

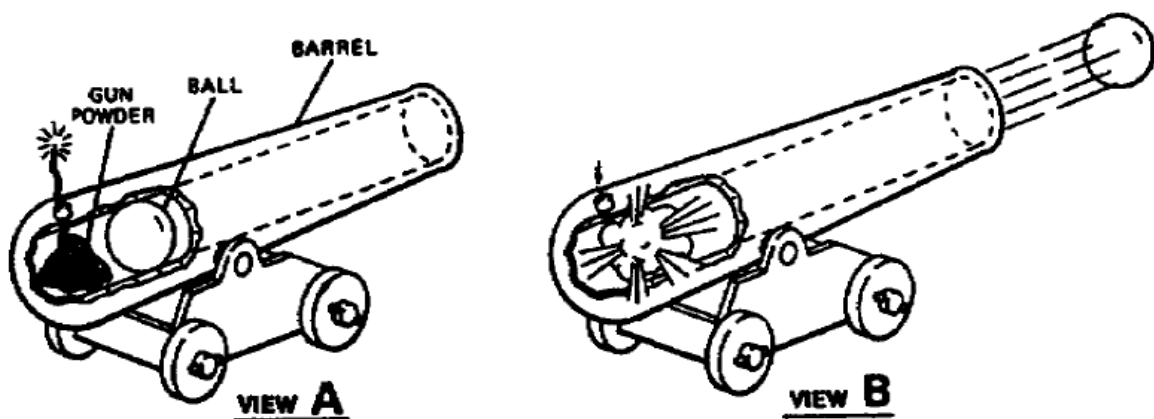
ME2041 - ADVANCED IC ENGINES

UNIT-I SPARK IGNITION ENGINES

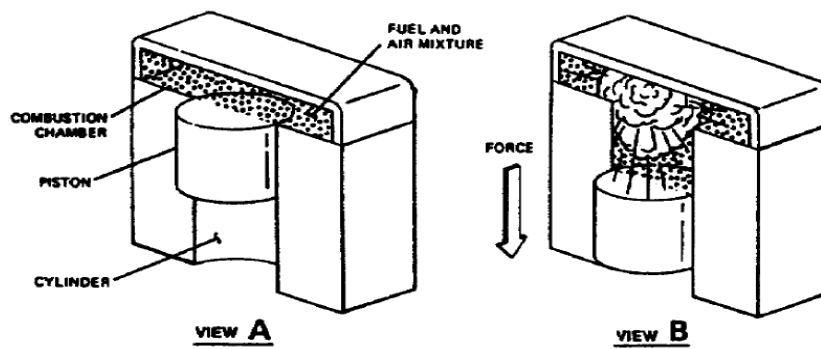
LECTURE – 1

Introduction about IC engines

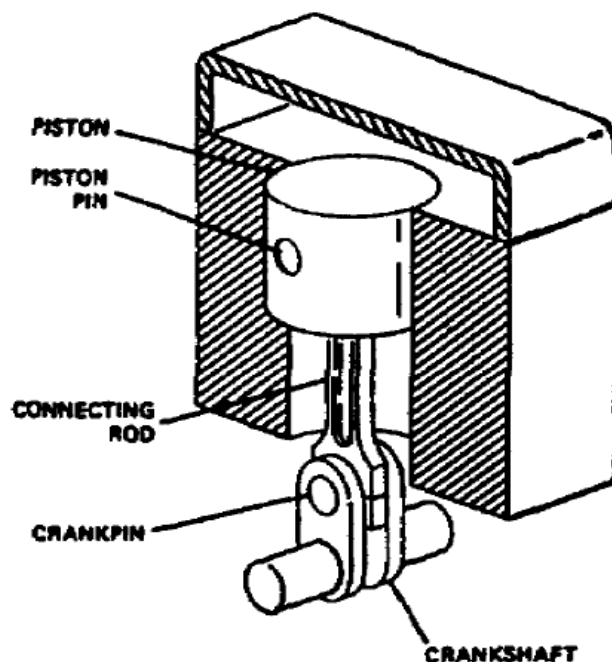
The operation of the piston engine can best be understood by comparing it to a simple cannon. In view A of figure 1 on the following page, a cannon barrel, charge of gunpowder, and a cannonball are illustrated. In view B of figure 1, the gunpowder is ignited. The gunpowder burns very rapidly and as it burns there is a rapid expansion of the resulting gases. This rapid expansion causes a tremendous increase in pressure that forces the cannonball from the barrel.



Now the cannon barrel has been replaced by a cylinder and a combustion chamber. The cannonball has been replaced by a piston. A mixture of vaporized fuel and air has replaced the gunpowder. In view B of figure, the gasoline is ignited. This time, the resulting force acts to push the piston downward.



The force of the piston acting in a downward motion is of little immediate value if it is to turn the wheels of a vehicle. In order to use this straight line or reciprocating motion, it must be transformed into rotary motion. This is made possible through the use of a crankshaft. The crankshaft is connected to the driving wheels of a vehicle through the drive train on one end. On the other end of the shaft is a crank with a crankpin offset from the shaft's centre. Figure below illustrates how the piston and the crankshaft are connected through the connecting rod and the crankpin. Figure below on the following page illustrates how reciprocating motion of the piston is changed to rotating motion of the crankshaft.



LECTURE – 2

Air-fuel ratio requirements

The task of the engine induction and fuel systems is to prepare from ambient air and fuel in the tank an air-fuel mixture that satisfies the requirements of the engine over its entire operating regime. In principle, the optimum air/fuel ratio for a spark-ignition engine is that which gives the required power output with the lowest fuel consumption, consistent with smooth and reliable operation. In practice, the constraints of emissions control may dictate a different air/fuel ratio, and may also require the recycling of a fraction of the exhaust gases (EGR) into the intake system. The relative proportions of fuel and air that provide the lowest fuel consumption, smooth reliable operation, and satisfy the emissions requirements, at the required power level, depend on engine speed and load. Mixture requirements and preparation are usually discussed in terms of the air/fuel ratio or fuel/air ratio and percent EGR. While the fuel metering system is designed to provide the appropriate fuel flow for the actual air flow at each speed and load, the relative proportions of fuel and air can be stated more generally in terms of the fuel/air equivalence ratio, which is the actual fuel/air ratio normalized by dividing by the stoichiometric fuel/air ratio. The combustion characteristics of fuel-air mixtures and the properties of combustion products, which govern engine performance, efficiency, and emissions, correlate best for a wide range of fuels relative to the stoichiometric mixture proportions. Where appropriate, therefore, the equivalence ratio will be used as the defining parameter. A typical value for the stoichiometric air/fuel ratio of gasoline is 14.6. Thus, for gasoline,

$$\phi \approx \frac{14.6}{A/F}$$

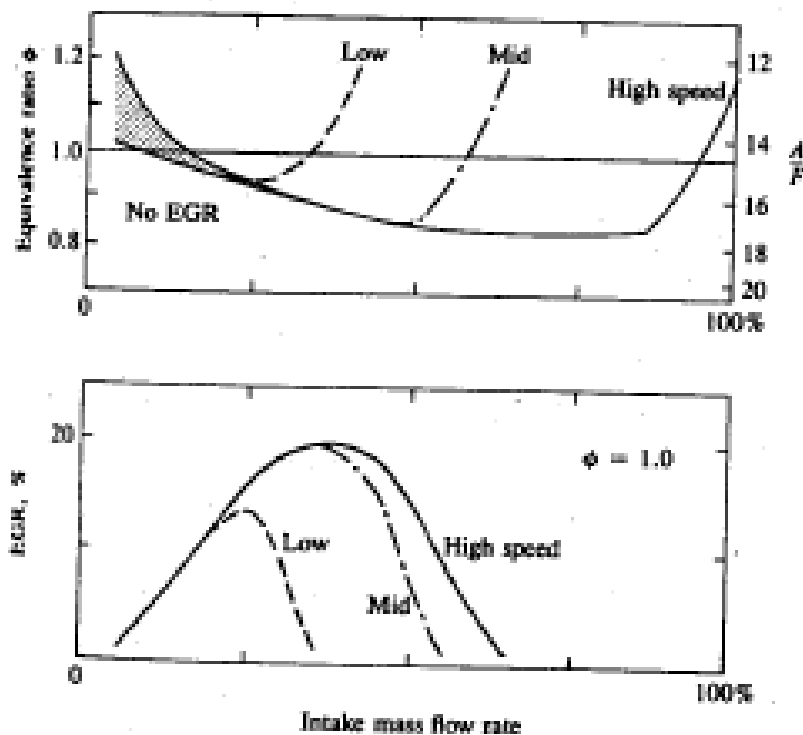
A brief summary is sufficient here. Mixture requirements are different for full-load (wide open throttle) and for part-load operation. At the former operating condition, complete utilization of the inducted air to obtain maximum power for a given displaced volume is the critical issue. Where less than the maximum power at a given speed is required, efficient utilization of the fuel is the critical issue. At wide-open throttle, maximum power for a given volumetric efficiency is obtained

