ADVANCED THEORY OF I.C. ENGINE

U6EAUA27 ADVANCED THEORY OF I.C. ENGINES

Unit – I Cycle Analysis 9

Otto, diesel, dual, Stirling and Brayton cycles, comparison of air standard, fuel air and actual cycles, simple problems on the above topics.

Unit – II Combustion 9

Combustion reactions and stoichiometry, heat of reaction, adiabatic flame temperature in constant pressure and constant volume systems, fuels for internal combustion engines and their properties, premixed and diffusion combustion as applicable to SI and CI engines, concepts of burning rate and flame velocity, fuel spray characteristics and combustion in diesel engines.

Unit – III Combustion Modelling 9

Basic concepts of engine simulation, governing equations, simulation of various engine processes for SI and CI engines. Adiabatic flame temperature, Heat release calculations. Thermodynamic and Fluid mechanic based models.

Unit – IV Advances in IC Engines 9

LHR engines, surface ignition concept and multi fuel engines, stratified charge and lean burn engines, performance and emission characteristics, merits and demerits.

Unit –V Electronic Engine Management 9

Computer control of SI & CI engines for better performance and low emissions, closed loop control of engine parameters of fuel injection and ignition

Text Books

1. Ganesan .V - "IC Engines" - Tata McGraw-Hill, 2003.
2. John B. Haywodd, "Internal Combustion Engine Fundamentals", McGraw Hill Automotive Technology Series ISBN 0-07-1000499-8, 1988.

Reference Books

1. Ganesan .V – 'Computer Simulation of Spark Ignition Processes' - Universities Process Ltd, Hyderabad - 1993.
2. Ganesan.V. – Computer Simulation of compression ignition engines – Orcent Longman – 2000.
3. Richard Stone – "Introduction to IC Engines" – 2nd edition – Macmilan – 1992.

UNIT – I

CYCLE ANALYSIS

PART – A

working cycle

For an engine to work continuously the cycle of operations, ie suction and compression of charge, ignition and combustion of charge, expansion and exhausting of products of combustion must be regularly repeated in the engine cylinder. A complete cycle of these operations is known as working cycle of the engine. As the working fluid undergoes a cycle of operations, power is produced inside the engine cylinder. The working cycle is repeated again and again and the engine works continuously.

Assumptions of air standard Otto cycle

1. Air is the working fluid
2. The engine operates in a closed cycle constant amount of working fluid hence mass is same.
3. Working fluid is homogenous thorayur and no chemical reaction.
4. Compression and expansion processes are alliabative
5. All the process are internally reversible and no mechanical or frictional losses to occur through at the process.
6. Combustion is replaced by heat addition process and exhaust is replaced by heat rejection process.

Thermo dynamic air standard cycle

Different IC engines work on various thermodynamic cycles. In order to compare the efficiencies of these thermodynamic cycles, it becomes necessary to eliminate the effect of the calorific value i.e., heat value of the fuels used. To do this, air is assumed to the working medium inside the engine cylinder.

Air is assumed to be heated during certain strokes by a hot body and then cooled during certain other strokes by the action of the cold body, applied to the cylinder end. Thus the air in the cylinder alternately absorbs and rejects heat during the cycle and the engine can be considered to be working as a hot air engine. Throughout the cycle, the working medium (air) is assumed t behave as a perfect gas. Further, the specific heats of air are considered as constant and no heat exchange takes place between the working medium and the engine walls during compression and expansion. These processes are assumed as adiabatic and reversible. A process is said to be thermodynamically reversible if it can be reversed and can return the medium and all other substances involved to their original condition existing before the process occurred.

The whole conception is theoretical only. The efficiency thus obtain is known as air standard efficiency. It is sometime called ideal efficiency.

Otto cycle.

The first successful engine embodying the principle of BEAU DE ROCHAS was built in 1876 by NIKOLAUS A OTTO, a German, from which came the term Otto cycle. This is the basic cycle for all engines working on spark ignition principle. The cycle is shown in figure.

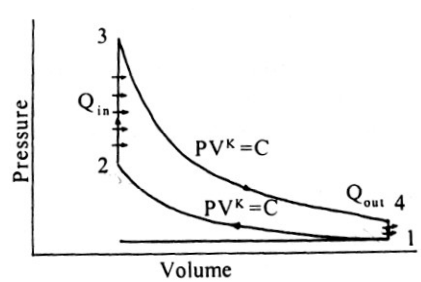


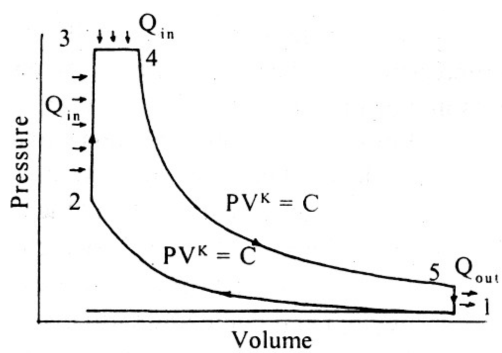
Fig. 2.1. OTTO CYCLE OR EXPLOSION CYCLE

In the air standard Otto cycle, air is compressed adiabatically and reversibly from 1 to 2. Heat is added to the compressed air during the constant volume heating process from 2 to 3. Adiabatic reversible expansion occurs from 3 to 4. The air is finally cooled from 4 to 1. This process returns the air to the initial condition.

PV diagram of diesel cycle constant pressure cycle and mention the process.

Diesel cycle

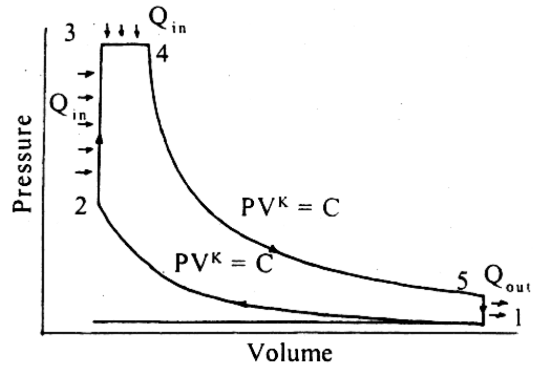
In 1892, Diesel, a German proposed compression of air alone until a sufficiently high temperature was attained to ignite the fuel which as to be injected at the end of the compression process. The cycle proposed by him is called Diesel cycle. This is the basic cycle for the slow speed compression ignition oil engines. The air standard diesel cycle is shown in fig.



In this cycle, air is compressed adiabatically and reversibly from 1 to 2. Heat is added to the compressed air from 2 to 3, at constant pressure. Adiabatic reversible expansion occurs from 3 to 4. Heat is rejected from 4 to 1. This process returns the air to the initial condition.

Mixed cycle, Dual cycle, Semi diesel cycle

Modern high speed diesel engines have the combustion process that lies between that of Otto engine and slow speed diesel engine. The cycle is a mixed cycle as shown in fig. In this air standard cycle, air is compressed adiabatically and reversibly from 1 to 2. Part of the heat is added at constant volume process from 2 to 3, and the balance at constant pressure from 3 to 4. Adiabatic reversible expansion occurs from 4 to 5. Heat is rejected at constant volume from 5 to 1. this process returns the air to tits initial condition.



Stirling Cycle?

This cycle was proposed by a Scottish scientist named Robert Stirling. The Stirling cycle consists of two isothermal and two constant volume (isochoric) processes, as shown in figure. The Stirling cycle has irreversible processes unlike the Carnot cycle. The amount of heat added and rejected during the constant volume process is the same.

Brayton or Joule cycle.

The Brayton or Joule cycle is the theoretical cycle for gas turbines. This cycle consists of two isentropic and two isobaric processes. An extended version with expanded isentropic compression and isochoric heat rejection is available.

Air standard η of an otto cycle whose compression ratio is 10. Take K = 1.4.

Ans:

$$\eta = 1-\left(\frac{1}{r}\right)^{K-1}$$

$$= 1\text{-}\left(\frac{1}{10}\right)^{1.4-1} = 1-(0.1)^{0.4}$$

$$= 0.6 = 60\%$$

Expression for air standard efficiency of an Otto cycle.

In the air standard Otto cycle, air is compressed adiabatically and reversibly from 1 to 2. Heat is added to the compressed air during the constant volume heating process from 2 to 3. Adiabatic reversible expansion occurs from 3 to 4. The air is finally cooled from 4 to 1. This process returns the air to the initial condition.

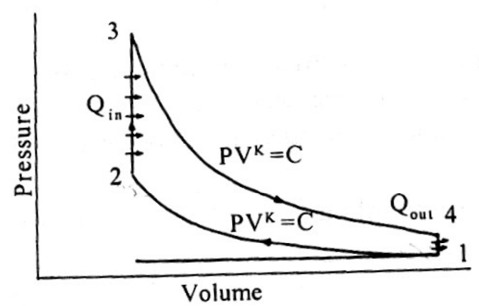


Fig. 2.1. OTTO CYCLE OR EXPLOSION CYCLE

Weight of fluid (w) = $\frac{P_1V_1}{RT_1} = \frac{P_2V_2}{RT_2} = \frac{P_3V_3}{RT_3}$

$= \frac{P_4V_4}{RT_4}$

Heat supplied, $Q_A = w\ C_v\ (T_3 - T_2)$

Heat rejected, $Q_R = w\ C_v\ (T_4 - T_1)$

Work done by the fluid = $\frac{P_4V_4 - P_3V_3}{(k-1)}$

Work done on the fluid = $\frac{P_2V_2 - P_1V_1}{(k-1)}$

Work output per cycle = $Q_A - Q_R$

$= w\ C_v\ [(T_3 - T_2) - (T_4 - T_1)]$

Cycle efficiency = $\frac{Q_A - Q_R}{Q_A} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$

We know that $P_1V_1 = mRT_1$ and $P_2V_2 = mRT_2$.

Where m is the mass of air.

Therefore, $$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

Also $P_1V_1^k = P_2V_2^k$

Therefore, $$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{1/k} \text{ or } \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k$$

$$\frac{T_2}{T_1} = \frac{P_2}{P_1}\left(\frac{V_2}{V_1}\right) = \left(\frac{V_1}{V_2}\right)^k\left(\frac{V_2}{V_1}\right) = \left(\frac{V_1}{V_2}\right)^{k-1}$$

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

Compression ratio, $$r = \frac{V_1}{V_2}$$

Expansion ratio $$= \frac{V_4}{V_3} = r(\text{in this case})$$

Therefore, $T_2 = T_1\, r^{\,k-1}$ and $T_3 = T_4\, r^{\,k-1}$

Efficiency $$\eta = 1 - \left(\frac{1}{r}\right)^{k-1}.$$

Expression for thermal efficiency of a diesel cycle.

Diesel cycle

In 1892, Diesel, a German proposed compression of air alone until a sufficiently high temperature was attained to ignite the fuel which as to be injected at the end of the compression process. The cycle proposed by him is called Diesel cycle. This is the basic cycle for the slow speed compression ignition oil engines. The air standard diesel cycle is shown in fig.

In this cycle, air is compressed adiabatically and reversibly from 1 to 2. Heat is added to the compressed air from 2 to 3, at constant pressure. Adiabatic reversible expansion occurs from 3 to 4. Heat is rejected from 4 to 1. This process returns the air to the initial condition.

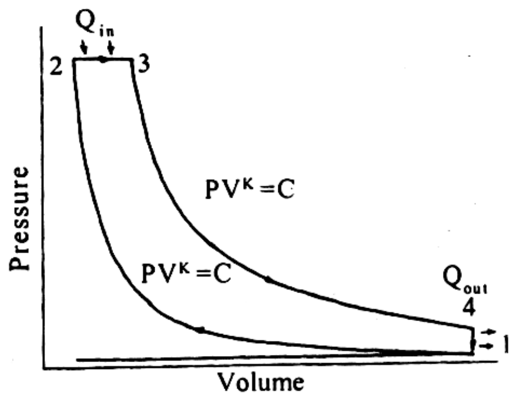


Fig. Diesel cycle for constant pressure cycle

Compression ratio, $r = \frac{V_1}{V_2}$

Cut off ratio, $\rho = \frac{V_3}{V_2}$

Heat supplied $Q_a = wC_p(T_3 - T_2)$

Heat rejected $Q_R = wC_v(T_4 - T_1)$

Weight of fluid (w) = $\frac{P_1V_1}{RT_1} = \frac{P_2V_2}{RT_2}$

Weight of fluid (w) = $\frac{P_3V_3}{RT_3} = \frac{P_4V_4}{RT_4}$

$P_1V_1^k = P_2V_2^k$ and $P_3V_3^k = P_4V_4^k$

Therefore $\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k = r^k$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = \rho$$

$$\frac{T_2}{T_1} = \frac{P_2V_2}{P_1V_1} = \left(\frac{V_1}{V_2}\right)^k \frac{V_2}{V_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

Work output = Q_A-Q_R

$$\text{Efficiency } \eta = \frac{(Q_A - Q_R)}{Q_A} = \frac{wC_p(T_3 - T_2) - wC_v(T_4 - T_1)}{wC_p(T_3 - T_2)}$$

$$= 1 - \frac{C_V}{C_p}\left[\frac{T_4 - T_1}{T_3 - T_2}\right]$$

$$T_2 = T_1 r^{k-1}, T_3 = T_2\rho = T_1\rho r^{k-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1}\left(\frac{V_4V_2}{V_2V_3}\right)^{k-1}$$

$$= \frac{r^{k-1}}{\rho^{k-1}}$$

$$T_4 = \frac{T_3\rho^{k-1}}{r^{k-1}} = \frac{T_1\rho r^{k-1}\rho^{k-1}}{r^{k-1}} = T_1\rho^k$$

$$\text{Efficiency } = 1 - \frac{1}{k}\left[\frac{T_1\rho^k - T_1}{T_1\rho r^{k-1} - T_1 r^{k-1}}\right] \qquad \because \frac{C_p}{C_v} = k$$

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

$$\text{Work output } = wC_v\left[\frac{C_p}{C_v}(T_3 - T_2) - (T_4 - T_1)\right]$$

$$= wC_v\left\{k\left[T_1\rho^k r^{k-1} - T_1 r^{k-1}\right] - \left[T_1\rho^k - T_1\right]\right\}$$

$$= wC_v T_1\left[kr^{k-1}(\rho - 1) - (\rho^k - 1)\right]$$

When ρ	1.25	1.50	2.00	3.00
Head supply % of stroke	2	3.9	7.7	11.5
Efficiency	63.6	62.0	59.2	55.8

The efficiency decreases as load (cut off ratio) on the engine increases. In other words as load decreases efficiency increases.

Functioning of a dual cycle and derive an expression for the thermal efficiency of dual cycle.

Modern high speed diesel engines have the combustion process that lies between that of Otto engine and slow speed diesel engine. The cycle is a mixed cycle as shown in fig. In this air standard cycle, air is compressed adiabatically and reversibly from 1 to 2. Part of the heat is added at constant volume process from 2 to 3, and the balance at constant pressure from 3 to 4. Adiabatic reversible expansion occurs from 4 to 5. Heat is rejected at constant volume from 5 to 1. this process returns the air to tits initial condition.

Compression ratio, $r=\frac{V_2}{V_1}$

Explosion ratio, $\alpha=\frac{P_3}{P_2}=\frac{T_3}{T_2}$

Cut off ratio, $\rho=\frac{V_4}{V_3}=\frac{T_4}{T_3}$

Weight of fluid (w) = $\frac{P_1V_1}{RT_1}=\frac{P_2V_2}{RT_2}=\frac{P_3V_3}{RT_3}=\frac{P_4V_4}{RT_4}=\frac{P_5V_5}{RT_5}$

Heat supplied = $Q_A=wC_v\left(T_3-T_2\right)+wC_p\left(T_4-T_3\right)=wC_v\left\{\left(\frac{T_3}{T_1}-\frac{T_2}{T_1}\right)+\frac{C_p}{C_v}\left(\frac{T_4}{T_1}-\frac{T_3}{T_1}\right)\right\}$

$\frac{T_3}{T_1}=\frac{T_3}{T_2}\frac{T_2}{T_1}=\alpha r^{k-1}\text{ since }\frac{T_2}{T_1}=r^{k-1}\text{and}\frac{T_3}{T_2}=\alpha$

$\frac{T_4}{T_1}=\frac{T_4}{T_3}\frac{T_3}{T_2}\frac{T_2}{T_1}=\rho\alpha r^{k-1}$

Heat supplied, $Q_A=wC_vT_1\left\{\left(\alpha r^{k-1}-r^{k-1}\right)+k\left(\rho\alpha r^{k-1}-\alpha r^{k-1}\right)\right\}$

Heat rejected, $Q_R=wC_v\left(T_5-T_1\right)=wC_vT_1\left(T_5/T_1-1\right)$

$$\frac{T_5}{T_1}=\frac{T_5}{T_4}\frac{T_4}{T_3}\frac{T_3}{T_2}\frac{T_2}{T_\pi}$$

$$\frac{T_5}{T_1} = \rho\alpha r^{k-1}\left(\frac{\rho}{r}\right)^{k-1} = \alpha\rho^k$$

Heat rejected $= wC_v T_1\{\alpha\rho^k - 1\}$

Work output $= Q_A - Q_R$

Efficiency $= \frac{Q_A - Q_R}{Q_A} = 1 - \frac{(\alpha\rho^k - 1)}{[(\alpha - 1) + k\alpha(\rho - 1)]r^{k-1}}$

When α=1, Dual cycle becomes diesel cycle.

