) **The pressure and temperature at the end of compression**
2) **The thermal efficiency**
3) **The mean effective pressure**

Take, Cp =1.005KJ/kgK Cv =0.118KJ/kgk , = 1.4

Given data:

P_1 = 1 barr = 12

T_1 =300K

K = 3% of Vs = 0.03Vs

P_3 = 70 bar

D = 25 cmL =30cm

Solution:

Specific volumes,:

$V_1\ RT_1/ P_1 = 287 \times 300/ 1 \times 10^5$

$= 0.861\ m^3 /kg$

$V_3 = V_2 = V_1/r = 0.861/12$

$= 0.07175 m^3/kg$

$V_4 – V_3 = 0.03 (V_1 –V_2) V_4 = 0.0954275\ m^3/kg$

$\rho = V_4 /V_3 = 0.054275/0.07175 \rho = 1.33$

1-2 isentropic compression process:

$P_2 = (r) \square \times P1 = (12)^{1.4} \times 1$

= 32.423 bar

$V_2 = (r)^{\square-1} \times T_1 = (12)^{1.4-1} \times 300 T_2 = 810.57K.$

2-3 constant volume heat addition process

$P_3/T_3 = P_2/T_2$

$T_3 = (P_3/P_2) \times T_2 = (70 / 32.423) \times$

$810.57 T_3 = 1750K$

3-4 constant pressure heat addition process:

$T_4 = (V4/V_3) \times T_3 = (0.0954275 / 0.07175)$

$\times 1750 T_4 = 2327.5$ K

Pressure ratio, K = (P_3/P_2) = 70/32.423 = 2.159

Net heat supplied to the cycle:

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

$= 0.718 (1750 - 810.57) + 1.005 (2327.5 - 1759)$

= 1254.9 KJ/kg

Efficiency of the cycle:

$\eta = 1 - 1/(r)^{\square-1} [(K \times P^{\square} - 1)/(k-1) + K\square(p-1)]$

= 61.92%

Net work done of the cycle:

$W = \eta \times Q_s$

= 0.6192 x 1254.9

= 777.1 KJ/kg

Mean effective pressure,

$P_m = W/V_1 - V_2$

= 777.1/ 0.361 – 0.07115

= 984.6 Kpa

P_m = 9.846 bar

5. The compression ratio of a dual cycle is 10. The pressure and temperature at the beginning of the cycle are 1 bar and 27°C. the maximum pressure of the cycle is limited to 70 bar and heat supplied is limited to 1675KJ/kg fair find thermal efficiency.

GIven data:

$r = 10$

$P_1 = 1$ bar

$T_1 = 27^0C = 300K$

$P_3 = 70$ bar

$Qs = 1675$ KJ/kg

Solution:

Specific volumes:

$V_1 = RT_1/P_1 = 287$ x

$300/1 \times 10^5 V_2 = V_1/r$

$= 0.861/10$

1-2 isentropic compression process:

$P_1 = (r)^{\square} \times P_1 = (10)^{1.4} \times 1 = 25.12$ bar

$T_2 = (r)^{\square-1} \times T_1 = (10)^{1.4-1} \times 300 = 753.57K$

2-3 constant volume heat addition process:

$T_3 = (P_3/P_2) \times T_2 = (70 / 25.12) \times 753.37 = 2100K$

Total heat supplied to the cycle:

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

$1675 = 0.718 (2100 -753.57) + 1.005 (T4 -$

$2100) T_4 = 2804.6$ K

Cut off ratio:

$\rho = V_4/V_3 = T_4/T_3 =$

$2804.6/2300 \rho = 1.3356$

Pressure ratio:

$K = P3/P2 = 70/25.12 = 2.787$

Efficiency of the cycle:

$\eta = 1- 1/ (r)^{\square-1} [(K \times P^{\square}-1)/(k-1) + K\square(p-1)]$

$= 59.13\%$

Explain the working principle of brayton cycle and draw the pv and T-s diagram .Also derive the air standard efficiency.

The Brayton cycle is also referred to as the Joule cycle or the gas turbine air cycle because all modern gas turbines work on this cycle. However, if the Brayton cycle is to be used for reciprocating piston engines, it requires two cylinders, one for compression and the other for expansion. Heat addition may be carried out separately in a heat exchanger or within the expander itself.

The pressure-volume and the corresponding temperature-entropy diagrams are shown in Figs 10 and 11 respectively.

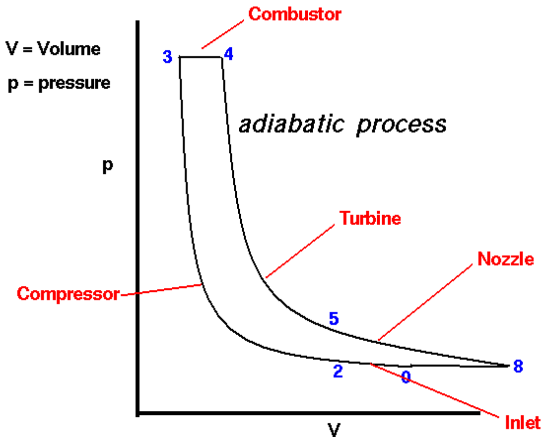


Fig. 10

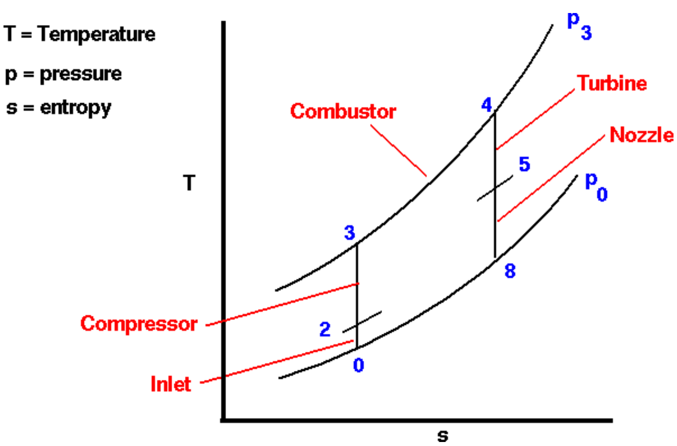


Fig. 11

The cycle consists of an isentropic compression process, a constant pressure heat addition process, an isentropic expansion process and a constant pressure heat rejection process. Expansion is carried out till the pressure drops to the initial (atmospheric) value.

Heat supplied in the cycle, Qs, is given by Cp(T3 – T2)

Heat rejected in the cycle, Qs, is given by Cp (T4 – T1)

Hence the thermal efficiency of the cycle is given by

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

$$= 1 - \frac{T_1}{T_2}\left\{\frac{\left(\frac{T_4}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)}\right\} \qquad (42)$$

$$\text{Now } \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4}$$

$$\text{And since } \frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \textit{we have} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Hence, substituting in Eq. 62, we get, assuming that r_p is the pressure ratio p_2/p_1

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

$$= 1 - \frac{1}{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}}$$

$$= 1 - \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}} \qquad (43)$$

This is numerically equal to the efficiency of the Otto cycle if we put

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

$$\text{so that } \eta_{th} = 1 - \frac{1}{r^{\gamma-1}} \qquad (43A)$$

where r is the volumetric compression ratio.

1. In a gas turbine plant working on the Brayton cycle, the air at the inlet is at 23^{0}C, 0.1MPa. The pressure ratio is 6.75 and the maximum temperature is 750^{0}C. The turbine expansion is divided into two stages with reheat to 750^{0}C. The efficiency of compressor and two turbines are 82 % and 86% respectively. Determine the maximum power that can be Obtained from this plant, if the mass flow rate of air is 5kg/sec.

Given data:

$p_1 = 0.1MPa = 1\ bar$

$T_1 = 23^0C = 296K$

$$R_p = \frac{p_2}{p_1} = 6.75$$

$T_3 = T_5 = 750^0C = 1023K$

$\eta_C = 82\% = 0.82$

$\eta_{T1} = \eta_{T2} = 86\% = 0.86$

Solution

For maximum work or power developed

$$p_4 = p_5 = p_4' = \sqrt{p_1 p_2} = \sqrt{1 \times 6.75} = 2.598 bar$$

From process 1-2 isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 296(6.75)^{\frac{1.4-1}{1.4}} = 510.78K$$

$$\eta_C = \frac{T_2 - T_1}{T_2' - T_1} = \frac{510.78 - 296}{T_2' - 296} = 0.82$$

$$T_2' = 557.9K$$

From isentropic expansion of air in first turbine 3-4

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{6.75}{2.598}\right)^{\frac{1.4-1}{1.4}} \qquad (\because P_2 = P_1 \times 6.75 = 6.73)$$

$$\eta_{T_1} = \frac{T_3 - T_4'}{T_3 - T_4} = \frac{1023 - T_4'}{1023 - 778.75} = 0.86$$

$T_4' = 812.94\ K$

Similarly, for turbine 2-process 5-6

$$\frac{T_5}{T_6} = \left(\frac{P_5}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{2.598}{1}\right)^{\frac{1.4-1}{1.4}} = 1.313$$

$$T_6 = \frac{T_5}{1.313} = \frac{1023}{1.313} = 778.76K$$

$$\eta_{T_2} = \frac{T_5' - T_6'}{T_5 - T_6} = \frac{1023 - T_6'}{1023 - 778.76} = 0.86$$

T_6'=812.95K

Work done by the turbine

$W_T = mC_p\ [(T_3 - T_4') + (T_5 - T_6')]$

$= 5 \times 1.005\ [\ (1023 - 812.94) + 1023 - 812.95)]$

$= 2111.1$ kJ/s

Work required by compressor

$W_C = m \times C_p(T_2' - T_1) = 5 \times 1.005\ (557.9 - 296)$

$= 1316$kJ/s

Net work,

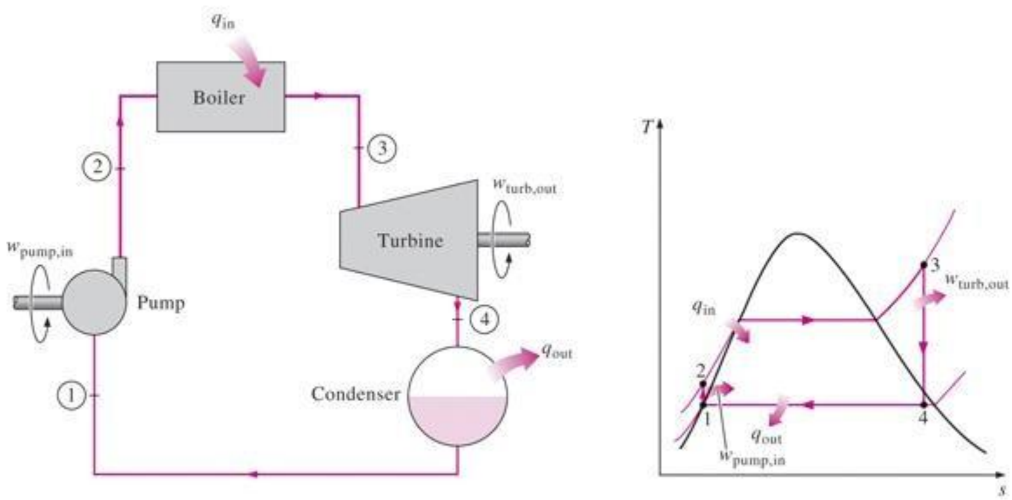
$W = W_T - W_C = 2111.1 - 1316$

$W = 795.1$kJ/s or kW Ans.

Rankine Cycle

Rankine cycle is the thermodynamic cycle for steam power plant. The various processes are

- 1-2: Isentropic compression
- 2-3: Reversible constant pressure heat addition
- 3-4: Isentropic expansion
- 4-1: Reversible constant pressure heat rejection



Work output of turbine $W_T = (h_3 – h_4)$

Work input to pump $W_P = (h_2 – h_1)$

Heat supplied in boiler $Q_s = (h_3 – h_2)$

Efficiency of the cycle $\eta = \dfrac{W_T - W_P}{Qs}$

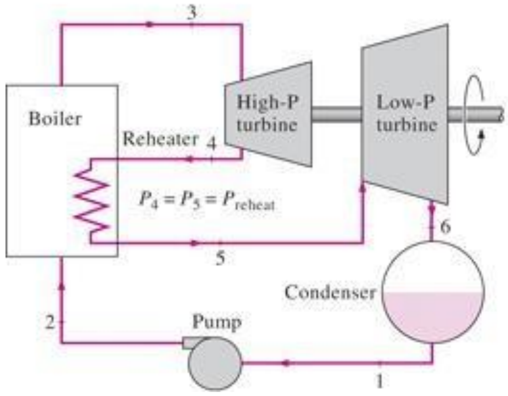
Steam rate $= \dfrac{3600}{W_T - W_P}$ kg/kW hr

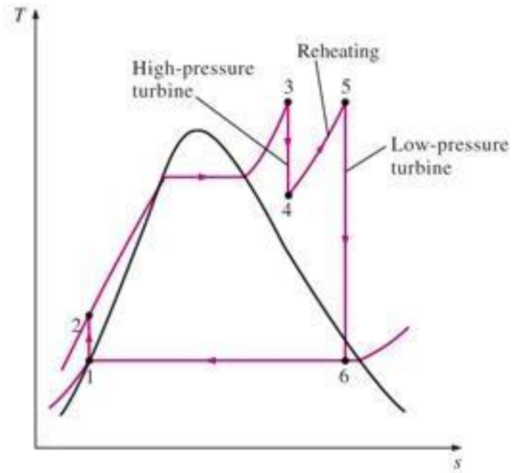
Heat rate $= \dfrac{3600 \times Qs}{W_T - W_P}$ kJ/kW hr

Reheat Cycle

The various processes in reheat cycle are:

- 1-2: Isentropic compression
- 2-3: Reversible constant pressure heat addition
- 3-4: Isentropic expansion in HP turbine
- 4-5: Reheating process
- 5-6: Isentropic expansion in LP turbine
- 6-1: Reversible constant pressure heat rejection





Work output of turbine $W_T = (h_3 – h_4) + (h_5 – h_6)$

Work input to pump $W_P = (h_2 – h_1)$

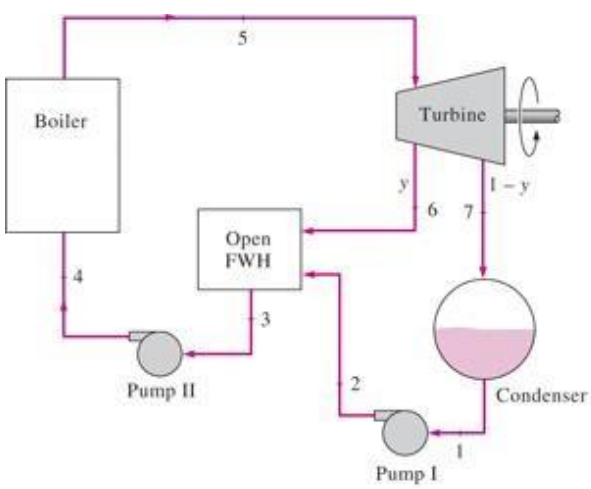
Heat supplied in boiler $Q_s = (h_3 – h_2) + (h_5 – h_4)$

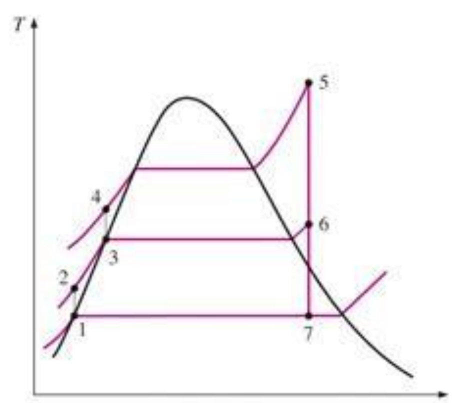
Efficiency of the cycle $\eta = \dfrac{W_T - W_P}{Qs}$

Steam rate $= \dfrac{3600}{W_T - W_P}$ kg/kW hr

Heat rate $= \dfrac{3600 \times Qs}{W_T - W_P}$ kJ/kW hr

Regenerative Cycle





The various processes are

1-2: Isentropic compression in Pump I

2-3: Reversible constant pressure heat addition in regenerator

3-4: Isentropic compression in Pump II
4-5: Reversible constant pressure heat addition in boiler
5-7: Isentropic expansion
7-1: Reversible constant pressure heat rejection

Work output of turbine $W_T = (h_5 – h_6) + (1 – y)(h_6 – h_7)$
Work input to pump $W_P = (1 – y) (h_2 – h_1) + (h_4 – h_3)$
Heat supplied in boiler $Q_s = (h_5 – h_4)$
Energy balance for regenerator $y h_6 + (1 – y) h_2 = h_3$ Where y = bleed steam

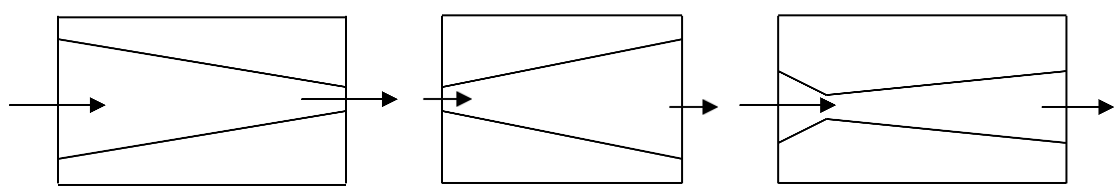
Efficiency of the cycle $\eta = \frac{W_T - W_P}{Qs}$

MODULE I

STEAM

NOZZLES

When a fluid flows through a passage or channel of varying cross section, its velocity varies from point to point along the passage. If the velocity increases, the passage is called a Nozzle. The steam nozzle is device designed to increase the velocityof steam. The fluid enters the nozzle at high pressure and expands to lower pressure. If the cross section of the nozzle decreases continuously fromthe entrance to exit, it is called Convergent nozzle. The maximum Mach number at the exit of the convergent nozzle is 1. If the cross section of the nozzle increases, it is called Divergent nozzle. If the cross section of the nozzle, first decreases and then increases, it is called Convergent-divergent nozzle. At the throat, i.e., at the narrowest cross section the Mach number is 1.



Convergent Divergent Convergent-Divergent

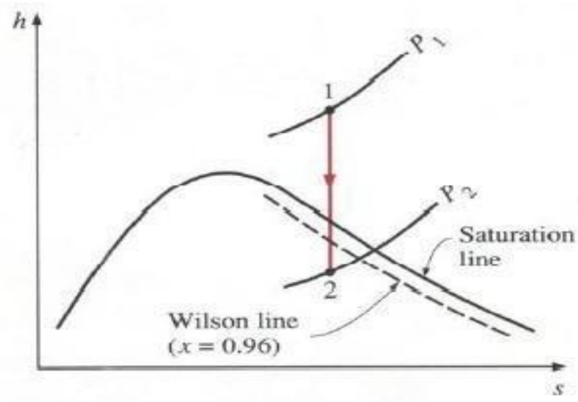
1. Describe about supersaturated flow or metastable flow in a nozzleand state effect of super saturation

As steam expands in the nozzle, its pressure and temperature drop, and it is expected that the Steam start condensing when it strikes the saturation line. But this is not always the case. Owingto the high velocities, the residence time of the steam in the nozzle is small, and there may not Sufficient time for the necessary heat transfer and the formation of liquid droplets. Consequently,the condensation of steam is delayed for a little while. This phenomenon is known as

Super saturation, and the steam that exists in the wet region without containing any liquid isKnown as supersaturated steam.

The locus of points where condensation will take place regardless of the initial temperature and

Approximated by the 4 percent moisture line. The super saturation phenomenon is shown on theh-s chart below:



The h-s diagram for the isentropic expansion of steam in a nozzle.

Effects of Supersaturation:

The following are the effects of supersaturation in a nozzle.

(a) The temperature at which the supersaturation occurs will be less than the saturation temperature corresponding to that pressure. Therefore, the density of supersaturated steam will be more than that of equilibrium condition which gives the increase in the mass of steam discharged.

(b) Supersaturation increases the specific volume and entropy of the steam.

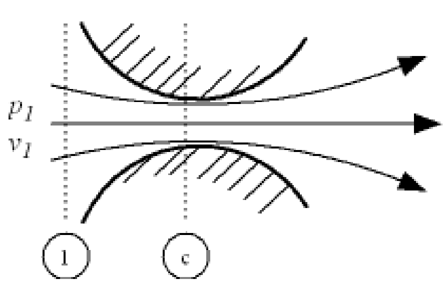
(c) Supersaturation reduces the heat drop. Thus the exit velocity of steam is reduced.

(d) Supersaturation increases the dryness fraction of the steam.

Critical Pressure Ratio:

The critical pressure ratio is the pressure ratio which will accelerate the flow to a velocityequal to the local velocity of sound in the fluid.

Critical flow nozzles are also called sonic chokes. By establishing a shock wave the sonic choke establish a fixed flow rate unaffected by the differential pressure, any fluctuations or changes in downstream pressure. A sonic choke may provide a simple way to regulate a gas flow.



The ratio between the critical pressure and the initial pressure for a nozzle can expressed as $p_c / p_1 = (2 / (n + 1))^{n / (n - 1)}$

where

p_c = critical pressure

(Pa) p_1 = inlet pressure

(Pa)

n = index of isentropic expansion or compression - or polytropic constant

For a perfect gas undergoing an adiabatic process the index - n - is the ratio of specific heats - k = c_p / c_v. There is no unique value for - n. Values for some common gases are

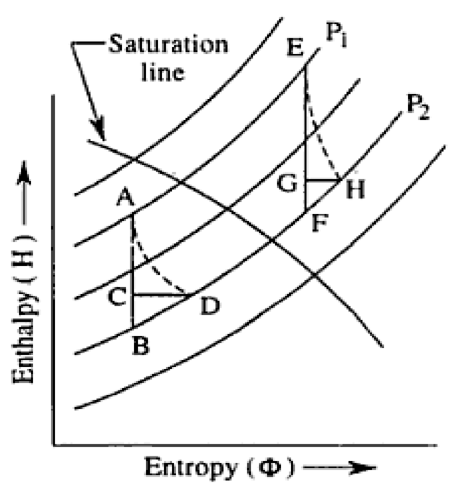
- **Steam where most of the process occurs in the wet region : n = 1.135**
- **Steam superheated: n = 1.30**
- **Air: n = 1.4**
- **Methane: n =**
- **1.31Helium: n = 1.667**

2. Describe the Effect of Friction on Nozzles.

1) Entropy is increased.
2) Available energy is decreased.
3) Velocity of flow at throat is decreased.
4) Volume of flowing steam is decreased.
5) Throat area necessary to discharge a given mass of steam is increased.

Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about 10 to 15 per cent. The velocity of steam will be then $V_2 = 44.72\sqrt{K(H_1 - H_2)}$ where K is the coefficient which allows for friction loss. It is also known as nozzle efficiency (η_n)

$$\therefore \quad V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n}$$



Velocity of steam at nozzle exit:

$$V_2^2 = 2000(H_1 - H_2) + V_1^2 \quad \therefore \quad V_2 = \sqrt{2000(H_1 - H_2) + V_1^2}$$

As the velocity of steam entering the nozzle is very small, V_1 can be neglected.

$$\therefore \qquad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

If frictional losses are taken into account then

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

3. Derive the expression for maximum discharge through a nozzle.

Mass of steam discharged through a nozzle:

$$m = A\sqrt{2000\frac{n}{n-1} \times \frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

Condition for maximum discharge through nozzle:

The nozzle is always designed for maximum discharge

$$\frac{m}{A} = \sqrt{2000\frac{n}{n-1} \times \frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

The mass flow per unit area will be maximum at the throat because the throat area is minimum.

It is seen from the above equation that the discharge through a nozzle is a function of $\frac{P_2}{P_1}$ only, as the expansion index is fixed according to the steam supplied to the nozzle.

Therefore, $\frac{m}{A}$ is maximum when

$$\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right] \text{ is minimum}$$

Values for maximum discharge:

$$m = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

we know $\dfrac{P_2}{P_1}=\left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$

Putting the value of $\dfrac{P_2}{P_1}$ in the above equation

$$m_{max} = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}-\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$m_{max} = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}-\frac{n+1}{n-1}}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{-1}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\frac{n+1}{2}-1\right]}$$

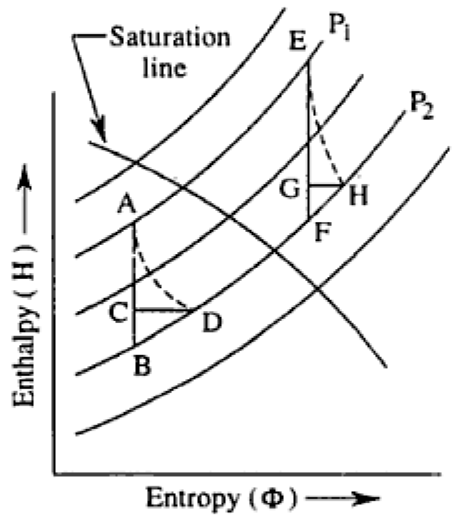
$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left(\frac{n-1}{2}\right)}$$

$$m_{max} = A\sqrt{1000n\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

W3here P_1 is the initial pressure of the steam in kpa and v_1 is the specific volume of the steam

Metastable expansion of steam in a nozzle with help of h-s diagram.

Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about 10 to 15 per cent. The velocity of steam will be then $V_2 = 44.72\sqrt{K(H_1 - H_2)}$ where K is the coefficient which allows for friction loss. It is also known as nozzle efficiency (η_n)



$$\therefore \quad V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n}$$

150m/s. The absolute

1) Velocity of steam at exit from the stage is 85 m/sat an angle of 80º from the tangential direction. Blade velocity coefficient is 0.82 and the rate of steam flowing through the stageis 2.5 kg/s. If the blades are equiangular, determine i) Blade angles and ii) Axial thrust. (MAY/JUNE 2016)

***(a)* Blade inlet angle**

By measurement from the velocity diagram, we find that the blade angle at inlet,

$\theta = 29^0$

***(b)* During force on the wheel**

We know that driving force on the wheel,

$F_x = m(V_w + V_{w1}) = 0.5\ (1130 + 190) = 660$ N

***(c)* Arial thrust on the wheel**

We know that axial thrust on the wheel,

$$F_\gamma = m\left(V_f - V_{f1}\right) = 0.5\left(410 - 310\right) = 50N$$

***(d)* Power development by the turbine**

We know that power development by the turbine,

$P = m(V_w + V_{w1})\ V_b$

$= 0.5(1130 + 190)\ 375 = 247\ 500$ W

$= 247.5$ Kw

STEAM NOZZLES PROBLEM

1) Dry saturated steam at a pressure of 8 bar enters a convergent divergent nozzle and leaves it at a pressure of 1.5 bar. If the flow is isentropic and if the corresponding expansion index is 1.133, find the ratio of cross sectional area at exit and throat for maximum discharge.

Heat drop between entrance and exit,

$$h_{d3} = h_1 - h_3 = 2775 - 2465 = 310 kJ/kg$$

∴ **Velocity of steam at throat,**

$$V_3 = \frac{\sqrt{h_{d3}}}{44.72} = \frac{\sqrt{310}}{44.72} = 787.4 m/s$$

an d $$m = \frac{A_2 V_2}{x_2 \upsilon_{g\,2}}$$

or $$A_3 = \frac{m x_3 \upsilon_{g3}}{V_3} = \frac{\mathbf{m \times 0.902 \times 1.159}}{\mathbf{787.4}} = \mathbf{0.001.33m}$$

∴ **Ratio of cross-sectional area at exit and throat,**

$$\frac{A_3}{A_2} = \frac{0.00133m}{0.000786m} = 1.7$$

2. Dry saturated steam at a pressure of 11 bar enters a convergent divergent nozzle and leaves at atpressure of 2 bar. If the flow is adiabatic and frictionless, determine

i) The exit velocity of steam and

ii) Ratio of cross section of exit and that at throat.

From steam tables, corresponding to a pressure of 10 bar, we find that enthalpy or total heat of dry saturatedsteam,

$$h_1 = h_{g1} = 2776.2 \text{ kJ/kg}$$

and

Corresponding to a pressure of 0.1 bar,

$$\mathbf{h_{f2} = 191.8 \text{ kJ/kg, and } h_{fg2} = 2392.9 \text{ kJ/kg}}$$

∴ **Enthalpy or total heat of steam of exit,**

$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$= 191.8 + 0.791 \times 2392.2$$

$$= 2084.6 \text{ kJ/kg}$$

and heat drop, $h_d = h_1 - h_2 = 2776.2 - 2084.6$

$$= 691.6 \text{ kJ/k}$$

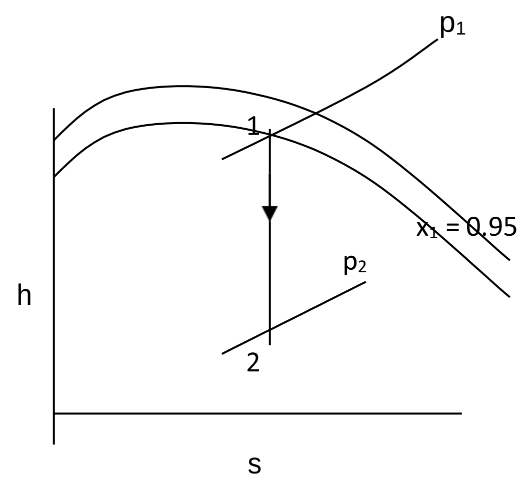
3 Steam approaches a nozzle with velocity of 250 m/s, pressure of 3.5 bar and dryness fraction of 0.95. If the isentropic expansion in the nozzle proceeds till the pressure of the exit is 2 bar, determine the change in enthalpy and the dryness fraction of steam. Calculate also the exit velocity from the nozzle and the area of the exit of the nozzle for the flow of 0.75 kg/s.

Given

Type = Convergent
Velocity of steam at inlet (V_1) = 250 m/s
Pressure at inlet (p_1) = 3.5 bar with x_1 = 0.95
Pressure at outlet (p_2) = 2 bar
Mass flow rate (m) = 0.75 kg/s

Required: ($h_1 - h_2$), x_2, V_2 & A_2

Solution



From Chart,

h_1 = 2625 kJ/kg h_2

= 2540 kJ/kg x_2

=**0.92 --- Ans**

∴ ($h_1 - h_2$) = 2625 – 2540 = **85 kJ/kg ----------------- Ans**

$(V_2^2 - V_1^2) / 2 = h_1 - h_2$

$(V_2^2 - 250^2) / 2 = 85 \times 10^3$

V_2 = **482.2 m/s**

A_2 = Area of the nozzle at outlet

$m = A_2 V_2 / v_2$

$v_2 = 0.8\ m^3/kg$ from chart at point (2)

$\therefore \quad 0.75 = A_2 \times 482.2 / 0.8$

$A_2 = 0.0012443\ m^2$ = **$12.443\ cm^2$---------- Ans**

4. Dry saturated steam at pressure of 8 bar flows through nozzles at the rate of4.6 kg/s and dischargesat a pressure of 1.5 bar. The loss due to friction occursonly in the diverging portion of the nozzle and its magnitude is 12 % of the total isentropic enthalpy drop. Assume the isentropic index of expansionn = 1.135, determine the cross sectional area at the throat and exit of the nozzles.

<u>Given</u>

Type	= Con-div type
Inlet pressure (p_1)	= 8 bar, dry
Mass flow rate of steam (m)	= 4.6 kg/s
Discharge pressure (p_3)	= 1.5 bar
$h_3 - h_{3'}$	= $0.12\ (h_1 - h_{3'})$
n	= 1.135

<u>Required:</u> A_2, A_3

<u>Solution</u>

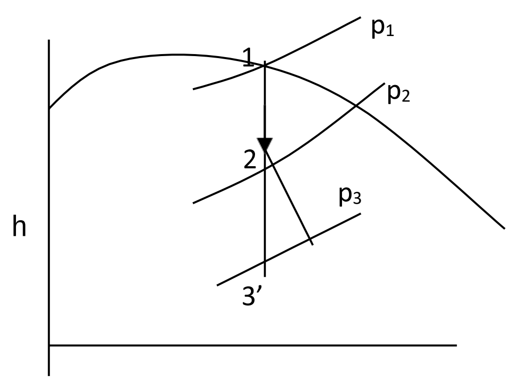
$$M = A_2V_2 / v_2 = A_3V_3 / v_3$$

Inlet velocity is not given, $\therefore\ V_1 = 0$

$$\therefore\ V_2^{\ 2}/2 = h_1 - h_2$$

From chart,

$h_1 = 2770$ kJ/kg



We can write,

$$P2/p1 = \left[\frac{2}{n+1}\right]^{n / (n+1)}$$

$$\therefore\ p_2 / 8 = [2 / (1.135 + 1)]^{1.135./ (1.135 - 1)}$$

$p_2 = 4.62$ bar

From Chart, $h_2 = 2545$ kJ/kg

$v_2 = 0.42$ m^3/kg

$\therefore \quad V_2^2/2 = (2770 - 2545) \times 10^3$

$V_2 = 670.8$ m/s

$\therefore \quad 4.6 = A_2 \times 670.8 / 0.42$

$A_2 = 0.00288$ m^2 **= 28.8 cm^2 ----------- Ans**

From chart, $h_{3'} = 2455$ kJ/kg

$v_{3'} = 1.1$ m^3/kg $\approx v_3$

$h_3 - h_{3'} = 0.12\,(h_1 - h_{3'})$

$h_3 - 2455 = 0.12\,(2770 - 2455)$

$h_3 = 2492.8$ kJ/kg

$V_3^2/2 = (2770 - 2492.8) \times 10^3$

$V_3 = 744.6$ m/s

$\therefore\ 4.6 = A_3 \times 744.6 / 1.1$

$A_3 = 0.0067956$ m^2 **= 67.956 cm^2 ------- Ans**

5. Steam at a pressure of 10 bar and dryness fraction of 0.98 is discharged through a convergent divergent nozzle to a back pressure of 0.1 bar. The massflow rate is 10 kg/kWh. If the power developed is 200 kW, determine, (a)Pressure at the throat (b) Number of nozzles required, if each nozzles has a throat of rectangular cross section of 5 mm x 10 mm and (c) exit area of nozzle if 10 % the overall isentropic enthalpy drop reheats the steam by friction in the divergent portion.

Given

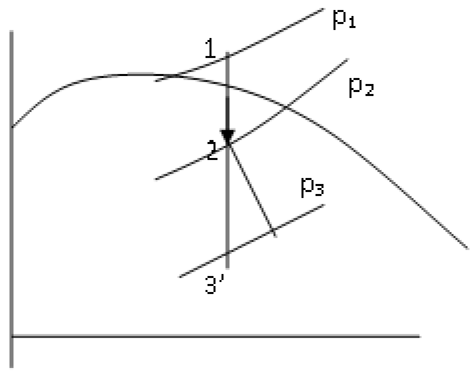
Type	= Con-div
Inlet pressure (p_1)	= 10 bar with $x_1 = 0.98$
Back pressure (p_3)	= 0.1 bar
Mass flow rate (m)	= 10 kg/kWh
Power (P)	= 200 kW
Size of nozzle	= 5 mm x 10 mm
$h_3 - h_{3'}$	= $0.1\,(h_1 - h_{3'})$

Required: (a) p_2

(b) Number of nozzles

(c) A_3

Solution



(a) Pressure at throat (p_2)

$$P2/p1 = \left[\frac{2}{n+1}\right]^{n/(n+1)}$$

$n = 1.035 + 0.1\ (x_1)$

$= 1.035 + (0.1 \times 0.98) = 1.133$

$\therefore\ p_2 / 10 = [2 / (1.133 + 1)]^{1.133./(1.133 - 1)}$

p_2 = **5.78 bar ------ Ans**

(b) Number of nozzles = Total area / Area per nozzle = A_2 / A_2 per nozzle

A_2/nozzle = 0.005 x 0.01 = 0.00005 m^2

$m = A_2V_2 / v_2$

$$m\ (kg/s) = \frac{m\ (kg/kWh) \times P}{3600}$$

= 10 x 200 / 3600 = 0.5556 kg/s

$V_2^2/2 = h_1 - h_2$

From chart, $h_1 = 2735$ kJ/kg

$h_2 = 2625$ kJ/kg

$v_2 = 0.32\ m^3/kg$

$\therefore\ V_2^2 = (27735 - 2625) \times 10^3$

$V_2 = 469.04$ m/s

$\therefore\quad 0.5556 = A_2 \times 469.04 / 0.32$

$A_2 = 0.00037905\ m^2$

$\therefore$ Number of nozzles = 0.00037905 / 0.00005 = 7.58 = **8------ Ans**

$m = A_3 V_3 / v_3$

$V_3^2/2 = h_1 - h_3$

From chart, $h_{3'} = 2055$ kJ/kg

$v_{3'} = 13\ m^3/kg = v_3$

$h_3 - h_{3'} = 0.1\ (h_1 - h_{3'})$

$h_3 - 2055 = 0.1 \times (2735 - 2055)$

$h_3 = 2123$ kJ/kg

$\therefore\quad V_3^2/2 = (2735 - 2123) \times 10^3$

$V_3 = 1106.3$ m/s

$\therefore\quad 0.5556 = A_3 \times 1106.3 / 13$

$A_3 = 0.0065276\ m^2$

reaction without the uses of flame to initiate the combustion, because the temperature is highthan self-ignition temperature.

20. What is meant by pre-ignition?

At very high temperature carbon deposits formed inside the combustion chamber ignites the air fuel mixture much before normal ignition occurred by spark plug. This is calledpre-ignition.

21. What are the factors affecting ignition lag?

Compression ratio, speed of the engine, Chemical nature of fuel and air fuel ration, andInitial pressure and temperature.

22. What is meant by knocking? How it occurs in diesel engines? Nov/Dec-2007

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.

23. What are the effects of knocking?

Ø The engine parts get overheated which may cause damage to the piston.
Ø It creates heavy vibration of engine and hence louder noise and roughness.Ø Decrease in power output and efficiency.
Ø More heat is lost to the coolant as the dissipation rate is rapid.
Ø The auto-ignition may over heat the spark plug and hence pre-ignition occursØ Carbon deposits.

24. What are the effects of rich mixture in petrol engine?

An engine might run rich especially when the weather is cold,whem under load or when it is accelerating

25. How are SI and Ci engione fuels rated?

SI engine fuels rating is done by Octane number and CI engine fuel rating is done by Cetane number

26. What is meant by ignition delay?

The Ignition delay period sub devided into two types.1.physical delay 2.Chemical delay.
The phycial delay period is the time between beginning of injection and attainment of chemical of chemical reaction conditions.Durimg this period,fuel is atomised,vapourised,mixed with air and raised to its self-ignitiontenperature.

During chemical delay ,reactions start slowly and then accelerate until the ignition takes place.Generally, thechemical delay is larger than physical delay.

27. What is the fuel injector?

Fuel injector is used in diesel engine to inject and atomize the diesel at the end of the compression stroke.

28. What is meant by SI engine? Why it is called so?

SI engine means spark ignition engine. In SI engine air fuel mixture is ignited by sparkplug hence it is called spark ignition engine. It is also called as petrol engine.

29. Give four major differences between two stroke and four stroke IC engine.

No Two stroke cycle engine Four Stroke cycle engine

piston or one revolution of the crank shaft. One cycle is completed in four stroke of the piston or two revolutionof the crank shaft.

1 For the same speed, twice the number of power strokes is produced than 4 stroke **engine. For the same speed, half of the number of power strokes is produced than 2 stroke engine.3 Turning moment is more uniform and hence lighter flywheel is used. Turning moment is not uniform and hence bigger flywheel is used.**

4 It contains ports which are operated by the piston movement. It contains valves which are operated by valvemechanism.

30. What is meant by CI Engine? Why it is called so?

CI engine means compression ignition engine. In CI engine the fuel is injected by a fuel injector in atomized form because of high compressed air it gets ignited automatically. Hence itis called as compression ignition engine.

31. What is a two stroke engine?

A two stroke engine is an engine in which one cycle of operation is completed in two stroke of the piston or one revolution of the crank shaft.

32. What is a four stroke engine?

A four stroke engine is an engine in which one cycle of operation is completed in four stroke of the piston or two revolution of the crank shaft.

33. Name the four strokes of an IC engine?

Suction, compression, power and exhaust stroke

34. Differentiate petrol and Diesel engines.

Petrol or SI engines	**Diesel or CI engine**
1. Combustion of air fuel mixture takes place by spark produced by sparkplug.	1. Combustion takes place by high compressed air.
2. Carburetor is used to mix the air fuel mixture.	2. Fuel injector is used to inject the fuel in Atomized form.
3. Compression ratio varies from 6 to 8.	3. Compression ratio varies from 12 to 18.
4.It works on Otto cycle.	4. It works on Diesel or Dual cycle.

45. What is the function of push rod and rocker arm in IC engine?

The function of push rod and rocker arm in IC engine is to transmit motion of the cam tothe valve.

46. What is the function of piston rings?

* It acts as air tight sealing between piston and cylinder to prevent gas leakages
* to wipe off the excess oil from the cylinder walls and also to return the excess oil to the sump through the slots provided on the rings.

47. What is 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.

48. What are the requirements of a fuel injection system of a diesel engine? Nov/Dec-2007

Ø To inject the fuel at correct moment, and quantity at various load conditionsØ To inject the fuel in a finely atomized condition.
Ø To distribute the fuel uniformly in the combustion chamber.Ø To control the rate of fuel injection.

49. List the requirements of ignition system.
Ø Ignition should takes place at the end of compression stroke.Ø There should be no missing cycle due to the spark failure.
Ø Ignition must add sufficient energy for starting and sub staining the charge burning
Ø Ignition system should supply the minimum required energy within a small volume in avery short time.

50. Which engine will have more cooling requirement two-stroke engine or four-stroke engine? Why?
Two stroke-engines will have more cooling requirements since power is developed for each revolution of crank. So, for each crank revolution, Combustion occurs and more heat willbe generated inside the cylin

STEAMTURBNE

The steam turbine is a prim-mover in which the potential energy of steam istransferred Into kinetic energy and later in its turn transferred into the mechanical energyofrotation of The turbine shaft.

Based on action of steam the steam turbines may be classifiedas

(i) Impulse turbine
(ii) Reaction turbine
(iii) Impulse and reaction turbine

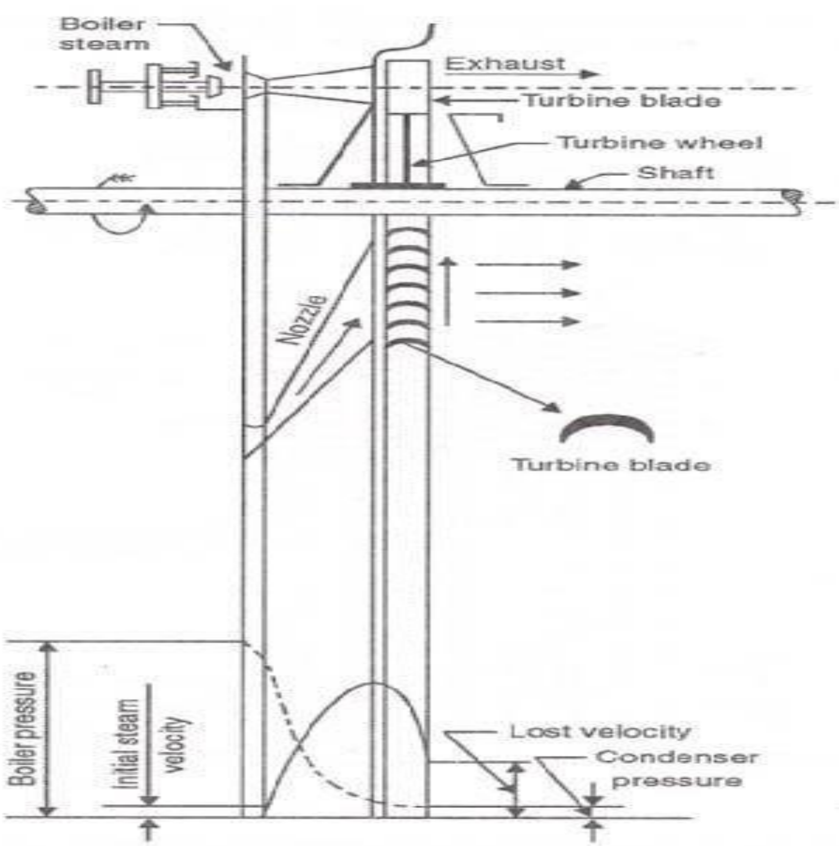
According to the direction of steam flow

(i) Axial flow turbine
(ii) Radial flow turbine

According to the number of stages

(i) Single stage turbine
(ii) Multi stage turbine

Simple Impulse turbine



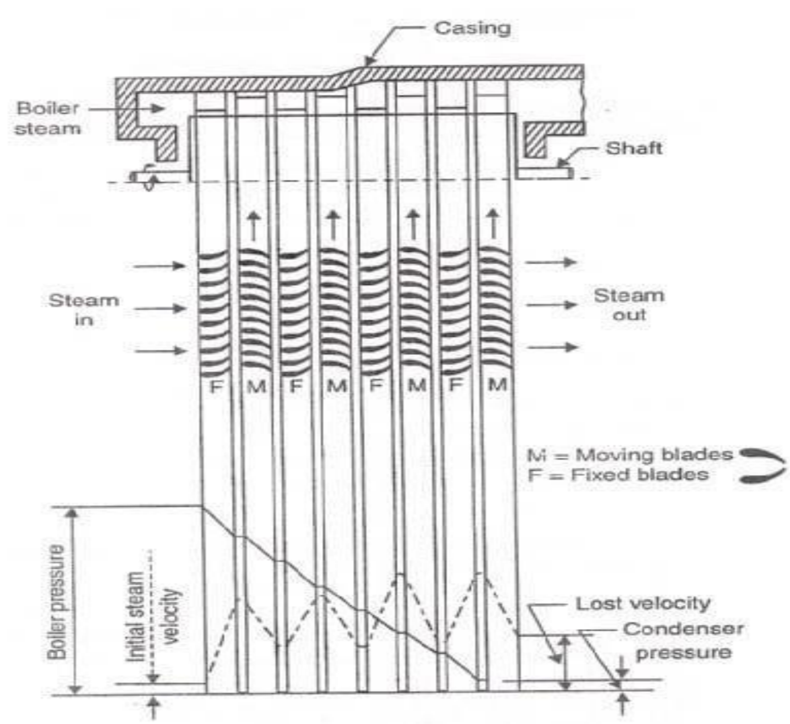
An impulse turbine runs by the impulse of steam jet. In this turbine the steam isfirst made to flow through a nozzle. Then the steam jet impinges on the turbine blades. The steam jet after Impinging on the rotor blades glides over the concave surface of the blades and finally Leaves the turbine.

A De-Laval turbine is the simple impulse turbine and is commonly used with fixed nozzles and a rotor with a ring of blades inside a casing. The surface of the blades are

Generallyverysmoothtominimizethefrictionallosses.Thebladesaregenerallymadeof Special steel alloys. Steam supplied to an impulse turbine expands completely in the nozzle. As the steam flows through the nozzle its pressure falls from steam chest pressure to Condenser pressure. Due to this relatively higher ratio of expansion of steam in the nozzlesthe Steam leaves the nozzle with a very high velocity. It can be observed that the velocity of the Steam leaving the moving blades is comparatively higher .The loss of energy due to this Higher exit velocity is called “Carry over loss” or “leaving loss”.This loss may amount to 3 to

% of the nozzle velocity. The moving blades of impulse turbine are ‘constant flow area profile type blades'. Therefore The pressure remains constant during the flow of steam through the moving bladesof impulse Turbine.

Reaction turbine



Inthistypeofturbine,thereisagradualpressuredropandtakesplacecontinuously over the fixed and moving blades. The function of the fixed blades is that they alter the direction of the steam as well as allow it expand to a larger velocity. As the steam passes over the moving blades its kinetic energy is absorbed by them. Instead of a set of nozzles, steam is admitted for whole of the circumference and therefore there is all round admission. In passing through the first row of fixed blades, the steam undergoes a small drop in pressure and its velocity is increased. It then enters the first row of moving blades and it suffers a change in direction and therefore momentum. This gives impulse to the blades. But the moving blades are of aerofoil type and hence there is also a pressure drop in the moving blades.

The reaction turbines which are used these days are really impulse-reaction turbines. Pure reaction turbines are not in general use. The expansion of steam and heatdropoccurbothinfixedandmovingblades.Thevelocityofsteaminthistypeofturbinesiscomparativ elylow,themaximumbeingaboutequaltobladevelocity.This type of turbine very successful lin practice. Itis also called “Parson’s Reaction Turbine”.

Difference between Impulse and Reaction turbines

S. No.	Particulars	Impulse turbine	Reaction turbine
1.	*Pressure drop*	Only in nozzles and not in moving blades.	In fixed blades (nozzles) as well as in moving blades.
2.	*Area of blade channels*	Constant.	Varying (converging type).
3.	*Blades*	Profile type.	Aerofoil type.
4.	*Admission of steam*	Not all round or complete.	All round or complete.
5.	*Nozzles / fixed blades*	Diaphram contains the nozzle.	Fixed blades similar to moving blades attached to the casing serve as nozzles and guide the steam.
6.	*Power*	Not much power can be developed.	Much power can be developed.
7.	*Space*	Requires less space for same power.	Requires more space for same power.
8.	*Efficiency*	Low.	High.
9.	*Suitability*	Suitable for small power requirements.	Suitable for medium and higher power requirements.
10.	*Blade manufacture*	Not difficult.	Difficult.