with rich-of-stoichiometric mixtures, 4 X 1.1. Mixtures that are richer still are sometimes used to increase volumetric efficiency by increasing the amount of charge cooling that accompanies

fuel vaporization, thereby increasing the inducted air density. At part-load (or part-throttle) operating conditions, it is advantageous to dilute the fuel-air mixture, either with excess air or with recycled exhaust gas. This dilution improves the fuel conversion efficiency for three reasons: (1) the expansion stroke work for a given expansion ratio is increased as a result of the change in thermodynamic properties of the burned gases—see Secs. 5.5.3 and 5.7.4; (2) for a given mean effective pressure, the intake pressure increases with increasing dilution, so pumping work decreases; (3) the heat losses to the walls are reduced because the burned gas temperatures are lower. In the absence of strict engine NO_x emission requirements, excess air is the obvious diluent, and at part throttle engines have traditionally operated lean. When tight control of NO_x, HC, and CO emissions is required, operation of the engine with a stoichiometric mixture is advantageous so that a three-way catalyst can be used to clean up the exhaust. The appropriate diluent is then recycled exhaust gases which significantly reduces NO_x emissions from the engine itself. The amount of diluent that the engine will tolerate at any given speed and load depends on the details of the engine's combustion process. Increasing excess air or the amount of recycled exhaust slows down the combustion process and increases its variability from cycle to cycle. A certain minimum combustion repeatability or stability level is required to maintain smooth engine operation. Deterioration in combustion stability therefore limits the amount of dilution an engine can tolerate. As load decreases, less dilution of the fresh mixture can be tolerated because the internal dilution of the mixture with residual gas increases. At idle conditions, the fresh mixture will not usually tolerate any **EGR** and may need to be stoichiometric or fuel-rich to obtain adequate combustion stability.

If stoichiometric operation and **EGR** are not required for emissions control, as load increases the mixture is leaned out from a fuel-rich or close-to-stoichiometric composition at very light load. As wide-open throttle operation is approached at each engine speed, the mixture is steadily enriched to rich-of-stoichiometric the maximum point. With the stoichiometric operating conditions required for three-way-catalyst-equipped engines, when **EGR** is used, the percentage of recycled exhaust increases from zero at light load to a maximum at mid-load, and then decreases to zero as wide-open throttle conditions are approached so maximum brake can be obtained. Combinations of these strategies are possible.

For example, lean operation at light load can be used for best efficiency, and



Typical mixture requirements for two common operating strategies: Top diagram shows equivalence ratio variation with intake mass flow rate (percent of maximum flow at rated speed) at constant low and high engine speeds. Bottom diagram shows recycled exhaust (EGR) schedule as a function of intake flow rate, for low, mid, and high speeds for stoichiometric operation.

Stoichiometric mixtures (with a three-way catalyst) and/or EGR can be used at mid loads to control NO_x emissions.

In practical spark-ignition engine induction systems, the fuel and air distribution between engine cylinders is not uniform (and also varies in each individual cylinder on a cycle-by-cycle basis). A spread of one or more air/fuel ratio between the leanest and richest cylinders over the engine's load and speed range is not uncommon in engines with conventional carburetors. The average mixture composition must be chosen to avoid excessive combustion variability in the leanest operating cylinder. Thus, as the spread in mixture nonuniformity increases, the mean equivalence ratio must be moved toward stoichiometric and away from the equivalence ratio which gives minimum fuel consumption.

LECTURE – 3

Design of carburettor

Carburetor Fundamentals

A carburetor has been the most common device used to control the fuel flow into the intake manifold and distribute the fuel across the air stream. In a carburetor the air flows through a converging-diverging nozzle called a venturi. The pressure difference set up between the carburetor inlet and the throat of the nozzle (which depends on the air flow rate) is used to meter the appropriate fuel flow for that air flow. The fuel enters the air stream through the fuel discharge tube or ports in the carburetor body and is atomized and convected by the air stream past the throttle plate and into the intake manifold. Fuel evaporation starts within the Carburetor and continues in the manifold as fuel droplets move with the air flow and as liquid fuel folks over the throttle and along the manifold walls. A modern carburetor which meters the appropriate fuel flow into the air stream over the complete engine operating range is a highly developed and complex device. There are many types of carburetors; they share the same basic concepts which we will now examine.

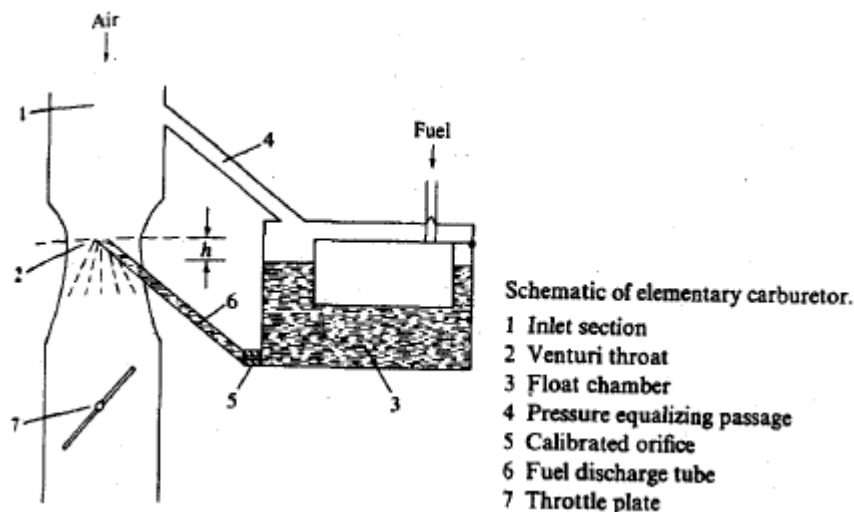


Figure shows the essential components of an elementary carburetor. The air enters the intake section of the carburetor (1) from the air cleaner which removes suspended dust particles. The air then flows into the carburetor venturi (a converging-diverging nozzle) (2) where the air

velocity increases and the pressure decrease. The fuel level is maintained at a constant height in the float chamber (3) which is connected via an air duct (4) to the carburetor intake section (I). The fuel flows through the main jet (a calibrated orifice) (5) as a result of the pressure difference between the float chamber and the venturi throat and through the fuel discharge nozzle (6) into the venturi throat where the air stream atomizes the liquid fuel. The fuel-air mixture flows through the diverging section of the venturi where the flow decelerates and some pressure recovery occurs. The flow then passes the throttle valve (7) and enters the intake manifold. Note that the flow may be unsteady even when engine load and speed are constant, due to the periodic filling of each of the engine cylinder which draws air through the carburetor venturi. The induction time, $1/(2N)$ (20 ms at 1500 rev / min) is the characteristic time of this periodic cylinder filling process. Generally, the characteristic times of changes in throttle setting are longer; it takes several engine operating cycles to re-establish steady-state engine operation after a sudden change in throttle position. It is usually assumed that the flow processes in the carburetor can be modelled as quasi steady.