ρ=1, Dual cycle becomes otto cycle.

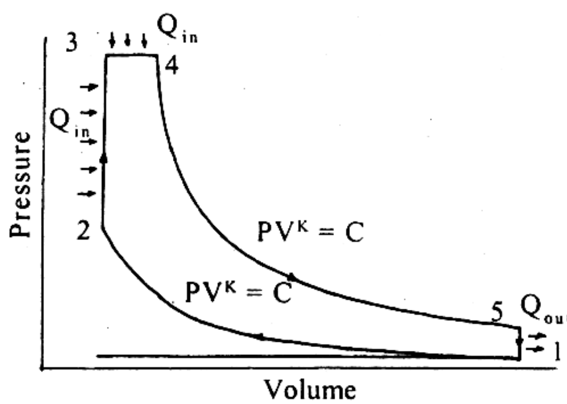


Fig. Mixed Cycle or Dual Cycle

Stirling Cycle

This cycle was proposed by a Scottish scientist named Robert Stirling. The Stirling cycle consists of two isothermal and two constant volume (isochoric) processes, as shown in figure. The Stirling cycle has irreversible processes unlike the Carnot cycle. The amount of heat added and rejected during the constant volume process is the same.

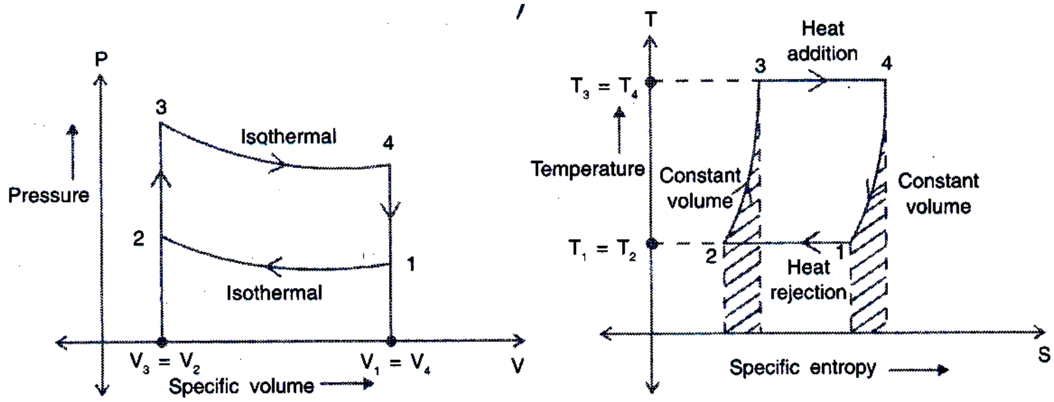


Figure: Stirling cycle on P-V and T-S diagrams

The efficiency of the Stirling cycle is given by

$$\eta_{th} = \frac{RT_3 \log_e\left(\frac{v_4}{v_3}\right) - RT_1 \log_e\left(\frac{v_1}{v_2}\right)}{RT_3 \log_e\left(\frac{v_4}{v_3}\right)}$$

But $v_3 = v_2$ and $v_1 = v_4$

$$\therefore\ \eta_{stirling} = \frac{T_3 - T_1}{T_3}$$

Stirling cycle was used for hot air engines and was later discarded. The heat addition and rejection has to be done by an efficient heat exchanger working at high temperatures. Design of such a heat exchanger was difficult. However with advances in metallurgy, such heat exchangers are possible. Thus the Stirling cycle has again gained prominence, especially in field of cryogenics. Practical Stirling cycle efficiency considering heat exchanger efficiency can be written as follows:

$$\eta_{stirling} = \frac{R(T_3 - T_1)\ln\ r}{RT_3 \ln\ r + (1-\in)C_v(T_3 - T_1)}$$

Where $\in$ = heat exchanger effectiveness

r = compression ratio

C_v = Specific heat at constant volume and

The processes of Stirling cycle are

1-2 Isothermal heat rejection
2-3 Isochoric compressions
3-4 Isothermal heat addition
4-1 Isochoric expansion

Expression for the efficiency and work out put of a Brayton cycle.

The Brayton or Joule cycle is the theoretical cycle for gas turbines. This cycle consists of two isentropic and two isobaric processes. An extended version with expanded isentropic compression and isochoric heat rejection is available. Refer Figure for the P-V and T-S diagrams of the extended Brayton cycle.

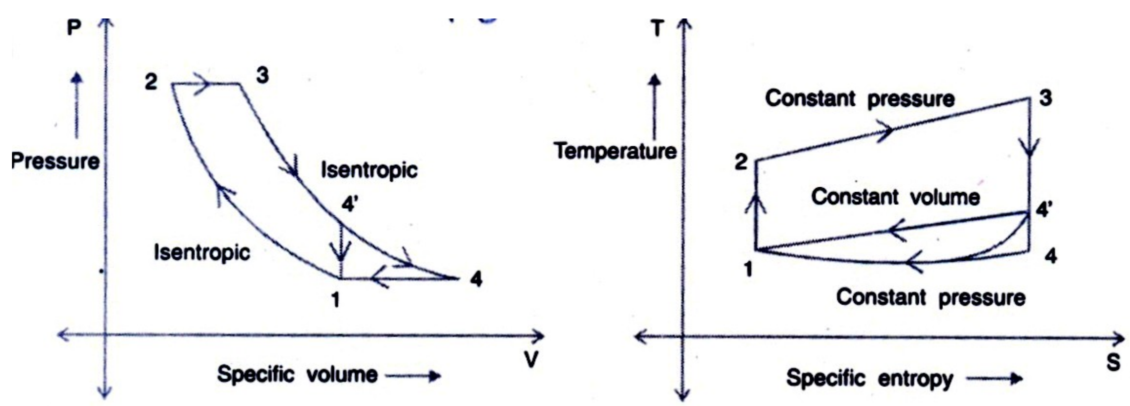


Figure: Extended Brayton cycle on P-V and T-S diagrams

The processes is this cycle are

1-2 Isentropic compression
2-3 Isobaric heat addition
3-4 Isentropic expansion
3-4′ Extended isentropic expansion (optional)
4-1 Isobaric heat rejection
4′-1 Isobaric heat rejection (optional)

Note that process 3-4′ and 4′-1 can be neglected. The Brayton cycle thus becomes as shown in figure with only 4 processes.

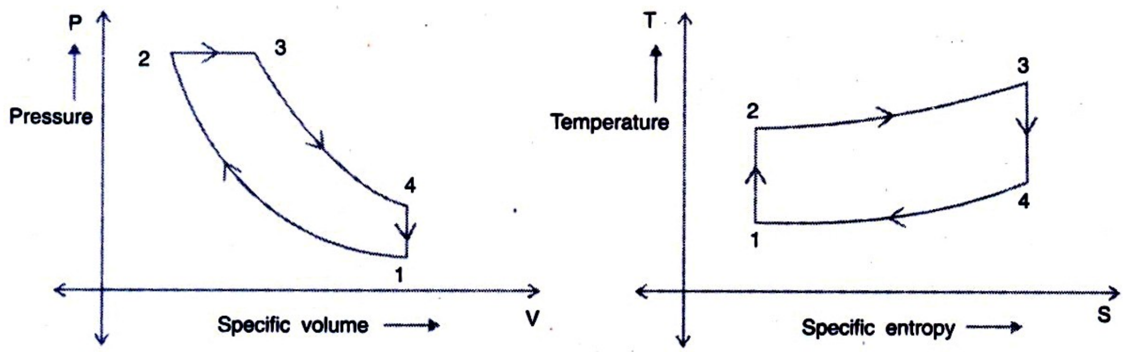


Figure: Brayton cycle on P-V and T-S diagrams

The efficiency of Brayton cycle,

$$\eta_{Brayton} = \frac{Q_S - Q_R}{Q_S} = \frac{mC_p(T_3 - T_2) - mC_p(T_4 - T_1)}{mC_p(T_3 - T_2)}$$

$$\therefore \quad \eta_{Brayton} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

If 'r' is the compression ratio, rp is the pressure ratio, then $r = \frac{V_1}{V_2}$ and $r_p = \frac{p_2}{p_1}$

$$\therefore \frac{T_3}{T_1} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} \left(\frac{V_1}{V}\right)^{\gamma-1} (\ )^{\gamma\text{-}1}$$

$$\therefore T_4 = \frac{T_3}{(r)^{\gamma-1}}; \text{Also } T_1 = \frac{T_2}{(r)^{\gamma-1}}$$

$$\therefore \eta_{Brayton} = 1 - \frac{(T_3/(r)^{\gamma-1}) - (T_2/(r)^{\gamma-1}}{T_3 - T_2}$$

$$\therefore \eta_{Brayton} = 1 - \frac{1}{r^{\gamma-1}}$$

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

$$\therefore \eta_{Brayton} = 1 - \frac{1}{\left(r_p^{\ 1/\gamma}\right)^{\gamma-1}} = 1 - \frac{1}{r_p^{\left(\frac{\gamma-1}{\gamma}\right)}}$$

Work output = W = Cp (T_3 - T_4) – Cp(T_2 – T_1)

$\therefore$ W = C_p ($T_3 - T_4 - T_2 + T_1$)

Thus the Brayton cycle efficiency depends on r and r_p whereas the work output depends on temperature T_1 to T_4.

Compare the Otto, Diesel and dual cycles on the basis of "same maximum pressure and work output

Otto, Diesel and Dual cycles can be compared on basis of four factors which are explained below.

(1) Same maximum pressure and work output

Efficiency η can be written as

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}}$$

Refer to T-S diagram given in figure.

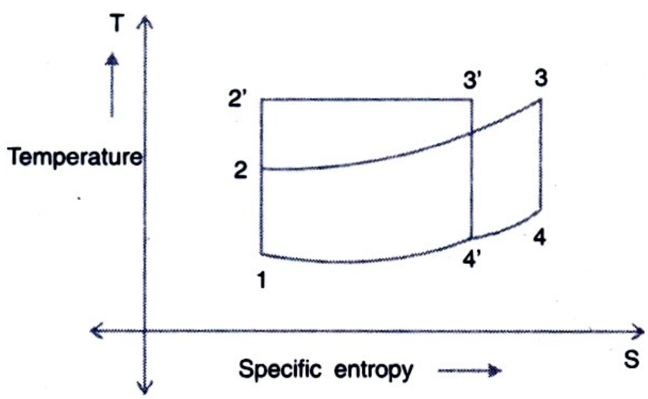


Figure: T-S diagram

For the same work output, area, 1-2-3-4 and area 1-2′-3′-4′ are same, however, heat rejection for Otto cycle is more than Diesel cycle. Thus, the Diesel cycle is more efficient than Otto and Dual cycle.

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

Compare the three cycles on the basis of "same maximum pressure and heat input

Refer to P-V diagram given in figure.

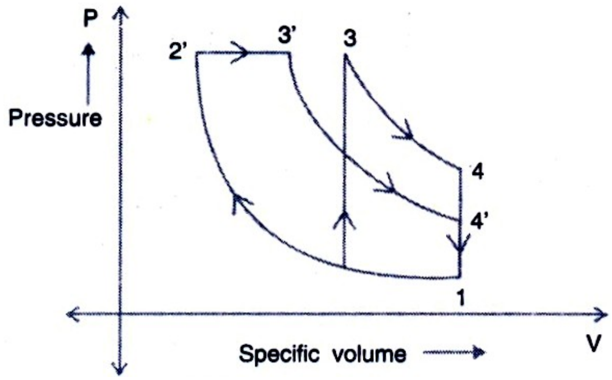


Figure: P-V diagram

It is evident from figure. The heat rejection for Otto cycle is higher than that of Diesel cycle. Also, the compression ratio for Diesel cycle is higher. Thus, for same heat input, work output of diesel cycle is higher. Dual cycle is in between Otto and Diesel cycle, in terms of efficiency.

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

Compare the three cycles on the basis of "same peak pressure, peak temperature and heat rejection"

Refer figure for the P-V diagram.

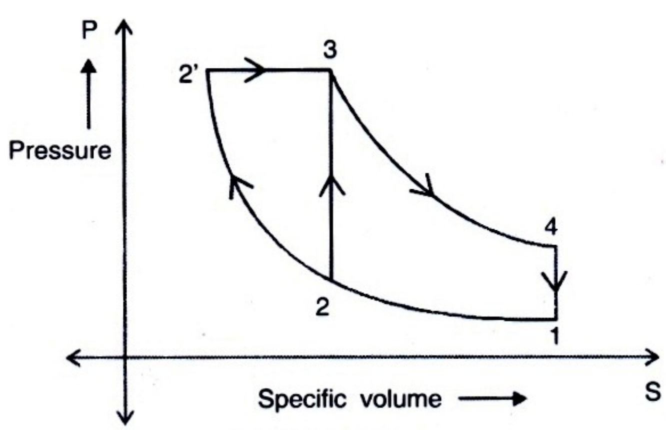


Figure: P-V diagram

The Otto cycle is 1-2-3-4 and Diesel cycle is 1-2′-3-4. The peak pressure and temperature are same.

$$\eta_{Otto} = 1 - \frac{Q_R}{Q_S} \text{ and } \eta_{Diesel} = 1 - \frac{Q_R}{Q'_S}$$

It is evident that $Q'_S > Q_S$. Thus the Diesel cycle is more efficient. The Dual cycle again lies between Otto cycle and Diesel cycle in terms of efficiency.

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

Compare the three cycles on the basis of "same compression ratio and heat addition"

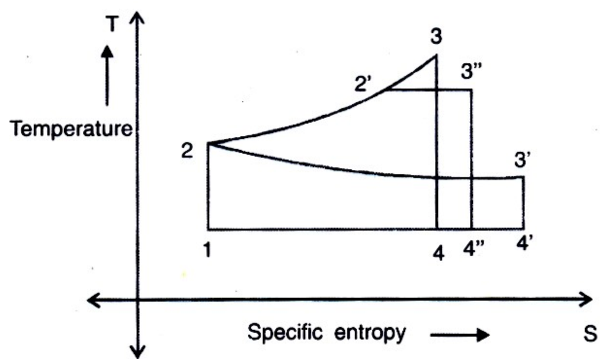


Figure: T-S diagram

Refer figure for the T-S diagram

The Otto cycle is represented by 1-2-3-4-1
The Diesel cycle is given by 1-2-3′-4′-1
The Dual cycle is given by 1-2-2′′-3′′-4′′-1
The heat input and compression ratio is same for all cycles.

It can be seen from figure, the heat rejection in Otto cycle is minimum and heat rejection in Diesel cycle is maximum. Dual cycle is somewhere in between. Thus Otto cycle has the highest efficiency as compared to Diesel and dual cycle.

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

10. An engine working as the Otto cycle has a compression ratio .85:1. The temperature and the pressure at the beginning of compression 93°C and 0.93 bar respectively. The max pressure in the cycle is 38 bar. Define the pressure and temperature at all the points to the cycle, air standard efficiency.