Methods of reducing rotor speed

Incaseofsimpleimpulseturbine,thesteamisexpandedfromtheboilerpressure to the condenser pressure in one stage only. Hence the speed of the rotorbecomes very high for practical purposes. In order to make the rotor speed practicablecompounding of steam turbine is done. Compounding is the method of reducing rotorspeed by adding stages to a simple impulse turbine without affecting the turbine workoutput.Therotor speedcanbe reduced by thefollowing methods.

(i) Velocity compounding

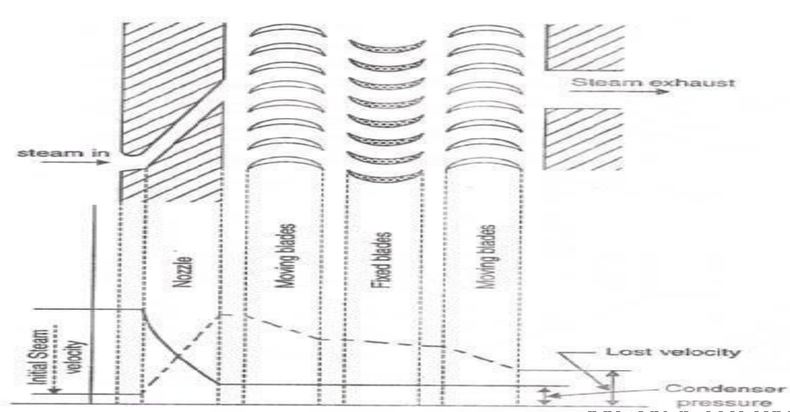
(ii) Pressure compounding

(iii) Pressure-Velocity compounding

(iv) Reaction turbine

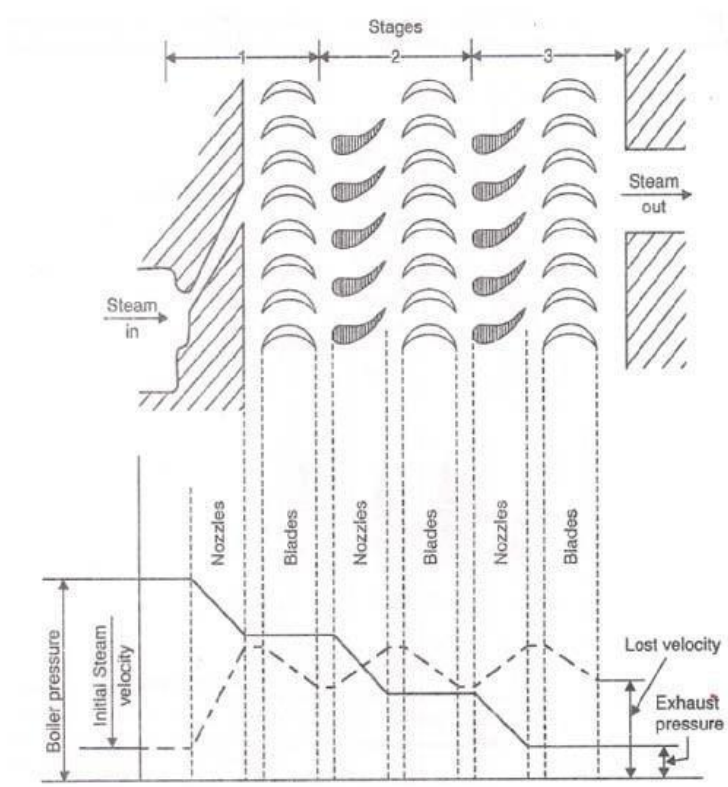
Velocity Compounding

Steamisexpandedthroughastationarynozzlefromtheboilerorinletpressur eto condenser pressure. So, the pressure in the nozzle drops, the kinetic energy of thesteamincreasesduetoincreaseinvelocity.Aportionofthisavailableenergyisabsorbed by a row of moving blades. The steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed. The function of these fixed blades is to redirect the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity. Fig shows a cut away section of such stage and changes in pressure and velocity as the steam passes through the nozzle, fixed and moving blades. Though this method has the advantage that the initial cost is low due to lesser number of stages yet its efficiency is low.



Pressure Compounding

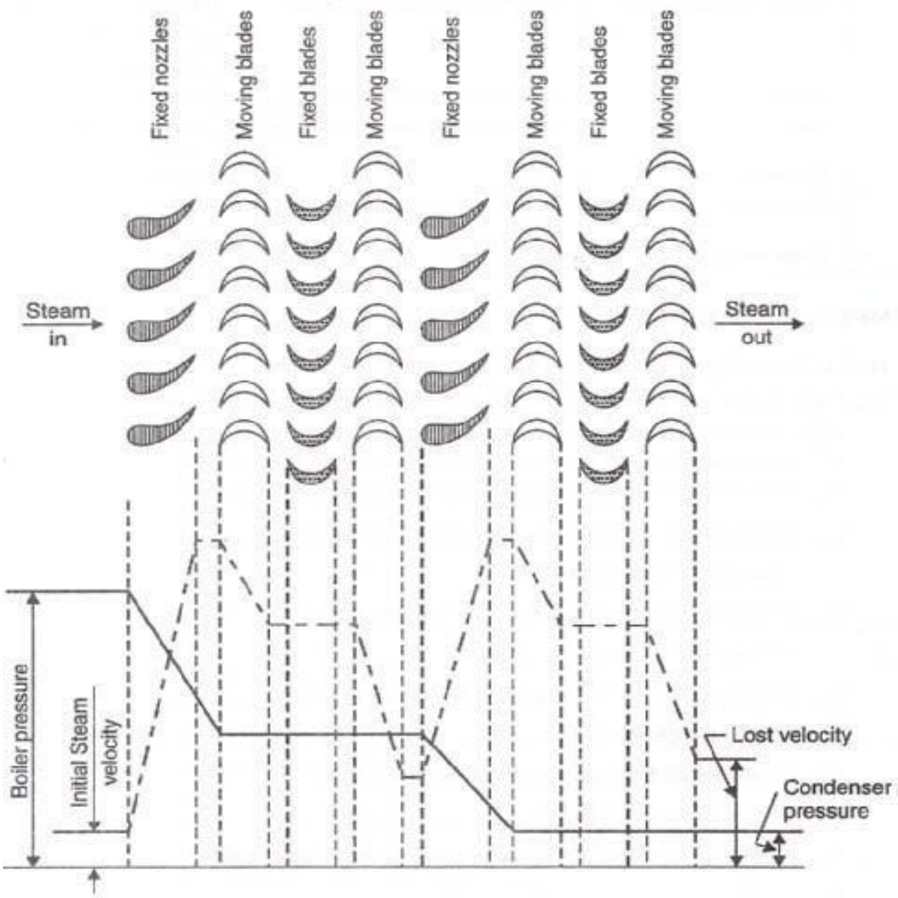
Fig shows rings of fixed blades incorporated between the rings of moving blades. The steam at boiler pressure enters the first set of nozzles and expands partially. The kinetic energy of the steam thus obtained is absorbed by the moving blades. The steam then expands partially in the second set of nozzles where its pressure again falls and thevelocityincreases;thekineticenergysoobtainedisabsorbedbythesecondringofmoving blades (stage-2). This is repeated in stage-3 and steam finally leaves the turbine at low velocity and pressure. The number of stages depends on the number of rows of nozzles through which the steam must pass.
This method of compounding is used in Rateau and Zoelly turbine. This is most efficient turbine since the speed ratio remains constant but it is expensive owing to a large number of stages



Pressure-Velocity Compounding

Thismethodofcompoundingisthecombinationofpressureandvelocitycompounding.

The total drop in steam pressure is divided into stages and the velocity obtained in each stage is also compounded. The rings of nozzles are fixed at the beginning of each stage and pressure remains constant during each stage. The changes in pressure and velocity are



shown. This method of compounding is used in Curtis and Moore turbine.

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.

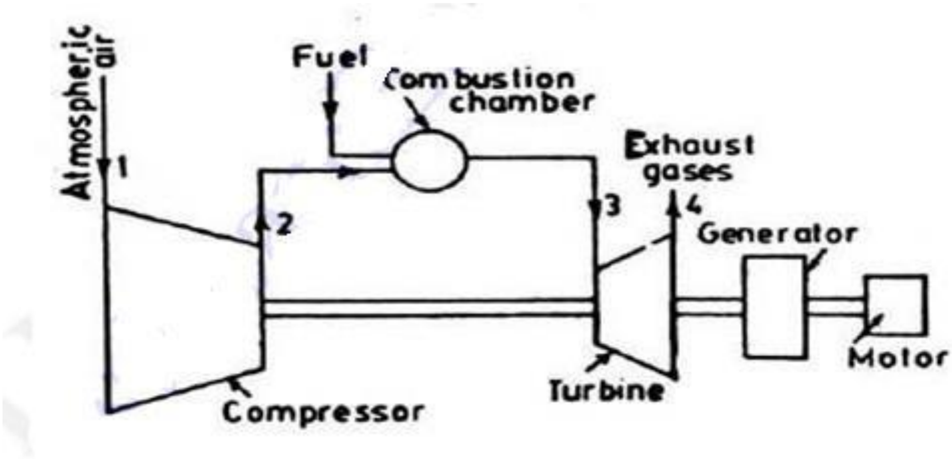
Comparison OF GAS TURBINES AND IC.ENGINES

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.

- Comparison Of Gas Turbines And SteamTurbines

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.

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, isentropic ally.
- 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:

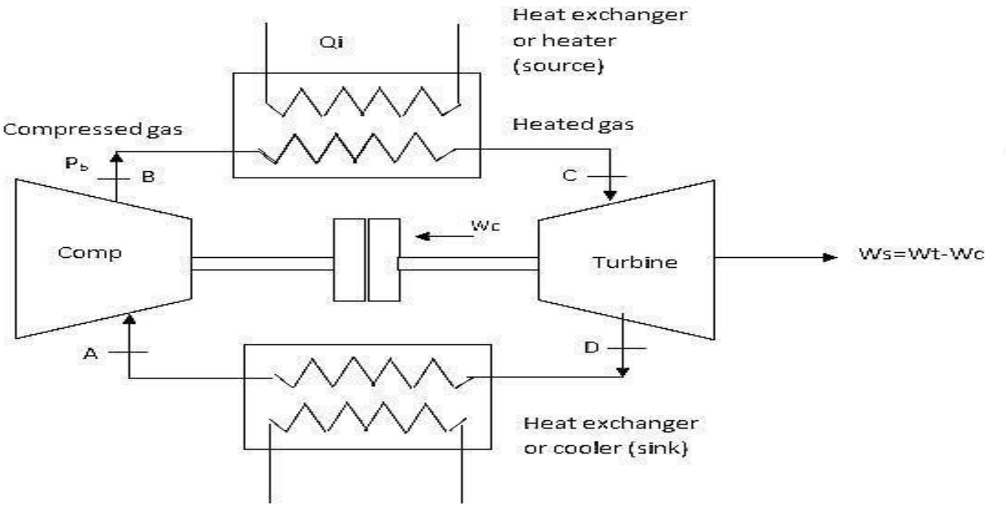
1. **Warm-up time:** Once the turbine is brought up to the rated speed by the starting motor and the fuel is ignited, the gas turbine will be accelerated from cold start to full load without warm-up time.
2. **Low weight and size:** The weight in kg per kW developed is less.
3. Fuels**:** Almost any 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.

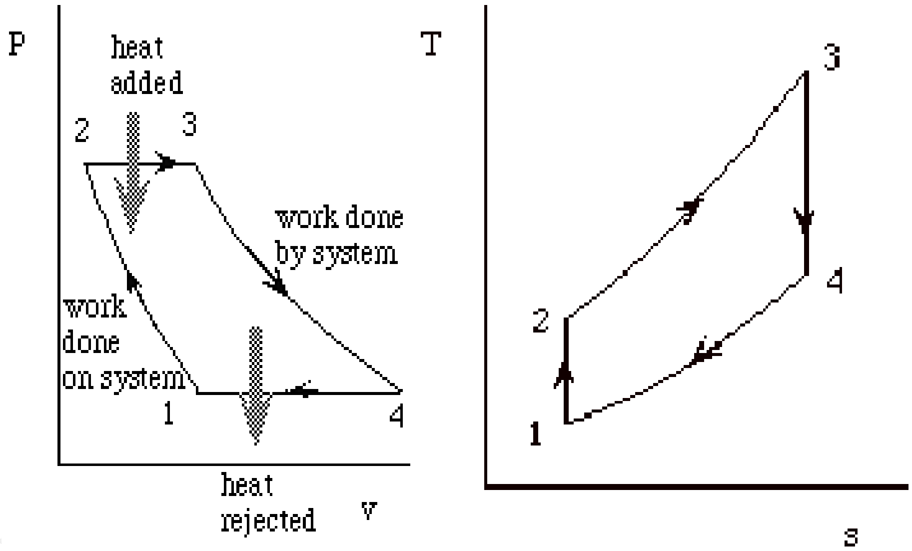
Disadvantages:

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. Sodamages 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.





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.

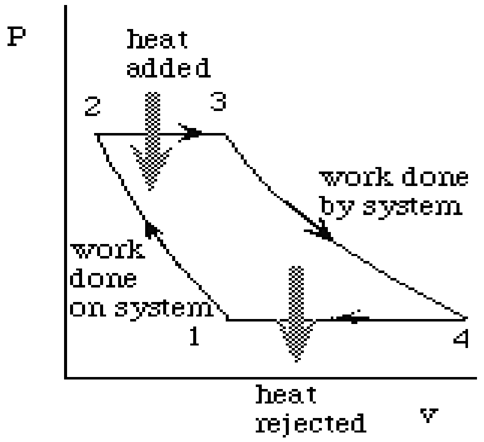
Disadvantages:

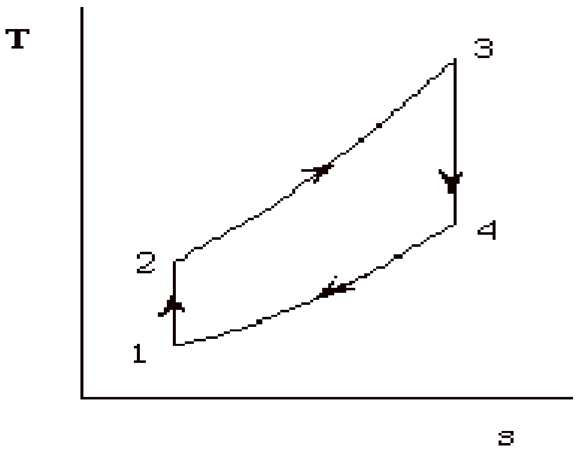
- ❖ Weight of system is high compared to open cycle.
- ❖ Large amount of water is required for cooling in cooler.
- ❖ System should be air tight when working substance other than air is used.
- ❖ If load on system increases then performance of system is poor.

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**

2-3 Process: Constant pressure heat addition

$$Q_S = mc_p(T_3 - T_2) \text{ kJ}$$

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

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

$$\mathbf{W_{net} = Q_s - Q_R \text{ kJ}}$$

Thermal Efficiency

Work done $\quad W_{net} \quad mc_p(T_3 - T_2) - mc_p(T_3 - T_4) \quad T_3 - T_4$

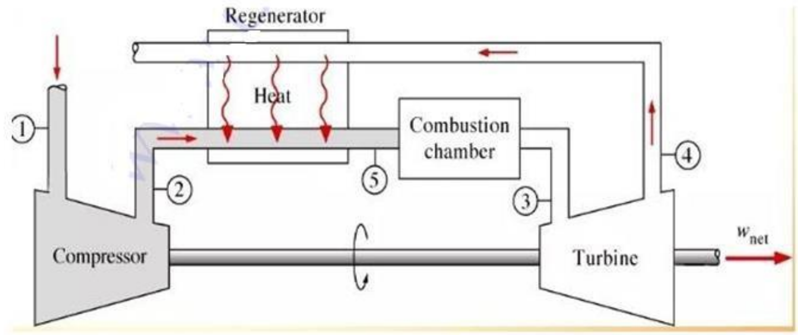
$$\eta = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

Turbine Work Ratio

$$W_R = \frac{W_T}{W_{net}} = 1 - \frac{T_1}{}\left(r^{\gamma-1}\right)$$

REGENERATIVE GAS TURBINE CYCLE

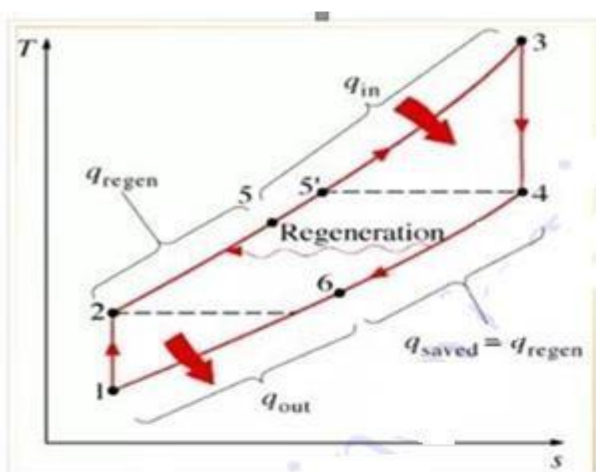
In this method, a regenerator (heat exchanger) is used for utilising heat of exhaust gases from turbine,in pre-heating the compressed air before it enters the combustion chamber. The preheating of the compressed air reduces the fuel consumption and consequently improves the thermal efficiency. Regeneration is shown in fig. As a result of regeneration, compressed air is preheated and exhaust gases are



Temperature of the exhaust gas leaving the turbine is higher than the temperature of the airleaving the compressor.

The air leaving the compressor can be heated by the hot exhaust gases in a counter-flow heatexchanger (a regenerator or recuperator) – a process called regeneration

The thermal efficiency of the Brayton cycle increases due to regeneration since less fuel is usedfor 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 heatexchangers 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 isutilized 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 andtemperature of the air is increased.

Now the air is passed to an intercooler which reduces the temperature of the compressedair to its original temperature, but keeping the pressure constant.

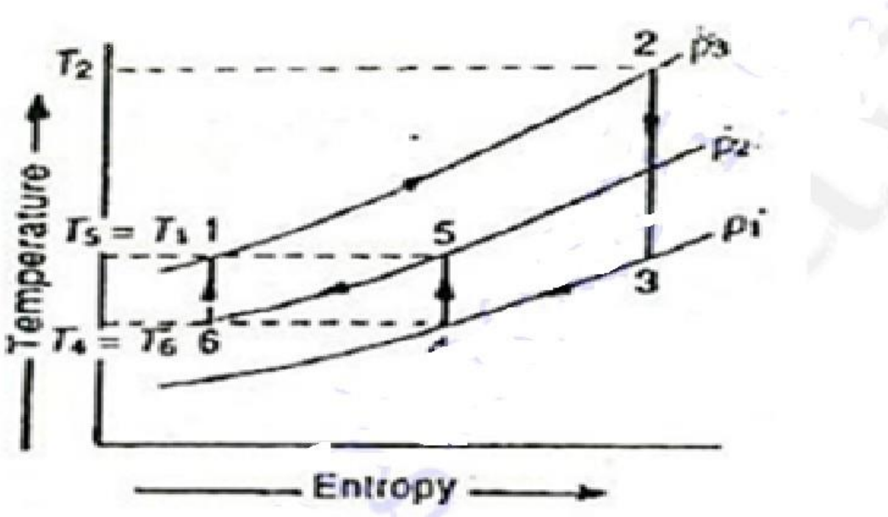
- **The process 1-2 shows heating of the air in heating chamber at constant**
- **pressure.The process 2-3 shows isentropic expansion of air in the turbine.**
- **The process 3-4 shows cooling of the air in the cooling chamber at constant**
- **pressure.The process 4-5 shows compression of air in the 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 turbine is improved by adopting multi-stage compression with intercoolingin between two stages as it reduces the work required to compress the air.

After that, the compressed air is once again compressed in the second compressor knownas 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 pressurecompressor as shown.

The process of intercooling the air in two stages of compression is shown on T-s diagram in Fig.



UNIT IV

INTERNAL COMBUSTION ENGINES – FEATURES AND COMBUSTION

INTRODUCTION

A heat engine is a device which converts the chemical energy of fuel into thermal energy and in turn into mechanical energy. The heat engines are classified into external combustion engines (EC Engines) and internal combustion engines (IC Engines).
In external combustion engine, the combustion of fuel takes place outside the engine cylinderand the heat produced is transferred to a second fluid which the working fluid of the cycle to produce the mechanical energy. Steam engine, steam turbine, Striling engine and closed cycle gas turbine are the examples of external combustion engine.
In internal combustion engine, the combustion of fuel takes place inside the engine and the products of combustion is used to produce mechanical energy. Open cycle gas turbine, Wankel engine, gasoline engine and diesel engine are examples of internal combustion engines.

ENGINE COMPONENTS

A cross section of a single cylinder internal combustion engine with overhead valves is shown in figure 3.1. The major components of the engine and their functions are briefly described below.

(i) Cylinder: As the name implies it is a cylindrical vessel or space in which the piston makes a reciprocating motion. The varying volume created in the cylinder during the operations of the engine is filled with the working fluid and subjected to different thermodynamic processes. The cylinder is supported in the cylinder block.

(ii) Piston: It is a cylindrical components fitted into the cylinder forming the moving boundary of the combustion system. It fits perfectly into the cylinder providing a gas tight space with the piston rings and the lubricant. It forms the first link in transmitting the gas forces to the output shaft

(iii) Combustion chamber: The space enclosed in the upper part of the cylinder, by the cylinder ahead and the piston top during the combustion process is called the combustion chamber. The combustion of fuel and the consequent release of thermal energy results in the building up pressure in the part of the cylinder.

(iv) Inlet manifold: The pipe which connects the intake system to the inlet valve of the engine and through which air or air-fuel mixture is drawn into the cylinder is called the inlet manifold.

(v) Exhaust manifold: The pipe which connects the exhaust system to the exhaust valve of the engine and through which the products of the combustion escape into the atmosphere is called the exhaust manifold.

(vi) Inlet and exhaust valves: Valves are commonly mushroom shaped puppet type. They are provided either on the cylinder head or on the side of the cylinder for regulating the charge coming into the cylinder (inlet valve) and for discharging the products of combustion (exhaust valve) from the cylinder.

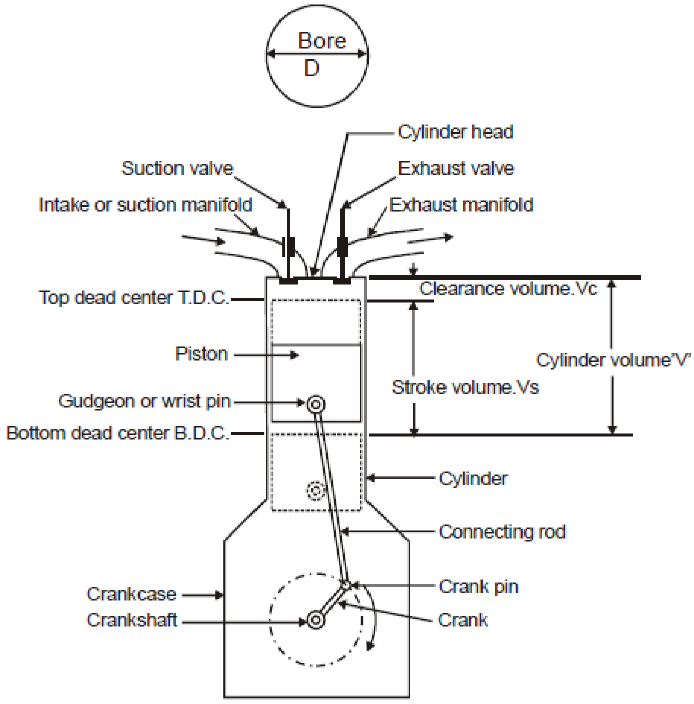


Figure 3.1 Cross section of an Internal Combustion Engine

(vii) Spark plug: It is a component initiates the combustion process in spark ignition engines and is usually located on the cylinder head.

(viii) Fuel injector: It is the component mounted on the cylinder head of compression ignition engine to inject the fuel at the end of compression stroke

(ix) Connecting rod: It interconnects the piston and the crank shaft and transmits the gas forces from the piston to the crankshaft.

(x) Crankshaft: It converts the reciprocating motion of the piston into useful rotary motion of the output shaft. In the crankshaft of a single cylinder engine there is a pair of crank arms and balance weights. The balance weights are provided for static and dynamic balancing of the rotating system. The crankshaft is enclosed in a crankcase.