LECTURE – 4

Venture size

Carburetor venturi size is usually designed by the conventional equation for better performance even though some modern design has changed over the performance of carburetor in the present scenario.

$$\dot{m}_a = \frac{C_{DT} A_T p_0}{\sqrt{RT_0}} \left(\frac{p_T}{p_0} \right)^{1/\gamma} \left\{ \frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{p_T}{p_0} \right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2}$$

where C_{DT} and A_T are the discharge coefficient and area of the venturi throat, respectively. If we assume the velocity at the carburetor inlet can be neglected, the above equation can be rearranged in terms of the pressure drop from upstream conditions to the venturi throat for the air stream, $\Delta p_a = p_0 - p_T$, as

$$\dot{m}_a = C_{DT} A_T (2\rho_{a0} \Delta p_a)^{1/2} \Phi$$

where

$$\Phi = \left[\left(\frac{\gamma}{\gamma - 1} \right) \frac{(p_T/p_0)^{2/\gamma} - (p_T/p_0)^{(\gamma+1)/\gamma}}{1 - (p_T/p_0)} \right]^{1/2}$$

and accounts for the effects of compressibility. For the normal carburetor operating range, where $\Delta p_a / p_o \leq 0.1$, the effects of compressibility which reduce Φ below 1.0 are small.

Fuel jet size

Since the fuel is a liquid and therefore essentially incompressible, the fuel flow rate through the main fuel jet is given by

$$\dot{m}_f = C_{D_o} A_o (2\rho_f \Delta p_f)^{1/2}$$

where C_{D_o} and A_o , are the discharge coefficient and area of the orifice, respectively, Δp_f , is the pressure difference across the orifice, and the orifice area is assumed much less than the passage area. Usually, the fuel level in the float chamber is held below the fuel discharge nozzle, as shown in Fig, to prevent fuel spillage when the engine is inclined to the horizontal (e.g., in a vehicle on a slope). Thus

$$\Delta p_f = \Delta p_a - \rho_f g h$$

where h is typically of order 10mm.

The discharge coefficient C_{D_o} in fuel flow rate equation represents the effect of all deviations from the ideal one-dimensional isentropic flow. It is influenced by many factors of which the most important are the following: (1) fluid mass flow rate; (2) orifice length / diameter ratio; (3) orifice/approach-area ratio; (4) orifice surface area; (5) orifice surface roughness; (6) orifice inlet and exit chamfers; (7) fluid specific gravity; (8) fluid viscosity; and (9) fluid surface tension. The use of the orifice Reynolds number, $Re_o = \rho V D_o / \mu$, as a correlating parameter for the discharge coefficient accounts for effects of mass flow rate, fluid density and viscosity, and length scale to a good first approximation. The discharge coefficient of a typical carburetor main fuel-metering system orifice increases smoothly with increasing Re_o .

Modern Carburetor Design

The changes required in the elementary carburetor so that it provides the equivalence ratio versus air flow distribution are

1. The fuel metering system must be compensated to provide essentially constant lean or stoichiometric mixtures over the 20 to 80 percent air flow range.
2. An idle system must be added to meter the fuel flow at idle and light loads.
3. An enrichment system must be added so the engine can provide its maximum power as wide-open throttle is approached.
4. An accelerator pump which injects additional fuel when the throttle is opened rapidly is required to maintain constant the equivalence ratio delivered to the engine cylinder.
5. A choke must be added to enrich the mixture during engine starting and warm-up to ensure a combustible mixture within each cylinder at the time of ignition.
6. Altitude compensation is required to adjust the fuel flow to changes in air density.

In addition, it is necessary to increase the magnitude of the pressure drop available for controlling the fuel flow. Two common methods used to achieve they are.

LECTURE – 5

COMBUSTION IN SI ENGINES

The combustion process of SI engines can be divided into three broad regions: (1) ignition and flame development, (2) flame propagation, and (3) flame termination. Flame development is generally considered the consumption of the first 5% of the air-fuel mixture (some sources use the first 10%). During the flame development period, ignition occurs and the combustion process starts, but very little pressure rise is noticeable and little or no useful work is produced (Fig. 7-1). Just about all useful work produced in an engine cycle is the result of the flame propagation period of the combustion process. This is the period when the bulk of the fuel and

air mass is burned (i.e., 80-90%, depending on how defined). During this

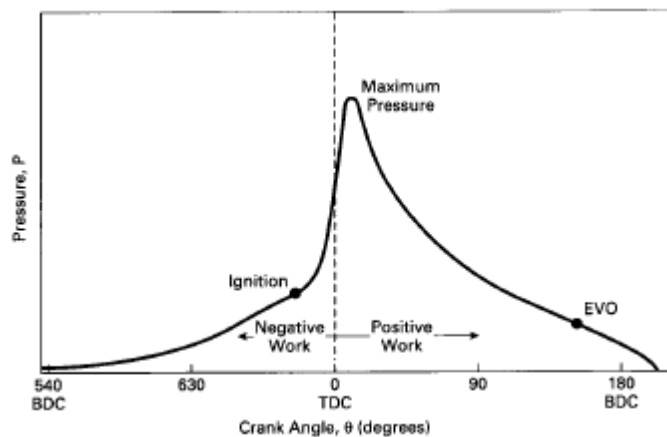


Figure Cylinder pressure in the combustion chamber of an SI engine as a function of crank angle. The increase in pressure rise is very slow after ignition during the flame development period. This results in a slow pressure force increase on the piston and a smooth engine cycle. Maximum pressure occurs 5° to 10° aTDC.

time, pressure in the expansion stroke. The final 5% (some sources use 10%) of the air-fuel mass which burns is classified as flame termination. During this time, pressure quickly decreases and combustion stops. In an SI engine, combustion ideally consists of an exothermic subsonic flame progressing through a premixed homogeneous air-fuel mixture. The spread of the flame front is greatly increased by induced turbulence, swirl, and squish within the cylinder. The right combination of fuel and operating characteristics is such that knock is avoided or almost avoided.

Types of combustion:

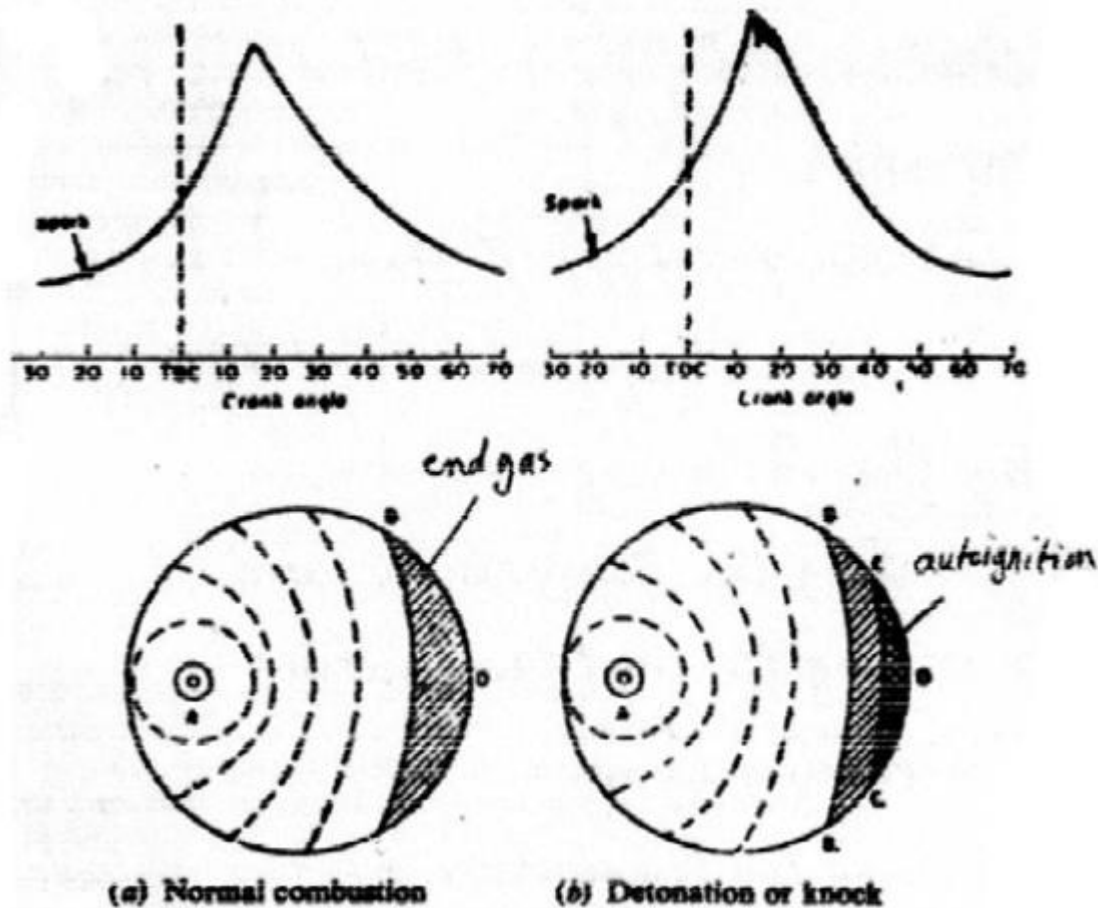
1. Normal Combustion.
2. Abnormal Combustion.