Given:

$$V_K = 8.5 = \frac{V_1}{V_2}$$

$$T_1 = 93^\circ C$$

$$= 93 + 273 = 366K$$

$$P_1 = 0.93 bar$$

$$= 0.93 \times 100 = 93 \text{ K pa}$$

$$P_3 = 38 bar = 38 \times 100 = 3800^\circ Kpa$$

To find: P & T at all points of the cycle

(2) Air std η (3) meP

Pressure 1-2 (Adiabatic compression)

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right); \; p_2 = p_1\left(\frac{V_1}{V_2}\right)^r = 8.5 \times 93$$

$$p_2 = 1860.67 Kpa$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{r-1}{r}} = \left(\frac{1860.67}{93}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 366\left(\frac{1860.67}{93}\right)^{\frac{0.4}{1.4}} = 366(1767.67)^{\frac{0.4}{1.4}}$$

$$= 366 \times (1767.67)^{0.285}$$

$$= 861.44 \text{ K}$$

Process 2-3 Constant volume heat addition

$$\frac{p_3}{p_2} = \frac{T_3}{T_2}\left[\frac{3800p_3}{1860.67p_2}\right] = \frac{T_3}{861.44}$$

$$\therefore T_3 = \frac{3800 \times 861.44}{1860.67}$$

$$= 1759.393K$$

Process 3-4 Adiabatic expansion

$$p_3V_3^r = p_4V_4^r;\ p_4 = \frac{p_3V_3^r}{V_4^r} = p_3\left(\frac{1}{r_K}\right)$$

$$V_3 = V_2,\ V_4 = V_1 = 3800 \times \left(\frac{1}{8.5}\right)^r = 189.932Kp$$

$$p_4 = 189.93Kpa$$

Process 4-1 constant volume heat rejection

$$\frac{p_4}{T_4} = \frac{p_1}{T_1};\ T_4 = \frac{p_4T_1}{p_1}$$

$$T_4 = \frac{189.93 \times 366}{93} = 747.474K$$

UNIT – II

COMBUSTION

Difficulty of combustion in CI engines

In the case of SI engines, the air and fuel are taken in during suction stroke in a properly mixed and vapourized form and compressed during compression stroke. At the end of compression a spark is produced in the combustion chamber by an electrical device. This spark initiates combustion. Since the charge is in the form of a homogenous mixture of air and fuel vapour, the flame spreads throughout the whole charge. There is little or no difficulty in achieving good combustion.

In the case of CI engines, air alone is taken in during suction stroke and compressed during compression stroke to a compression ratio of 14 to 18. The temperature and pressure of air increase. At the end of compression, fuel is injected into the combustion chamber. The hot air ignites the fuel and hence combustion takes place.

Usually fuel is injected around 10 to 20^o before TDC and terminated at about 10^o after TDC. As such, the whole combustion process occupies about 30^o of crankshaft rotation around TDC. If the engine is running at 1200 rpm, then the time available for combustion will be equal to 60 x 30/1200x360 i.e. 1/240 sec. Within this small interval of time whatever fuel that has been injected must mix thoroughly with the air, get itself vapourized and burn in the most efficient form. Hence, combustion in a CI engine is a much more difficult and complicated affair when compared to the combustion in a SI engine. The problem becomes still aggravated in the case of high speed engines.

Delay period in CI engines

As already explained, delay period is the time interval (measured in milliseconds) between the commencement of fuel injection and the beginning of ignition and combustion. The start of combustion is indicated by the deviation point of the pressure curve above the normal compression pressure. In practice, this actual time is as low as 0.006 seconds. The delay period consists of (i) physical delay period and (ii) chemical delay period.

Flame Front

The flame is started by the spark at the spark plug terminals. The flame spreads from there to the remotest points of the combustion chamber. At any instant, the flame has a definite front or boundary (surface area) called flame front. The flame front separates the burned charge from the unburned charge.

Flame velocity

The speed with which has flame front travels affects combustion phenomena, development of pressure and production of power. The mass rate of burning of the mixture depends upon the flame velocity and the shape or contour of the combustion chamber. The mass rate of burning decides the rate of pressure rise in the engine cylinder and smoothness of engine operation. The propagation of the flame through the mixture in the combustion chamber, although very rapid, takes some time. This time is influenced by flame speed. The time between mixture ignition and complete combustion is roughly two milliseconds.

Four stages of combustion in a CI engine

Herry Ricardo has investigated the combustion in a compression ignition engine and divided the same into the following four stages:

1. Ignition delay or delay period.
2. Uncontrolled combustion.
3. Controlled combustion.
4. After burning.

After burning

At the last stage, i.e. between E and F the fuel that is left in the combustion space when the fuel injection stops is burnt. This stage of combustion is called After burning (burning on the expansion stroke). In the indicator diagram, after burning will not be visible. This is because the downward movement of the piston causes the pressure to drop inspired of the heat that is released by the burning of the last portion of the charge.

Increasing excess air, or air motion will shorten after burning i.e. reduce the quantity of fuel that may undergo after burning).

Ignition limits

Experimentally it has been proved that ignition of charge is only possible with certain limits of fuel-air ratio. These “ignition limits” correspond approximately to the A/F ratios (mixture ratios) at lean and rich ends of the scale. Beyond these limits the heat released by spark is insufficient to initiate the combustion process. It is agreed that flame will propagate only if temperature of the burned gases exceeds 1500 K. The stoichiometric A/F ratio for gasoline is approx 15:1, hence the ignition limits are 7:1 to 30:1 respectively. The lower and upper ignition limit are dependent on mixture ratio and temperature. The ignition limits are wider at higher temperatures due to higher thermal diffusivity.

Premixed flame

This region is of very low luminosity, which can be made visible by adding copper additives to the fuel. The flames are normally bluish-green in color.

Diffusion flame

This region is bright due to burning high temperature carbon particles in the flame. The flame appears as yellow or orange in colour.

Desirable properties of IC engine fuels

A good IC engine fuel should fulfill the following requirements:

1. It must have good thermal stability
2. It should be non-toxic
3. It should have high energy density
4. It should be easy to handle and store.
5. It should be flammable with good combustion properties.
6. It should be safe to handle.
7. It should be easily available and should be cheaper in cost.
8. It should have low deposit forming tendency.
9. It should not react chemically with engine components.
10. It should be volatile in nature.
11. It should be mixable with air.
12. Products of combustion should not be corrosive.
13. It should produce lower emissions and hence should limit pollution of the environment.
14. The process of combustion should be rapid with higher amounts of heat release.
15. It should be pure and should not be contaminated.

Classifications of IC engine fuels

Fuels can be classified as solid, liquid and gaseous. For IC engines only liquid or gaseous fuels can be used. As far as gaseous fuels are concerned, they include (1) natural gas (methane) – CNG and (2) hydrogen. Gaseous fuels are advantageous from the point of view of easy compressions, easy transport and lack of freezing in winter. However, they have disadvantages like high cost, large requirement of storage volume and the increase engine size. Liquid fuels are thus most commonly used.

Crude (natural petroleum) oil is the single largest source of IC engine fuels. Petrol and diesel are two main liquid fuels, which are obtained from crude petroleum. Boiling range of petrol is 30°C to 200°C and that of diesel is 200°C to 375°C. Petroleum can be further classified into four classes of fuels.

(1) Paraffins having formula – C_nH_{2n+2}
(2) Olefins having formula - C_nH_{2n}
(3) Napthenes having formula - C_nH_{2n}
(4) Aromatics having formula - C_nH_{2n-6}

Spray characteristics

Quantity of fuel injected and rate of fuel injection decide the power output and performance of the engine (smooth or rough) respectively. Combustion process, rate of pressure rise and engine operation are influenced by the spray characteristics of the fuel jet. The spray characteristics include injection timing atomization, penetration and dispersion aspects.

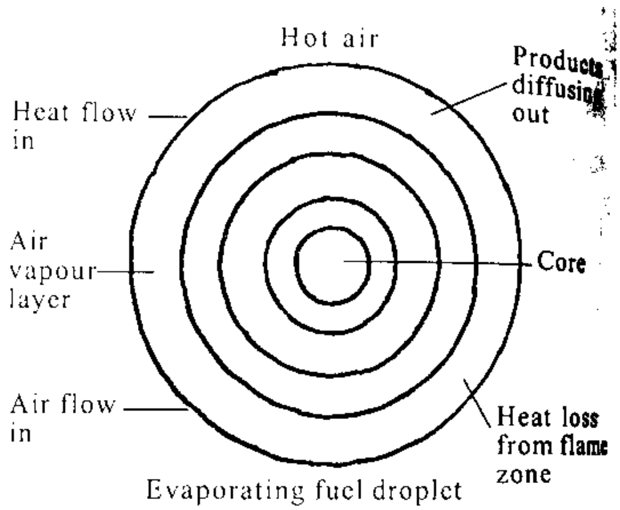
Ignition and Combustion of a fuel drop.

In a compression ignition engine, air is drawn into the cylinder during suction stroke. Air is compressed during compression stroke. Due to this compression, pressure and temperature of air increase. During the latter part of compression, air is given a certain amount of motion. Just before the piston reaches TDC, liquid fuel is injected in the form of a fine spray. The spray contains drops of different sizes. These drops get well dispersed in the air mass and auto ignite. The manner in which these drops burn, decides the performance of CI engines.

COMBUSTION OF A FUEL DROP

The fuel drops may be assumed to be spherical in shape. These drops exist at a lower temperature compared to that of air. Air temperature is about 480 to 500° C, which is higher than the self ignition temperature of the fuel.

For better understanding of droplet combustion, let us imagine that a drop is made up of liquid layers as shown in fig. The duel drop comes in contact with hot air and absorbs heat from air.



Now the following changes take place:

1. Temperature of the drop increases.
2. Drop begins to vapourize. Vapourization starts at the surface of the drop and proceeds to wads the core. During vapourization, latent heat of vaporization is supplied by the hot air. This heat transfer lowers local air temperature momentarily to some extent.
3. Air diffuses into the fuel vapour layer and mixes with it. A low oxidation starts in the mixture layer and proceeds until the temperature of the mixture reaches self ignition temperature.
4. Once the air fuel ratio in the vapour air layer is about the stoichiometric value and the layer attains self ignition temperature, a quick chemical reaction takes place. The vapour air layer now starts burning.
5. The chemical reaction and the consequent burning of the outermost layer cause an increase of pressure and temperature.
6. The heat of combustion of the outer vapour air layer hastens the vaporization of the interior layers of the fuel drop. The chemical reaction now spreads to these

interior layers. When the air vapour mixture in the inner second layer attains stoichiometric mixture strength and self ignition level, it starts burning. This process continues until the core of the fuel drop is consumed.

7. Each fuel drop in the combustion chamber undergoes the above steps of combustion. In other words, combustion takes place at number of locations within the combustion chamber. This type of combustion is called HETROGENOUS COMBUSTION.

For the oxidation and chemical reactions to take place in the interior layers of the fuel drop there is a condition. The products of combustion formed in the surrounding layers should be scored away as soon as they are formed. Only this will allow the interior layers to come in contact with the molecules of oxygen.

The burning of the fuel drop occupies a certain time though it is extremely small. This aspect becomes important particularly in high speed engines, where the time available for the various processes is limited.

Factors influencing flame velocity

The following factors affect the speed of the flame front i.e., flame velocity, with in the combustion chamber.

1. Inlet pressure and temperature conditions
2. Turbulence prevailing and temperature conditions
3. Engine (crankshaft rotational)speed
4. Residual gas content i.e., products of combustion left in the cylinder at the end of the exhaust process.
5. Compression ratio.
6. Spark timing i.e., crank angle with respect to TDC at which spark occurs.
7. Mixture strength i.e air fuel ratio.
8. Fuel (physical and chemical) characteristics.

The velocity of flame propagation is influenced greatly by the air fuel ratio and reaches a maximum at 85 to 90% of the theoretical air, below and above this amount, the velocity decreases, and with a theoretical air it is about 10% lower. Investigations reveal that at certain upper and lower limits the mixture is no longer explosive and only slow combustion can take place.

Turbulence and flame velocity – Air fuel movement in the cylinder will normally speed up and improve combustion efficiency. Turbulence will mix and stir the air fuel mixture and expose more of the unvapourized (unbroken) fuel droplets to the combustion flame. Inlet port, inlet valve, piston and combustion chamber shape, all these affect flow and turbulence of the incoming charge and hence affect combustion.

Turbulence created before combustion starts which includes squish and swirl is called primary turbulence. In an engine cylinder, the working mixture is in a sufficiently intensive turbulent motion consisting of directed vortices and random pulsation of the velocities of the gas steams. As such, the flame front does not have a smooth shape but irregular. Turbulence causes wrinkling or breaking up the flame front. This aspect increases the actual surface of (combustion) the flame front many times.

The influence of mixture turbulence in the cylinder upon the rate of combustion in a petrol engine can be seen in fig. it is evident that excessive turbulence causes a steep pressure rise but reduces peak pressure, probably due to greater heat losses.

The Ricardo turbulent head combustion chamber can be seen in fig. The effect of the variation of the head clearance in the Ricardo turbulent head on pressure can be seen in fig. Reduction in the head clearance increases intensity of turbulence in the mixture and as such deteriorates engine performance.

The flame propagation velocity ranges from 10 to 40m/sec. The average velocity is about 25m/sec, depending mainly upon the character of turbulence. The character of turbulence depends to a great extent on the engine speed, and improves with an increase of speed. If the engine speed is increased, combustion usually gets completed in a shorter time interval. However, combustion duration corresponds approximately the same crank travel.

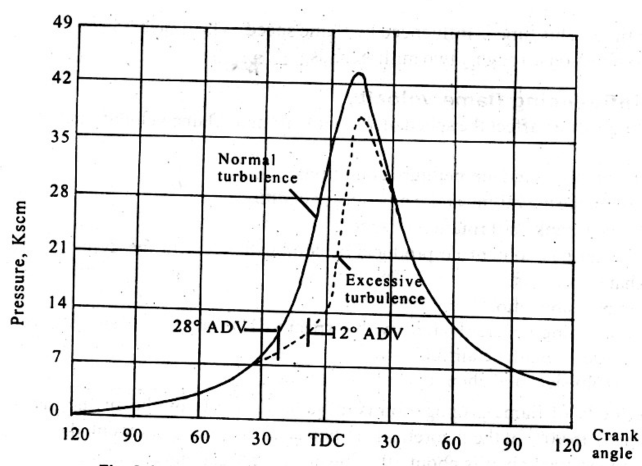


Fig. 3.4. INFLUENCE OF MIXTURE TURBULENCE IN THE CYLINDER ON THE RATE OF COMBUSION

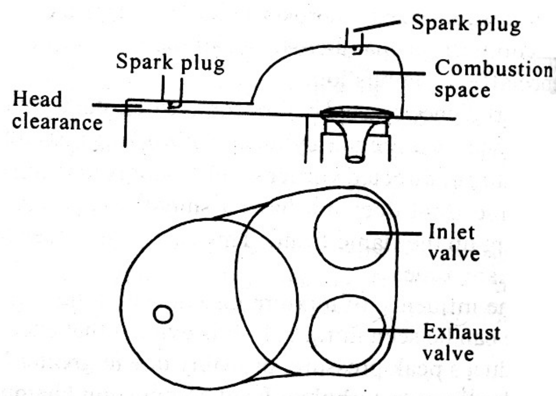


Fig. 3.5. ELEVATION AND PLAN OF RICARDO TURBULENT HEAD

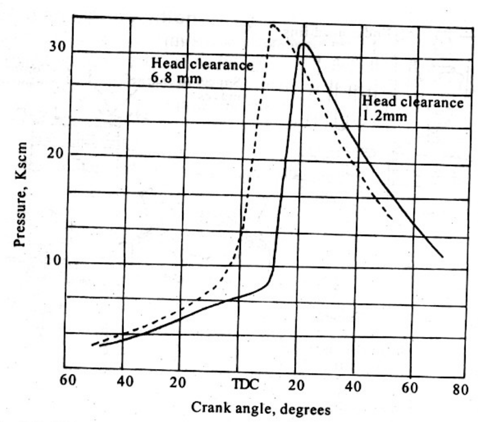


Fig. 3.6. INDICATOR DIAGRAMS SHOWING THE EFFECT OF THE VARIATION OF THE HEAD CLEARANCE IN THE RICARDO TURBULENT HEAD

Surface ignition

Surface ignition is the ignition of the air fuel mixture by one or more of the following

1. Incandescent (glowing hot) piece of carbon in the combustion chamber.
2. Overheated engine from improper operation of the cooling system.
3. Exhaust valve overheated by lean air fuel mixture supply (due to lean carburetor setting, clogged injector strainer, vacuum leak, stuck EGR valve etc).
4. Overheated spark plug central electrode (heat range too high).
5. Exhaust valve overheated by gas leakage (insufficient tappet clearance, weak valve spring, sticking valve etc).
6. Sharp edges in the combustion chamber (over heated threads on spark plug, edge of cylinder head gasket, sharp machined parts etc).
7. Excessively dry and hot atmospheric conditions or an air filter clogged.

Surface ignition may occur before the spark plug ignites the charge (preignition) or after normal ignition (post ignition). It may produce a single flame or many flames. Surface ignition may result in knock if it occurs after the spark.

Four stages of combustion in a CI engine

Herry Ricardo has investigated the combustion in a compression ignition engine and divided the same into the following four stages:

5. Ignition delay or delay period.
6. Uncontrolled combustion.
7. Controlled combustion.
8. After burning.

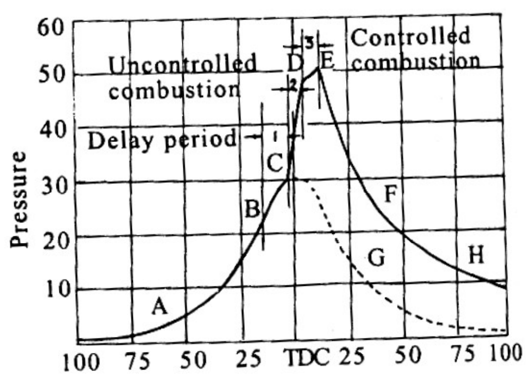


Fig: Pressure time diagram illustrating in a compression ignition engine.