(xi) Piston rings: Piston rings, fitted into the slots around the piston , provides a tight seal between the piston and the cylinder wall thus preventing leakage of combustion gases.

(xii) Gudgeon pin: It forms the link between the small end of the connecting rod and the piston.

(xiii) Cam shaft: The camshaft and its associated parts control the opening and closing of the two valves. The associated parts are push rods, rocker arms, valve springs and tappets. This shaft also provides the drive to the ignition system. The camshaft is driven by the crank shaft through timing gears.

(xiv) Cams: These are made as integral parts of the camshaft and are designed in such a way to open the valves at the correct timing and to keep them open for the necessary duration.

(xv) Flywheel: The net torque imparted to the crank shaft during one completed cycle of operation of the engine fluctuates causing a change in the angular velocity of the shaft. In order to achieve a uniform torque an inertia mass in the form a wheel is attached to the output shaft and this wheel is called the flywheel. The variation of the net torque decreases with

increase in the number of cylinders in the engine and thereby the size of the flywheel also becomes smaller. This means that a single cylinder engine will have a larger flywheel whereas a multi cylinder engine will have a smaller flywheel.

The list of engine parts and material used are given in table 3.1

Table: The list of engine parts and material used	
Name of the part	**Material used**
Cylinder	Cast iron, alloy steel
Cylinder head	Cast iron, alluminium alloy
Piston	Cast iron, alluminium alloy
Piston rings	Silicon cast iron
Wrist pin	Steel
Valves	Alloy steel
Connecting rod	Steel
Crankshaft	Alloy steel, SG iron
Crankcase	Alluminium alloy, steel, cast iron
Cylinder liner	Cast iron, nickel alloy steel
Bearing	White metal, leaded bronze

ENGINE TERMINOLOGIES

The various terminologies used in an internal combustion engines are:

(i) Cylinder bore: The nominal inner diameter of the working cylinder is called the cylinder bore and is designated by the letter (d) and is usually expressed in millimeter.

(ii) Piston area: At the area of the circle of diameter equal to the cylinder bore is called the piston area and is designated by the letter (A) and usually expressed in square centimeter.

(iii) Stroke: The nominal distance through which a working piston moves between two successive reversals of its direction of motion is called the stroke and is designated by the letter L and expressed in millimeter

(iv) Dead center: The position of the working piston and the moving parts which are mechanically connected to it, at the moment when the direction of the piston motion is reversed at either end of the stroke is called the dead center. There are two dead centers in an engine, top dead center (TDC) and bottom dead center (BDC). The TDC is the dead center when the piston is farthest from the crankshaft. The BDC is the dead center when the piston is nearest to the crankshaft.

(v) Displacement or swept volume (V_s): The nominal volume swept by the piston when traveling from one dead center to the other is called the displacement volume. It is expressed in terms of the cubic centimeter

$V_s = (\pi/4)\ d^2L$

(vi) Clearance volume: The nominal volume of the combustion chamber above the piston when it is at the top dead center is the clearance volume. It is designated as V_c and expressed in cubic centimeter.

(vii) Compression ratio: It is the ratio of the total cylinder volume when the piston is at the bottom dead center to the clearance volume. It is designated by the letter r.

$r = (V_s + V_c)/V_c$

CLASSIFICATION OF INTERNAL COMBUSTON ENGINES

Internal Combustions engines are classified under different headings, some of theclassifications are as follows.

(i) According to number of strokes

- **Four stroke cycle engine and**
- **Two stroke engine**

(ii) According to cycle of operation

- **Otto cycle engine and**
- **Diesel cycle engine**

(iii) According to type of fuel used

- **Engine using volatile liquid fuels like gasoline, alcohol, kerosene**
- **Engine using gaseous fuel like natural gas, petroleum gas, blast furnace gas**
- **Engine using solid fuel like char coal, powered coal**
- **Engine using viscous fuels like diesel oil**
- **Engine using dual fuel**

(iv) According to Method of charging

- **Naturally aspirated engine and**
- **Supercharged engine**

(v) According to type of ignition

- **Spark ignition engine and**
- **Compression ignition engine**

(vi) According to cooling method

- **Air cooled engine and**
- **Water cooled engine**

(vii) According to governing method

- **Hit and miss governed engine**
- **Quality governed engine**
- **Quantity governed engine**

(viii) According to speed of the engine

- **Low speed engine**
- **Medium speed engine**
- **High speed engine**

(xi) According to their use

- **Stationary engine**
- **Portable engine**
- **Marine engine**
- **Automobile engine**
- **Aero engine**

(x) According to number of cylinders

- **Single cylinder engine**
- **Multi cylinder engine**

(xi) According to Cylinder arrangement

- **Inline engine**
- **V- Engine**
- **Opposed cylinder engine**
- **Opposed piston engine**
- **Radial engine etc.,**

WORKING PRINCIPLE OF ENGINES

For an engine is to work successfully then it has to follow a cycle of operation in a sequential manner. This sequence is quite rigid and cannot be changed. In the following sections the working principles of both SI and CI engines is described. Even though both engines have much in common there are certain fundamental differences. The credit of inventing the spark ignition engine goes to Nicolas Otto (1876) whereas compression ignitionengine was invented by Rudolf Diesel (1892). Therefore, they are often referred to as Otto engine and Diesel engine.

Working of Four Stroke Spark Ignition Engine

In a four stroke engine, the cycle of operations is completed in four strokes of the piston or two revolutions of the crankshaft. During the four strokes, there are five events to completed, viz., suction, compression, combustion, expansion and exhaust. Each stroke consists of 180 degrees of crank rotation and hence a four stroke cycle is completed through 720 degrees of crank rotation.

The cycle of operation for an ideal four stroke SI engine consists of (i) Suction stroke (ii) Compression stroke (iii) Expansion or power stroke and (iv) Exhaust stroke as shown in figure.

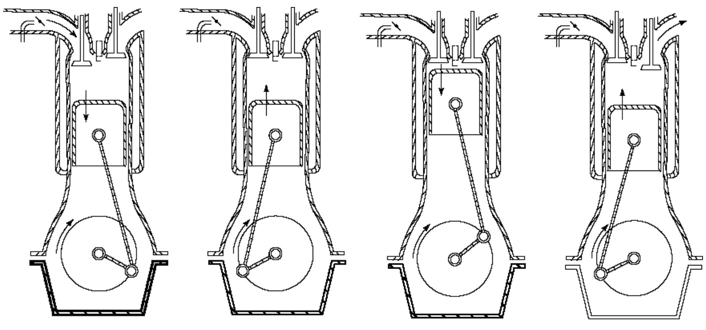


Figure Four Stroke Cycle SI Engine

(i) Suction or intake stroke: Suction stroke starts when the piston is at the top dead center and about to move downwards. The inlet valve is open at this time and the exhaust valve is closed. Due to the suction created by the motion of the piston towards the bottom dead center, the charge consisting of fuel air mixture is drawn into the cylinder. When the piston reaches the bottom dead center, the suction stroke ends and the inlet valve closes.

(ii) Compression stroke: The charge taken into the cylinder during the suction stroke is compressed by the return stroke of the piston. During this stroke both inlet and exhaust valves are in the closed position and the piston moves from BDC to TDC. The mixture which fills the entire cylinder volume is now compressed into the clearance volume. At the end of the compression stroke the mixture is ignited with the help of a spark produced by a spark plug located on the cylinder head. Burning takes place almost instantaneously when the piston is the top dead center and hence the burning process can be approximated as heat addition at constant volume. During the burning process the chemical energy of the fuel is

converted into heat energy producing a temperature rise of about 2000 °C. This pressure at the end of the combustion process is considerably increased due to the heat release.

(iii) Expansion stroke: The high pressure of the burnt gases forces the piston towards the BDC with both inlet and exhaust valves remaining closed. Thus power is obtained during this stroke. Both pressure and temperature decreases during expansion.

(iv) Exhaust stroke: At the end of the expansion stroke the exhaust valve opens and the inlet valve remains closed. The piston moves from BDC to TDC. The pressure falls to atmospheric level and the burnt gases are sent out of the cylinder. The exhaust valve is closed at the end of the stroke some residual gases are left in the clearance space of the engine cylinder. These residual gases mix with the fresh charge coming in during the following cycle, forming its working fluid. Each cylinder of a four stroke engine completes the above four operations in two engine revolutions, one revolution of the crankshaft occurs during the suction and compression strokes and the second revolution during the power and exhaust strokes. Thus, for one complete cycle there is only one power stroke while the crankshaft turns by two revolutions.

Working of Four Stroke Cycle Compression Ignition Engine

The four stroke CI engine is similar to the four stroke SI engine but it operated at a much higher compression ratio. The compression ratio of a SI engine varies from 6 to 10 while for a CI engine it is from 16 to 20. In the CI engine during suction stroke air is inducted due to the high compression ratio employed the temperature at the end of the compression stroke is sufficiently high to self -ignite the fuel which is injected into the combustion chamber.

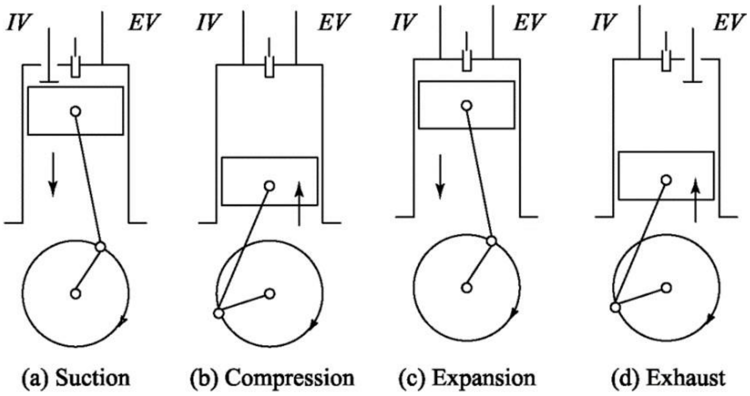


Figure Four Stroke Cycle CI Engine

Figure shows the four stroke cycle CI engine. The ideal sequence of operations for the four stroke CI engine is as follows

(i) Suction stroke: During this stroke intake valve is open and exhaust valve is closed. The piston moves from TDC to BDC. Air alone is inducted during the suction stroke.

(ii) Compression stroke: Both valves remain closed and piston moves from BDC to TDC during this stroke. Air inducted during the suction stroke is compressed into the clearance volume. The pressure and temperature at the end of compression stroke is very high.

(iii)Expansion stroke: Fuel injection stars nearly at the end of the compression stroke and combustion of fuel takes place. The products of combustion expand with both valves remain closed during the expansion stroke.

(iv)Exhaust stroke: The piston is travelling from BDC to TDC pushes out the products of combustion. The exhaust valve opens and the intake valve is closed during this stroke.

Comparison of SI and CI Engines

The detailed comparison of SI and CI engines is given in table

Table Comparison of SI and CI engines		
Description	**SI engine**	**CI engine**
Basic cycle	Otto cycle or constant volume heat addition cycle	Diesel cycle or constant pressure heat addition cycle
Fuel	Gasoline, a high volatile fuel. Self-ignition temperature is high	Diesel oil, a non-volatile fuel. Self-ignition temperature is comparatively low
Ignition	Requires an ignition system with spark plug in the combustion chamber. Primary voltage is provided by a battery or magneto	Self-ignition occurs due to high temperature of air because of the high compression. Ignition system and spark plug are not necessary
Load control	Throttle controls the quantity of mixture introduced	The quantity of fuel is regulated in the pump. Air quantity is not controlled
Speed	Due to light weight and also due to homogeneous combustion they are high speed engines	Due to heavy weight and also due to heterogeneous combustion, they are low speed engines
Compression ratio	6 to 10 upper limit is fixed by antiknock quality of the fuel	16 to 20 upper limit is limited by weight increase of the engine
Weight	Lighter due to lower peak pressures	Heavier due to higher peak pressures
Thermal efficiency	Low because of lower compression ratio	High because of the higher compression ratio
Introduction of fuel	A gaseous mixture of fuel and air is introduced during the suction stroke. A carburetor is necessary to provide the mixture	Fuel is injected directly into the combustion chamber at high pressure at the end of the compression stroke. A fuel pump and injector are necessary

Working of Two Stroke Cycle Engine

In two stroke cycle engine, the cycle is completed in one revolution of the crank shaft.The main difference between two stroke and four stroke engines is in the method of fillingthe fresh charge and removing the burnt gases from the cylinder. In the four stroke engine these operations are performed by the engine piston during the suction and exhaust strokes respectively. In a two stroke engine, the filling process is accomplished by the charge compressed in crankcase or by a blower. The induction of the compressed charge moves out the product of combustion through exhaust ports. Therefore, no piston strokes are required for these two operations. Two strokes are sufficient to complete the cycle, one for compressing the fresh charge and the other for expansion or power stroke.

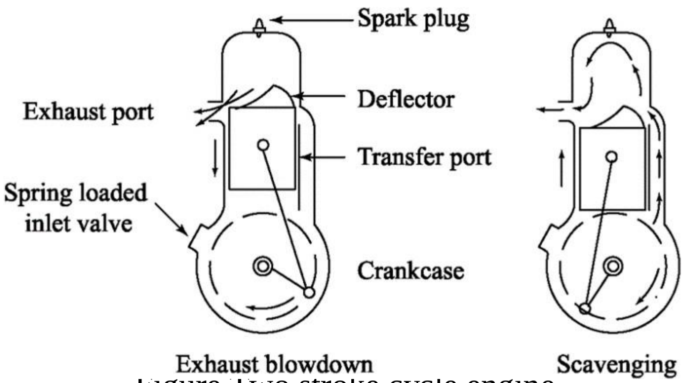


Figure Two stroke cycle engine

Figure shows the two stroke cycle engine. When the piston is in the TDC, compressed charge available at the top of the piston is ignited and combustion of the fuel takes place, the intake port is kept opened and the fresh charge is inducted into the crankcase. The fresh charge is compressed in the crank case when the piston forced towards the BDC by the expansion of the gases and power is obtained during this stroke. When the piston is nearing the BDC, first the exhaust port is opened and the pressure inside the cylinder decreased to atmospheric pressure and the burnt gases are leaving to the atmosphere. Then, the transfer port is opened and the fresh charge is sent to the cylinder. When the piston is at BDC the cylinder is completely filled with fresh charge. As the piston moves towards the TDC, the fresh charge in the engine cylinder is compressed and at the end of compression the ignitionis initiated and combustion of the fuel takes place and the cycle is repeated.

Comparison of Four Stroke and Two Stroke Cycle Engines

The comparison of four stroke and two stroke cycle engines are given in table

Table Comparison of four stroke and two stroke engines

Four stroke engine	**Two stroke engine**
The thermodynamic cycle is completed in four strokes of the piston or in two revolutions of the crank shaft. Thus, one power stroke is obtained in every two revolutions of the crank shaft	The thermodynamic cycle is completed in two stokes or one revolution of the crank shaft. Thus, one power stroke is obtained in every revolution of the crank shaft
Because of the above turning moment is not uniform and hence heavier flywheel is required	Turning is more uniform and hence lighter flywheel can be used
Because of one power stroke in two revolutions of the crank shaft lesser cooling and lubrication requirement. Lower rate of wear and tear	Because of one power stroke in one revolution greater cooling and lubrication requirements. Higher rate of wear and tear
Power produced by same size of the engine is less	Power produced by the same size of the engine is more
The four stroke engine contains valves and valve actuating mechanism to open and close the valves	Two stroke engines have no valves but only ports.

Initial cost of the engine is more	Initial cost of the engine is less
Volumetric efficiency is more due to more time of induction	Volumetric efficiency is low due to lesser time for induction
Thermal efficiency is higher and part load efficiency is better	Thermal efficiency is lower and part load efficiency is poor
Used where efficiency is important e.g., cars, buses, tractors, power generation	Used where low cost, compactness and light weight are important, e.g., mopeds, scooters , motor cycles , hand sprayers.

THEORITICAL AND ACTUAL P-V DIAGRAMS

Four stroke Otto cycle engine

The theoretical P-V diagram of a four stroke Otto cycle engine is shown in figure and the processes are given in table.

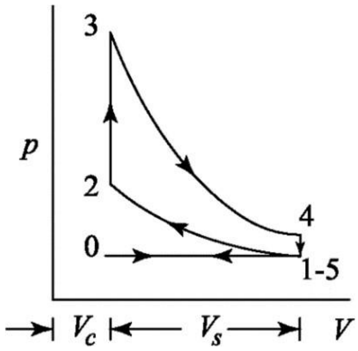


Figure Theoretical P-V diagram of four stroke Otto cycle engine

Table				
	Inlet valve	**Exhaust valve**	**Piston movement**	**Process**
Suction stroke (5 – 1)	Opened	Closed	TDC to BDC	The air-fuel mixture is drawn into the cylinder at constant pressure.
Compression stroke (1 – 2)	Closed	Closed	BDC to TDC	The air fuel mixture is compressed isentropically. At the end of compression process an electric spark is produced which initiate the combustion process (2 – 3). The combustion takes place at constant volume.
Expansion stroke (3 – 4)	Closed	Closed	TDC to BDC	The hot gases produced by combustion pushes the piston towards BDC and work is obtained in this stroke.
Exhaust stroke (1 – 5)	Closed	Opened	BDC to TDC	When the exhaust valve open the pressure inside the cylinder drops as shown in 4–1. Then the waste gases are exhausted to the atmosphere at constant

				pressure.

The actual P-V diagram of four stroke Otto cycle engine is shown in figure. During the suction process the pressure inside the cylinder is lower than atmospheric pressure to enable the fresh charge enter into the cylinder effectively and during exhaust process the pressure inside the cylinder is slightly higher than the atmospheric pressure for effective exhaust of the gases. The loop 4-5-1 is called negative loop and 12-3-4 is called as positive loop and the net work of cycle is obtained by subtracting the area 4-5-1 from 1-2-3-4

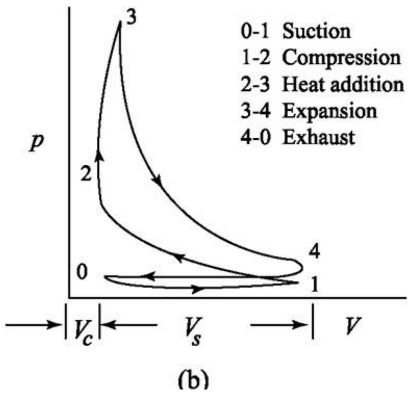


Figure Actual P-V diagram of four stroke Otto cycle engine

Four stroke diesel cycle engine

The theoretical P-V diagram of a four stroke Diesel cycle engine is shown in figure and the processes are given in table

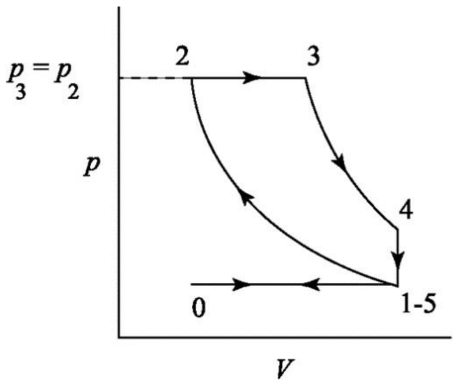


Figure Theoretical P-V diagram of four stroke Diesel cycle engine

Table

	Inlet valve	Exhaust valve	Piston movement	Process
Suction stroke (5–1)	Opened	Closed	TDC to BDC	The air is drawn into the cylinder at constant pressure.
Compression stroke (1–2)	Closed	Closed	BDC to TDC	The air is compressed isentropically. At the end of compression process, fuel is injected and it is ignited using the high temperature at the end of compression stroke. The combustion takes place at constant pressure (2-3)

Expansion stroke (3–4)	Closed	Closed	TDC to BDC	The hot gases produced by combustion pushes the piston towards BDC and work is obtained in this stroke.
Exhaust stroke (1–5)	Closed	Opened	BDC to TDC	When the exhaust valve open the pressure inside the cylinder drops as shown in 4–1. Then the waste gases are exhausted to the atmosphere at constant pressure.

The actual P-V diagram of four stroke Diesel cycle engine is shown in figure. The actual P-V diagram of four stroke Diesel cycle engine is shown in figure 3.8. During the suction process the pressure inside the cylinder is lower than atmospheric pressure to enable the fresh charge enter into the cylinder effectively and during exhaust process the pressure inside the cylinder is slightly higher than the atmospheric pressure for effective exhaust of the gases. The loop 4-5-1 is called negative loop and 12-3-4 is called as positive loop and the net work of cycle is obtained by subtracting the area 4-5-1 from 1-2-3-4.

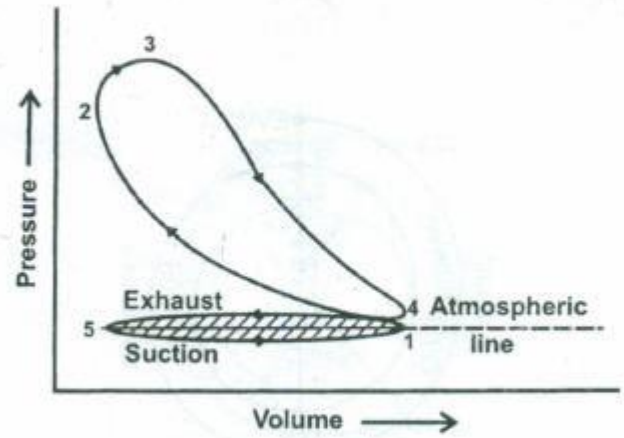


Figure Actual P-V diagram of four stroke Diesel cycle engine

Two stroke cycle engine

The P-V diagram of two stroke cycle engine is shown in figure for the main cylinder or top of the piston. The processes compression and expansion are shown and suction and crank case compression are not shown in the diagram.

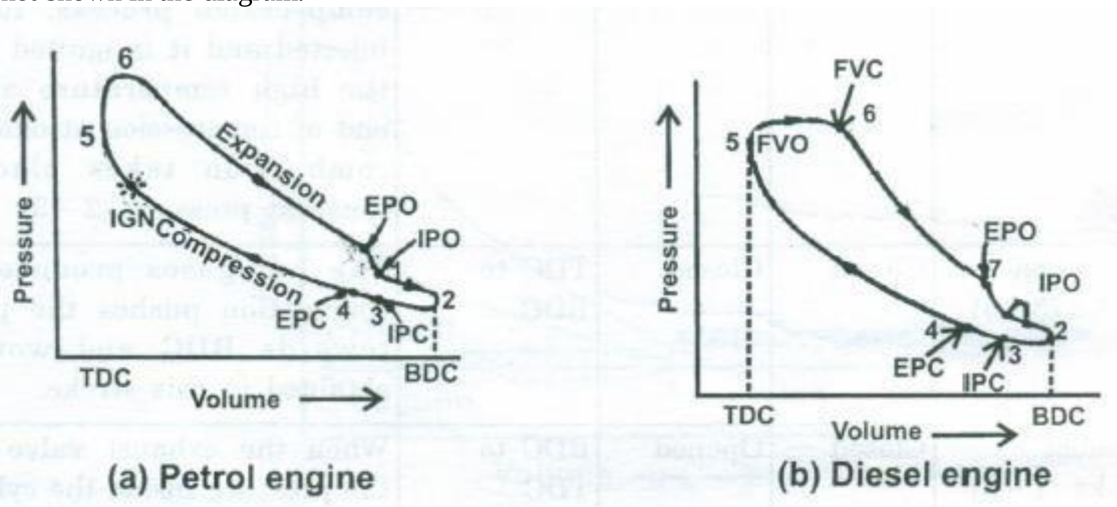
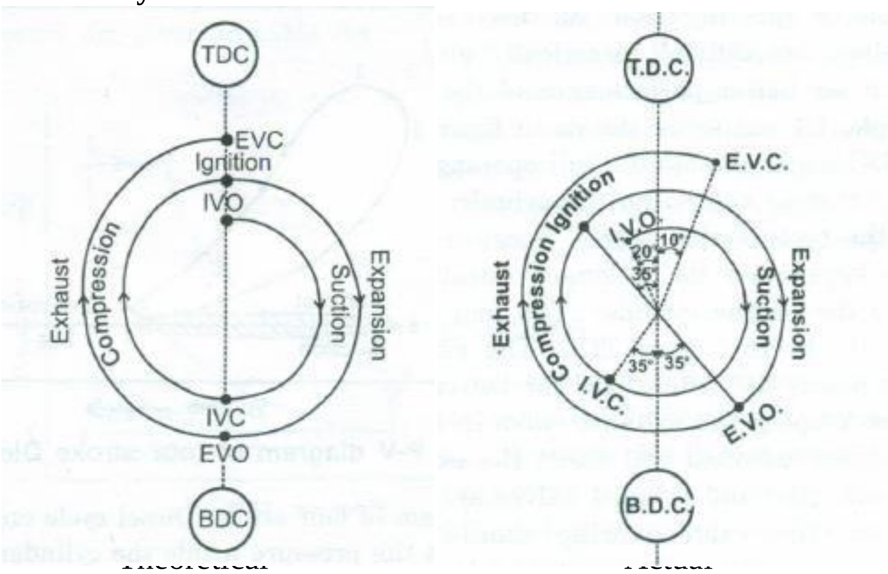


Figure P-V diagram of two stroke cycle engine

VALVE AND PORT TIMING DIAGRAMS

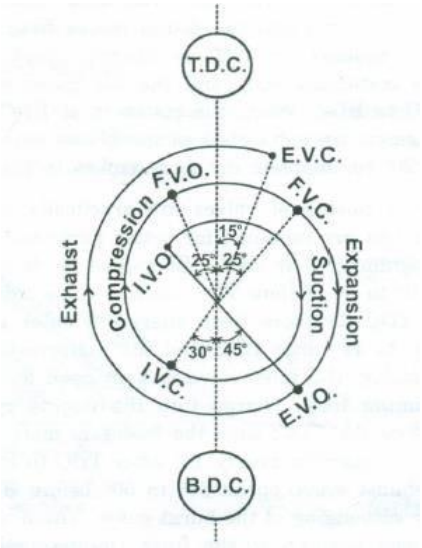
Valve timing diagram of four stroke cycle SI engine

Figure (theoretical)below shows the theoretical valve timing diagram of four stroke cycle spark ignition engine. The inlet valve opens (IVO) when the piston is at TDC and suction process occurs as piston moves towards BDC. The inlet valve closes (IVC) at BDC. The compression process occurs when the piston moves from BDC to TDC. At the end of the compression process, at TDC an electric spark is produced and the combustion takes place instantaneously and the hot gases are expanded and the piston moves from TDC to BDC. When the piston is at BDC the exhaust valve is opened and the waste gases are exhausted as the piston moves from BDC to TDC. The crank rotates by 720° to complete the four strokes in the cycle.