Normal Combustion:

Normal combustion rarely occurs in a real engine without some trace of autoignition appearing. After ignition, the flame front travels across the combustion chamber. The gas ahead of the flame front called the "end gas". The end gas receives heat due to compression by expanding gases and by radiation from the advancing flame front, therefore, its temperature and density increases. If the temperature exceeds the self-ignition

temperature and the un-burnt gas remains at or above this temperature for a period of time equal to or greater the delay period, spontaneous ignition (or auto ignition) will occur at various locations. Shortly afterwards an audible sound called **knock** appears.

If the end gas does not reach its self-ignition temperature, the combustion will be normal.



Lecture – 6

Abnormal Combustion:

In Internal combustion engines, abnormal combustion is a significant phenomenon associated with the combustion processes on which the life and performance of the engine depends. The two important abnormal combustion phenomena are **1. KNOCK** and **2. 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.

KNOCK

Knock is the most important abnormal combustion phenomenon. It is important because it puts a limit on the compression ratio at which an engine can be operated, which in turn controls the efficiency and to some extent power output. It got the name "**Knock**" because of the noise that results from the auto ignition of a portion of fuel air mixture ahead of the advancing flame. As the spark is ignited there is a formation of flame front and it starts propagating. As the flame propagates across the combustion chamber, speed of flame front is about **15-30 m/s**; the unburned charge ahead of the flame called the **END GAS** is compressed, raising its pressure, temperature and density. In case of abnormal combustion the end gas fuel air mixture undergoes fast chemical reactions, which results in auto ignition prior to normal combustion (i.e. the flame front reaching it). During auto ignition a large portion of end gas releases its chemical energy rapidly and spontaneously at a rate **5 to 25** times as in case of normal combustion. This spontaneous ignition of the End gas raises the pressure very rapidly and causes high frequency oscillations inside the cylinder resulting in a high pitched metallic noise characterized as **KNOCK**. During this knocking phenomenon pressure waves of very large amplitudes propagate across the combustion chamber and very high local pressures are produced which are as high as **150 to 200 bars**. Local 5% of the total charge is sufficient to produce a very violent severe knock. The velocity reached during knock is of the order of **300 to 1000 m/s**.

Basically knock depends on the outcome of shorter of two different processes (i.e. Least time taken by one of the two processes) they are:

1. The advancing flame front grabbing all the fuel air mixture.
2. The pre combustion reaction in the unburned end gas. The time taken in this preparative phase of auto ignition (i.e. pre combustion reaction) is called "**Ignition delay**". Knock will not occur if the ignition delay is so long that the flame front consumes all the end gas and auto ignition takes place i.e. normal combustion occurs. Knock will occur if the pre combustion reaction produces auto ignition before the flame front arrives. Auto ignition when it occurs repeatedly, the phenomenon is called "Spark Knock". Spark knock is controllable by spark

advance: advancing the spark increases the knock intensity and retarding the spark decreases the knock.

CHARACTERISTICS OF KNOCK :

- ❖ Knock in S.I. engine generally occurs at the end of combustion process during which the end gas charge is trapped between the advancing flame front and the engine. Cylinder periphery.
- ❖ Knock primary occurs under wide open throttle operating condition thus it is a direct constrain on engine performance.
- ❖ It also constrains engine efficiency, as it limits the temp and pressure of the end gas and thus limits the compression ratio of the engine
- ❖ The impact of knock depends on its intensity and duration. If knock is short duration usually called "**Acceleration Knock**" it is unlikely to cause damage. But "**Constant Speed Knock**" however can lead to engine damage.

SURFACE IGNITION

The other important abnormal combustion phenomenon is surface ignition. Surface ignition is ignition of fuel air charge by overheated valves or spark plugs, by glowing combustion chamber deposits, or by any other hot spot in the engine combustion chamber. It is the initiation of flame front by a hot surface other than the spark plug. Mostly surface ignition is due to carbon deposits.

Surface ignition may occur before the spark plug ignites the charge (pre ignition) or after normal ignition (post ignition). It may produce a single flame or multiple flames. Uncontrolled combustion is most evident and the results are most severe in case of pre ignition. Surface ignition may also cause knock. As due surface ignition there is a rapid and high rise of pressure and temperature than in case of normal combustion

because the flame either starts propagating sooner, or it does from more than one sources, which likely results in Knocking.

Knocking is also an outcome of Surface ignition. To identify whether surface ignition causes knock or not, the term "**knocking surface ignition**" and "**non-knocking surface ignition**" are used. Knocking surface ignition usually occurs due to pre ignition caused by glowing combustion chamber deposits and can't be controlled by retarding the spark timing since knock is not due to spark ignited flame. While the non-knocking surface ignition occurs late in the operating cycle.

Different surface ignition phenomenon is wild ping, run on, run away, rumble, etc.

Knocking surface ignition may give rise to "**Wild Ping**" and non-knocking surface ignition to "**Rumble**". While both knocking and non-knocking surface ignition may give rise to "**Run-On**" and "**Run-Away**".

Wild Ping

Wild ping is a variation of knocking surface ignition, which produces sharp cracking noise in bursts. Probably it results from early ignition of the fuel air mixture in the combustion chamber by glowing loose deposit particles. It disappears when the particles are exhausted and reappears when fresh particles break loose from the combustion chamber surfaces.

Rumble

Rumble is a relatively stable low frequency noise (600-1200 Hz) phenomenon associated with deposit -caused surface ignition in high compression ratio engines. The pressure rises rapidly to a high value and resulting in engine vibration. Rumble and knock can occur together.

Run-on

Run-on occurs when the fuel air mixture within the cylinder continues to ignite when the ignition system is switched off. During run-on the engine usually produces knock like noises. It is probably caused by the compression ignition of fuel air mixture (assisted by surface ignition).

Run-away

Run-away is surface ignition which occurs earlier and earlier in the cycle. It is usually caused by overheated sparkplugs or valves. It is the most destructive type of surface ignitions which may lead to disastrous results—seizure or melting of piston or the engine catching fire. It can lead to serious overheating and structural damage to the engine.