1. Ignition delay
2. Uncontrolled combustion
3. Controlled combustion
4. After burning.

The details of these stages of combustion are given below:

Pressure V_s crank angle of a CI engine in a simplified from is shown in fig. The curved line ABCG represents compression and expansion of the air charge in the engine cylinder when the engine is being motored, without fuel injection. This curve is mirror symmetry with respect to TDC line. The curve ABCDEFH shows the pressure trace of an actual engine.

Delay period

In an actual engine, fuel injection beings at the point B during the compression stroke. The injected fuel does not ignite immediately. It takes some time to ignite. Ignition sets in at the point C. During the crank travel B to C pressure in the combustion chamber does not rise above the compression curve. The period corresponding to the crank angle B to C is called delay period or ignition delay (about 0.001 seconds).

During ignition delay, the following events take place. The injected spray enters the combustion chamber and slowly (at about 55 m/min) bores hole in the air mass, while the fuel particles are stripped away. Some of these particles are vapourized. Thus, the main body of the spray is surrounded by vapour liquid particle air envelope. In small combustion chambers, the spray body may impinge on the walls. Some of the impinged fuel may bounce off the surface, while the rest may glide on the walls. Vapourization of fuel particles tends to lower the compression pressure and temperature slightly. At the same time, the energy released in the preflame reactions tends to raise the pressure. Now in the outer envelope of the spray, ignition nuclei are formed. Mostly, the nuclei are cool flame reactions, on the verge of autoignition. By oxidation or cracking reactions, luminescent carbon particles are formed.

Uncontrolled combustion

At the end of the delay period i.e. at the point C, fuel starts burning. At this point, a good amount of fuel would have already entered and got accumulated inside the combustion chamber. This fuel charge is surrounded by hot air. The fuel is finely divided and evaporated. Majority of the fuel burns with an explosion like effect. This instantaneous combustion is called uncontrolled combustion. This combustion causes a rapid pressure rise.

During uncontrolled combustion the following take place. Flame appears at one or more locations and spreads turbulently, with glowing luminosity. Flame of low luminosity marks regions of vapourized fuel and air (premixed flame. Flames of higher luminosity mark regions of liquid droplets and air (diffusion flame). The initial spreading of non luminous and luminous flame arises from autoignition and flame propagation. This is the knock reaction with a high rate of energy release and correspondingly high rate of pressure rise.

Combustion during crank travel C to D is called uncontrolled combustion. This is because no control over this combustion is possible by the engine operator. Since this combustion is more or les instantaneous, it is also called rapid combustion.

If more fuel is present in the cylinder at the end of delay period, and undergoes rapid combustion when ignition sets in, the rate of pressure rise and the peak pressure attained will be greater. During this combustion the piston is around TDC, and is almost stand still. Too rapid a pressure rise and severe pressure impulse at this position of the piston will result in combustion noise called Diesel Knock.

The severity of the knock reactions is in proportion to the mass enflamed. The regions of premixed flame are probably hotter (and older) than the regions where liquid droplets are present. As such, the knock reaction may be propagated mainly in the low luminosity state of the flame.

The rate at which the uncontrolled combustion takes place will depend upon the following:

1. The quantity of fuel in the combustion chamber at the point C. This quantity depends upon the rate at which fuel is injected during delay period and the duration of ignition delay.
2. The condition of fuel that has got accumulated in the combustion chamber at the point C.

The rate of combustion during the crank travel C to D and the resulting rate of pressure rise determine the quietness and smoothness of operation of the engine.

Controlled combustion

During controlled combustion, following thing happen. The flame spreads rapidly (but less than 135 m/min), as a turbulent, heterogeneous or diffusion flame with a gradually decreasing rate of energy release. Even in this stage, small autoignition regions may be present. The diffusion flame is characterized by its high luminosity. Bright, white carbon flame with a peak temperature of 2500^{o} C is noticed. In this stage, radiation plays a significant part in engine heat transfer.

During the period D to E, combustion is gradual. Further by controlling the rate of fuel injection, complete control is possible over the rate of burning. Therefore, the rate o pressure rise is controllable. Hence, this stage of combustion is called Gradual combustion or Controlled combustion.

The period corresponding to the crank travel D to E is called the period of controlled combustion.

The rate of burning during the period of controlled combustion depends on the following:

1. Rate of fuel injection during the period of controlled combustion.
2. The fineness of atomization of the injected fuel.
3. The uniformity of distribution of the injected fuel in the combustion chamber.
4. Amount and distribution of the oxygen left in the combustion space for reaction of the injected fuel.

At the point E, injection of fuel ends, the period of controlled combustion ends at this point. When the load on the engine is greater, the period of controlled combustion is also greater.

During controlled combustion, the pressure in the cylinder may increase or remain constant or decrease. Usually during this period, the combustion is more or less at constant pressure (on a PV diagram) because the downward movement of the piston (i.e. increase in volume) compensates for the effect of heat release and the consequent pressure rise.

After burning

At the last stage, i.e. between E and F the fuel that is left in the combustion space when the fuel injection stops is burnt. This stage of combustion is called After burning (burning on the expansion stroke). In the indicator diagram, after burning will not be visible. This is because the downward movement of the piston causes the pressure to drop inspired of the heat that is released by the burning of the last portion of the charge.

Increasing excess air, or air motion will shorten after burning i.e. reduce the quantity of fuel that may undergo after burning).

Detonation in SI engine and diesel knock

Detonation in the spark ignition engine and knock in the compression ignition engine have essentially the same basic cause, that is, compression ignition followed by a rapid pressure rise. However, in SI engine the reaction and compression ignition is in the last part of the charge to burn, while in CI engine it occurs in the first part of the charge to burn. Comparison of time of detonation in SI engine and knock in CI engine with respect to pressure V_s crank angle can be seen in fig.

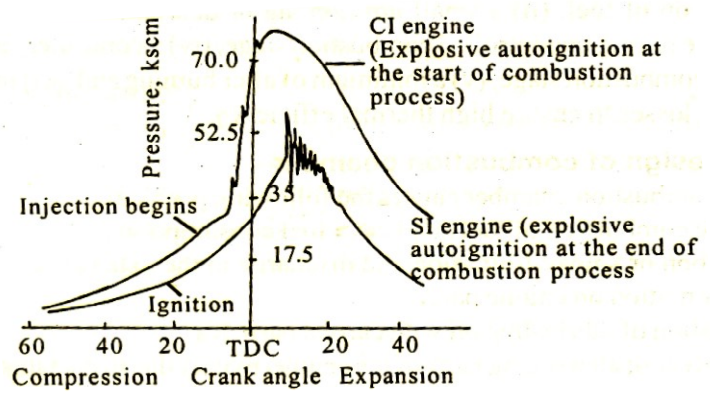


Fig. Detonation in S.I. Engine and C.I. Engine

In order to avoid detonation in the SI engine, it is necessary to prevent compression ignition from taking place at all. In the CI engine, the earliest possible compression ignition is necessary. As such, many of the methods of reducing detonation are exactly opposite for the two types as indicated below,

Characteristics tending to reduce detonation or diesel knock:

CHARACTERISTIC	SI ENGINES	CI ENGINES
Ignition temperature of fuel	High	Low
Compression ratio	Low	High
Inlet temperature	Low	High
Inlet pressure	Low	High
Combustion chamber temperature	Low	High
Speed of engine	High	Low
Cylinder size	Small	large

HOMOGENOUS MIXTURE FORMATION IN SI ENGINES

The fuel and air are homogeneously mixed together in the carburetor. The homogenous mixture enters the engine through the intake manifold, where it mixes with residual gases and is compressed. The combustion is initiated by a spark at the end of the compression stroke. A turbulent flame develops following the ignition, propagating through the premixed fuel-air charge. The flame front spreads with a certain velocity. In the homogenous mixture, the fuel and oxygen molecules are uniformly distributed. The flame from propagates through the gas and is quenched at the walls. Heat transfer and diffusion of burning fuel contribute to the flame movement in the adjacent layers of unburned mixture. The velocity at which the flame moves is called the flame velocity.

If the homogeneous mixture is such that it has a balanced ratio of air and fuel or in other terms chemically correct mixture or exact amount of air for burning all the fuel completely, then it is known as a stoichiometric mixture. The A/F ratio in this case is called stoichiometric air to fuel ratio. If the homogeneous mixture is such that it has more (excess) air than what is required to completely burn the fuel, then it is known as a lean mixture. If the homogenous mixture is such that it has less air than what is required to burn the fuel completely, it is known as a rich mixture.

The equivalence ratio ϕ is defined as the ratio of actual fuel to air ratio to the stoichiometric fuel to air ratio.

Always $0 < \phi < 1 \rightarrow$ lean mixture $\phi = \dfrac{(F/A)_{actual}}{(F/A)_{stoich}}$

$\phi = 1 \rightarrow$ stoichiometric mixture

$\phi > 1 \rightarrow$ rich mixture

The excess air factor λ is defined as the ratio of stoichiometric fuel to air ratio to the actual fuel to air ratio.

$$\therefore\ \lambda = \frac{1}{\phi}; \quad \therefore\ \lambda = \frac{(F/A)_{stoich}}{(F/A)_{actual}}$$

Always $0 < \lambda < 1 \rightarrow$ rich mixture

$\lambda = 1 \rightarrow$ stoichiometric

$\lambda > 1 \rightarrow$ lean mixture

In the SI engine maximum flame speed is achieved when mixture is slightly rich, i.e., ϕ lies between 1.1 and 1.2. The flame speed can be increased by turbulence.

Combustion stages in SI engine

A typical pressure-crank angle (P-θ) diagram showing the engine compression, combustion and expansion process in an ideal SI engine is shown in figure.

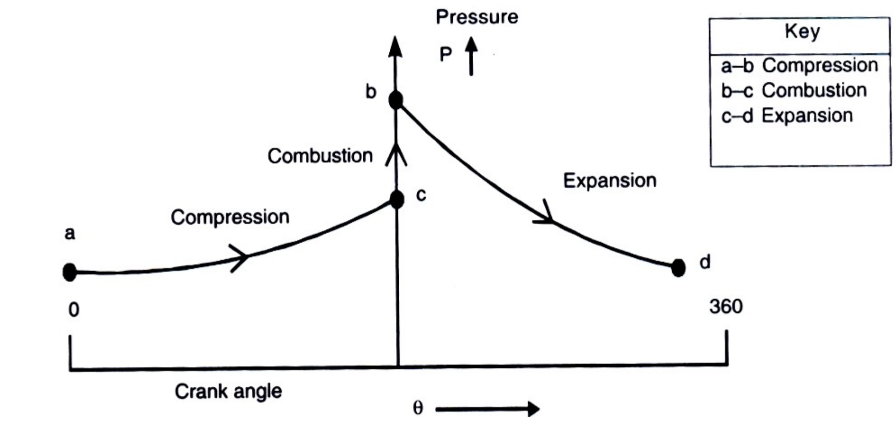


Figure: Ideal P-θ diagram for an SI engine

In an ideal engine the entire pressure rise during combustion takes place at constant volume, i.e., at TDC. However, in an actual engine, constant volume combustion is not possible.

The SI engine process has three stages. Refer figure for the actual P-θ diagram of an SI engine.

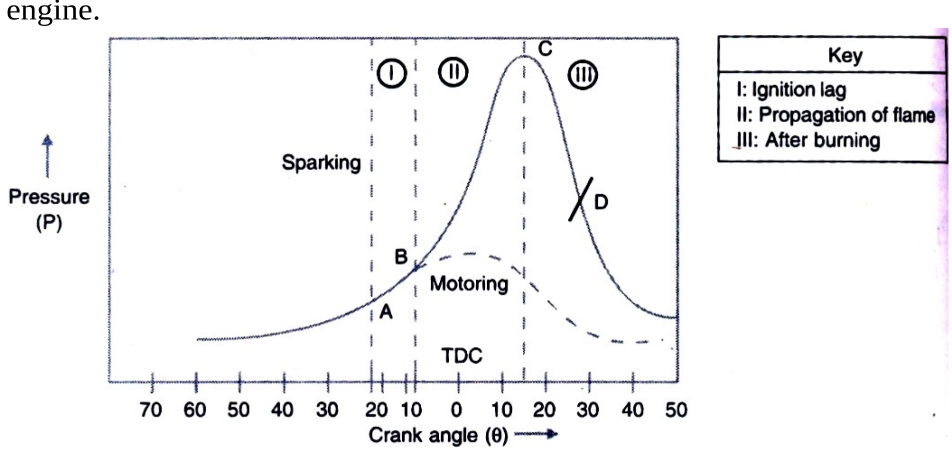


Figure: Actual P- θ diagram

The actual P-θ diagram shows the pressure variation in the engine during actual combustion process. Point A is the point of passage of spark (20° BTDC), point B is the point at which beginning of pressure rise can be detected (8° BTDC) and point C is the attainment of peak pressure. Thus AB is the first stage, BC is the second stage and CD is the third stage of combustion.

The first stage (A-B) is called the ignition lag. This is a preparation phase in which development of the flame nucleus takes place. This is a chemical process dependent on factors such as temperature, pressure, nature of fuel and proportion of exhaust residue.

The second stage (B-C) is a physical process of propagation of flame in the combustion chamber. The starting point of the second stage is when the first measurable rise of pressure is seen on the indicator diagram. At this point the line of combustion departs from the compression line. This can be seen by the deviation of the motoring curve as shown in figure. During second stage the flame propagates at constant velocity. The rate of heat release depends upon mixture composition, turbulence intensity and the reaction rate. Also the rate of pressure rise is proportional to heat release rate as piston is at TDC and combustion chamber volume is constant.

The third stage (C-D) is the after burning phase. The starting point of this phase is the instant at which maximum pressure is reached on the indicator diagram (Point C). The flame velocity reduces and rate of combustion becomes low as a result. No pressure rise is possible in this stage as the piston moves away from TDC and expansion stroke commences.

Disintegration of fuel jet in CI engine

In CI engines, only air is compressed by a piston at high compression ratios between 17:1 to 22:1, thereby raising its temperature and pressure. Fuel is injected by jets into this highly compressed air in the combustion chamber. Here the fuel jet disintegrates into a core of fuel surrounded by a spray envelope of air and fuel particles. This spray envelope is shown in figure and is created by atomization and vaporization of fuel.

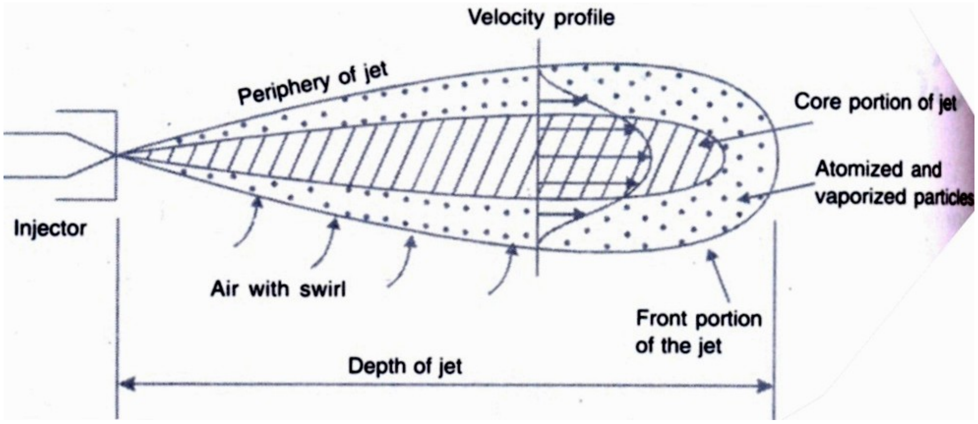


Figure: Disintegration of fuel jet in CI engine

The turbulence of air in the combustion chamber passing across the jet separates the fuel particles from the core. A mixture of air and fuel is thus formed at the periphery of jet. The droplets of fuel vaporize by absorbing latent heat from surrounding air. As soon as the fuel-air mixture reaches the auto-ignition temperature the ignition takes place. There is a certain delay period before the ignition takes place. As the fuel droplets are not uniformly distributed throughout the combustion chamber, the fuel air mixture is heterogenous.

Four flame structures in CI engines

The phenomenon of heat release rate is essential to understand the flame structure in CI engines. Heat release rate is defined as the rate at which chemical energy of the fuel is released by the combustion process. It can be calculated from the cylinder pressure crank angle (P-θ) curve.