Instant opening and closing of valves are practically difficult and the valve opening and closing timings are modified for better performance of the engine. The actual valve timing diagram of four stroke SI engine is shown in figure (actual). The inlet valve is opened 10°to 30° before TDC which enable the full opening of valve when the piston is at TDC for more fresh charge to enter into the cylinder. The inlet valve is closed 30° to 40° after BDC and the compression starts. Because the change in inlet valve timing, the inlet valve is kept open for a longer duration for the admission of maximum fresh charge into the engine cylinder. The spark is produced 30° to 40° before the TDC; thus the fuel gets more time to burn and the cylinder pressure will be maximum nearly 10° after TDC to extract maximum work from the gases. The exhaust valve opens 30° to 50° before BDC and closed nearly 10° after TDC for better scavenging of the burnt gases. The next cycle begins keeping the exhaust valve in open position so the fresh charge coming in via the intake manifold will assist the exhaust of the burnt gases. For certain period of time both inlet and exhaust valves are kept open and the period is known as valve overlap. The valve overlap should be optimum for better performance of the engine.

Valve timing diagram of four stroke cycle CI engine

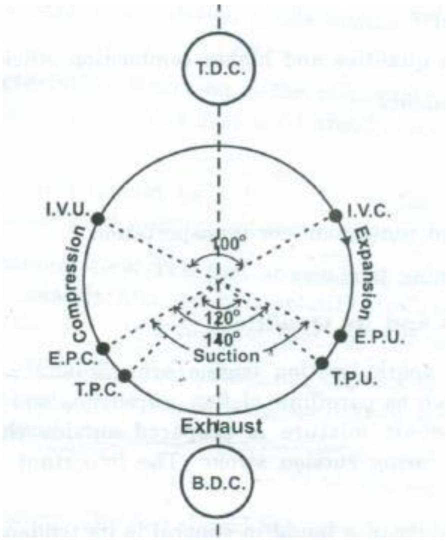


Actual valve timing diagram of CI engine

The theoretical valve timing diagram of four stroke CI engine is similar to figure petrol engine but the ignition is replaced by fuel injection. As discussed for SI engine, instant opening and closing of valves are difficult practically and the valve opening and closing timings are modified for better performance of the engine. The actual valve timing diagram of four stroke CI engine is shown in figure above. The inlet valve is opened 10º to 25º beforeTDC which enable the full opening of valve when the piston is at TDC for more fresh charge to enter into the cylinder. The inlet valve is closed 25º to 40º after BDC and the compression starts. Because the change in inlet valve timing, the inlet valve is kept open for a longer duration for the admission of maximum fresh charge into the engine cylinder. The fuel injection begins 5º to 10º before TDC and continues 10º to 15º after TDC. The exhaust valve opens 30º to 50º before BDC and closed nearly 10º after TDC for better scavenging of the burnt gases. The next cycle begins keeping the exhaust valve in open position so the fresh charge coming in via the intake manifold will assist the exhaust of the burnt gases. For certain period of time both inlet and exhaust valves are kept open and the period is known as valve overlap. The valve overlap should be optimum for better performance of the engine.

Port timing diagram of two stroke cycle engine

The port timing diagram of two stroke cycle engine is shown in figure below. The inlet port opens before TDC and closes after TDC and the total duration of inlet port opening is approximately 100º. The exhaust port opens before BDC and closes after BDC and the total duration of exhaust port opening is approximately 140º. The transfer port opens before BDC and closes after BDC and the total duration of opening is approximately 120º. In a two stroke cycle engine all the four operations viz., suction, compression, expansion and exhaust are taking place in one revolution of crankshaft i.e. 360º of crank rotation. The port timing diagram of petrol and diesel engines are same excepting the fuel injection in diesel engine and spark production in petrol engine. In petrol engine the spark is produced just before the TDC and in diesel engine the fuel injection valve open just prior to TDC and closes just after TDC.



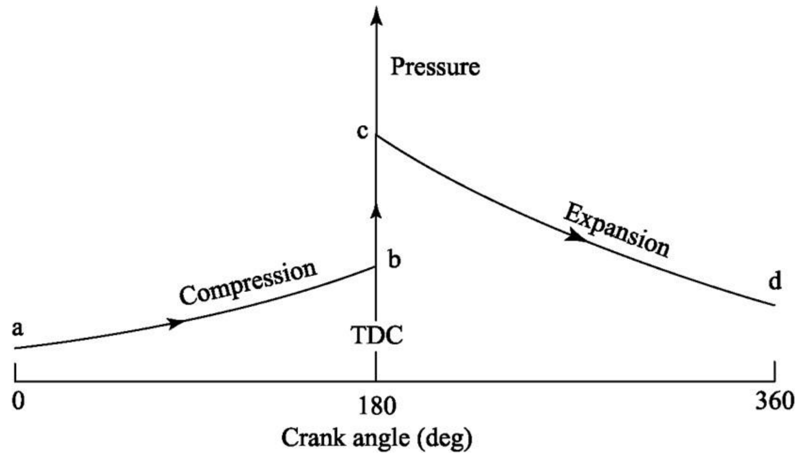
Port timing diagram of two stroke cycle engine

Learning Outcome Assessment Questions

1. What is meant by valve overlapping period? (AU-Nov 2014)
2. Draw the port timing diagram with fuel injection of a two stroke diesel engine and explain the salient points (AU-Nov 2009, May 2010)
3. Draw and explain the port timing diagram of two stoke cycle diesel engine (AU-May 2013)
4. Draw and describe the four stroke SI engine valve diagram (AU-May 2009)

COMBUSTION IN SI ENGINE

Combustion is a chemical reaction in which the carbon and hydrogen elements of the fuelcombine with oxygen in the air liberating heat energy and causing an increase in temperatureof the gases. The process of combustion in engines generally takes place either in ahomogeneous or a heterogeneous fuel vapour air mixture depending of the type of the engine.In a conventional SI engine the fuel and air are homogeneously mixed together in the intake system and inducted through the intake valve into the cylinder where it mixes withresidual gases and is then compressed. Under normal operation conditions a spark isproduced at the end of the compression process which initiates the combustion. Once the fuelvapour air mixture is ignited at a point, a flame front appears and rapidly spreads in themixture. The flame propagation is caused by heat transfer and diffusion of burning fuelmolecules from the combustion zone to the adjacent layer of fresh mixture. The flame front isa narrow zone separating the fresh mixture from the combustion products. The velocity withwhich the flame front moves with respect to the unburned mixture in a direction normal to itssurface is called the normal flame velocity. In a homogeneous mixture the equivalence ratiois approximately 1.1 and the normal flame velocity is 40 cm/sec. Higher flame velocity is obtained when the mixture is slightly rich.



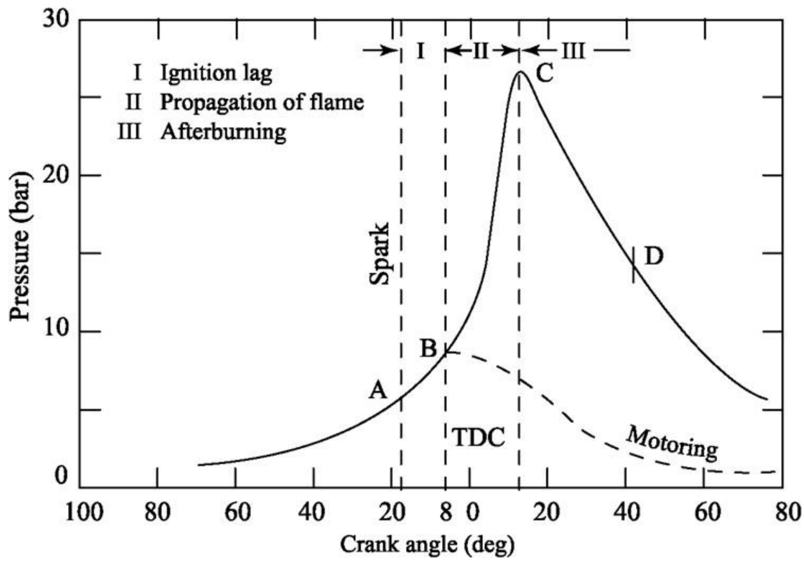
Stages of combustion in SI engine

The combustion in a SI engine is divided into three stagesStage I: Ignition lag period (A – B) Stage II: Flame propagation period (B – C)Stage III: After burning stage(C - D)

I stage: It is the ignition lag or preparation phase period during which growth and development of a self-propagating nucleus of flame takes place. This is a chemical process period depends on both temperature and pressure, nature of the fuel and proportion of exhaustresidual gas.

II stage: It is the flame propagation period during which the flame spreads throughout the combustion chamber. The flame propagates at a constant velocity. The heat release rate depends on the intensity of turbulence and the reaction rate which depends of the mixture composition.

III stage: In the after burning stage, the flame velocity decreases. The rate of combustion becomes low due to lower flame velocity and reduced flame front surface. Since the expansion stroke starts before this stage of combustion with the piston moving away from the TDC there can be no pressure rise during this stage.



Flame front propagation:

For efficient combustion the rate of propagation of the flame front within the cylinderis quite critical. The two important factors which determine the rate of movement of the

flame front across the combustion chamber are the reaction rate and the transposition rate. The reaction rate is the result of a purely chemical combination process in which the flame eats its way into the unburned charge. The transposition rate is due to the physical movement of the flame front relative to the cylinder wall and is also the result of the pressure differential set up between the burning and unburnt gases in the combustion chamber.

Factors influencing the flame speed:

The various factors influencing the flame speed are

Turbulence: Turbulence is swirls produced in the combustion chamber which improves the proper mixing of air and fuel. In a combustion chamber, many minute swirls are preferred than larger swirls. The turbulence in the combustion chamber increases the heat flow to the cylinder wall, accelerate the chemical reaction and the combustion duration is minimized which reduces the tendency of abnormal combustion.

Fuel air ratio: The fuel air ratio has a very significant influence on the flame speed. The highest flame velocities are obtained with somewhat richer mixture. When the mixture is made leaner or richer with respect to the figure the flame speed decreases. Less thermal energy is released in the case of lean mixtures resulting in lower flame temperature. Very rich mixtures lead to incomplete combustion which result again in the release of less thermal energy.

Temperature and pressure: Flame speed increases with increase in intake pressure and temperature. This is possible because of an overall increase in density of charge.

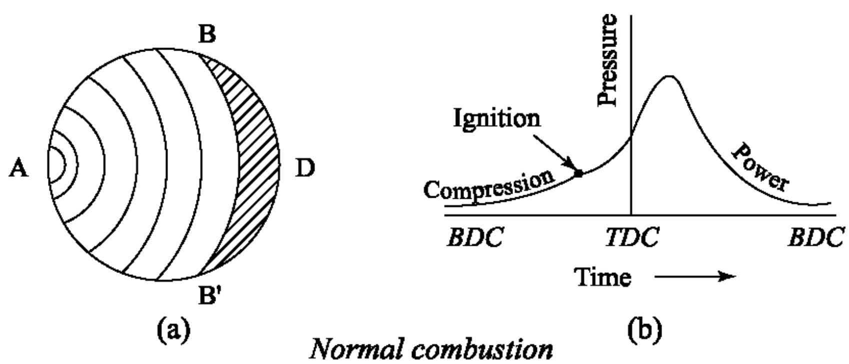
Compression ratio: Higher the compression ratio the flame speed will be high.

Engine output: The cycle pressure increases when the engine output is increased resulting in increased flame speed.

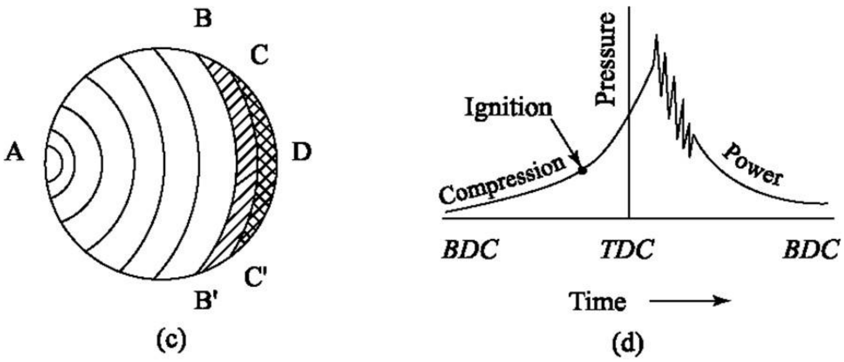
Engine speed: Flame speed increases almost linearly with engine speed since increase in engine speed increased the turbulence inside the cylinder

Normal and Abnormal combustion:

In a spark ignition engine combustion which is initiated between the spark plug electrodes spreads across the combustible mixture. A definite flame front which separates thefresh mixture from the products of combustion travels from the spark plug to the other end of the combustion chamber. Heat release due to combustion increases the temperature and consequently the pressure of the burned part of the mixture above those of the unburned mixture. In order to effect pressure equalization the burned part of the mixture will expand and compress the unburned mixture adiabatically thereby increasing its pressure and temperature. This process continues as the flame front advances through the mixture and the temperature and pressure of the unburned mixture are increased further.



Normal combustion



Combustion with detonation

If the temperature of the unburnt mixture exceeds the self-ignition temperature of the fuel and remains at or above this temperature during the period of preflame reactions, spontaneous ignition or auto ignition occurs at various pin point locations. This phenomenon is called knocking. The process of auto ignition leads towards engine knock.

The normal and abnormal combustion are shown in the figures. In the normal combustion the flame is initiated near the spark plug and it propagates at constant velocity towards other end of the combustion chamber. In the abnormal combustion, because of the auto ignition another flame front starts traveling in the opposite direction to the main flame front. When the two flame front collides a severe pressure pulse is generated. The gas in the chamber is alternatively compressed and expanded by the pressure pulse until pressure equilibrium is restored. This disturbance can force the walls of the combustion chambers to vibrate at the same frequency as the gas. The abnormal combustion causes loss of power and mechanical damage to the engine.

Effect of engine variables on knock:

(a) Density factor: Any factor which reduces the temperature of unburned charge should reduce the possibility of knocking.

Any factor which reduces the density of unburned charge should reduce the possibility ofknocking.

- **Increasing the compression ratio will increase the tendency of knocking**
- **Reduction in mass of charge inducted will reduce the tendency of knocking**
- **Increasing the temperature of mixture at inlet will increase the tendency of knocking.**
- **Hot spots in the combustion chamber wall should be avoided to reduce the tendency of knocking**
- **By retarding the spark timing from the optimized timing the peak pressure are reached farther down on the power stroke and are thus of lower magnitude. This might reduce the knocking.**
- **A decrease in output of the engine will reduce the tendency of knocking.**

(b)Time factor: Increasing the flame speed or reducing the time of exposure of the unburned mixture to auto ignition conditionwill tend to reduce knocking.

- **Increasing the flame speed increases flame speed and reduces the tendency of knocking**
- **An increase in engine speed increases the turbulence of the mixture**
- **The knocking tendency is reduced by shortening the time required for the flame front to traverse the combustion chamber. Engine size, combustion chamber size and sparkplug position are the factors governing the flame travel distance.**

- **Larger the engine size has greater tendency for knocking**
- **In order to have a minimum flame travel, the spark plug is centrally located in the combustion chamber resulting in minimum knocking tendency.**

(c)Composition factor: The composition of fuel like fuel air ratio and octane value are also influences the knocking tendency of the engine. The flame speed, flame temperature and reaction time are different for different fuel air ratio.

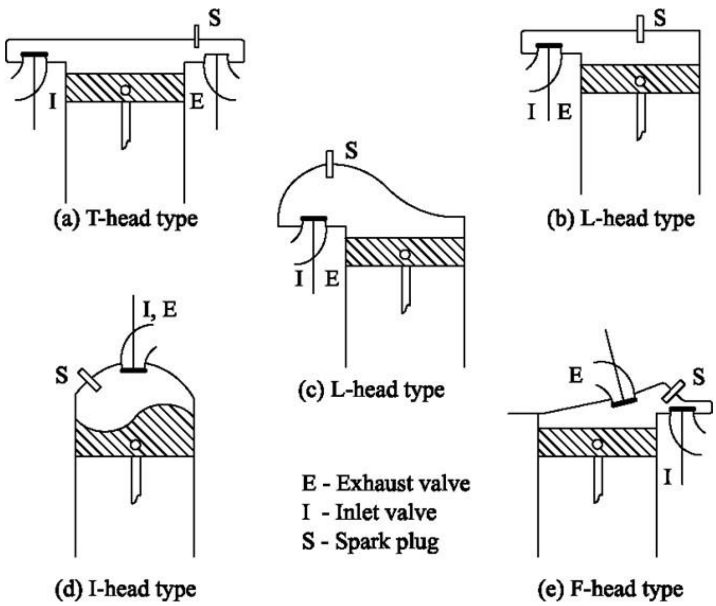
Combustion chamber for SI engine:

The design of combustion chamber for an SI engine has an important influence on the engine performance and its knocking tendencies. The design involves the shape of the combustion chamber, location of spark plug and location of inlet and exhaust valves. The important requirements of an SI engine combustion chamber are to provide higher power output, high thermal efficiency and smooth engine operation. Smooth engine operation is achieved by reducing the possibility of knocking by

- ⊕ **Reducing the distance of flame travel by centrally locating the spark plug.**
- ⊕ **Satisfactory cooling of spark plug points and of exhaust valve area**
- ⊕ **Reducing the temperature of the last portion of charge by transferring heat to the cylinder wall**

High power output and thermal efficiency is achieved by

- ⊕ **High degree of turbulence in the combustion chamber**
- ⊕ **High volumetric efficiency (i.e) inducing more charge during intake stroke results in high output**
- ⊕ **Compact combustion chamber reduces the heat loss during combustion and increases the thermal efficiency.**



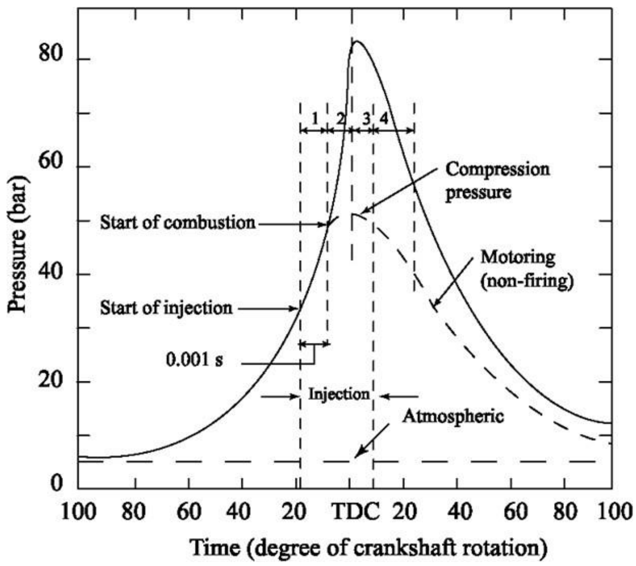
The different types of combustion chamber used in SI engine are

- ⊕ **T head type**
- ⊕ **L head type**
- ⊕ **I head type**
- ⊕ **F head type**

COMBUSTION IN CI ENGINE

The combustion in CI engine is considered to be taking place in four phases. It is divided into the ignition delay period, the period of rapid combustion, the period of controlled combustion and the period of after burning. The details are explained below.

1. **Ignition delay period**: The ignition period is also called the preparatory phase which is counted from the start of injection to the start of combustion. The combustion delay period is very important because it has effect on the combustion rate, knocking, engine starting ability and presence of smoke in the exhaust. The ignition delay period is divided into two parts, the physical delay and chemical delay. The physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions. During this period the fuel is atomized, vaporized, mixed with air and raised to its self-ignition temperature. This physical delay depends on the type of fuel. The physical delay is greatly reduced by using high injection pressures and high turbulence to facilitate breakup of the jet and improving evaporation. The chemical delay is the period for the chemical reaction till the ignition takes place. Generally the chemical delay is larger than the physical delay.



2. **Period of rapid combustion:** The period of rapid combustion also called the uncontrolled combustion during which the pressure raise is rapid. The rate of heat release is the maximum during this period.
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 is already quit high. Hence the fuel droplets injected during the second stage burn faster with reduced ignition delay as soon as they find the necessary oxygen and any further pressure rise is controlled by the injection rate. The period of controlled combustion is assumed to end at maximum cycle temperature.
4. **Period of after burning**: 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 period is called as after burning period.

Factors affecting the delay period:

(a) Compression ratio: The compression ratio should be increased to decrease the ignition delay period

(b) Engine speed: When the engine speed is increased the delay period decreases during variable speed operation

(c) Output: with an increase in engine output, operating temperature increased and hence delay period decreases.

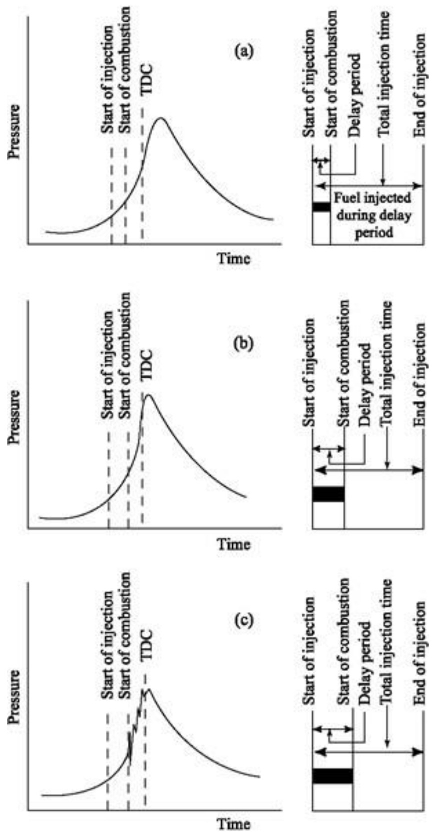
(d) Atomization and duration of injection: Higher the fuel injection pressure increases the degree of atomization. The fines of atomization reduce the ignition lag.

(e) Injection timing: Optimum angle of injection advance generally 20^0bTDC

(f) Quality of fuel: Self ignition temperature is important property which affects the delay period. A lower self-ignition temperature results in a lower delay period. Other properties which affect the delay period are volatility and viscosity.

(g) Intake temperature and pressure: Increase in intake pressure and temperature will reduce the delay period.