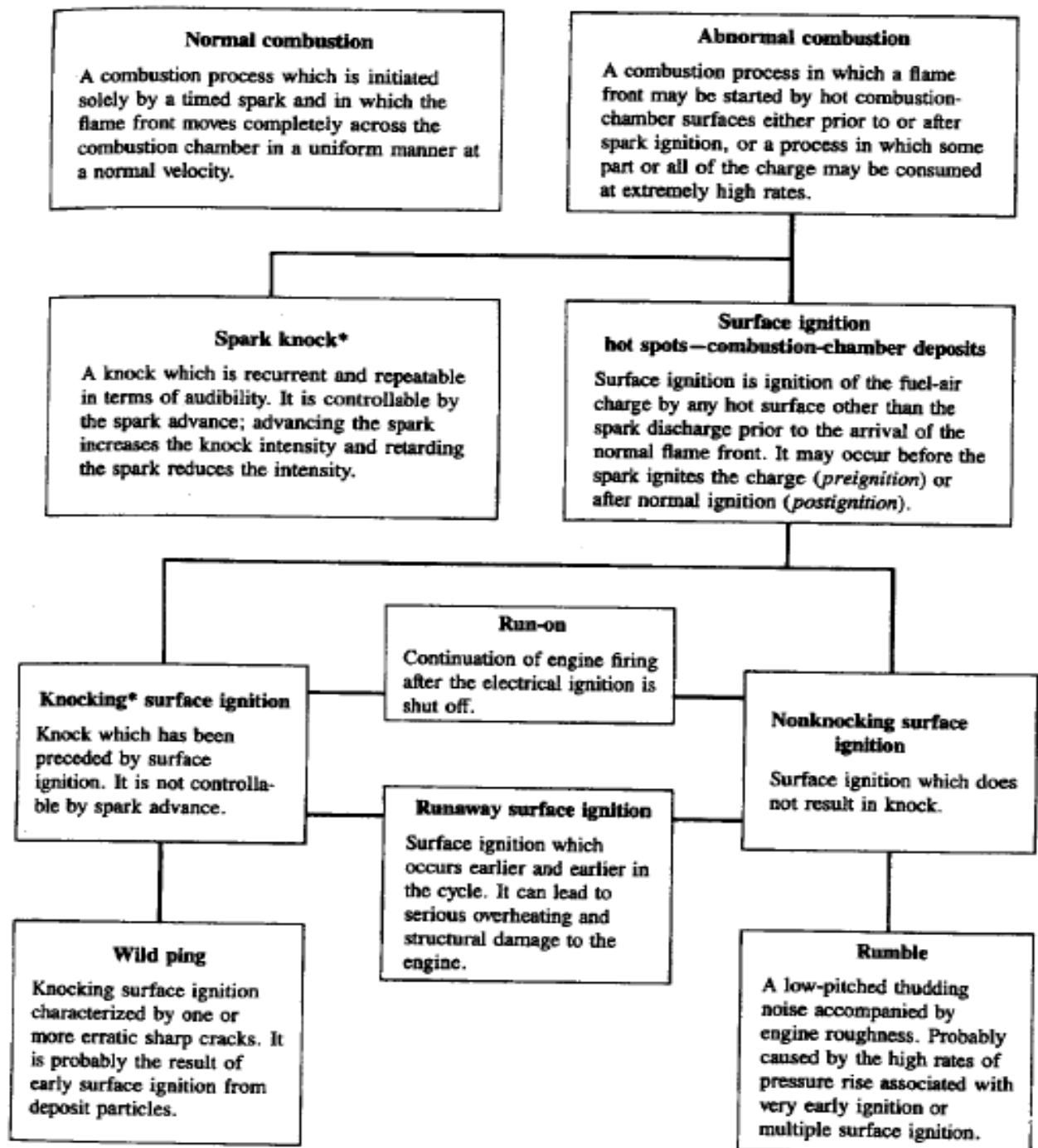
PRE-IGNITION

As discussed earlier, pre ignition is the phenomenon of surface ignition before the passage of spark. The usual cause is an overheated spot, which may occur at spark plugs, combustion chamber deposits, or exhaust valves. Mostly it is due to spark plug. Exhaust valve usually runs hot and sometimes when there is an increase in heat load for these valves there will be an increase in the temperature and may cause pre ignition. Heat transfer principles indicate that the surface of the deposits is hotter than the metal surface to which the deposits are attached. Hence, sufficient deposits result in hot enough surfaces to cause pre ignition.

Pre ignition is potentially the most damaging surface ignition phenomenon. The effect of pre ignition is same as very advanced ignition timing. Any process that advances the start of combustion that gives maximum torque will cause higher heat rejection because of the increased burned gas pressures and temperatures (due to the negative work done during the compression stroke). Higher heat rejection causes higher temperature components thus the pre ignition damage is largely thermal which is evidenced by the fusion of spark plugs, piston and destruction of piston rings.

Lecture – 7

Factors affecting knock



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

What is Knocking?

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.

Effect of Knock:

1. Knock has the following effects on engine operation:
2. Noise and Roughness.
3. Mechanical damage: increase in engine wear, cylinder head and valves may be pitted.
4. Carbon deposits.
5. Increase in heat transfer.
6. Decrease in power output and efficiency.
7. Pre-ignition: combustion Occurs before the spark.

Effect of engine variables on Knock:

To prevent Knock in the S.I. engine the end gas should have:

- A- Low temperature.
- B- Low density.
- C- Long ignition delay.
- D- Non- reactive combustion.

When the engine conditions are changed, the effect of the change may be reflected by more than one of the above variables.

A- Temperature factors:

The temperature of the unburned mixture is increased by the following factors:

1. Raising the compression ratio.
2. Supercharging.
3. Raising the inlet temperature.
4. Raising the coolant temp.

5. Increasing load.
6. Advancing the spark.
7. Raising the temperature of the cylinder and combustion chamber walls.

B- Density factors:

Increasing density by any of the following methods, will increase the possibility of Knock:

1. Increasing load.
2. Increasing compression ratio.
3. Supercharging.
4. Advancing the spark.

C- Time factors:

Increasing the time of exposure of the unburned mixture to auto-ignitions by any of the following factors will increase tendency to knock:

1. Increasing the distance of the flame travel.
2. Decreasing the turbulence of mixture.
3. Decreasing the speed of the engine.

D- Composition:

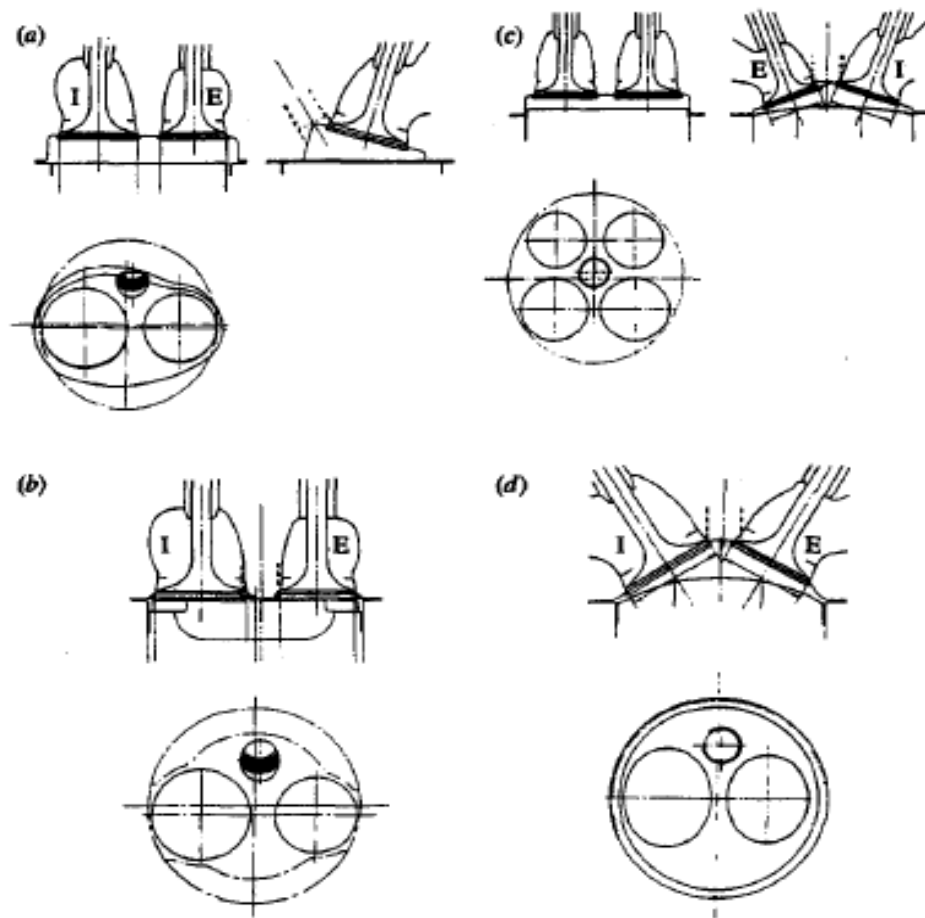
The probability of Knock in S.I. engines is decreased by:

1. Increasing the octane rating of the fuel.
2. Either rich or lean mixtures.
3. Stratifying the mixture.
4. Increasing the humidity of the entering air.