High speed photography analysis of CI combustion chamber reveals following features of the CI combustion:

(1) Fuel sprays: The fuel droplets reflect light and define extent of atomization in the combustion chamber before vaporization.
(2) Premixed flame: This region is of very low luminosity, which can be made visible by adding copper additives to the fuel. The flames are normally bluish-green in colour.
(3) Diffusion flame: This region is bright due to burning high temperature carbon particles in the flame. The flame appears as yellow or orange in colour.
(4) Rich flame: This is a brown coloured flame. This is a fuel rich region surrounded by a diffusion layer and is the reason for soot production.

The flame is produced by first penetration of fuel in the hot air and its subsequent rapid vaporization and then ignition. The flame then spreads rapidly throughout the spray to the spray tip. The flame eventually engulfs the entire spray.

The white-yellow flame is near the injector. The rich region is at the far end combustion chamber. Spray tips are al so affected by the swirl in the combustion chamber.

Physical and Chemical properties of SI engine fuels

Gasoline (petrol) is used as the main SI engine fuel. It is a blend of several low boiling paraffins in varying proportions. Following are some of its important properties.

(1) Volatility: This is one of the important properties of gasoline, which gives information about the fraction of fuel that evaporates at room temperature. Volatility decides the suitability of gasoline for use in SI engines. Since gasoline is a mixture of different hydrocarbons, volatility depends on fractional composition of fuel. ASTM methods are used to measure volatility of fuels. ASTM is the acronym for American Society for Testing Materials. Fuel volatility is measured by distillation and boiling of fuel.

(2) Cold starting and warm up: It is necessary that some part of gasoline should vaporize at room temperature for easy starting of the engine. Low distillation temperatures for gasoline are desired for engine warm up.

(3) Operating range performance: In order to obtain good vaporization of gasoline, low distillation temperatures are preferred in engine operating range. Better vaporization provides uniform distribution of fuel to cylinders.

(4) Crankcase dilution: Liquid fuel in the cylinder causes loss of lubricating oil by washing off the cylinder walls. This deteriorates quality of lubricant and increases wear and tear of engine. Thus, at high temperatures, entire gasoline should be vaporized before combustion.

(5) Vapour lock characteristics: High volatility rates of gasoline can stop fuel flow to the engine by setting up a vapour lock, i.e., obstruction in fuel passages. Thus, due to this tendency high boiling temperature hydrocarbons are preferred. However, a compromise has to be made between this requirement and other contrasting requirements (properties 1-3).

(6) Antiknock properties: Abnormal burning or knocking can cause damage to the engine due to high energy releases. Thus, fuel should resist the tendency of knocking and this property is called as antiknock property. The antiknock property of fuel depends on the self-ignition characteristics of the mixture of air and fuel. Higher the antiknock property for a fuel, higher is its thermal efficiency and power output.

(7) Gum deposits: Reactive hydrocarbons and impurities in fuels oxidize upon storage and form gummy solid substances. This gum will deposit on valves, piston rings and manifolds, damaging them. The gum will choke carburetor jets. Thus, gum forming of tendency of gasoline should be low upon exposure to sunlight and oxygen.

(8) Sulphur content: The sulphur content of gasoline should be low. This is due to the fact that, sulphur is corrosive in nature. It can corrode fuel lines, carburetors and injection pumps. It may also combine with moisture to form sulphuric acid, which can corrode metallic parts of the engine. Presence of sulphur reduces self-ignition temperature and promotes knocking. Thus, sulphur content is undesirable.

Physical and Chemical properties of CI engine fuels

Diesel is used as the main CI engine fuel. It is mostly a blend of paraffins like cetane and also some napthalenes. Following are some important properties of CI engine fuels.

(1) Knock characteristics: Knocking occurs in CI engines due to the ignition lag in the combustion of the fuel. The ignition lag is the time period between injection of fuel and actual burning of fuel. The lag causes accumulation of fuel, which ignites suddenly causing violent energy release and pressure rise leading to knocking. A good fuel should have short ignition lag i.e., it should ignite quickly.

(2) Volatility: The fuel should be sufficiently volatile in the operating range of temperature to produce good mixing and combustion.

(3) Starting characteristics: The fuel should help in starting the engine easily. This requirement demands high volatility to form combustible mixture. The self ignition temperature should be low.

(4) Smoking and odour: The fuel should not promote smoke or odour in the engine exhaust. The combustion should be complete and no fuel should be left unburned.

(5) Corrosion and wear: The presence of sulphur and other solid impurities in the fuel should be low. Thus the fuel should not cause corrosion and wear of the engine components.

(6) Viscosity: Viscosity of fuel should be lower. Viscosity depends on temperature.

The fuel should be able to flow through the fuel system at low temperature easily.

(7) Handling ease: The fuel should have a high flash point and fire point, i.e., it should be safe for handling.

UNIT – III

COMBUSTION MODELLING

Two schemes of available in solution algorithm

Two schemes are available: (1) Implicit solver (2) Explicit solver. In implicit solver, the variables are interconnected and a system of simultaneous equations is formed. The computation is thus laborious and time consuming. In explicit solver, there are no relations between the variables. The computation speed is thus higher. However, stability conditions limit the time step for solving.

Navier-Stoke′s equation in a compact form for fluid flow

The Navier-Stoke′s equations can be written in compact form as,

$$\frac{Df}{Dt} \equiv \frac{\partial(\delta t)}{\partial t} + \frac{\partial}{\partial x_j}(\delta u_j f)$$

Here, f = internal energy; δ = density; t = time; u_j = x component of velocity

Arrhenius equation

Arrhenius equation $Rf = A\delta^2 x_f^a \; x_{0x}^b \; \exp\left(-\frac{E_A}{RT}\right)$

where, Rf = rate of burning of fuel
A = Arrhenius constant
E_A = activation energy
δ = density of fuel
a, b = coefficients
x_f = unburned fuel mass fraction
x_{0f} = unburned oxygen mass fraction
R = universal gas constant
T = temperature of reaction

Zero Dimensional computer models

The numerical models range from the simplified phenomenological model to the comprehensive fluid dynamics based model to study the engine flow and combustion process in details.

Zero-dimensional: Uses a simple model to estimate the residual gas fraction.

One-Dimensional: Uses the method of characteristics to study the flow exchange process within the entire engine system.

Multi-dimensional: Includes the flow, spray and combustion model development and applications using codes such as KIVA, STAR-CD and VECTIS.

Uses of modeling process of IC engines

Modeling is a technique used in engineering to develop, using assumptions and equations, a model to analyze the critical features of an entity or process. In this chapter we shall focus on the modeling of IC engines, which has the following benefits.

(1) Modeling helps to understand completely various engine processes under study including chemical and physical phenomena contributing to the processes.
(2) Modeling identifies key controlling variables in engine processes and provides guidelines for development effects thereby reducing dependence on experimental methods and related cost.
(3) Modeling helps to predict engine behavior over a wide range of operating conditions to help establish trends.
(4) Modeling provides a rational basis for design activities.
(5) Modeling helps identify critical areas of engine performance and provides an opportunity to the engine designer to play with variables and establish an optimum configuration, which is not possible with experimental methods.

Types and uses of Engine models

These models are used to study thermodynamics, fluid flow, heat transfer, combustion, lubrication and pollutant formation in IC engines. For processes that govern engine performance and emissions two models are important:

(1) Thermodynamic model
(2) Fluid dynamic model

Thermodynamic models are energy conservation based, requiring additional inputs about geometric features.

Fluid dynamic models are multidimensional models due to their ability to provide geometric information about flow fields based on momentum equations.

Before modeling any process, the objective of the model should be clearly defined. Empirical relations may be included with some experimental inputs.

Fluid mechanics based models of Engine flow

The purpose of fluid mechanics based models is to determine the details of flow field within engines and heat transfer and combustion process dependent on the flow fields. The Navier-Stoke's equations are the governing equations for fluid flow and they are solved numerically. The flows through intake and exhaust manifolds are considered as 1-D unsteady flows while the flows inside the cylinder are considered as 3-D unsteady flows. The output of fluid mechanics model describes gas flow patterns and information about pressure, temperature and velocity fields, in the engine.

The principal components of an engine flow model are:

(1) Equations to describe fluid flow (usually partial differential equations)
(2) Discretization strategies to convert partial differential equations to algebraic form.
(3) Solution algorithm to solve the algebraic equations.
(4) Computer software to translate the solution in graphic or tabular format.

The Navier-Stoke's equations can be written in compact form as,

$$\frac{Df}{Dt} \equiv \frac{\partial(\delta t)}{\partial t} + \frac{\partial}{\partial x_j}(\delta u_j f)$$

Here, f = internal energy; δ = density; t = time; u_j = x component of velocity.

Turbulence phenomena also need to be accounted for in flow models of engine. Two approaches, (1) full field modeling (FFM) and (2) large-eddy simulation (LES) are available for that purpose. The difference between models is in their definition of turbulence. In FFM, the turbulence is the derivation of flow at any instant from the average over many cycles of flow. It accounts for cycle by cycle flow variations. In LES turbulence is defined in terms of variations of a local average. It does not account for cycle by cycle flow variations as the calculations are done for a single cycle.

K-∈ Turbulence model

This is a widely used turbulence model. This assumes a Newtonian relationship between the turbulence stresses and mean strain rates and computes the turbulent viscosity from local turbulent kinetic energy (k) and the dissipation rate (∈). The governing equation is:

$$\frac{Dk}{Dt} = \delta(p - \in) - \frac{\partial}{\partial x} J_k$$

Here, J_k = diffusive transport

k = turbulent kinetic energy

∈ = dissipation rate

t = time

δ = density

p = rate of turbulence production per unit mass

The relation between k and ∈ is given by:

$$\in \alpha \frac{k^{3/2}}{l}$$

Where l = turbulent length.

Numerical computing mesh with diagram.

The requirements of a computing mesh are:

(1) It adequately fits the geometry of the entity being modeled.
(2) It allows control of the local resolution to obtain maximum accuracy with a given number of grid points.
(3) Each interior grid point should be connected to same number of neighboring points.

The first requirement is due to need for accuracy in modeling engine geometry. The second requirement is due to need of reducing computing time by optimizing grid points. The third requirement is due to the need for rectangular well defined mesh. In early engine models, a polar-cylindrical frame was used to define the coordinate system for the grid. This was not accurate enough to model engine geometry. Nowadays, flexible “body fitting” coordinate frame is used whose surface is as per the geometry of the entity to be modeled. Figure shows the flexible mesh for a diesel engine combustion chamber.

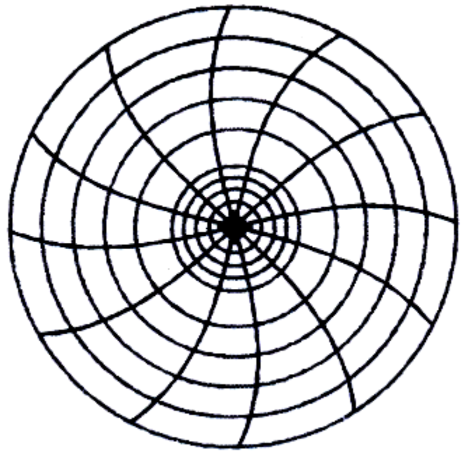


Figure: Flexible mesh for a diesel engine combustion chamber.

Graphical output diagrams of spray simulation with experimental results and diesel fuel spray combustion model.

The solution of equations provides a large amount of information on many fluid flow and state variables, which is generated with each calculation. The processing and presentation of this data is an enormous task. Fluid flow results are presented in terms of gas velocity vectors at each grid point of the mesh in appropriately selected planes. Arrows are used to indicate magnitude and direction of each vector. Figure shows the graphical output from various engine models.

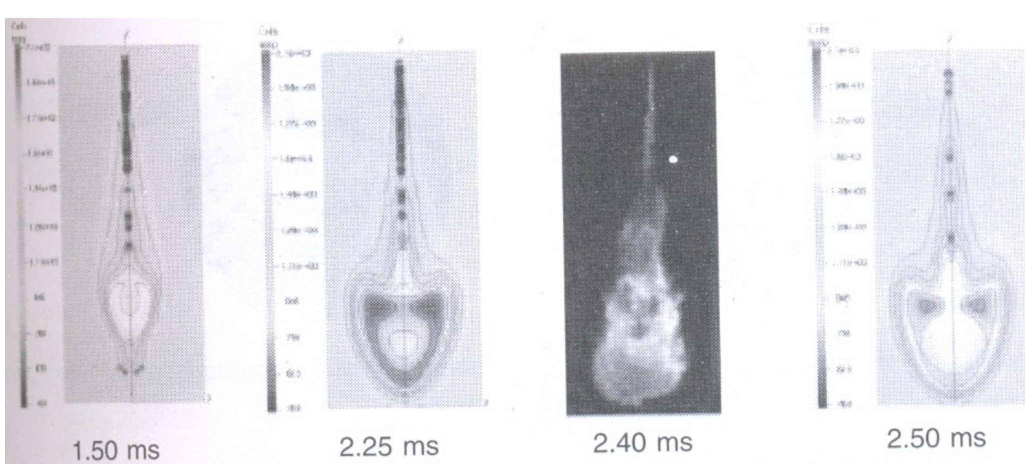


Figure: Comparison of spray simulation with experimental results.

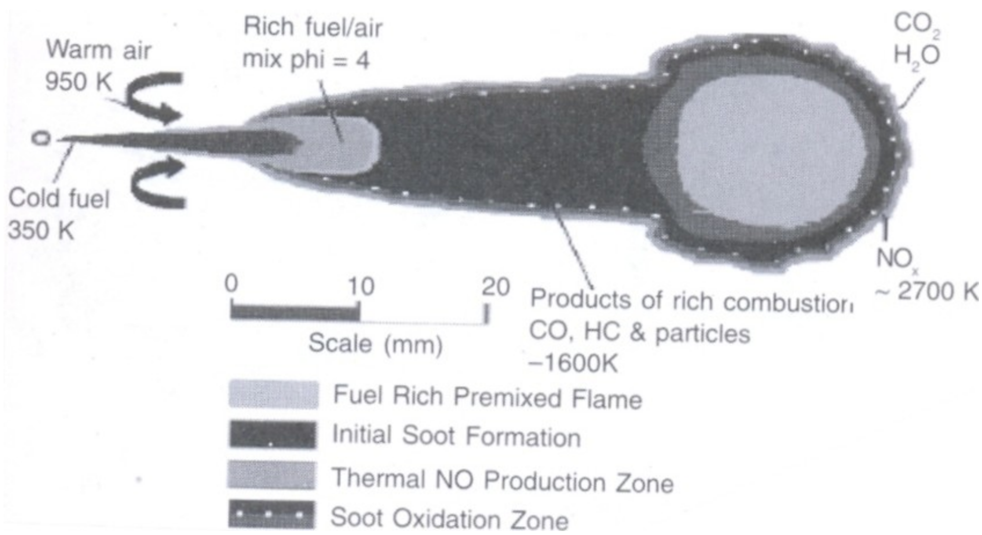


Figure: Diesel fuel spray combustion model.

Combustion modeling with governing equations

For modeling reactive flows like combustion flows, simplified reaction schemes are used due to computing restrictions. Also, reaction schemes are available for only select species like methane, propane or butane. Scheme for complex molecules are not available. Chemical kinetics plays a major role in emission formation and is one area which is the least understood by IC engine designers. The basis of modeling combustion reaction rates is the simple Arrhenius equation.

$$Rf = A\delta^2 x_f^a \; x_{0x}^b \; \exp\left(-\frac{E_A}{RT}\right)$$

where, Rf = rate of burning of fuel
A = Arrhenius constant
E_A = activation energy
δ = density of fuel
a, b = coefficients
x_f = unburned fuel mass fraction
x_{0f} = unburned oxygen mass fraction
R = universal gas constant
T = temperature of reaction

The predictions from this equations match experimental data with reasonable accuracy. The constants A, a and b can be used to fit the model data to the experimental result. This is called constant adjusting. The Arrhenius equation does not account for chemical kinetics or turbulence effects. Some mixing models are available which assume turbulent mixing is the reaction rate controlling process. The reaction rate is inversely proportional to the turbulent mixing time, which is equal to $k/\in$. Combustion models of SI engines can also predict flame speeds.

CFD simulation of Engine process.

The numerical models range from the simplified phenomenological model to the comprehensive fluid dynamics based model to study the engine flow and combustion process in details.

Zero-dimensional: Uses a simple model to estimate the residual gas fraction.
One-Dimensional: Uses the method of characteristics to study the flow exchange process within the entire engine system.
Multi-dimensional: Includes the flow, spray and combustion model development and applications using codes such as KIVA, STAR-CD and VECTIS.