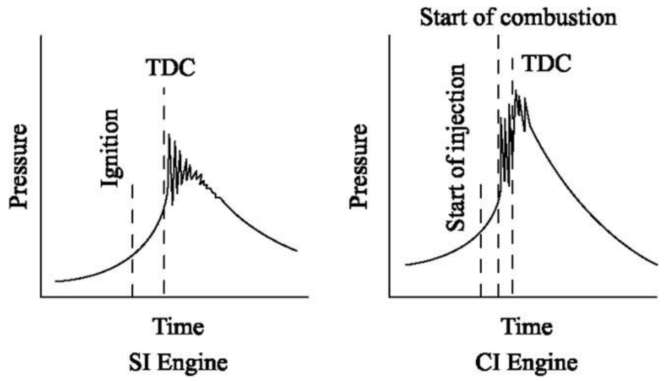
The phenomenon of knock in CI engine:



In CI engine the injection process takes place over a definite interval of time. Consequently, as the first few droplets to be injected are passing through the ignitiondelay period, additional droplets are being injected into the chamber. If the ignition delay of the fuel being injected is short, the first few droplets will commence the actualburning phase in a relatively short time after injection and a relatively small amount for fuel will be accumulated in the chamber when actual burning commences. As a result, the mass rate of mixture burned will be such as to produce a rate of pressure rise that will exert a smooth force on the piston. If on the other hand the ignition delay is longer the actual burning of the first few droplets is delayed and a greater quantity of fuel droplets a gets accumulated in the chamber.

When the actual burning commences, the additional fuel can cause too rapid rate of pressurerise, resulting in a jamming of forces against the piston and rough engineoperation.

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 the extreme pressure differentials andviolent gas vibrations known as knocking and is evidenced by audible knock.



The phenomenon is similar to that in the SI engine. However, in the SI engine knocking occurs near the end of combustion whereas in the CI engine knocking occurs near the beginning of combustion. In order to decrease the tendency of knock it is necessary to start the actual burning as early as possible after the injection begins. In other words it is necessaryto decrease the ignition delay and thus decrease the amount of fuel present when the actual burning of the first few droplets starts.

Combustion chamber for CI engines

The CI engine combustion chambers are classified into two categories

a. Direct injection type: This type of combustion chamber is also called an open combustion chamber. In this type the entire volume of the combustion chamber is located in the main cylinder and the fuel is injected into this volume
b. Indirect injection type: In this type of combustion chamber, the combustion space is divided into two parts, one part in the main cylinder and the other part in the cylinder head. The fuel is injected usually into that part of the chamber located in the cylinder head.

These chambers are classified further into

- Swirl chambers in which compression swirl is generated
- Pre combustion chamber in which combustion swirl is induced
- Air cell chamber in which both compression and combustion swirl are induced

Direct injection chambers:These chambers mainly consist of space formed between a flat cylinder head and a cavity in the piston crown in different shapes. The fuel is injected directly into this space. The injector nozzles used for this type of chamber are generally of multi-hole type working at a high pressure (200 bar)

The main advantages of this type of chambers are

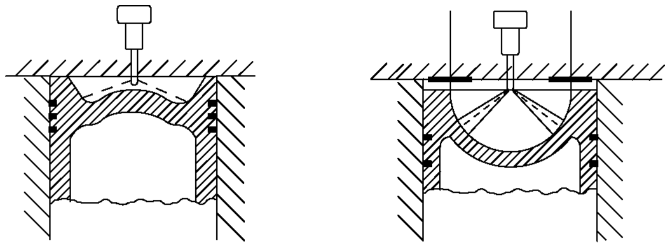
- Minimum heat loss during compression because of lower surface area to volume ration and hence, better efficiency
- No cold starting problem
- Fine atomization because of multi hole nozzle

The drawbacks of this combustion chamber are:

- High fuel injection pressure required and hence complex design of fuel injection pump
- Necessity of accurate metering of fuel by the injection system, particularly for small engine

Shallow depth chamber: In shallow depth chamber the depth of the cavity provded in the piston is quite small. This chamber is usually adopted for large engines running at low speeds. Since the cavity diameter is very large, the squish is negligible

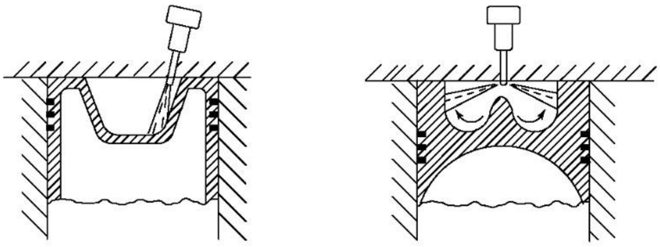
Hemispherical chamber: This chamber also gives small squish. However, the depth to diameter ratio for a cylindrical chamber can be varied to give any desired squish to give better performance



(a) Shallow depth chamber (b) Hemispherical chamber

Cylindrical chamber: This design was attempted in recent diesel engines. This is a modification of the cylindrical chamber in the form of a truncated cone with base angle of 30°The swirl was produced by masking the valve for nearly 80° of circumference. Squish canalso be varied by varying the depth.

Torodial chamber: The idea behind this shape is to provide a powerful squish along with theair movement, similar to that of the familiar smoke ring, within the toroidal chamber. Due to powerful squish the mask needed on inlet valve is small and there is better utilization of oxygen. The cone angle of spray for this type of chamber is 150° to 160°



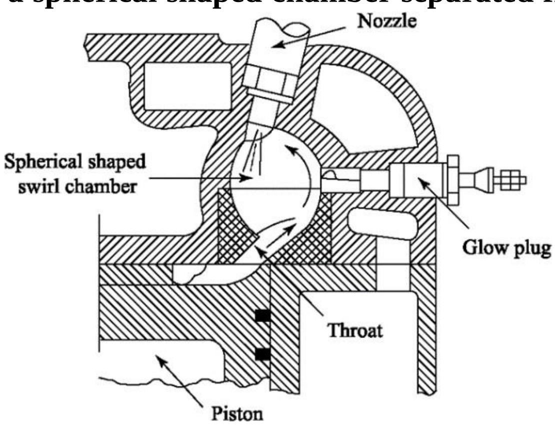
(c) Cylindrical chamber (d) Toroidal chamber

Indirect injection chambers:

The various types of indirect injection chambers are dealt here

Swirl chamber:

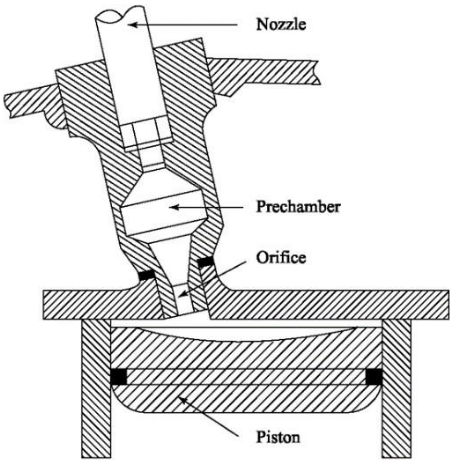
Swirl chambers consist of a spherical shaped chamber separated from the engine cylinder



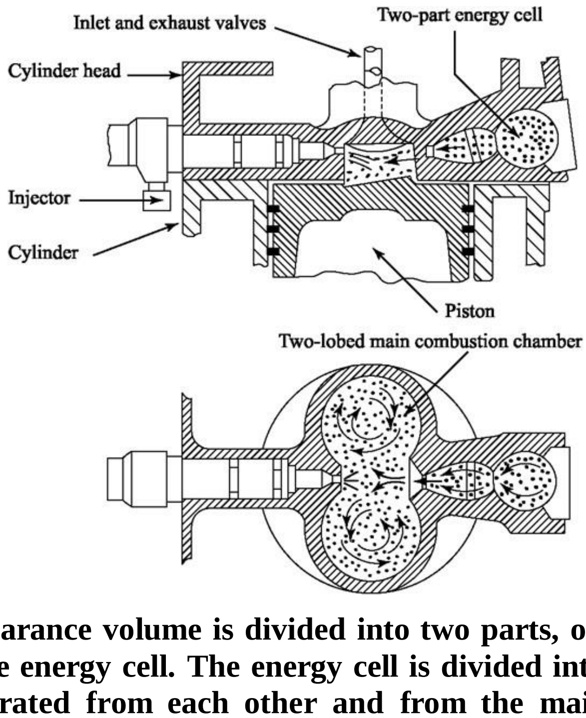
andlocated in the cylinder head. Into this chamber, about 50% of the air is transferred during the compression stroke. A throat connects the chamber to the cylinder which enters the chamber

in a tangential direction so that the air coming into this chamber is given a strong rotary movement inside the swirl chamber and after combustion, the products rush back into the cylinder through the same throat at much higher velocity. This causes considerable heat loss to the walls of the passage which can be reduced by employing a heat insulated chamber. This type of combustion chamber finds application where fuel quality is difficult to control, where reliability under adverse conditions is more important than fuel economy.

Precombustion chamber: A typical Precombustion chamber consists of an antichamber connected to the main chamber through a number of small holes. The Precombustion chamber is located in the cylinder head and its volume accounts for about 40% of the total combustion space. During the compression stroke the piston forces the air into the Precombustion chamber. The fuel is injected into the prechamber and the combustion is initiated. The resulting pressure rise forces the flaming droplets together with some air and their combustion products to rush out into the main cylinder at high velocity through the small holes. Thus it creates both strong secondary turbulence and distributes the flaming fuel droplets though out the air in the main combustion chamber where bulk of combustion takes place. About 80% of energy is released in main combustion chamber.



Air cell chamber



In this chamber the clearance volume is divided into two parts, one in the main cylinder and the other called the energy cell. The energy cell is divided into two parts, major and minor, which are separated from each other and from the main chamber by narrow orifices.During

combustion the pressure in the main chamber is higher than that inside the energy cell due to restricted passage area between the two. At the TDC, the difference in pressure will be high and air will be forced at high velocity through the opening into the energy cell and this moment the fuel injection also begins. Combustion starts initially in the main chamber where the temperature is comparatively higher but the rate of burning is very slow due to absence ofany air motion. In the energy cell, the fuel is well mixed with air and high pressure is developed due to heat release and the hot burning gases blow out through the small passage into the main chamber. This high velocity jet produces swirling motion in the main chamber and thereby thoroughly mixes the fuel with air resulting in complete combustion.

The main advantages of indirect injection combustion chamber are

- Injection pressure required is low
- Direction of spraying is no very important

The drawbacks of this type of chambers are

- Poor cold starting performance requiring heater plugs
- Specific fuel consumption is high

Learning Outcome Assessment Questions

1. Four operations, namely suction, compression, expansion and exhaust, in a four stroke cycle are completed in the number of revolutions of crank shaft equal to
   a. One
   b. Two
   c. Three
   d. Four
2. The four operations, namely suction, compression, expansion and exhaust, in a four stroke cycle are completed in the number of revolutions of cam shaft equal to
   a. One
   b. Two
   c. Three
   d. Four
3. In two stroke cycle engine, the processes suction, compression, expansion and exhaust is completed in the number of revolutions of crank shaft equal to

a. One
b. Two

4. In spark ignition four stroke cycle engine the cam shaft runs
   a. At twice the speed of crank shaft
   b. At the same speed of crank shaft
   c. At half the speed of crank shaft
   d. At any speed irrespective of crank shaft speed
5. In a four stroke cycle diesel engine, during suction stroke
   a. Only air is sucked in
   b. Only fuel is sucked in
   c. Mixture of fuel and air sucked in
   d. None of the above
6. The following is a spark ignition reciprocating engine
   a. Petrol engine
   b. Diesel engine

UNITV

INTERNALCOMBUSTIONENGINEPERFORMANCEANDAUXILIARYSYSTEMSIGNITIONSYSTEM

Pollutantsemissionfromdieselengines

Nowadays, pollution from automobiles becomes the main concern for our lovely environment. Pollutants emitted from diesel engines and petrol engines are dangerous for our environment and our precious health. Both the petrol engines and diesel engines emit pollutants in their workingconditions. But the Concentration of pollutants from diesel engines and petrol engines are different. Here we are going to see the emission of different pollutants from diesel engines

Pollutantsemissionfromdieselengines:

1. Hydrocarbons(HC)
2. CarbonMonoxide(CO)
3. OxidesofNitrogen(NO2)
4. SmokeandParticulate Matter
5. Aldehydes
6. Otheremissions

1. Hydrocarbons:

Hydrocarbonsarethesignificantcontributorsinthedieselengines emission.Blacksmoke fromdiesel engineexhaustcausedbythepresenceofcarbonparticles alsoknownassootinthefluegases.Sootis hazardous for the health of living animals.

2. Carbon Monoxide:

CarbonMonoxideisapresenceinanexhaustofbothdieselenginesandgasoline/petrolengines.But the concentration of Carbon Monoxide in the diesel engines is less as compared to petrol engines. If the petrol/gasoline engine produces 5% of Carbon Monoxide, then diesel engines will produce only 2% of it.

3. Oxides of Nitrogen:

When atmospheric O2 and N2 combine inside the combustion chamber at very high temperature, it produces Oxides of Nitrogen. Diesel engines produce very high Oxides of Nitrogen during the accelerationperiodbecausetemperaturereachesthehighestduetocompletecombustionofthefuelin the chamber.

4. SmokeandParticulate Matter:

Color of exhaust smoke may be white smoke or black smoke. The presence of smoke at the exhaust containsvisibleproductsproducedduetopoorcombustionoffuelandair.Ifwhitesmokeobserved in a diesel engine at cold starting and no load or low load running of the engine, it indicates piston rings of diesel engines are worn out and it needs a replacement. Black smoke in the diesel engine is caused due to incomplete combustion of fuel. The amount of black smoke is directly proportional to the load on the engine.

The ignition system is provided with SI engine, which is used to produce an electric spark at the endof the compression stroke to ignite the air fuel mixture. In the design of an ignition system, the following factors should be taken into account

- Enginedesign
- Enginespeed
- Inletmanifoldpressure
- Mixture composition

Requirementofanignitionsystem:

Asmoothandreliablefunctioningofanignitionsystemisessentialforreliableworkingofan engine.The requirements of such an ignition system are:

⊕Itshould provide a goodsparkbetween theelectrodeoftheplugsatthecorrect timing

⊕It shouldfunctionefficientlyovertheentirerange ofengine speed

⊕It should belight, effectiveand reliablein service

⊕Itshouldbecompactand easyto maintain

⊕Itshould be cheapandconvenientto handle

⊕Theinterferenceofthehighvoltagesourceshouldnotaffectthefunctioningofradioand television receiver inside the automobile

Batteryignitionsystem:

Theessential componentsof thebatteryignition system are

Battery: This provides the energy for ignition.The battery used may be lead acid battery or aalkaline battery.The battery is charged by the dynamo provided in the engine. The battery must be mechanically strong to withstand the strains to which it is constantly subjected to.It converts the chemical energy into electrical energy due to the electrochemical reaction taking place inside the battery

Ignition switch: Battery is connected to the primary winding of the ignition coil through an ignition switch and ballast resistor.With the help of the ignition switch the ignition system can be turned on or off.

Ballast resistor:A ballast resistor is provided in series with primary winding to regulate the primary current.The objective of this is to prevent injuryto the spark coil byoverheating if the engine should by operated for a long time at low speed. The coil is made of iron wire.

Ignition coil: The ignition coil consists of a magnetic core of soft iron sheet and two insulated conduction coils called primary and secondary windings.The secondary coil consists about 21000 turnsof38-40 gaugeenamelled copper wire sufficientlyinsulated towithstandthehighvoltage.It is wound close to the core with one end connected to secondary terminal and other end is grounded to the metal case.The primary winding located outside the secondary coil generally formed 200-300 turns of 20 gauge copper wire.More heat is generated in primary than the secondary and with the primary coil wound over the secondary coil, it is easier to dissipate the heat.The entire unit is enclosed in a metal container and forms a neat and compact unit.

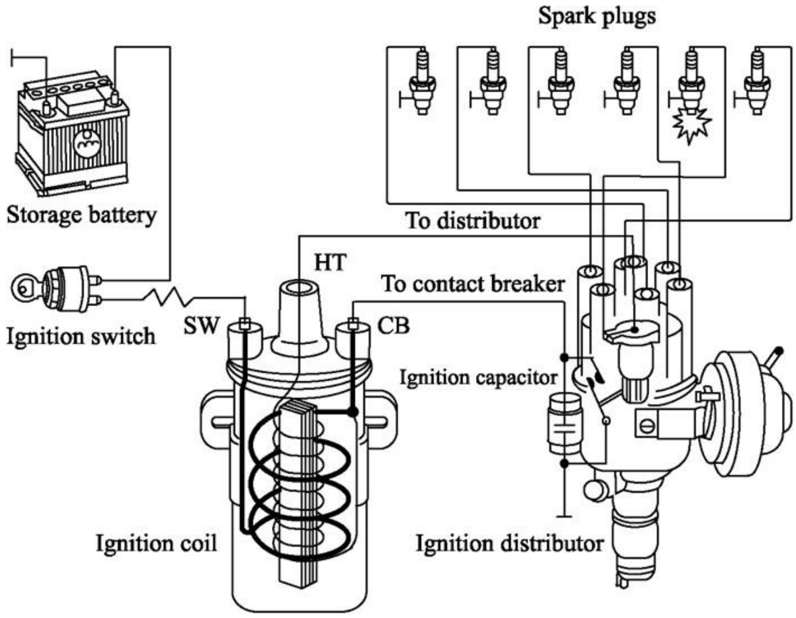


Figure:BatteryIgnitionsystemforasixcylinderengine

Contact breaker: This is a mechanical device for making and breaking the primary circuit of the ignition coil.The metal used for contact surfaces usuallytungsten and diameter of contact surfaces is about 3 mm
Capacitor: The principle of construction of a ignition capacitor is the same as that of electrical capacitor.Two strips ofaluminum foil and several layersofspecial capacitorpaper, are rolled up in a solid roll.Then the roll is inserted into a metal shell for protectionagainst moisture and damage.
Distributor: The function of the distributor is to distribute the ignition current to the individual spark plugs in correct sequence and at the correct instant of time.There two types of distributors. Brush type and gap type

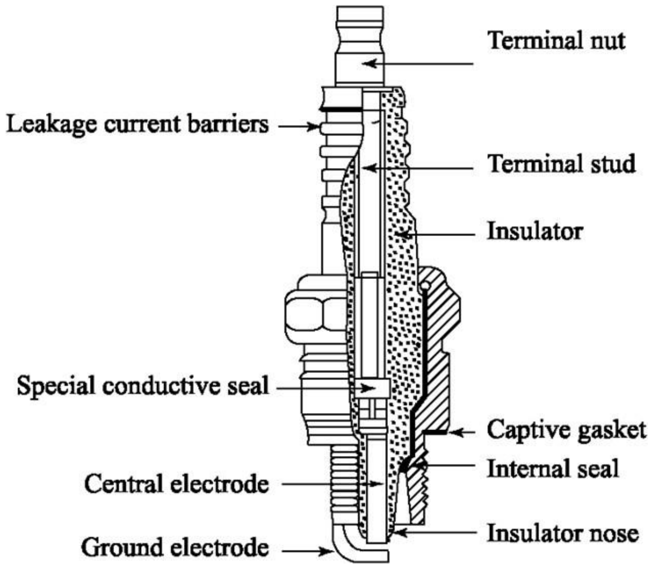


Figure:SparkPlug

Spark plug: The spark plug provides the two electrodes with a proper gap across which the high potential discharges to generate a spark and ignite the combustible mixture with in the combustion chamber.The spark plug consists of a steel shell, an insulator and two electrodes.The central electrode to which the high tension supplyfrom the ignition coil is connected, is well insulated with porcelain or other ceramic material.The other electrode is

welded to steel shell of the plug.The electrodesareusuallymadeofhighnickelalloytowithstandthesevereerosionandcorrosionto

which they are subjected in use.The spark plugs are usually classified as hot plugs or cold plugs depending upon the relative operating temperature range of the tip of the high tension electrode

Operationof abatteryignition system:

When the ignition switch is closed, the primary winding of the coil is connected to the positive terminal post of the storage battery. If the primary is closed through the breaker contacts, a current flows, the so called primary current.This current, flowing through the primary coil, which is wound on a soft iron core, produces a magnetic field in the core.A cam driven by the engine shaft is arranged to open the breaker points whenever an ignition discharge is required. When the breaker points open, the current which had been flowing through the points now flows into the condenser, which is connected across the points.As the condenser becomes charged, the primary current falls and the magnetic field collapses.The collapse of the field induces a voltage in the primary winding, which charges the condenser to a voltage much higher than battery voltage.The condenser then discharges into the battery, reversing the direction of both the primary current and the magnetic field. The rapid collapse and reversal of the magnetic field in the core induce a very high voltage in the secondary winding of the ignition coil.The secondary winding consists of a large number of turns of very fine wire wound on the same core with the primary.The high secondary voltage is led to the proper spark plug by means of a rotating switch called the distributor.This ignition system is used in cars and commercial vehicles.

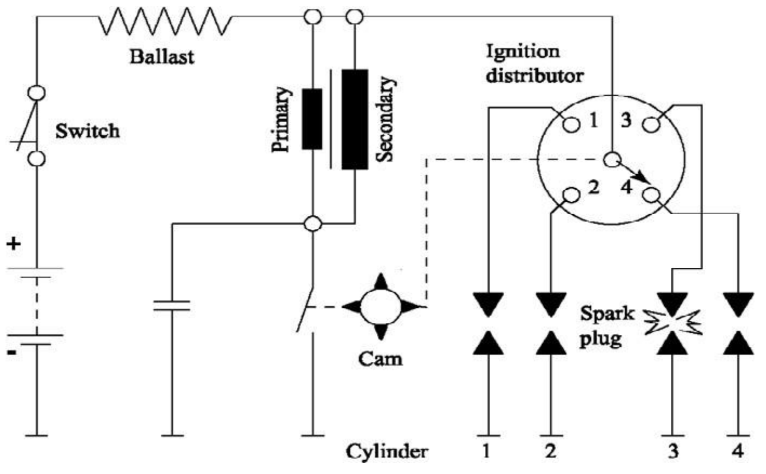


Figure:WorkingofBatteryIgnitionSystem

MagnetoIgnitionSystem

Magneto is a special type of ignition system with its own electric generator to provide the necessary energyforthesystem.It ismountedontheengineand replaces allthecomponentsofthecoilignition system except the spark plug.A magneto when rotated by the engine is capable of producing a very high voltage and does not need a battery as a source of external energy. Magneto may be rotating armature type or rotating magnet type. The working principle of the magneto ignition system is exactlythe same as that of the coil ignition system. With the help of a cam, the primarycircuit flux is changed and a high voltage is produced in the secondary circuit.

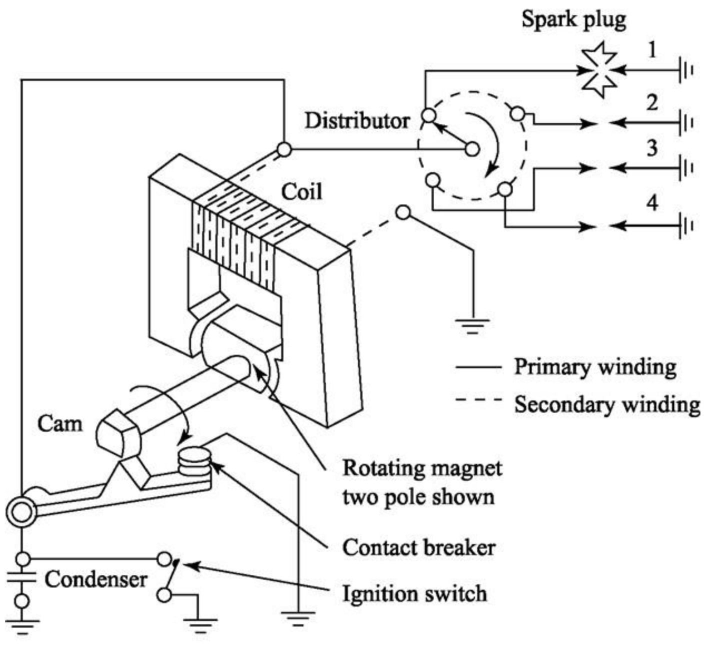


Figure: Magneto Ignition

System

Comparisonofbatteryignitionsystemandmagnetoignitionsyste

m

Batteryignitionsystem	**Magnetoignitionsystem**
Battery is necessary.Difficult to start the engine when battery is discharged	No battery is needed and therefore there is no problem of battery discharge
Maintenanceproblemaremoredueto battery	Maintenanceproblemarelesssincethereis no battery
Current for the primary circuit is obtained from the battery	Therequiredelectriccurrentisgenerated by the magneto
A good spark is available at the spark plug even at low speed	Duringstarting,qualityofsparkispoor due to low speed
Efficiencyof the system decreases with the reductioninsparkintensityasenginespeed rises	Efficiency of the system improves as the engine speed rises due to high intensity spark
Occupiesmorespace	Occupieslessspace
Commonlyemployedincarsandlight commercial vehicles	Mainlyusedinracingcars andtwo wheelers

FUELSUPPLYSYSTEMFOR CIENGINE

The fuel injection system is the most vital component in the working of CI engines.The engine performance is greatly dependent on the effectiveness of the fuel injection system. The injection system has to perform the important duty of initiating and controlling the combustion process.The schematic diagram of the fuel injection system of CI engine is shown in figure.