LECTURE – 8**Combustion chambers**

There are basically four types of combustion chambers they are,

- a. Bathtub and wedge.
- b. Blow in piston.
- c. Four valve pent proof.
- d. Hemispherical.



Types of Combustion chambers

The major combustion chamber design objectives which relate to engine performance and emissions are: (1) a fast combustion process, with low cycle-by-cycle variability, over the full engine operating range; (2) a high volumetric efficiency at wide-open throttle; (3) minimum heat loss to the combustion chamber walls; (4) a low fuel octane requirement.

Many methods for producing a "fast burn" have been proposed. These include ways of making the combustion chamber shape more compact, moving the spark plug to a more central location within the chamber, using two plugs, and increasing in-cylinder gas motion by creating swirl during the induction process or during the latter stages of compression. A faster combustion process relative to more moderate burn rate engines does result in a direct engine efficiency gain, other factors being equal. The magnitude of this direct gain is relatively modest. Experimental studies of the effect of an increase in burn rate from moderate to fast at constant engine load, speed, and mixture composition show that this effect is a few percent at

most.²³ Computersimulations of the engine operating cycle confirm these experimental observations: while a decrease in total burn duration from 100 to **60°** (slow to moderate burn) does result in a 4 percent decrease in bsfc, a decrease in burn duration from 60 to 20° gives only a further 1.5 percent bsfc decrease. Of greater importance is the fact that the faster burn process is more robust and results in the engine being able to operate satisfactorily with much more EGR, or much leaner, without a large deterioration in combustion quality. Faster burning chamber designs exhibit much less cycle-by-cycle variability. This ability to operate with greater dilution at part load while maintaining a short burn duration and low cycle-by-cycle variability, permits much greater control of NO_x within the engine with 20 or more percent **EGR** without any substantial increase in HC emissions (see Fig. 11-29), or permits very lean operation. In both cases the efficiency gain relative to moderate burn rate engines, *which must operate with less dilution*, is sizeable.²⁴ High volumetric efficiency is required to obtain the highest possible power density. The shape of the cylinder head affects the size of valves that can be incorporated into the design. Effective valve open area, which depends on valve diameter and lift, directly affects volumetric efficiency. Swirl is used in many modern chamber designs to speed up the burning process and achieve greater combustion stability. Induction-generated swirl appears to be a particularly stable in-cylinder flow. Swirl results in higher turbulence inside the chamber during combustion, thus increasing the rate of flame development and propagation. Generating swirl during the intake process decreases volumetric efficiency. Heat transfer to the combustion chamber walls has a significant impact on engine efficiency. It is affected by cylinder head and piston crown surface area, by the magnitude of in-cylinder gas velocities during combustion and expansion, by the gas temperatures and the wall temperatures. The heat-transfer implications of a combustion chamber should be included in the design process. Knock effectively limits the maximum compression ratio that can be used in any combustion chamber; it therefore has a direct impact on efficiency. Knock is affected by all the factors discussed above. It is the hardest of all the constraints to incorporate into the design process because of its obvious complexity. Knowledge of the fundamentals of spark-ignition engine combustion, in-cylinder gas motion, and heat transfer has developed to the point where a rational procedure for evaluating these factors for optimum combustion chamber development and design can be defined. The next two sections develop such a procedure.

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Knowledge of the fundamentals of spark-ignition engine combustion, in-cylinder gas motion, and heat transfer has developed to the point where a rational procedure for evaluating these factors for optimum combustion chamber development and design can be defined. The next two sections develop such a procedure. Burning chamber designs exhibit much less cycle-by-cycle variability. This ability to operate with greater dilution at part load while maintaining a short burn duration and low cycle-by-cycle variability, permits much greater control of NO_x. Within the engine with 20 or more percent **EGR** without any substantial increase in HC emissions, or permits very lean operation. In both cases the efficiency gain relative to moderate burn rate engines, *which must operate with less dilution*, is sizeable. High volumetric efficiency is required to obtain the highest possible power density. The shape of the cylinder head affects the size of valves that can be incorporated into the design. Effective valve open area, which depends on valve diameter and lift, directly affects volumetric efficiency. Swirl is used in many modern chamber designs to speed up the burning process and achieve greater combustion stability. Induction-generated swirl appears to be a

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LECTURE – 9

Thermodynamic analysis of SI engine combustion process

Burned and Unburned Mixture States

The gas pressure, temperature, and density change as a result of changes in volume due to piston motion. During combustion, the cylinder pressure increases due to the release of the fuel's chemical energy. As each element of fuel-air mixture burns, its density decreases by about a factor of four. This combustion-produced gas expansion compresses the unburned mixture ahead of the flame and displaces it toward the combustion chamber walls. The combustion-produced gas expansion also compresses those parts of the charge which have already burned, and displaces them back toward the spark plug. During the combustion process, the unburned gas elements move away from the spark plug; following combustion, individual gas elements move back toward the spark plug. Further, elements of the unburned mixture which burn at different times have different pressures and temperatures just prior to combustion, and therefore end up at different states after combustion. The thermodynamic state and composition of the burned gas is, therefore, non-uniform. A first law analysis of the spark-ignition engine combustion process enables us to quantify these gas states. Work transfer occurs between the cylinder gases and the piston (to the gas before TC; to the piston after TC). Heat transfer occurs to the chamber walls, primarily from the burned gases. At the temperatures and pressures typical of spark-ignition engines it is a reasonable approximation to assume that the volume of the reaction zone where combustion is actually occurring is a negligible fraction of the chamber volume even though the thickness of the turbulent flame may not be negligible compared with the chamber dimensions (see Sec. 9.3.2). With normal engine operation, at any point in time or crank angle, the pressure throughout the cylinder is

close to uniform. The conditions in the burned and unburned gas are then determined by conservation of mass:

$$\frac{V}{m} = \int_0^{x_b} v_b dx + \int_{x_b}^1 v_u dx$$

and conservation of energy:

$$\frac{U_0 - W - Q}{m} = \int_0^{x_b} u_b dx + \int_{x_b}^1 u_u dx$$

where V is the cylinder volume, m is the mass of the cylinder contents, v is the specific volume, x_b is the mass fraction burned, U_0 is the internal energy of the cylinder contents at some reference point θ_0 , u is the specific internal energy, W is the work done on the piston, and Q is the heat transfer to the walls. The subscripts u and b denote unburned and burned gas properties, respectively. The work and heat transfers are

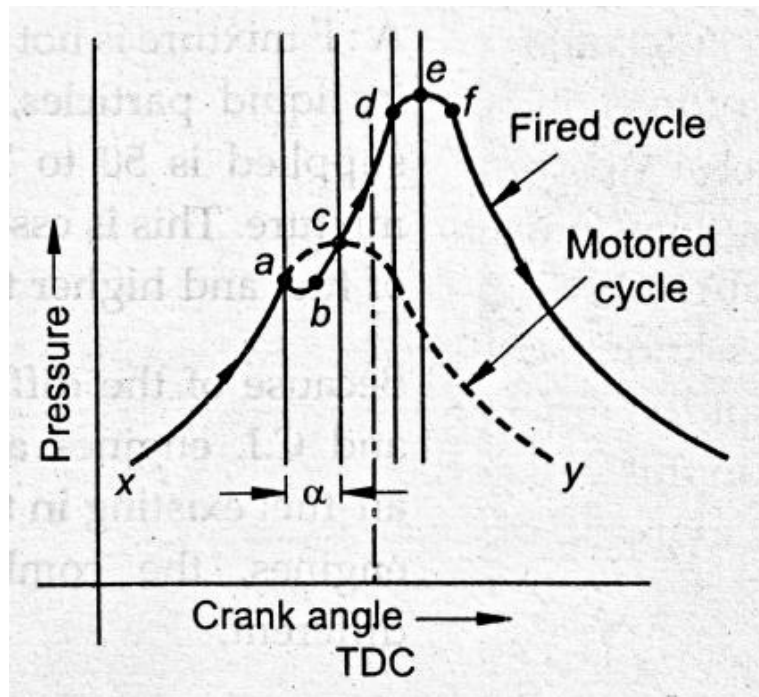
$$W = \int_{V_0}^V p dV' \quad Q = \int_{\theta_0}^{\theta} \left(\frac{\dot{Q}}{360N} \right) d\theta$$

where \dot{Q} is the instantaneous heat-transfer rate to the chamber walls. To proceed further, models for the thermodynamic properties of the burned and unburned gases are required. Several categories of models are described in Chap. 4. Accurate calculations of the state of the cylinder gases require an equilibrium model (or good approximation to it) for the burned gas and an ideal gas mixture model (of frozen composition) for the unburned gas. However, useful illustrative results can be obtained by assuming that the burned and unburned gases are different ideal gases, each with constant specific heat.