It has proven to be successful to combine the use of two or more of these models. For instance, the zero dimensional model may be used to estimate initial values for a multidimensional simulation. The one-dimensional model, which uses the method of characteristics, gives higher accuracy than the zero-dimensional model but is more difficult to use. Multidimensional models, such as the spray and combustion models, give a detailed picture of the engine processes of interest for improving engine performance and reducing emissions. Multidimensional codes are being used for the development of engine technology necessary to meet new and more stringent emissions and performance standards. An example of such codes is the KIVA family of codes developed at the Los Almos National Laboratory, which is specifically designed for performing internal combustion engine calculations. Modeling of internal combustion engines, in particular direct-injection diesel and gasoline engines, present many challenges: for example, moving boundaries, two-phase chemically reacting turbulent flows. Thus, even though the current predictive capability is good, the development of improved models is necessary to increase reliability of these codes. In addition to the KIVA code, several commercial engine simulation packages including STAR-CD and VECTIS are available. The physical and chemistry submodels are being implemented into the above package to improve the capability of predicting the complex engine processes.

Heat release rate

Studies of photographs of diesel engine combustion, combined with analyses of engine cylinder pressure data, have led to a widely accepted descriptive model of the compression-ignition engine combustion process. The concept of heat-release rate is important to understanding this model. It is defined as the rate at which the chemical energy of the fuel is released by the combustion process. It can be calculated from cylinder pressure versus crank angle data, as the energy release required to create the measured pressure, using the techniques described in section. The combustion model defines four separate phases of diesel combustion, each phase being controlled by different physical or chemical processes. Although the relative importance of each phase does depend on the combustion system used, and engine operating conditions, these four phases are common to all diesel engines.

Different governing equations for the thermodynamic models with notation

The governing equations for thermodynamic models are: (1) Conservation of mass (2) Conservation of energy.

Thermodynamic models which are based on these two equations are used to model an engine region as an open thermodynamic system where free flow of engine mass and energy takes place. For such models the gas composition is assumed to be uniform and time and crank angle is sued as an independent variable. Examples of thermodynamic models are modeling of intake and exhaust manifold or engine cylinder.

Conservation of mass can be written in equation form as.

$$\dot{m} = \sum_{j} mj$$

Where, $\dot{m}$ = the rate of change of mass of the system

$\sum mj$ = the sum of mass flow in add out of the system

Usually, mass flow into the system is considered positive and mass flow out of the system is considered negative. The fuel fraction ‘f’ can be defined as ratio of mass of fuel to the total mass in the system.

$$\therefore f = \frac{mf}{m}$$

$$\therefore \dot{m}f = \text{fuel flow rate} = \frac{d}{dt}(mf) = \sum_j \dot{m}_j f_j$$

The fuel to air equivalence ratio = ϕ

$$\phi = 1/\lambda = \frac{1}{\text{excess air ratio}}$$

The equivalence ratio is related to the fuel fraction by the equation,

$$\phi = \frac{1}{(F/A)_s} \cdot \frac{f}{(1-f)^2}$$

Where, $(F/A)_s$ is the stoichiometric fuel to air ratio

Conservation of energy is the first law of thermodynamics and can be written as,

$$\dot{E} = \dot{Q}_w - \dot{W} + \sum_j \dot{m}_j h_j$$

Where,

$\dot{E}$ = energy rate of change

$\dot{Q}_w$ = heat transfer rate to and from the system

$\dot{W}$ = work transfer rate out of the system across the boundary = PV

In this case, piston produces the work cross the engine cylinder boundary.

$\sum m_j h_j$ = heat energy produced by combustion of fuel

The heat produced during combustion depends on enthalpies of various reactants.

The equation of state is PV = mRT.

$$\therefore P = \delta RT$$

If the internal energy is 'u' and enthalpy is 'h' then,

$$E = \frac{d}{dt}(mu) = \frac{d}{dt}(mh) - \frac{d}{dt}(PV)$$

For simplification of thermodynamics models, effects of variable composition of gas, dissociation phenomena are neglected.

Thermodynamic models may be multizonal, i.e., having multiple zones to model specific areas. For example is SI engines, a two zonal thermodynamic model may be used for combustion, which includes two zones; one for unburned mixture and the other for the combusted region. We shall now study some specific thermodynamic models.

UNIT – IV

ADVANCES IN IC ENGINES

LHR engine

To increase the efficiency of internal combustion engines and generate higher chamber temperatures, low heat rejection concepts are being investigated.

In gasoline engines, the thermal insulation will increase the wall temperature which will lead to unwanted detonation. Because of this, insulation of the combustion chamber could be done only in diesel engines.

This has two important purposes; one to reduce the size of the coolant system and second to increase the exhaust energy available for turbo charging and thereby increasing the power and efficiency.

Stratified charge

Stratification of charge means to provide variable air to fuel ratio mixtures at different places in the combustion chamber of SI engines. Particularly, rich mixture is provided in rest of the cylinder in layers with varying A/F ratio. The segregation in mixture can be obtained by carburetor or fuel injection. The rich mixture near the spark plug forms a strong flame, which burns the lean mixture completely. Thus, SI engines were able to run on lean mixtures especially at part load conditions, which was not possible earlier. This stratified charge concept saved precious saved precious fuel and eliminated emissions resulting from the additional fuel. The emissions of NO_x, HC and CO were reduced due to stratification of charge.

Advantages of stratified change engine

(1) Possibility of explosion near the spark plug reduced due to presence of lean mixture surrounding the rich mixture.
(2) Good part load efficiency
(3) Resistance to knocking is high
(4) Can burn low grade fuels.
(5) Emissions are less (NO_x and CO)
(6) Easy to start
(7) Can burn various fuels (multi-fuel capability)

(8) Thermodynamic efficiency is high for a stratified charge engine.

Disadvantages of stratified charge engine

(1) Complex design of the engine due to variable A/F ratio of charge.
(2) The stratified engine is noisy in operation.
(3) Throttling losses are high.
(4) Possibility of misfire of charge due to charge stratification.
(5) Possibility of higher HC (hydrocarbon) emissions due to incomplete combustion of charge.

Recent advances in IC engines

In this chapter we shall focus on some recent developments in the field of IC engines. A lot of research is constantly being done in the field and newer engines are being developed. The emphasis is on promoting environment friendliness and performance characteristics of the engines. This is achieved by reducing emissions and reducing the dependence of engines on traditional fossil fuels. Several alternative fuels have been developed for IC engines. Newer pollution reduction techniques are being used, e.g. higher loading catalytic converters for IC engines. Considerable modifications have been done to fuel injection systems including the development of Direct Injection technique. Newer engines being developed are as follows:

(1) Dual fuel engine
(2) Variable compressions ratio engine
(3) Free piston engine

Advantages of dual fuel engine

(1) It can use cheaper gaseous fuels like natural gas, producer gas, biogas, etc.
(2) Exhaust of engine is relatively clean, so air pollution is avoided.
(3) Wear and tear of engine is less and lubricant consumption is low.
(4) Changeover of fuels is quick.
(5) Waste energy is utilized in these engines due to combustion of waste gaseous fuels like biogas.
(6) Duel fuel engine can be used to produce synthetic gas (e.g., $CO + H_2$) by combustion of gaseous fuels (e.g., Methane (CH_4))
(7) Dual fuel engines are best suited for agricultural and automobile applications.
(8) Operation is flexible.

Surface ignition hotspots – combustion-chamber deposits

Surface ignition is ignition of the fuel air charge by any hot surface other than the spark discharge prior to the arrival of the normal flame front. It may occur before the spark ignites the charge (preignition) or after normal ignition (postignition).

Lean burn engine

Lean burn engine is a layout of otto cycle engine designed to permit the combustion of lean air fuel mixtures and to obtain simultaneously low emission values as well as high fuel economy.

Generally, all SI engines operate with mixtures nearby stoichiometric air fuel ratio about 12:1 to 16:1. But the lean burn engine is designed to operate effectively in the air fuel ratio 14:1 ~ 16:1 to 20:1 ~ 22:1.

By optimizing the compression ratio, combustion chamber shape, ignition system, lean limit can be extended. When these parameters are successfully optimized, the engine is referred to as a lean burn engine.

Working of a stratified changed engine with diagram

Several designs of stratified charge engines are available including Ricardos system, Hessleman system, Broderson system, Volkswagen system, Texaco system and so on. We shall only study typical engine and that is the Volkswagen PCI stratified charge engine.

Refer figure of the stratified charge engine.

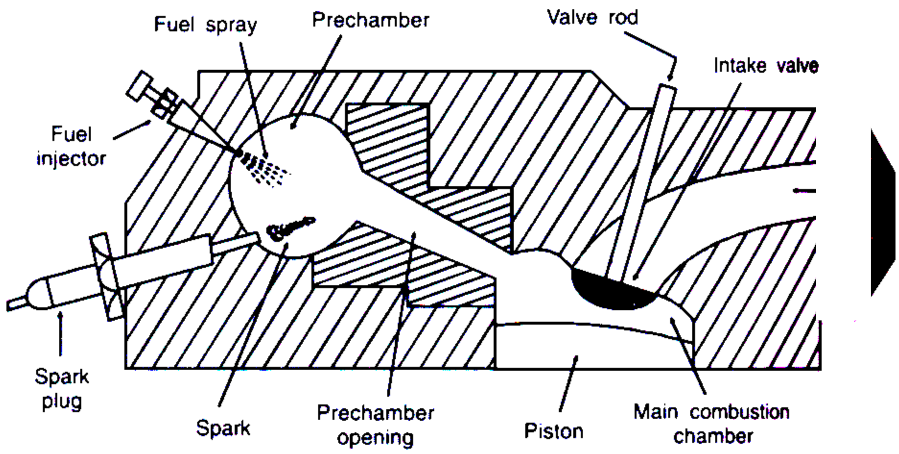


Figure: Stratified Charge Engine (Volkswagen PCI)

The engine consists of a spherical unscavanged prechamber consisting of approximately 25-30 percent of compression volume. It is linked to the main combustion chamber by a relatively large flow transfer passage. The main combustion chamber contains no slant surfaces and is disc shaped. A slight swirl to the charge is included by the intake port. The injection nozzle and spark plug are arranged in sequence in the flow direction, so that the spark plug receives a mixture produced by blending of the incoming air with fuel dispersed in air. This avoids over enrichment of charge at the spark plug.

The total fuel volume is injected partly in the main combustion chamber intake manifold and rest in the prechamber. There is a rich mixture maintained near the spark plug for all operating conditions. Load regulation is achieved primarily by adjusting mixture strength introduced in the main combustion chamber.

One major advantage of this system is that unlike other charge stratifications systems, the fuel injection timing is not required to be varied. This simplifies the design of the engine. The use of prechamber means that there is some loss of thermal efficiency due to throttling. Also, costly sophisticated fuel injection is required to give good distributed charge at high loads.

Construction of a duel fuel engine

The dual fuel engine was developed from the diesel engine to take advantage environment friendly fuels like biogas, producer gas and natural gas. The idea was to run the engine on conventional fuel as well as a second (alternative) fuel. These engines use high compression ratio and run with high air to fuel ratio. The gaseous fuel is ignited by injecting pilot diesel fuel into the heated air fuel mixture. The gaseous fuel has self ignition temperature higher than that of diesel.

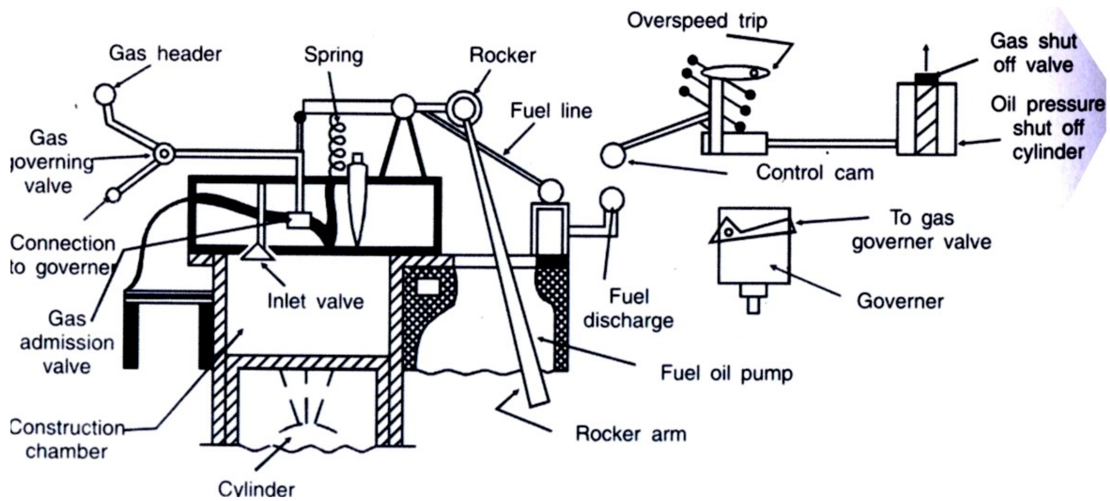


Figure: Dual-Fuel Engine and Governor Assembly

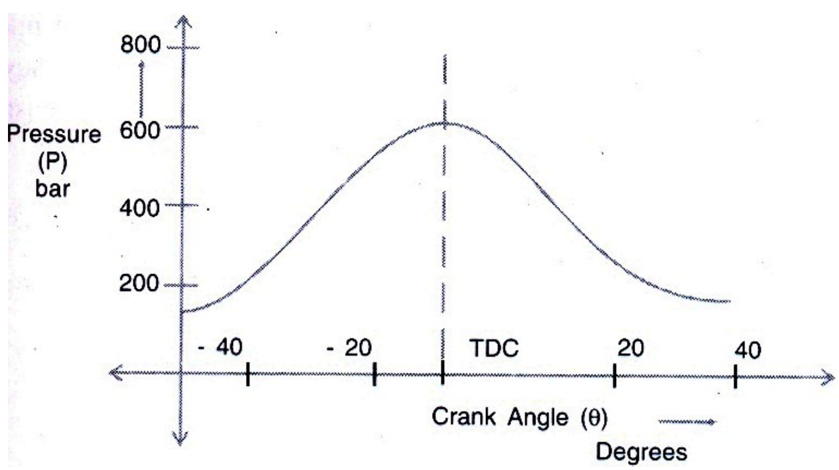


Figure: P-θ diagram for dual-fuel engine

Nowadays shortage of liquid fuels and availability of gaseous fuels have led to increased research on dual fuel engines. These engines can use conventional fuels in case of emergency and switchover from alternative fuel to conventional fuel is fast and easy. Refer figure for the construction of the engine and figure for the P-θ diagram.