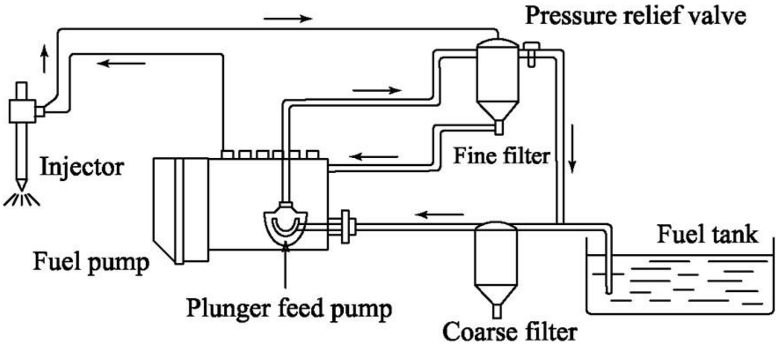


Figure:FuelsupplysystemforCIengine

The fuel from the tank is taken bythe plunger feed pump through the coarse filter and flow to the fine filter then to the fuel pump.The coarse and fine filters remove all kinds of impurities from the fuel. The fuel pump raises the pressure of the fuel to the injection pressure approximately 200 bar and supplies the fuel to the fuel injector which injects the fuel in fine droplets.These droplets vaporize due to the heat transfer from the compressed air and form a fuel-air mixture.The temperature of the fuel reaches a value higher than its self-ignition temperature and the combustion process is initiated spontaneously.A pressure relief valve is also provided for the safety of the system.For proper running and better performance of the engine, the fuel injection system must meet the following requirements.

- **Accuratemeteringoffuelinjectedper cycle**
- **Timingthe injection offuel correctlyin thecycle**
- **Propercontrolofrateofinjectionsothatdesiredheatreleasepatternisachievedduring combustion**
- **Proper atomizationoffuel intoveryfine droplets**
- **Proper spraypattern to ensurerapid mixingof fuel and air**
- **Uniformdistributionoffuelthroughoutthe combustion chamber**
- **To supplyequal quantityof injection in thecaseofmulti cylinderengine**

The fuel injection used in the CI engine is classified into two types: (i) air injection systems and (ii)solid injection systems.They are briefly explained below.

AirInjectionSystem

In this system, the fuel is forced into the cylinder by means of compressed air.This system is little used nowadays, because it requires a bulky multistage air compressor.This causes an increase in engine weight and reduces the brake power output further.One advantage of this type is good mixing of fuel with the air.Another advantage is the ability to utilize fuels of high viscosity.

SolidInjectionSystem

In this system thefuel is directlyinjected into thecombustion chamberwithout theaid ofcompressed air.This system is further classified into four types

(a) Individualpump andnozzle system
(b) Unitinjectorsystem
(c) Commonrailsystem

(d) Distributor system

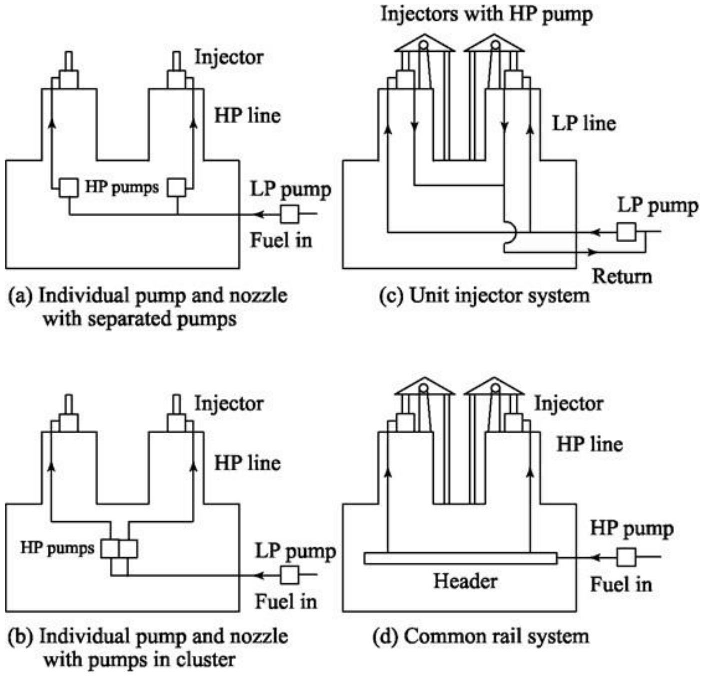


Figure:Solidinjectionsystems

(a) Individual pump and nozzle system:(Refer figure a&b) In this system, each cylinder is provided with one pump and one injector.In this arrangement a separate metering and compression pump is provided for each cylinder.The pump may be placed close to the cylinder (figure a) or they maybe arranged in a cluster (figure b).The plunger of high pressure pump is actuated by a cam and producesthefuelpressurenecessarytoopenthe injectorvalve atthe correcttime.The amountof fuel injected depends on the effective stroke of the plunger.

(b) Unit injector system: (Refer figure c) In this system the pump and injector nozzle are combinedin one housing.Each cylinder is provided with one of these unit injectors.Fuel is brought up to the injector by alow pressure pump, at correct time a rocker arm actuates the plunger and thus injectsthe fuel into the cylinder.The amount of fuel injected is regulated by the effective stroke of the plunger.

(c) Common rail system: (Refer figure d) In the common rail system a HP pump supplies fuel under high pressure to a fuel header.From the header the fuel supplied to the individual cylinders through pipes.The amount of fuel entering the cylinder is regulated by varying the length of the push rod stroke.

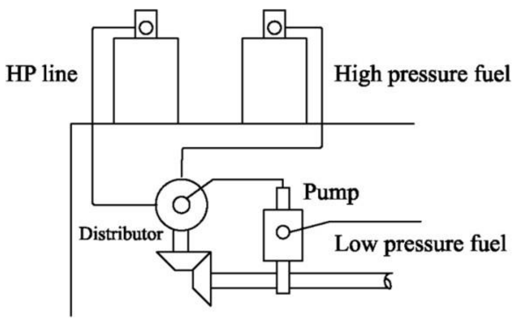


Figure:Distributorsystem

(d) **Distributorsystem**: (Refer figure) In this system the pump which pressurizes the fuel also meters and times it.The fuel pump after metering the required amount of fuel supplies it to a rotating distributor at the correct time for supply to each cylinder.

FuelInjectionPump

The main objective of the fuel injection pump is to deliver accurately metered quantity of fuel under high pressure at the correct instant to the injector fitted on each cylinder.Injection pumps are generally of two types

Jerktypepump:Figure showstheJerk typefuelinjectionpump.

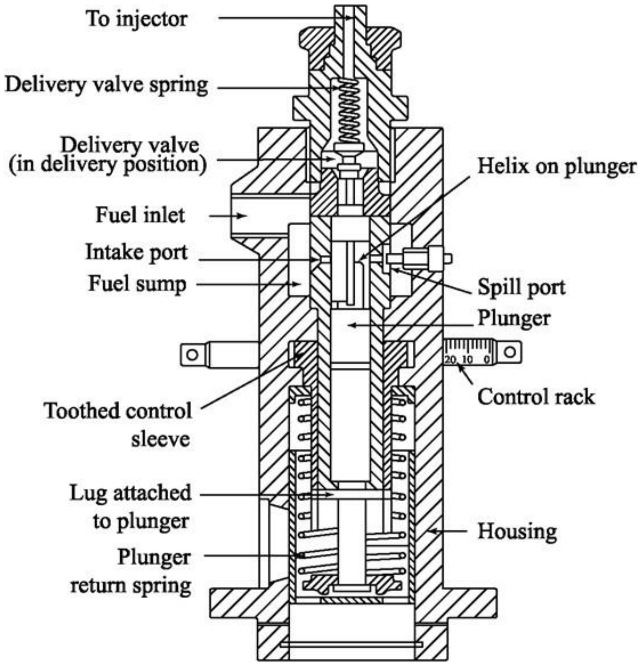


Figure:Jerktypefuelinjectionpump

In the Jerk type pump, the fuel is pressurized bythe plunger movement.The reciprocating movement is given to the plunger by means of the cam and the rotating movement is given by the rack mechanism.When the plunger is in the bottom most position, through the intake port the fuel is admitted into the cylinder of the pumpand during the upward movement of the plunger the fuel is pressurized and delivered to the injector though the opening at the top of the pump against thedeliveryvalvespringpressure.Whenthecontrol rack rotatestheplunger,thehigh pressurefuelinthe top of the cylinder is exposed to spill port through the helix on the plunger so that the pressure of the fuel decreases and injection is stopped.It is important to remember here that though the axialdistance traversed by the plunger is same for every stroke, the rotation of the plunger by the rack determining the length of the effective stroke and thus the quantity of the fuel injected.

Distributortype pump: Figureshows the distributor type fuel injection pump.This pump has onlya single pumping element and the fuel is distributed to each cylinder by means of a rotor.There is a central longitudinal passage in the rotor and also two sets of radial holes (each equal to the number of engine cylinders) located at different heights. One set is connected to pump inlet via central passage whereas the second set is connected to deliverylines leading to injectors of the various cylinders. The fuel is drawn into the central rotor passage from the inlet port when the pump plunger moves away from each other. Wherever, the radial deliverypassage in the rotor coincides with the deliveryport for any cylinder the fuel is delivered to each cylinder in turn.The main advantage of this type is small size and light in weight.

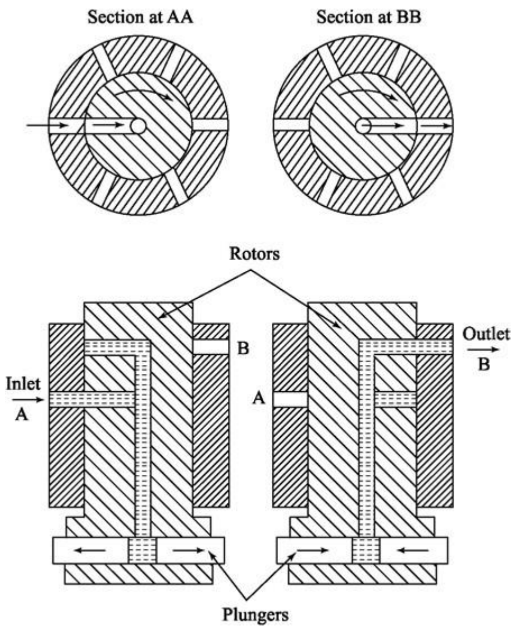
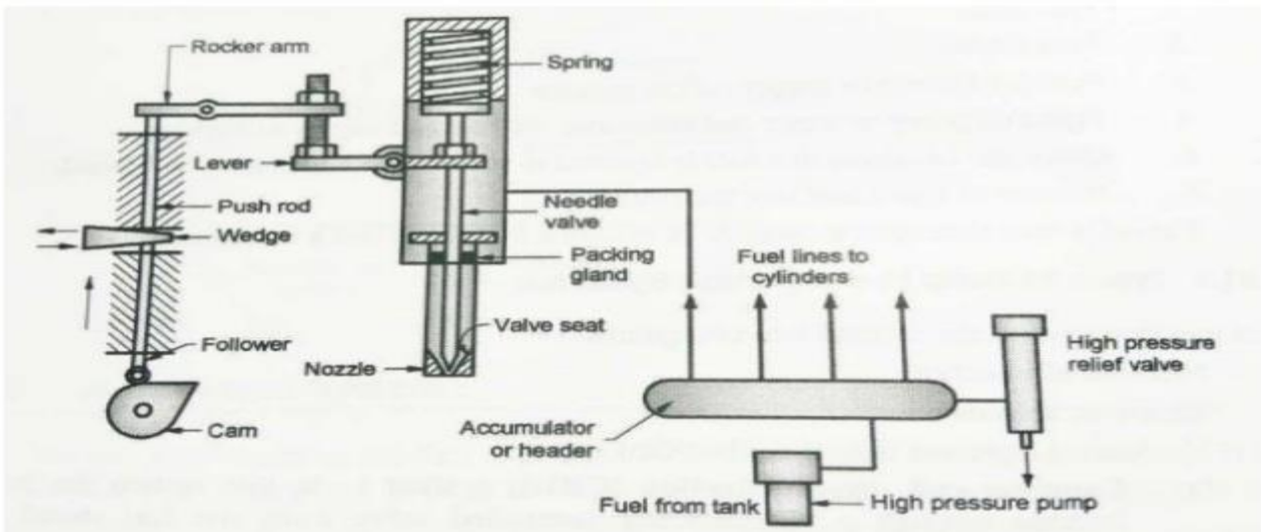


Figure:Distributortypefuelinjection pump

CommonRailDirectInjectionorCRDISystem:

There are two different types of injection systems in the diesel engines or compression ignition (C I Engines).OneistheAirInjectionSystemandanotheroneisAirlessorSolidInjectionSystem.Inthis post, we are going to learn about the Common Rail Direct Injection System CRDI System which comes under a solid injection system.



Workingof CRDISystem or CommonRail DirectInjection:

1. Asyou canseeinthediagramoftheCRDIsystem,thehigh-pressurepumpisusedtosupply fuel to the accumulator or the header from the fuel tank. In case pressure in the accumulator increases beyond the limit, the high-pressure relief valve which is connected to the accumulator helps to reduce the pressure.

2. Now,thisfuelfromtheaccumulatorsuppliedtoenginecylindersusingfuellineswiththehelp of solid injectors.
3. Anotherspring-loadedhigh-pressurerelief valveusedtomaintaintheconstantpressureinthe systemforsmoothoperations. Italsoreturnstheextrafueloftheaccumulatortothefueltank.
4. In the diagram, you can see the needle valve. It is used to control the opening and closing of thenozzlewhileitspraysthefuelintothecylinders.Theupwardanddownwardmotionofthe nozzle is measured by the cam.
5. Cam is connected to the spring with the help of a rocker arm and lever. During the dwell periodofthecam,springwiththehelpoftheneedlevalvepreventstheinjectionofthefuel into the cylinder.
6. Thepackingglandensuresthelevelofthefuelabovethevalveseatforbetterinjectionofthe fuel into the cylinders.
7. Thewedgeplaysthemainroleinthissystem. Itcontrolstheamountoffueltobeinjectedinto thecylinderinaccordancewiththepower requiredfortheengine.Thewedgeisoperatedbya governor or it can be operated manually as per requirement.

AdvantagesofCRDISystem:

- CRDIsystem cancontrolthe flowoffuel inaccordancewith theloadand speedof the engine.
- Thissystemrequires onlyonefuelpump formultiplecylinders.
- CRDIsystemisbeneficialfortheenvironmentasitreducesnoise,smokeandparticulate matter.
- It giveshighpoweroutput atlowrpm.
- ThemainadvantageoftheCRDIsystemisfueleconomy.

Disadvantages of CRDI System:

- Thissystemiscomplexthan MPFIsystem andneedsgoodengineering work.
- TheCRDIsystemcannotsuit ordinaryengines.

3.10.4Fuel Injector

Quick and complete combustion is ensured bya well-designed fuel injector.Byatomizing the fuel into very fine droplets, it increases the surface area of the fuel droplets resulting in better mixing and subsequent combustion.Atomization is done by forcing the fuel through a small orifice under high pressure.The injector assembly consists of a needle valve, a compression spring, a nozzle andan injector body as shown in figure 3.24. When the fuel is supplied by the injection pump it exerts sufficient force against the spring to lift the nozzle valve and the fuel is sprayed into the combustion chamber in a finely atomized particles.After fuel from the delivery pump gets exhausted the spring pressure pushes the nozzle valve back on its seat.For proper lubrication between nozzle valve and its guide a small quantity of fuel is allowed to leak through the clearance between them and then drained back to fuel tank through leak off connection.The spring tension and hence the valve opening pressure is controlled by adjusting the screw provided at the top.

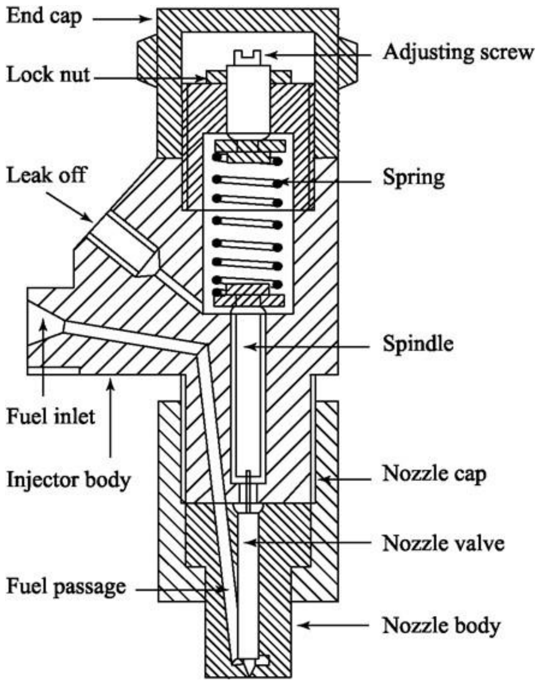


Figure:Fuel Injector

FuelInjection Nozzle

Nozzle is that part of an injector through which the liquid fuel is sprayed into the combustion chamber.The nozzle should fulfil the following functions:

- **Atomization:This is veryimportant function sinceit is thefirst phaseinobtainingpropermixing of the fuel and air in the combustion.**
- **Distributionoffuel:**Distributionoffueltotherequiredareaswiththecombustionchamber. **Factorsaffectingthisareinjectionpressure,densityof airandphysicalpropertiesoffuel.**
- **Preventionof impingement on walls: Prevention of the fuel from impinging directly on the walls of combustion chamber or piston.This is necessary because fuel striking the walls decomposes and produces carbon deposits.This causes smoky exhaust as well as increase in fuel consumption.**
- **Mixing: Mixing the fuel and air in case of no turbulent type of combustion chamber should be taken care of by the nozzle**

Varioustypes of nozzles used in CIengine areshown in figure3.25 and brieflyexplained below.

(a) **The pintle nozzle**: The stem of the nozzle valve is extended to form a pin or pintle which protrudesthroughthemouthofthenozzle.Thesizeandshapeofthepintlecanbevariedaccordingto the requirement.It provides a spray operating at low injection pressures of 8 to 10 MPa.The spray coneangleis generally60^0Advantageofthis nozzleis thatit avoids weak injection and dribbling.It prevents the carbon deposition on the nozzle hole.

(b) Single hole nozzle: At the centre of the nozzle body, there is a single hole which is closed by the nozzle valve.The size of the hole is usually of the order of 0.2 mm.Injection pressure is of orderof 8 to 10 MPa and spraycone angle is about 15^oMajor disadvantage with such nozzle is that theytend to dribble. Besides their spray angle is too narrow to facilitate good mixing unless higher velocitiesare used.

(c) Multi hole nozzle: It consists if a number of holes bored in the tip of the nozzle. The number of holes varies from 4 to 18 and the size from 35 to200 µm.The hole angle maybe form 20^oupwards. Thesenozzlesoperateathighinjectionpressuresoftheorderof18MPa.Theiradvantageliesinthe

abilitytodistributethefuelproperlyevenwithpowerairmotionavailableinopencombustion chamber.

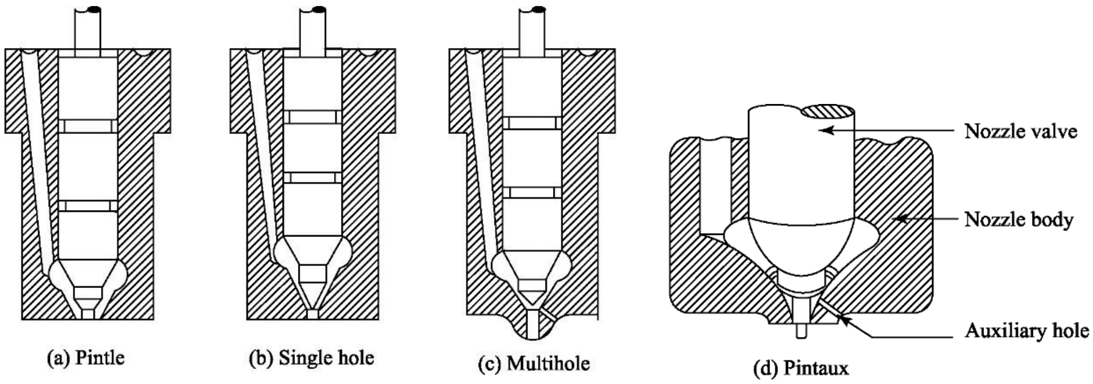


Figure:Typesofnozzles

(e) **Pintauxnozzle**: Itisatypeofpintlenozzlewhichhasanauxiliaryholedrilledinthenozzlebody. It injects a small amount of fuel through this additional hole in the upstream direction slightly before the main injection. The needle valve does not lift fully at low speeds and most of thefuelin injected through the auxiliary hole.Main advantage of this nozzle is better cold starting performance.

(f) **IndicatedPower:**Thepowerisproducedinsidetheenginecylinderiscalledindicatedpower. The indicated power is calculated from the indicator diagram.

(g) Theindicatedpoweriscalculatedby: IP=P_{im}×A×L×n×k kW

(h) WhereP_{im}= Indicatedmeaneffective pressure

(i) $$P_{im} = \frac{\textbf{Areaoftheindicator diagram}}{\textbf{Length ofindicator) diagram}} \times \textbf{Springconstant}(\text{N/m}^2/\text{m}$$

A=Areaofthecylinder(m2), L = Stroke length (m)

n=Number ofexplosions per second(n=Nfortwo strokeengine, n=N/2forfourstrokeengine), N=Speedoftheengineinrps,

k=Number of cylinders

Brakepower: Thepoweravailableatenginecrankshaftforexternalworkiscalledbrakepower.The brake power is determined by Prony brake dynamometer, Rope brake dynamometer and Hydraulic dynamometer.

(j) **Pronybrakedynamometer:**BP= 2×π×N×T kW

WhereT =Torque= W×R,

W=Loadapplied (kN),

R=Radius of brakedrum (m)

(k) **Ropebrakedynamometer:**BP = π×D×N×(W–S) kW

WhereD=Diameterofbrakedrum(m),

W = Weight applied at the end of the rope(kN), S= Spring scale reading (kN)

(l) **Hydraulicdynamometer:**BP=WN/2000 kW

WhereW=brakeload(kN)

(m) **Frictionpower:**Frictionpoweristhedifferencebetweenindicatedpowerandbrakepowerofan engine.

FP= IP–BP

(n) Thefrictionpoweristhemeasureofinternallossesintheengine.Thefrictionpoweris determined by Willan’s line method, Morse test, Motoring test and Retardation test.

(o)

(p) **Brakespecificfuelconsumption(BSFC)**istheamountoffuelconsumedperkWofbrake power developed.

$$BSFC=\frac{FC}{BP} \quad \text{kg/kWs}$$

(q) **MechanicalEfficiency** $\eta_{mech}=\dfrac{\text{Brake Power}}{\text{IndicatedPower}}$

(r) **Brakethermalefficiency**: $\eta_{br.the}=\dfrac{BP}{FC\times CV}$

(s) **Indicatedthermalefficiency** $\eta_{in.the}=\dfrac{IP}{FC\times CV}$

(t) **Relativeefficiency** $\eta_r=\dfrac{\textbf{Brakethermalefficiency}}{\textbf{Air standardeffificney}}$ ***or*** $\dfrac{\textbf{Indicatedthermalefficiency}}{\textbf{Airstandard effificney}}$

(u) **Volumetricefficiency** $\eta=\dfrac{m_{actv}}{m_{th}}$

Quantityof FuelInjected

Thequantityoffuelinjected percycleisgiven by

$$Q=\left(\frac{\pi}{4}d^2n\right)\times V_f\times\left(\frac{\theta}{360}\times\frac{60}{N}\right)\times\left(\frac{N_i}{60}\right)$$

Wheren=numberofholesinnozzle, d

= diameter of orifice,

V_f= velocityoffuel,

θ=durationoffuelinjection incrank angle,

N = speed of the engine, and N_i= number of injection per min (N/2 for four stroke engine and N for two stroke engine)

Afourstroke,fourcylindergasolineenginehasaboreof60mm andastrokeof100mm.Ontestitdevelopsa 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=42000kJ/kg,VC=60CC,ηRbt=0.5

Solution:

BrakePower:BP=2πNT60BP=2×π×3000×66.560BP=20891.59W

BP= **20**. **89**kW

BrakeMeanEffectivePressureBP=PBMEPLAnk

20.89=PBMEP×0.1×π×0.0624×30002×60×4**P**

BMEP=738.83**kN**$\boldsymbol{m}$**2**

FuelConsumption:ηBT=BPṁf×CVṁf=20.89×36000.25×42000

ṁf= **7**. **16**$\boldsymbol{kghr}$

BrakeThermalEfficiency:ηRbt=ηbtidealηbt=ηRbt×idealηbt=0.5×0.5 **ηbt**= **0**.**25**Ideal Efficiency: ηIdeal =

1 − 1 $rc\gamma$−1 ηIdeal = 1 − 1 5.7421.4−1 **ηIdeal** = **0**. **5**

Where,rC =V1V2=VS+$VCVC$rC=**2**.**83**×**10**−**3**+**60**×**10**−**660**×**10**−**6r**$\boldsymbol{C}$=**5**.**742**VS=πD24× L

VS= π×0.0624×0.1 $\boldsymbol{VS}$=**2**.**83** ×**10**−**3**$\boldsymbol{m}$

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 ofstroke,indexofexpansion=1.3,indexofcompression=1.35,L/D=1.5.Assumethepressureandtemperatureof the air at inlet are 1 bar and 40˚C respectively.