UNIT-II COMPRESSION IGNITION ENGINES

LECTURE – 1

Stages of combustion:



Stages of combustion

Stages of combustion can be divided into four stages namely,

1. Pre-flame combustion
2. Uncontrolled combustion
3. Controlled combustion and
4. After burning.

Pre flame combustion:

In actual engine cycle, the fuel injection starts at the point 'a' shown in fig. As soon as the fuel jet is known into a fine spray, the fuel starts absorbing heat from the surrounding high temperature air and vaporization of fuel starts. But in the absence of flame, therefore it is known as pre-flame reaction. At the beginning of pre-flame combustion, the energy release rate is very less than rate of heat absorption by the fuel because the amount of fuel vapour is small. As a result, the pressure in the cylinder decreases with the progressive fuel

vaporization. This decrease in pressure attains a maximum value when the energy release due to pre-flame reaction is equal to the rate of heat absorption by the fuel. This process of fuel vaporization and subsequent decrease in pressure in the cylinder is shown on fig by paths 'ab'.

As the energy release rate due to pre-flame reaction is more than the rate of heat absorption, the pressure inside the cylinder starts increasing. This rising pressure intersects the pressure curve without the fuel injection at the point 'c'. At point 'c', the pressure drop caused by the fuel vaporization is completely recovered by the energy released due to preflame combustion. The pressure inside the cylinder after the point 'c' rapidly increases as the ignition takes place somewhere around the point 'c' and flame appears. The actual flame (actual combustion) starts at the point 'c' where as the fuel injection starts at point 'a'. The time required to start the actual combustion of after starting the fuel injection is known as "delay period" and the crank angle required for this is known as "delay period angle" and it is shown in the fig by an angle α .

Uncontrolled Combustion:

The time and place where ignition will stop is not fixed by anything in compression ignition engine as in SI engines.

The air fuel mixture in the combustion chamber before starting the combustion is very heterogeneous and the concentration of the fuel may vary from 0 to 100%. The first ignition (flame) generally occurs in the region of chemically correct A:F mixture because it requires minimum reaction time. Once the ignition takes place, the flame formed propagates through the mixture of air and vaporized fuel and ignites the adjacent part of the charge or it may initiate the auto ignition in the part of A:F mixture away from the flame front by transferring the heat by radiation.

A considerable amount of fuel is accumulated in the combustion chamber during the relay period (time between the start of injection of fuel and start of ignition of fuel). This accumulated fuel burns very rapidly causing a steep raise in the cylinder pressure. The rate of pressure raise increases with the increase in delay period because of the amount of fuel taking part in this combustion increase with an increasing delay period. This phase of combustion causing rapid pressure raise in the cylinder is known as "period of uncontrolled combustion".

Controlled combustion:

All the accumulated fuel during the delay period generally burns during the period of controlled combustion. The fuel injected after this (after point d) burns at the same rate at which it is injected because, the vaporisation of fuel, mixing with the air and burning takes place almost instantaneously as the fuel leaves the nozzle. This is because, the temperature and pressure inside the cylinder are sufficiently high and sufficient turbulence is created due to previous burning, thus the delay period for the fuel injected after point “d” is almost zero. This period of combustion is known as “controlled combustion” because the rate of burning can be controlled by controlling the rate of injection. This is confirmed until the supply of fuel ceases. This process is shown by the path “de” on the fig.

After burning:

The thermal decomposition of the part of fuel takes place during uncontrolled and controlled combustion. The decomposed fuel molecules contain enough number of hydrocarbons and carbon particles which has lower reaction rates. Some carbon and hydrocarbon, decomposed from fuel are left in the combustion product because the rate of decomposition during uncontrolled and controlled combustion is more than the rate of reaction of these molecules during that period. These unburned hydrocarbons and carbon generally burn after stopping the fuel injection during the expansion stroke. This process of combustion of decomposed carbon atoms is known as “after burning”.

LECTURE – 2**Abnormal Combustion****Delay period:**

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 of 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 the following:

1. Physical delay period.
2. Chemical delay period.

Physical delay period:

Physical delay period or the mixing period is the time elapses between the beginning of fuel injection and the beginning of pre flame reactions. During this period, heating, and vaporization of the fuel drop and diffusion of air into the vapour layer takes place. This results in the formation of suitable mixture of fuel vapour and air ready for chemical reaction.

Chemical delay period:

Chemical delay period or integration period is the time that elapses between the beginning of chemical and beginning of ignition. During this period, the fuel vapour is being oxidized at an ever increasing rate until ignition occurs. In other words, this is the period taken up by the chemical reactions to attain the point of burning or self-ignition. In addition to the above, the delay period is also influenced by the degree of atomization i.e. disintegration of fuel jet injected and characteristics of combustion chamber. The delay is the more or less constant in time units. In high speed diesel engines, ignition delay is in the order of 0.0012 to 0.0018 seconds.

Factors influencing delay period:

1. Characteristics of fuel (self-ignition temperature, volatility and viscosity).
2. Temperature and pressure of compressed air.
3. Degree of atomization of injected fuel.
4. Air motion / turbulence present in the combustion space.
5. Engine speed.
6. Injection timing.
7. Characteristics of the combustion chamber.

LECTURE – 3& 4**Factors affecting knock**

Knocking is violent gas vibration and audible sound produced by extreme pressure differentials leading to the very rapid rise during the early part of uncontrolled second phase of combustion.

In C.I. engines the injection process takes place over a definite interval of time. Consequently, as the first few droplets injected are passing through the ignition lag period, additional droplets are being injected into the chamber. If the ignition delay is longer, the

actual burning of the first few droplets is delayed and a greater quantity of fuel droplets gets accumulated in the chamber. When the actual burning commences, the additional fuel can cause too rapid a rate of pressure rise, as shown on pressure crank angle diagram above, resulting in Jamming of forces against the piston (as if struck by a hammer) and rough engine operation. If the ignition delay is quite long, so much fuel can accumulate that the rate of pressure rise is almost instantaneous. Such, a situation produces extreme pressure differentials and violent gas vibration known as knocking (diesel knock), and is evidenced by audible knock. The phenomenon is similar to that in the SI engine. However, in SI Engine knocking occurs near the end of combustion whereas in CI engine, knocking that occurs near the beginning of combustion.

Delay period is directly related to Knocking in CI engine. An extensive delay period can be due to following factors:

- A low compression ratio permitting only a marginal self-ignition temperature to be reached.
- A low combustion pressure due to worn out piston, rings and bad valves
- Low cetane number of fuel
- Poorly atomized fuel spray preventing early combustion
- Coarse droplet formation due to malfunctioning of injector parts like spring
- Low intake temperature and pressure of air

FACTORS AFFECTING KNOCKING

Injection timing

At normal operating conditions min ignition delay (ID) occurs with start of injection at 10 to 15 OCA BTDC.

Cylinder temperature and pressure drops if injection is earlier or later (high at first but decrease as delay proceeds).