The working of the dual fuel engine is similar to diesel engine except that there is a separate gas admission valve to admit gas along with air in the engine. The gas valve always opens after the air valve and closes before the air valve. The simultaneous flow of air gas form a homogenous mixture which burns easily and smoothly. The mixture is ignited by pilot fuel. The governor and fuel injection systems have been modified to take care of part loads. A separate gas valve operated by a cam is provided and is kept open for 140 degree rotation. The air:gas ratio changes from 20:1 at full load to 40:1 at part load. However it is difficult to burn lean mixtures and unburned gases are sometimes exhausted. Conventional fuel injectors incorporating nozzles are used for metering the fuel. Plunger design is slightly modified from the base engine design. In dual-fuel engine, the fuel governor regulates gas and diesel simultaneously. Governor linkages are arranged for manual transfer from one fuel to another.

dual fuel engine

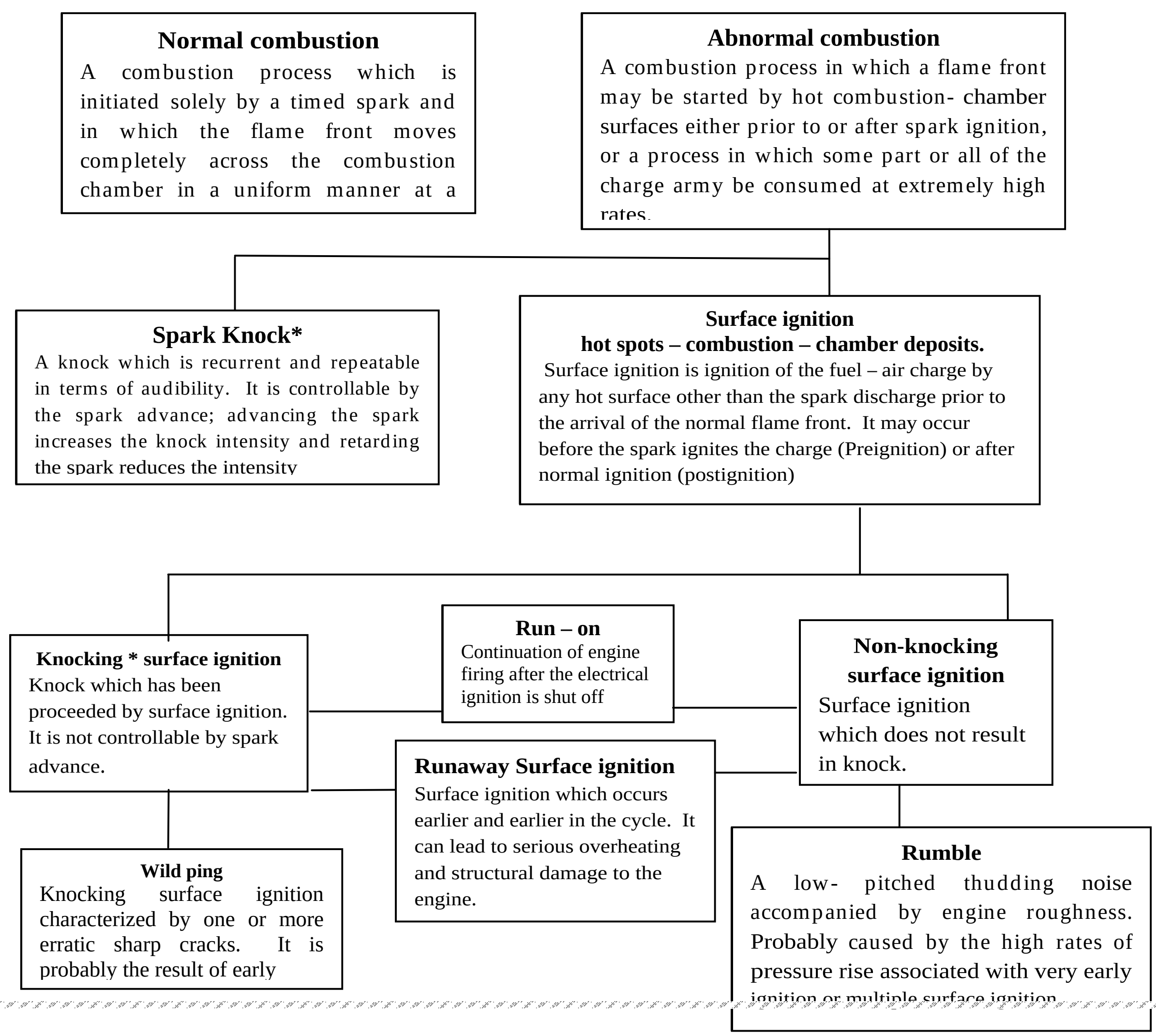
The gaseous fuel is supplied to the air inducted by the engine. The air and gas mixture is then compressed in the cylinder similar to air compression in diesel engine. Near TDC, small amount of diesel fuel (5 to 7% of fuel at full load) is supplied to the engine. This pilot fuel acts as a source ignition. The air and gas mixture around the injected spray ignites simultaneously at number of places establishing a number of flame fronts and combustion flame continues smoothly and rapidly. The combustion in a dual-fuel engine starts similar to CI engine, but propagates by flame fronts similar to SI engine.

The dual-fuel engines are capable of producing same power as that of diesel engines. Thermal efficiency of dual-fuel engine is low at part load conditions and specific fuel consumption is high because of increased delay periods. To remedy this, additional fuel needs to be injected at part loads. The same speed and same capacity dual-fuel engines give higher efficiency than diesel engine at full loads. This is due to the fact that gaseous fuels fill the entire combustion chamber and allow more air to take part in the combustion reaction.

Phenomena of surface ignition

Description of Phenomena: Abnormal combustion reveals itself it many ways. Of the various abnormal combustion processes which are important in practice, the two major phenomena are Knock and surface ignition. These abnormal combustion phenomena are of concern because: (1) when severe, they can cause major engine damage: and (2) even if not severe, they are regarded as an objectionable source of noise by the engine or vehicle operator. Knock is the name given to the noise which is transmitted through the engine structure when essentially spontaneous ignition of a portion of the end – gas- the fuel, air, residual gas, mixture ahead of the propagating flame- occurs. When this abnormal combustion process takes place, there is an extremely rapid release of much of the chemical energy in the end-gas, causing very high local pressures and the propagation of pressure waves of substantial amplitude across the combustion chamber. Surface ignition is ignition of the fuel – air mixture by a hot spot on the combustion chamber walls such as an overheated valve or spark plug. Or glowing combustion chamber deposit: i.e., by any means other than the normal spark discharge. It can occur before the occurrence of the spark (Post ignition), Following surface ignition, a turbulent flame develops at each surface – ignition location and starts to propagate across the chamber in an analogous manner to what occurs with normal spark ignition.

Because the spontaneous ignition phenomenon that causes knock is governed by the temperature and pressure history of the end gas, and therefore by the phasing and rate of development of the flame, various combination of these two phenomena – surface ignition and Knock – can occur. These have been categorized as indicated in Fig. When auto ignition occurs repeatedly, during



*Knock: The noise associated with AutoNation of a portion of the fuel – air mixture ahead of the advancing flame front. Auto ignition is the spontaneous ignition and the resulting very rapid reaction of a portion or all of the fuel – air mixture.
FIGURE:

Definition of combustion phenomena – normal and abnormal (knock and surface ignition) – in a spark – ignition engine. (Courtesy Coordinating Research Council.) otherwise normal combustion events, the phenomena is call *spark – knock.*

Advantages of Lean burn engine
Lean mixture is preferred n SI engine

(1) Lower pollutants
Investigations show that in the lean mixture range all the three major pollutants (CO, HC and NO_x) show descending trend.
(2) Good fuel economy.
(3) The ratio of specific heats (K) approaches that of air with lean mixtures.
(4) Heat transfer loss to the cooling medium are reduced
(5) Higher compression ratios can be used which in turn will improve thermal efficiency.

The following are some of the modifications to be made to convert an existing engine as a lean burn engine:

(1) Increasing the compression ratio of the engine to accelerate flame propagation. Since lean mixtures are resistant to knocking, and preignition, this measure can be used.

(2) Increasing the swirl and turbulence of the mixture in order to increase flame speed. This is brought about by inlet port design, and by suitable combustion chamber design.
(3) Minimizing the heat losses from the combustion chamber and locating the combustion chamber and spark plug near the hot regions of the engine in order to bring about some mild preheating of the mixture.
(4) By using an ignition system with high sparking energy and prolonged spark duration.

(5) Catalytic activation of the charge in the combustion chamber.

Low Heat rejection engines

To increase the efficiency of internal combustion engines and generate higher chamber temperatures, low heat rejection concepts are being investigated.

In gasoline engines, the thermal insulation will increase the wall temperature which will lead to unwanted detonation. Because of this, insulation of the combustion chamber could be done only in diesel engines.

This has two important purposes; one to reduce the size of the coolant system and second to increase the exhaust energy available for turbo charging and thereby increasing the power and efficiency.

In the insulated engines, the wall temperature increases volumetric efficiency. To maintain the volumetric efficiency, the insulated engines are usually turbo charged.

Insulation of the combustion chamber is done by coating it with ceramics. Partially stabilized zirconia and aluminium titanate are used for coating. The coating is mainly done by plasma spraying.

The spraying parameters are very important in determining the reliability of the coating. A 2mm layer of zirconia will reduce the heat rejection to the coolant by 48%, while a 8mm layer is required to reduce the heat flow by 78%.

The Engine components which are commonly coated are cylinder head, valves, liners and piston crown.

The thermal conductivities of typical ceramics can be seen in figure. The thermal expansion characteristic of these ceramics can be seen in figure.

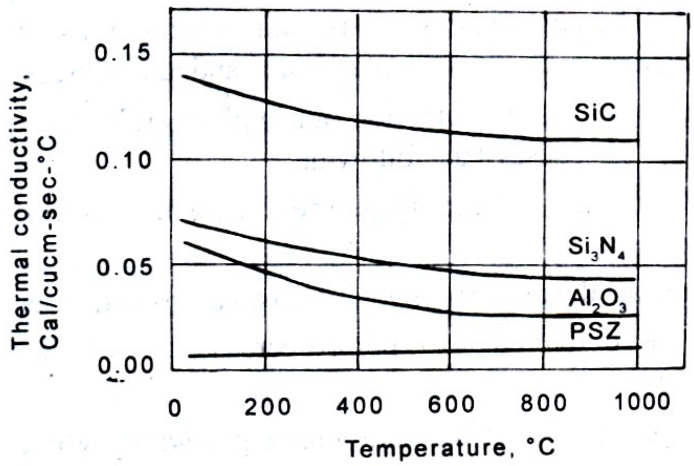


Figure: Thermal conductivities of a typical ceramics

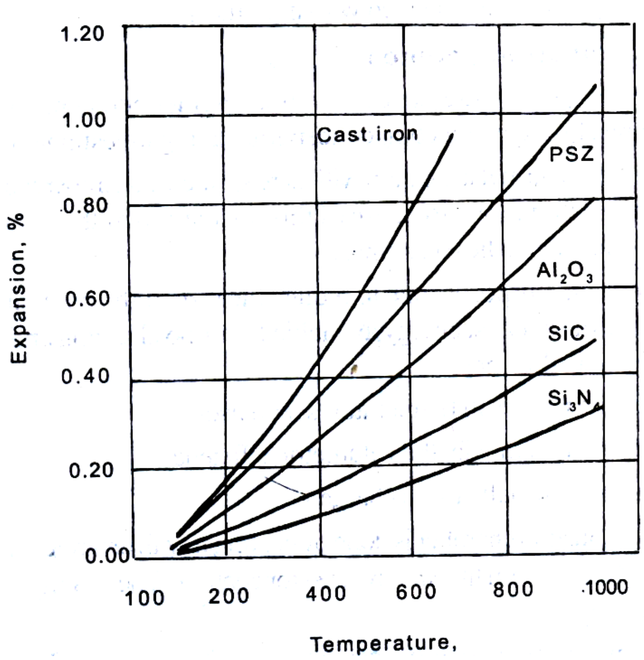


Figure: Thermal expansion of typical ceramics

It is expected that an insulated diesel engine should easily out perform the standard cooled engine. However, experiments reveal that in many cases, the performance of LHR engine is only slightly better, in a few cases worse than the cooled version of the same engine. A more likely cause for the performance changes is that the introduction of insulation in the cylinder could significantly alter the combustion process.

The changes in the combustion process due to insulation also affect exhaust emissions. Higher gas temperatures are supposed to reduce the concentration of incomplete combustion at the expense of increase in nitric oxide.

An air gap insulated piston can be seen in figure. This piston reduces heat losses and thus improves combustion.

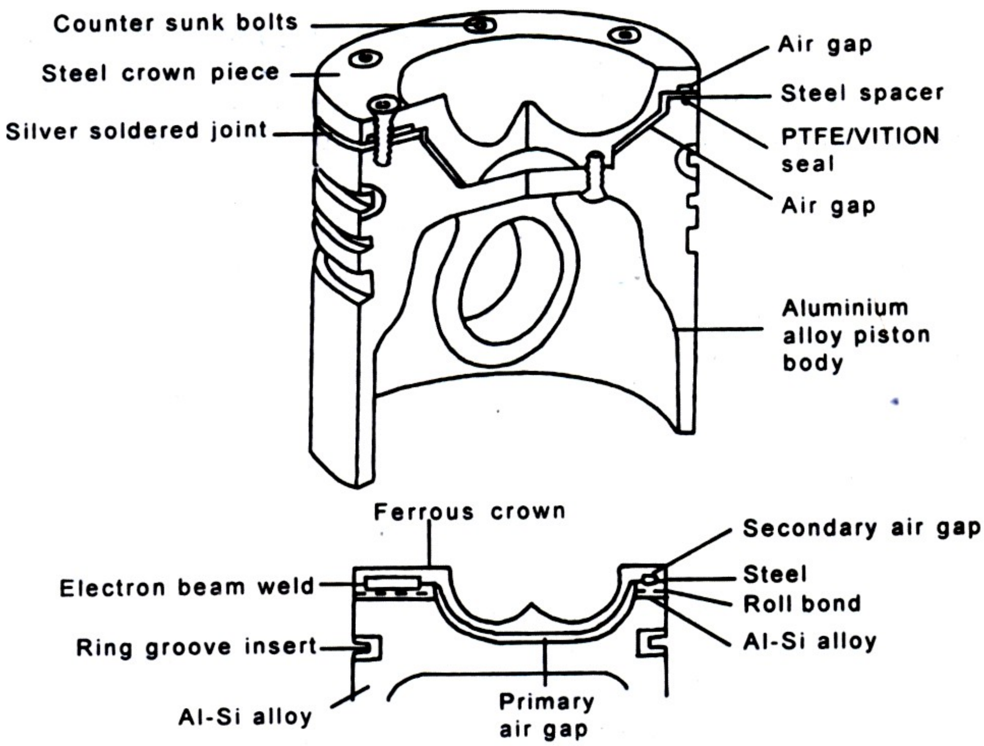


Figure: Air gap insulated piston.

Working of a lean burn Engine

The AVL high compression lean burn (HCLB) combustion system can be seen in figure.

The wide open throttle (WOT) performance for different 2.3-2.4 L.4 cylinder engines can be seen in Figure. The performance characteristics of SI engines having 2 valves per cylinder and 4 valves per cylinder can also be seen in this figure. Multiple valve engine gives higher BMEP and lower SFC. The performance characteristics of a typical IDI diesel engine can also be seen in this figure.

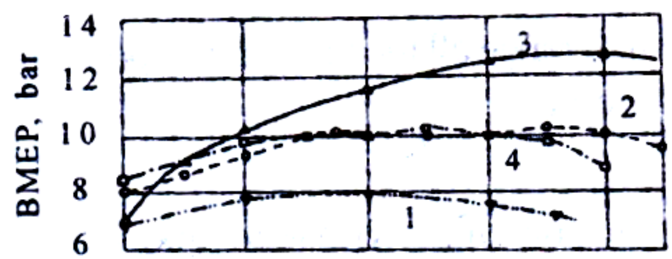
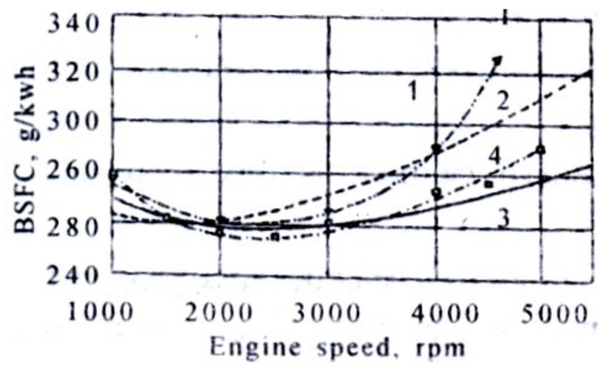


Figure: Wide open throttle (wot) performance for different 2.3-2.5L, four cylinder engines



1. IDI Diesel Engine
2. SI Engine, 2-valve, RON = 98
3. SI Engine, 4-valve, RON = 98
4. Lean burn engine, RON = 91

The effect of EGR (exhaust gas recirculation) in the AVL-HCLB-2.3L engine (compression ratio: 11, Swirl: 1.40, rpm : 2000) for different load conditions can be seen in figure. With EGR, HC and NO_x emissions are found to be lesser, though there is no variation of CO emissions.

The influence of different modes of fuel injection, namely, high pressure direct injection, low pressure semi direct injection and low pressure intake manifold injector.

On BSFC and BMEP can be seen in figure. High pressure (in cylinder) direct injection is found to yield better results.

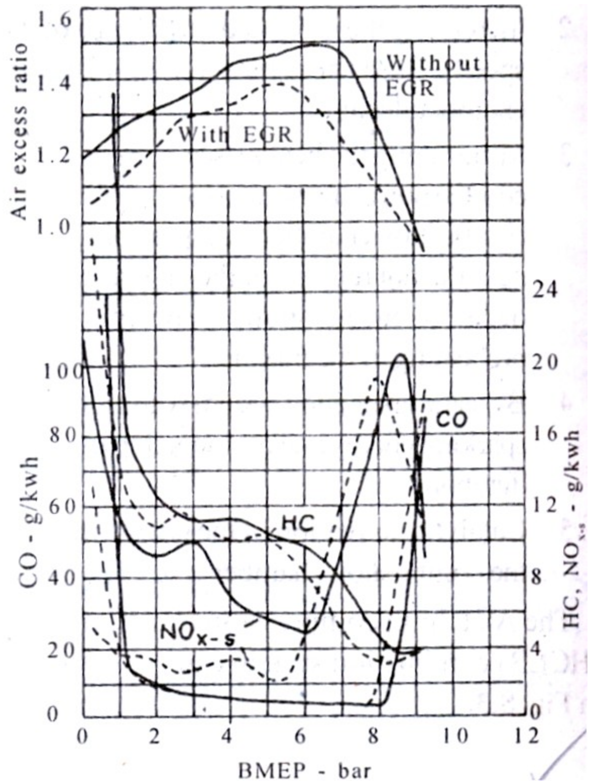


Figure: Exhaust gas recirculation (EGR) on the performance of hclb 2.3 L engine

UNIT – V

ELECTRONIC ENGINE MANAGEMENT

Objectives of electronic control system for IC engines

Any engine electronic control system should be rugged and reliable. The main objectives of an electronic control system for IC engines should be to:

1) Improve driving comfort
2) Reduce exhaust emissions
3) Lower fuel consumption
4) Improve power and torque of engine
5) Lower noise levels
6) Prolong life of the engine

Important components of electronic engine management system.