IP=18.75kW,N=220rpm,CR=14,L/D=1.5,P1=1bar,T1=40℃

CutoffRatio:V3−V2=0.05(V1−V2)→0.05=ρ−1 rc−1 $\boldsymbol{\rho}$=**1**. **65**

MeanEffectivePressure:

Pm=P1rcγ[γ(ρ−1)−rc1−γ(ργ−1)](γ−1)(rc−1)

Pm=1×141.3[1.3(1.65−1)−141−1.3(1.651.3−1)](1.3−1)(14−1)

Pm=**3**. **4ba**

IndicatedPower:IP=PIMEPLAnk

18.75=3.4×102×1.5D ×π×D24×2202×60 ×1D =0.294m

LengthofStroke:L

D=1.5L=1.5× 0.294

L =0.589

During an experiment on four stroke single cylinder engine the indicator diagram obtained hasaverage height of 1 cm while indicator constant is 25 kN/m2per mm. The engine run at 300 rem and the swept volume is 1.5 × 104 cm3The 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 6kg/min. The cooling water enters at 35o C and leaves at 70oC. Exhaust gases leaving have energy of30 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.

Indicator Diagram Height =1cm,

IndicatorConstant=25kN/mmpermm,

N=300rpm,

VS =1.5 ×104cm3

PIMEP=Ind.diagramheight×IndicatorConstant

PIMEP = 10mm × 25 kNmm×m2

PIMEP=250Kn

Heat Lossdueto Brake power:

P =2πNT60

BP=4.62 kJs×60BP =277. 2kJmin

UnaccountedLoss:

Qua=QS− Qw+Qg+QBP Qua=5040−877.8+1800 +277.2 qua=2085kJmin

%Qw=Qw/QS

%Qw =877.8/5040

%Qw=17. 42

%Qg=Qg/QS

%Qg=1800/5040

%Qg=35. 71

%QBP=QBP/QS

%QBP=277.2/5040

%QBP=5.5

%Qun=Qun/QS

%Qun=2085/5040

%Qun=41. 37

MECHANICALSUPERCHARGING

Superchargers are the main category of forced induction systems. Superchargers are compressors that are driven by mechanical means. Typically, they are driven by the crankshaft of an engine with the help of belts and pulleys. They are coupled directlyto the engine and this does not allow for anydelayto exist between the engine and the compressor. Superchargers are classified into two categories such as positive displacement pumps (Eg. Lysholm, Roots, Eaton, Scroll, Vane) and rotodynamic pumps (Eg. Centrifugal).

PositiveDisplacementType

Mechanical supercharging is probably the oldest way of boosting the IC engine [Ainsdale, 1980]. Positive displacement supercharger is simpler in construction. They are easy to install and doesn't require complicated controls in most cases. It draws power through mechanical connection to the crankshaft, and its revolution (rotational speed) is directly proportional to the speed of the engine. The engine-compressor matching is relatively easy, and the boost pressure is almost constant over the entire range of engine operating speeds. Therefore, the torque curve is flat, and the turbo-lag problem is completely overcome. In effect, a supercharged engine behaves as a naturally aspirated engine with a larger displacement volume. It is this linearity that makes designing and predicting its boosting characteristics relativelyeasier than turbo charging. However, a supercharged engine consumes more fuel than a turbocharged engine with comparable power since the supercharger draws power directly from the engine crankshaft.

A supercharger has two rotors, a male and a female, forming a set of chambers between themselves and the housing. The chamber's volume is changing during the rotationandthuscompressestheairinternally.Sincethepowerneededtodrivethe

superchargeristakendirectlyfromthecrankshaft,mostdevelopmentworkhasbeen invested to increase the efficiency and to minimize the parasitic losses when the supercharger is not needed (i.e. part load of the engine).

The most common types of the supercharger in the market today are the Lysholm compressor and the Roots blower. Figure 2.1 & 2.2 shows the cut-away view of the Lysholm screw and roots type supercharger [Heinz, 1995]. Both are displacement pumps and from a first glance theylook verysimilar. Theydiffer on one bigpoint; the Lysholm screw has internal compression while the Roots do not have internal combustion. The Lysholm Compressor works with internal compression.

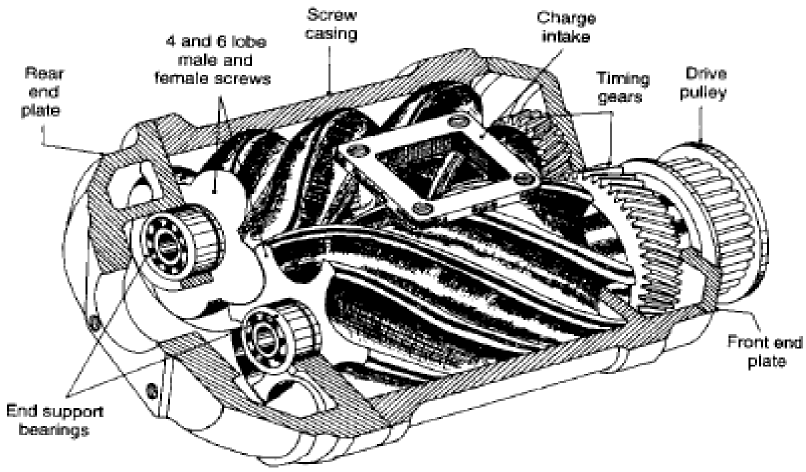


Figure2.1Cut-wayview oftheLysholmscrewsupercharger[Heinz, 1995].

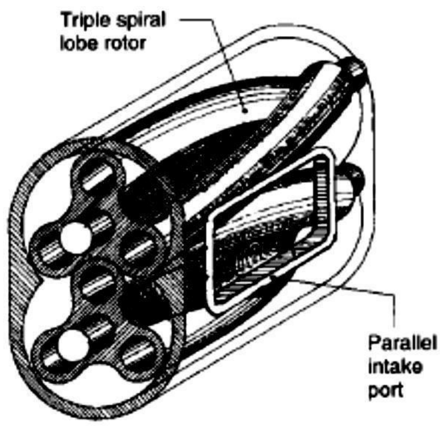


Figure2.2 Cut-wayview ofthe rootstypesupercharger[Heinz, 1995].

The Roots on the other hand works without internal compression, the compression takes place as the air is discharged from the blower outlet instead of inside the supercharger.This means that thecompression takes placeat isochoricconditions (i.e. constant volume). This process is known to be more power consuming and heat producing than the adiabatic process.

CentrifugalCompressorasaSupercharger

The centrifugal compressor has an isentropic efficiency that can match,and sometimes exceed, the efficiency of a Lysholm screw compressor. It is a dynamic machine where the rotor increases the internal energy of the air, both through increased density and increased velocity. The velocity is then carefully diffused to recoverthekineticenergyasstaticpressure.Consequently,thecentrifugalcompressor has internal compression. Figure 2.3 shows the cut-away view of the centrifugal supercharger.

Unfortunately, the flow vs. speed characteristics of the centrifugal compressor is very non-linear. A fixed gear ratio between the compressor and the crankshaft results in a very peaky boost pressure delivery. Despite this, it has been used in cars, during the 1930s and 1950s. The speed of the centrifugal compressor is above 100000rpm and it is up to 10 times higher than roots and about 3 to 5 times the speed of a screw compressor. The need for a gearbox caused trouble for the designer. High speeds can lead to both sound problems if gears are used and the torque needed for acceleration can be rather high even if the moment of inertia is not particularly high.

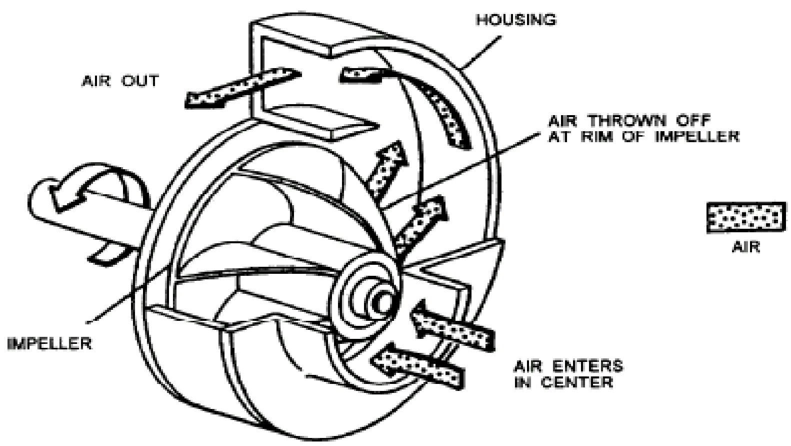


Figure2.3 Cutviewof centrifugal supercharger

ParasiticLosses

The largest problem of using a mechanical supercharger on a downsized engine is not the top-end performance, but the parasitic losses on part load. Miyagi et al., (1996) showed that for a Lysholm compressor only20-30%oftheparasiticlosses come from mechanical losses and the rest from losses in the airflow, i.e. unnecessary pumping. Therefore, it is most important to try to minimize the airflow losses.

Clutch

Themostobviouswayoflimitingtheparasiticlossesistohaveaclutchto engageand disengagethe compressor.Using aclutch has the advantageofreducingboth losses in the air and the mechanical losses, assuming that the clutch is positioned on the crankshaft end of the belt. Since the compressor has a non-negligible inertia and thatit rotates with high speed, it is necessary to apply large amounts of torque in order to accelerate the compressor to working speeds within reasonable time. These torque impulses will result in comfort problems if the work has to be taken from the crankshaft.

TURBOCHARGER

Turbochargers are commonlyused in engines because theyextract some of the energy from the exhaust gases that would have otherwise been lost [Corky Bell, 1997]. Turbochargers consist of a turbine (the component that is being spun by the passing exhaust gases) and the compressor (the component increasing the intake pressure) which is coupled to the turbine by a rotor. Figure 2.4 shows schematic layout of turbocharging system in a diesel engine.

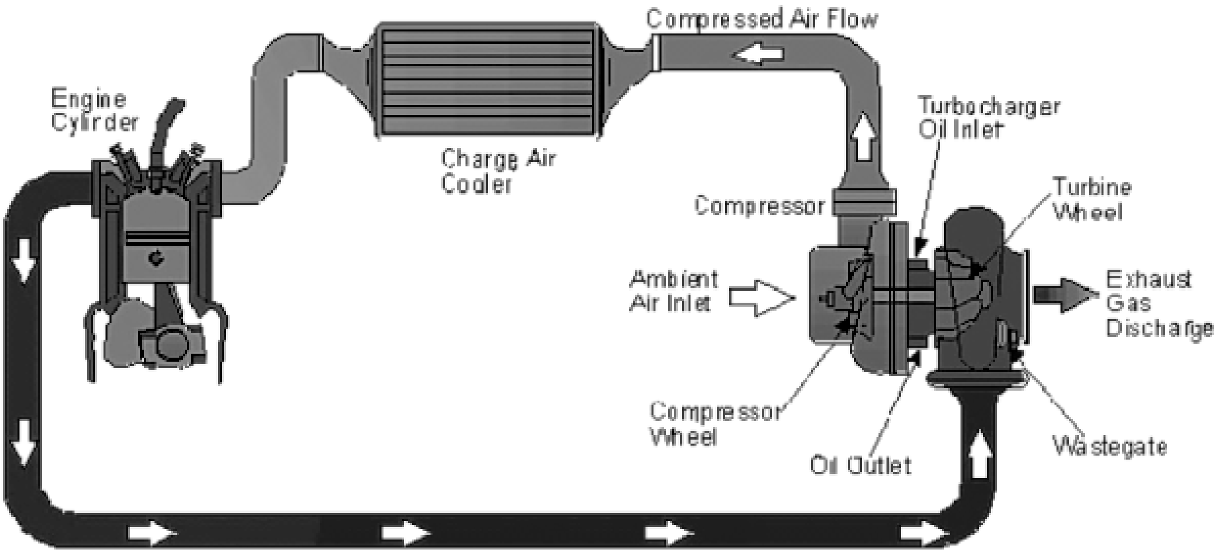


Figure2.4Typicalschematiclayoutofturbochargersystem ofanICengine

One of the advantages of turbochargers is that they are able to recuperate some of the energy of the exhaust gases. This makes turbochargers suitable for use in engines where efficiency is important, as well as for diesel engines that are otherwise not able to produce a high power. Another reason why turbochargers are preferred in diesel engines is because of the lower exhaust temperature [Cengel and Boles, 2006], which doesn't damage the turbine blades (unlike gas powered engines). Typically, turbines are very delicate and require special grades of oils because of their veryhigh rotational speeds (sometimes exceeding 150000 rpm); as long with this ingas powered engine, the higher temperature of exhaust gases tends to melt the tips of the turbineblades which dramaticallydecreases theefficiencyoftheturbinewhichcauses a significant decrease in engine power. In road going vehicles and race cars, turbochargers are well known for the turbo lag. The turbo lag is an unwanted effect and is caused because of the high turbine spool time due to the moment of inertia of the turbine and compressor. As well, it is due to the fact that the driving force of the turbine comes from the exhaust gases whichare compressible. The turbo lag is in other words a delay in the turbine response

WasteGateOperation

Some turbochargers are equipped with a waste gate [Figure 2.5]. This device allows

someoftheexhaustgasestobypasstheturbinerotorathigherenginespeeds.With

this arrangement, the turbocharger can be designed to be more effective at lower engine speeds. The waste gate consists of a valve, actuator, and connecting linkage. The actuator consists of a diaphragm and spring enclosed in canister housing. The valve is located in an exhaust bypass line. Under low boost conditions, the spring pushes against the diaphragm moving the linkage to close the waste gate valve. Turbo boost pressure is directed against the other side of the diaphragm. As boost pressure increases with increased engine speed, the diaphragm movesagainst spring pressure to open the valve and allow a portion of the exhaust gases to bypass the turbine wheel through a connecting line. As boost pressure drops, spring pressure moves the diaphragm and linkage to close the valve. The waste gate is preset at the factory and no adjustment can be made.

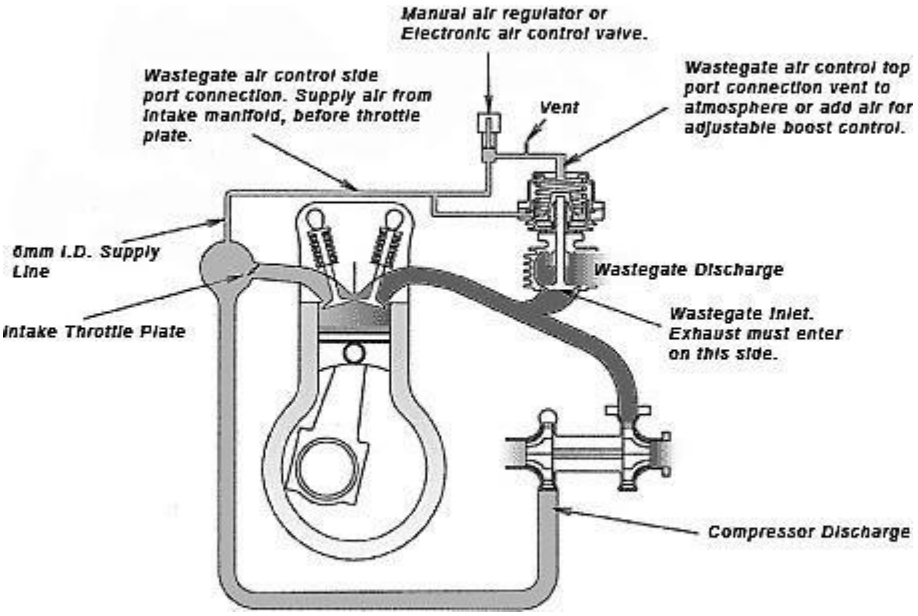


Figure2.5Wastegatearrangementsinaturbocharged engine.

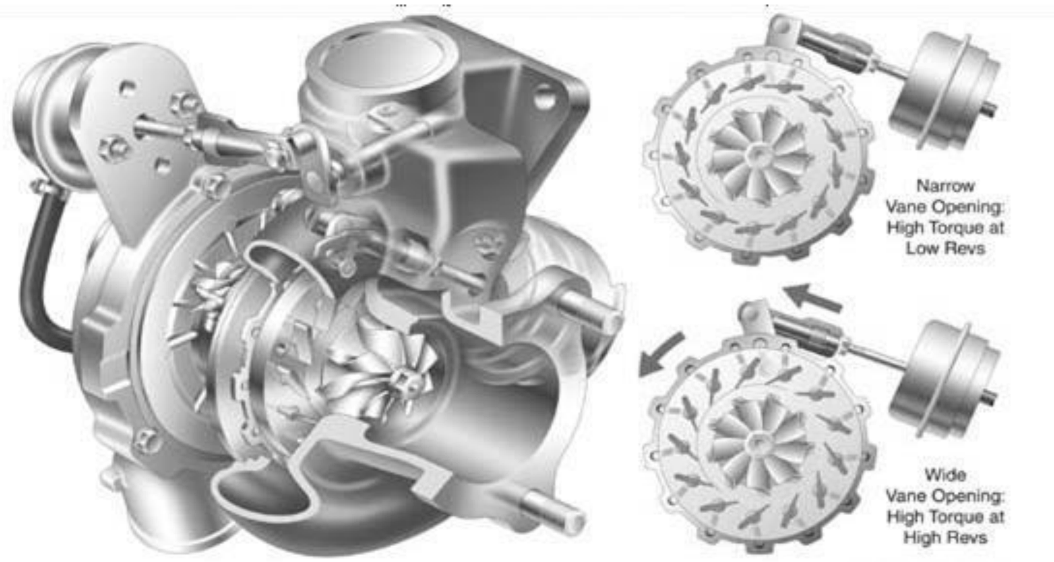
TrendofTurboChargingTechnologies

The goals that a turbocharger must satisfy are the ability to provide high-pressure turbocharging at low enginespeeds, ahigh transient response, and high efficiencyat a high pressure ratio. A fixed geometry turbine is not capable of supplying enough power to the compressor for the boost pressure required for low speed and during transient conditions. On the other hand, it offers a higher turbine inlet pressure which

leads to increased fuel consumption if the turbocharger characteristics are optimized for low engine speeds. In addition, the flow range of a centrifugal compressor is a limiting factor, and if higher boost pressures are demanded, it will be even more difficult to achieve satisfactory width of the usable range since the width of the compressor map becomes narrower as the boost pressure approaches its maximum.

Different turbocharger designs such as the variable geometryturbocharger,electrically assisted turbocharger, and two-stage turbocharger have been developed as a means of achieving the above-mentioned performance improvements. Also, the Variable-NozzleTurbocharger (VNT), which is capableofchangingtheflowcapacity of the turbine, is already in widespread use in diesel passenger cars. The electrically assisted turbocharger improves the transient response and, because it is excellent at achieving high-pressure turbo charging at low speeds, manufacturers are putting considerable effort into its development.

For aturbocharged enginewith theVariable NozzleTurbine (VNT), as is widelyused in diesel engines, theboost pressurecan beraised bycontrollingthevariablenozzleat low enginespeeds. Figure2.6 shows an exampleofaVNTthat was developed foruse in diesel engines. In addition, improving the turbocharger efficiency in the region where the pressure ratio is high is important to reduce the turbine inlet pressure for high-pressure turbo



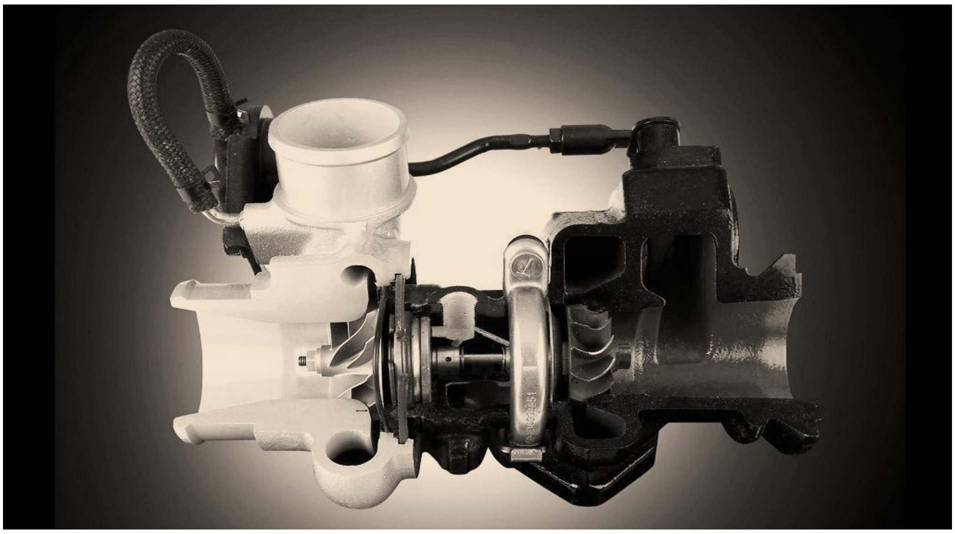
charging.

The VNT has rapidly gained popularity in Japan and Europe despite its higher cost because it offers the advantages of low-end torque, transient response, and lower turbine inlet pressures at high speeds, relative to conventional turbochargers. At present, the VNT is the only technology available that allows diesel engines to satisfy current emissions regulations. On the other hand, the VNT is not compatible with the high gas temperatures of gasoline engines because of its complicated structure and links. However, the amount by which the pressure can be increased at low engine speeds is limited dueto thelow exhaust energy, such that engine back pressurearises.

The Motor-Assist turbocharger (MAT), however, is able to raise the boost pressure at low engine speeds.So, by adding motor assistance, torque characteristics on a par with a large-displacement engine can be attained. A MAT can also recover thermal exhaust energy by acting as a dynamo at high engine speeds [Figure 2.7]. The high- speed motor has its permanent magnet installed on the shaft of the rotor while the stator is in the bearing housing. Because the motor is sensitive to heat, the cooling method is an important aspect of the development.

Moreover, the outer diameter of the permanent magnet cannot be made much bigger than already present because the combined strength of the permanent magnet and the shaft is low. To increase the power of the motor, therefore, the length ofthe permanent magnet has to be extended. Unfortunately, this leads to a reduction in the critical speed of the rotor shaft, vibration, and a risk of damage when operating athigh speeds. Because the surge phenomenon of the compressor sometimes leads to damage to the rotor, it has been the subject of research for some time and by many different manufacturers.

Many kinds of casing treatments have been investigated to improve the surge characteristic and have been put commercialized in large-scale turbochargers. To obtain a high boost pressure over the wide operating range of a turbocharged engine, the turbocharger has to operate at a high pressure ratio and high rotational speed overa wide flow range. On the other hand, the compressor has a surge limit that is related to the flow rate and therefore cannot be operated at



FigureSchematiclayoutofa MAT.

To overcome this, the rotational speed of the turbocharger has to be increased by controlling the variable nozzle vane angle or the power of the motor assistance, while shifting the surge limit of the compressor towards a lower flow rate, or eliminating it altogether, in order to increase the boost pressure at low engine speeds.

There are two ways of eliminating the surge limit. The first is the application of two- stage turbo charging whereby the small compressor of the high-pressure turbocharger is used at low engine speeds. Unfortunately, a disadvantage of two-stage turbo charging is that the system is more complex and larger than single-stage turbo charging. The second method involves bypassing the compressor discharge air to the compressor inlet so as to increase the flow rate of the compressor. This, however, causes an increase in the turbine inlet pressure due to the increase in the compressor power. As a result, the fuel consumption of the engine deteriorates.Therefore, a means of improving thesurge limit of the compressor is an essential technology. There are several means of improving the surge limit of a centrifugal compressor.Oneeffectivemeansisto re-circulatepartof the airthatis compressedbytheimpeller to the impeller inlet byusing a casing treatment on the shroud wall.

The surge flow rate can be reduced by using a compressor with a variable inlet g