The electronic control system essentially consists of an ECU, i.e., electronic control unit along with various sensors and actuators. Using the sensors, the ECU obtains feedback from the engine and regulates engine parameters such as air fuel mixture distribution in cylinders, air-fuel ratio, ignition timing in case of SI engines, injection timing in case of CI engines, injection duration for CI engines, idling control and fuel cut off in case of over speeding. Electronic control systems also provide services such as engine speed governing in case of fluctuating speed and load.

Open loop control systems

In the open loop control system, if there is an error or deviation the error signal does not reach the ECU. This means there is no feedback from the engine process parameters to the ECU. Such systems are not efficient from fuel economy point of view, but have lower cost. Open loop systems are used when engine is cold or has low vacuum or is in acceleration mode.

Closed loop control system

For closed loop control system, an oxygen sensor is used for providing feedback. This oxygen sensor is located in the exhaust manifold and is also termed as "λ" sensor or excess air flow factor sensor. It senses the oxygen content in the exhaust gas and provides a feedback signal to the ECU.

Sensor

A sensor is an input device that converts one form of energy to another. Since a computer can only read voltage signals, an automobile sensor must convert motion, pressure, temperature and light to voltage signal. Sensors for IC engines include timers, resistors, transformers, switches and generators. Sensors are digital or analog. Sensors are also called as transducers as they convert one form of energy to another.

Types of engine sensors

The main sensors used in the engine are:

(1) Exhaust gas oxygen (I) sensor
(2) Fuel metering (flow) sensor
(3) Manifold air pressure sensor
(4) Engine vacuum sensor
(5) Air intake temperature sensor

(6) Throttle position sensor
(7) Crankshaft position sensor
(8) Engine speed sensor
(9) Exhaust gas recirculation sensor
(10) Air conditioning sensor
(11) Detonation sensor
(12) Ignition timing sensor
(13) Vibration sensor
(14) Force sensor
(15) Humidity sensor
(16) Coolant temperature sensor

Ignition Timing sensor

This sensor is also of the hall effect type. Here the change in magnetic field due to movement of piston causes generation of a square pulse. This square pulse is fed to the ECU, which activates the ignition coils and spark is produced at the appropriate time in the cycle.

Engine Data Acquisition System.

Engine data acquisition systems have come a long way from simple hand recording on "data sheets" or "test sheets" to sophisticated DAQ or data acquisition servers. During this transition several devices such as chart recorders were used for data logging. Primitive data loggers were also developed. With the advent of the digital age and the computer revolution, computerized data acquisition systems have become the standard norm. Virtual instrumentation was developed and massive amount of data were recorded, stored and analyzed in computers.

Electronic Control Unit (ECU)

Electronic control units are also called as engine computers or engine CPUs. Like any computer, the ECU has four basic functions:

(1) Input – Receives signals from various sensors in form of voltage
(2) Processing-computes input data to make decisions and perform output functions
(3) Storage-for storage of information, conditions or signals

(4) Output-After processing, ECU sends signals to actuators or display devices to execute the decision. Actuators are solenoid valves or motors which perform necessary movements.

All ECUs are digital and input or output signal is in terms of yes/no or high/low or on/off. The analog input from sensors is converted to digital signal by an analog to digital converter (A/D converter). The microprocessors process data in the form of "bits", i.e., binary digits, using logic networks made up of several MOSFET transistors. The ECU interface with the operator is in terms of a software.

ECU Software Program

The program instruction that the ECU uses to understand its data and perform calculations and send its output command, consists of:

- ❖ Mathematical instructions in binary form for processing data
- ❖ Information on engine constants such as number of cylinders, bore, stroke, compression ratio, etc.
- ❖ Information on engine variables such as engine speed, air flow rate, air to fuel ratio, temperature, manifold pressure, EGR flow, etc

To load the variable values in the ECU memory a process known as "Engine Mapping" is used. This engine mapping is done after extensive trials in an engine test bed, where all variables are measured for best performance and lowest fuel consumption and emissions. This information is then stored in tables known as "Look up Tables". These tables can be modified by the engine operator as and when desired. This process of tuning the engine is called "Engine Optimization". Usually the parameters of most importance are the air fuel ratio, ignition timing, EGR, etc. Fig. shows a typical 3-D engine map.

ECU control-systems-open loop vs closed loop

Every ECU control system has two operating modes(1) open loop mode or (2) closed loop mode.

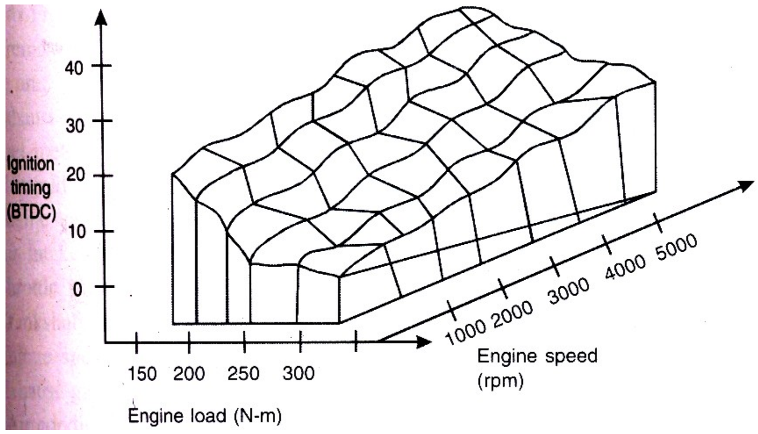


Figure: Engine map

Open loop control system

In the open loop control system, if there is an error or deviation the error signal does not reach the ECU. This means there is no feedback from the engine process parameters to the ECU. Such systems are not efficient from fuel economy point of view, but have lower cost. Open loop systems are used when engine is cold or has low vacuum or is in acceleration mode. Nowadays the preference is for closed loop control.

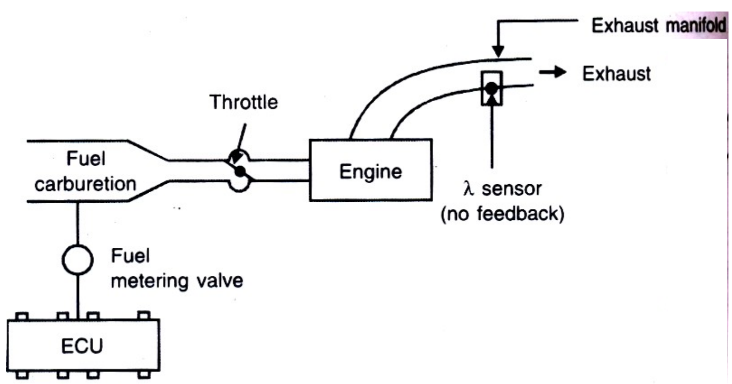


Figure: Open loop control system

Closed loop control system

For closed loop control system, an oxygen sensor is used for providing feedback. This oxygen sensor is located in the exhaust manifold and is also termed as "λ" sensor or excess air flow factor sensor. It senses the oxygen content in the exhaust gas and provides a feedback signal to the ECU.

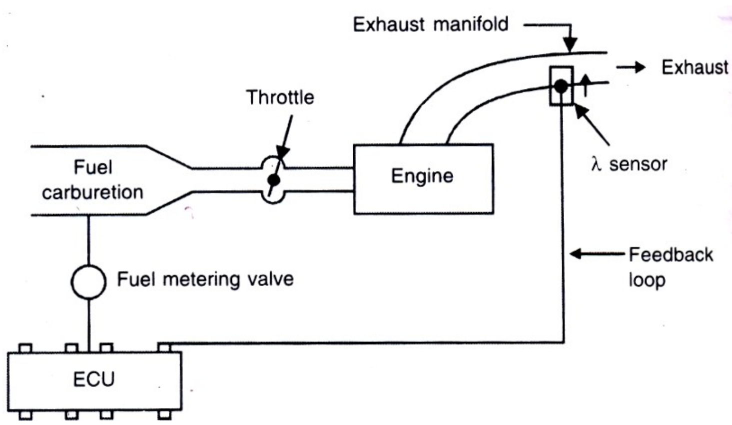


Figure: Closed loop control system

If the λ sensor provides the feedback that the mixture is too lean, then the ECU adjusts the fuel metering valve so as to release more fuel, i.e., rich mixture. If the λ sensor provides the feedback that the mixture is too rich, the ECU adjusts the fuel metering valve to release less fuel, i.e., lean mixture. The whole purpose of using a λ sensor is to maintain the fuel-air mixture at stoichiometric conditions (λ-1).

Other feedback controls include turbo boost pressure control to avoid detonation and idle speed control. Thus, the main difference between open loop and closed loop control system is the incorporation of the feedback loop as shown in fig.

Engine Sensors

Refer to fig. for a block diagram displaying all engine sensors used for engine variable measurement and control. All sensors are connected to the ECU. The main sensors used in the engine are:

(1) Exhaust gas oxygen (I) sensor
(2) Fuel metering (flow) sensor
(3) Manifold air pressure sensor
(4) Engine vacuum sensor
(5) Air intake temperature sensor
(6) Throttle position sensor
(7) Crankshaft position sensor
(8) Engine speed sensor
(9) Exhaust gas recirculation sensor
(10) Air conditioning sensor
(11) Detonation sensor

(12) Ignition timing sensor
(13) Vibration sensor
(14) Force sensor
(15) Humidity sensor
(16) Coolant temperature sensor

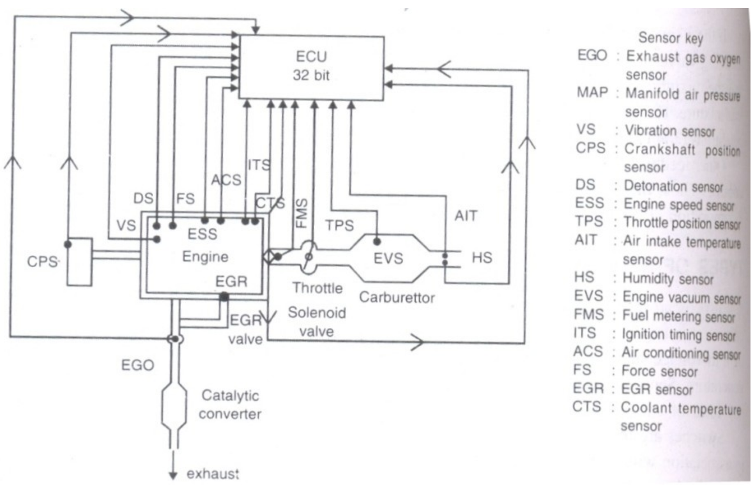


Figure: ECU with engine sensor

(1) Exhaust gas oxygen (I sensor or EGO sensor)

Fig. shows the construction of a typical EGO sensor. The λ sensor is unique voltage generator that measures exhaust oxygen content. It generates analog signals from 0 to 1 volt by comparing difference between oxygen in the exhaust and oxygen in the ambient air. EGO sensor is also called 'λ' sensor as it is based on Lambda or excess-air ratio measurement. Lambda (λ) is the ratio of excess air to stoichiometric air. λ=1 for stoichiometric combustion and EGO sensors provide feedback to maintain λ at unity.

The EGO sensor works as a galvanic battery to generate voltage of 0.1 to 0.9 volts. When oxygen content in exhaust gases is high the corresponding output voltage is high (0.4 to 0.9 volts) and when oxygen content is low the output voltage is low (0.1 to 0.4 volts). The EGO sensor must be warmed up to 350°C before measurement and works best at 800°C.

The construction of the sensor is as follows. It consists of two platinum electrodes, separated by zirconium dioxide ceramic electrolyte (Zr O_2). This electrolyte attracts negatively charged oxygen ions. One electrode is exposed to ambient air and attracts ambient oxygen ions, whereas the other electrode is exposed to the exhaust gases and collects exhaust gas oxygen ions. Thus, one electrode becomes negative (ambient electrode) and other is relatively positive (exhaust gas electrode).

This causes a simple current to flow between the electrodes. More the difference in ions, higher is the collected current. The voltage developed is directly proportional to amount of oxygen in the exhaust.

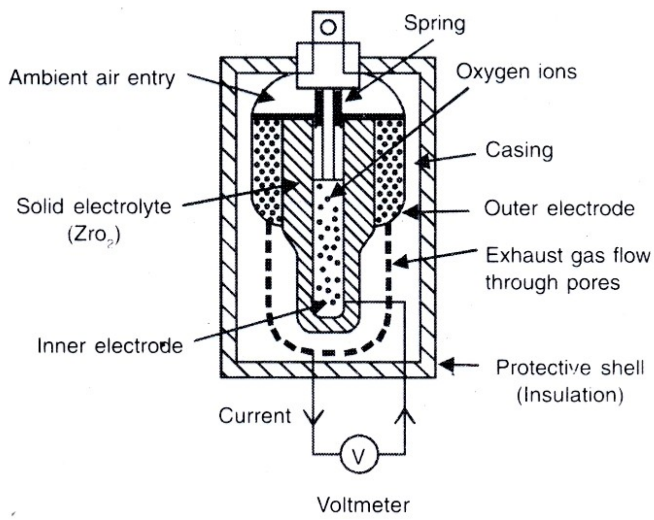


Figure: EGO sensor construction

Heated EGO sensors are used for better accuracy. The sensor is heated using battery current passing through a wire. The current warms the sensor faster during engine starting and helps to maintain accurate voltage signals at all conditions. EGO sensors are always installed in exhaust manifolds or tailpipes. However, the position may vary as per engine requirement. Care has to be taken to mount the sensor away from the exhaust valve as high temperature of exhaust gases can damage the sensor. The response time of a typical EGO sensor is of the order of 50 milliseconds.

Computerized data acquisition

The sensors used in engines sense various quantities and the information needs to be recorded in information hubs known as “channels”. As the number of channels increase computerized control becomes important. A typical engine has a 40 channel data acquisition system. Typically the following quantities are required to be recorded:

- Engine speed, torque, power and fuel consumption
- Engine air, coolant and oil temperature
- Turbocharger speed and pressure
- Exhaust gas temperature
- Emission measurement (HC, NO_x, CO, CO_2, O_2, etc.)

All the signals are analog in nature, which need to be converted to digital form using an A/D convertor. A signal conditioning unit is used to condition and linearize the signals. Data collection and signal conditioning (AT) cards are available with 8 or 12 or 16 bit resolution. The signals are fed to a PC which displays them on a monitor. The readings are recorded continuously at an interval predetermined by the user (usually every 2-5 minutes). The stored data can be retrieved and displayed in graphical format.

Desirable features of a data acquisition system:

- It should have provision for at least 32 channels
- It should provide fast data logging
- It should have ability to record data from peripheral instrumentation such as fuel flow meters
- It should have large data storage capacity
- Ability to vary the sampling frequency6
- Live graphical display of channels
- Live graphical display of channels
- Menu driven calibration routines for transducers
- Fully configurable four level alarms
- Multi level password protection of data
- Facilities for linking to intranet or internet
- Remote access facility

Fig. shows the schematic of a typical data acquisition system

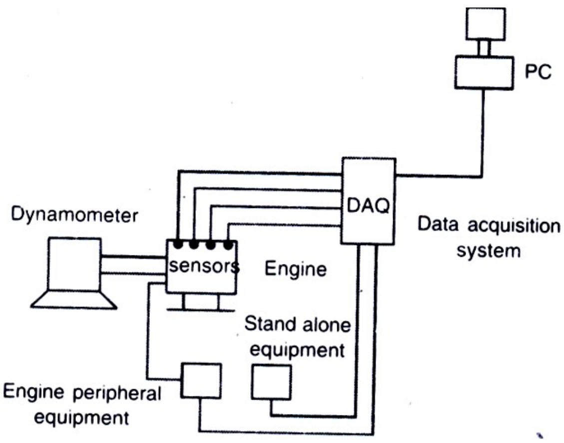


Figure: Computerized data acquisition system

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