Injection quantity (load)

Reducing engine load changes AFR, cools down the engine, reduces wall temperatures, reduces residual gas temperatures and increases ID

Droplet size, injection velocity and rate

Ignition quality within practical limits does not have significant effect on ID including in injection produces only modest decreasing in ID Injector nozzle diameter effects droplet size but has no significant effect on ID

Intake air temperature and pressure

Reducing intake air T and p increases ID.

Strong dependence of ID on charge temperature below 1000 K – above this value effect of intake air conditions is not significant.

Engine speed

Increase in engine speed increases the air motion and turbulence, reduces ID time slightly (in ms), in terms of CA degrees ID increases almost linearly. A change in engine speed, changes “temp~time” and “pressure~time” relationships.

Combustion chamber design

Spray impingement on the walls effect fuel evaporation and ID

Increase in compression ratio, increases pressure and temperature and reduces ID

Reducing stroke volume, inc surface area to volume ratio, increases engine cooling and increases ID

Swirl rate

Change evaporating rate and air-fuel mixing - under normal operating conditions the effect is small.

At start-up (low engine speed and temperature) more important, high rate of evaporation and mixing is obtained by swirl

Oxygen concentration

Residual gases reduce O₂ concentration and reducing oxygen concentration increases ID.

SN	Variable	Effect on delay period	Reason
1.	Cetane number of fuel	Reduces	Reduces self-ignition temperature
2.	Injection pressure	Reduces	Greater surface volume ratio, less physical delay
3.	Injection advance angle	Increases	Pressures and temperatures lower when injection begins
4.	Compression ratio	Reduces	Increases air temperature and pressure and reduces auto-ignition temperatures.
5.	Intake temperature	Reduces	Increases air temperature
6.	Intake Pressure	Reduces	Increase in density reduces auto-ignition temperature
7.	Speed of engine	Reduces in milli-seconds but increases in crank angle	Less loss of heat more crank angle in a given time
8.	Load(fuel-air-ratio)	Decreases	Operating temperature increases

METHODS OF CONTROLLING DIESEL KNOCK

We have discussed the factors which are responsible for the detonation in the previous sections. If these factors are controlled, then the detonation can be avoided.

- Using a better fuel. Higher CN fuel has lower delay period and reduces knocking tendency.
- Controlling the Rate of Fuel Supply. By injecting less fuel in the beginning and then more fuel amount in the combustion chamber detonation can be controlled to a certain extent. Cam shape of suitable profile can be designed for this purpose.
- Knock reducing fuel injector: This type of injector avoids the sudden increase in pressure inside the combustion chamber because of accumulated fuel. This can be done by arranging the injector so that only a small amount of fuel is injected first. This can be achieved by using two or more injectors arranged in out of phase.
- By using Ignition accelerators : C N number can be increased by adding chemical called dopes. The two chemical dopes used are ethyl-nitrate and amyle –nitrate in concentration of 8.8 gm/Litre and 7.7 gm/Litre. But these two increase the NO_x emissions.
- Increasing Swirl : Knocking can be greatly reduced by increasing swirl (or reducing turbulence). Swirl helps in knock free combustion.

LECTURE – 5

Direct injection systems

Diesel injection engines are divided into two categories and they are

- a. Direct – injection engines.
- b. Indirect – injection engines.

Direct – injection engines have a single open combustion chamber into which fuel is injected directly.

Indirect – injection engines have chambers divided into two regions and the fuel is injected into the “prechamber” which is connected to main chamber via nozzle, one or more orifices. IDI engines are only used in small engine sizes.

Direct – Injection systems:

In largest size engines, where mixing rate requirement are less stringent, quiescent direct – injection systems of the type are used. The momentum and energy of the injected fuel jets are sufficient to achieve adequate fuel distribution and rates of mixing with the air. Additional organized air motion is not required. The combustion chamber shape is usually a shallow bowl in the crown of the piston, and a central multihole injector is used.

As engine size decreases, increasing amounts of air swirl are used to achieve faster fuel – air mixing rates. Air swirl is generated by suitable design of the inlet port. The swirl rate can be increased as the piston approaches TC by forcing the air toward the cylinder axis, into a bowl-in-piston type of combustion chamber.

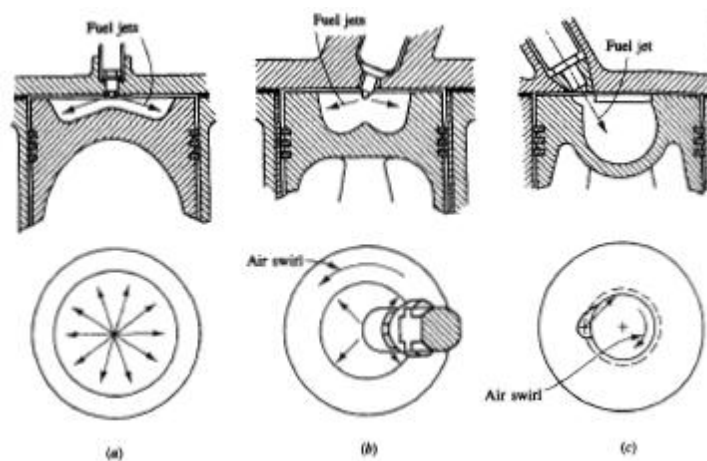


Fig b and c shows the two types of DI engine with swirl in combustion use. Fig b shows a DI engine with swirl, with centrally located multi hole injector nozzle. Here the design is to hold the amount of liquid fuel which impinges nozzle. Here the design goal is to hold the amount of liquid fuel which impinges on the piston bowl walls to a minimum. Fig c shows the M.A.N. “M system” with its single-hole fuel injection nozzle, oriented so that most of the fuel is deposited on the piston bowl walls. These two types of designs are used in

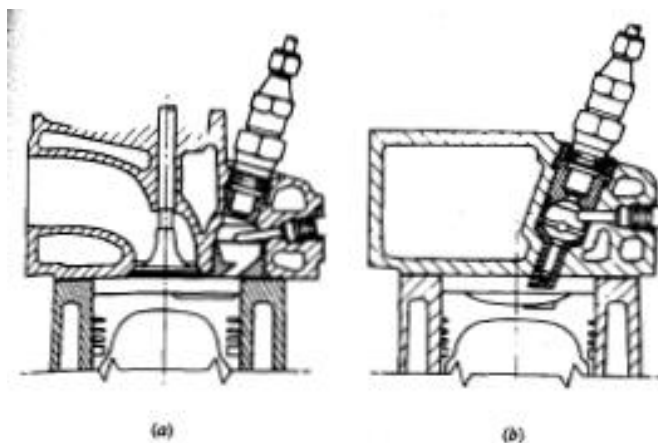
medium-size (10 to 15cm bore) diesels and with increase swirl, in small size (8 to 10cm bore) diesels.

LECTURE - 6

Indirect injection systems

Inlet generated air swirl, despite amplification in the piston cup, has not provided sufficiently high fuel-air mixing rates for small high-speed diesels such as those used in automobiles. Indirect-injection or divided-chamber engine systems have been used instead, where the vigorous charge motion required during fuel injection is generated during the compression stroke. Two broad classes of IDI system can be defined (1) swirl chamber systems and (2) prechamber systems, as illustrated in fig a and b, respectively. During compression, air is forced from the main chamber above the piston into the auxiliary chamber, through the nozzle or orifice (or set of orifices). Thus, toward the end of the compression, a vigorous flow in the auxiliary chamber is set up: in swirl chamber systems the connecting passage and chamber are shaped so that this flow within the auxiliary chamber rotates rapidly.

Fuel is usually injected into the auxiliary chamber at the lower injection-system pressure than is typical of DI systems through a pintle nozzle as a single spray, as shown in fig. Combustion starts in the auxiliary chamber: the pressure rise associated with combustion forces fluid back into the main chamber where the jet issuing from the nozzle entrains and mixes with the main chamber air. The glow plug shown on the right of the pre chamber is a cold starting aid. The plug is heated prior to starting the engine to ensure ignition of fuel early in the engine cranking process.

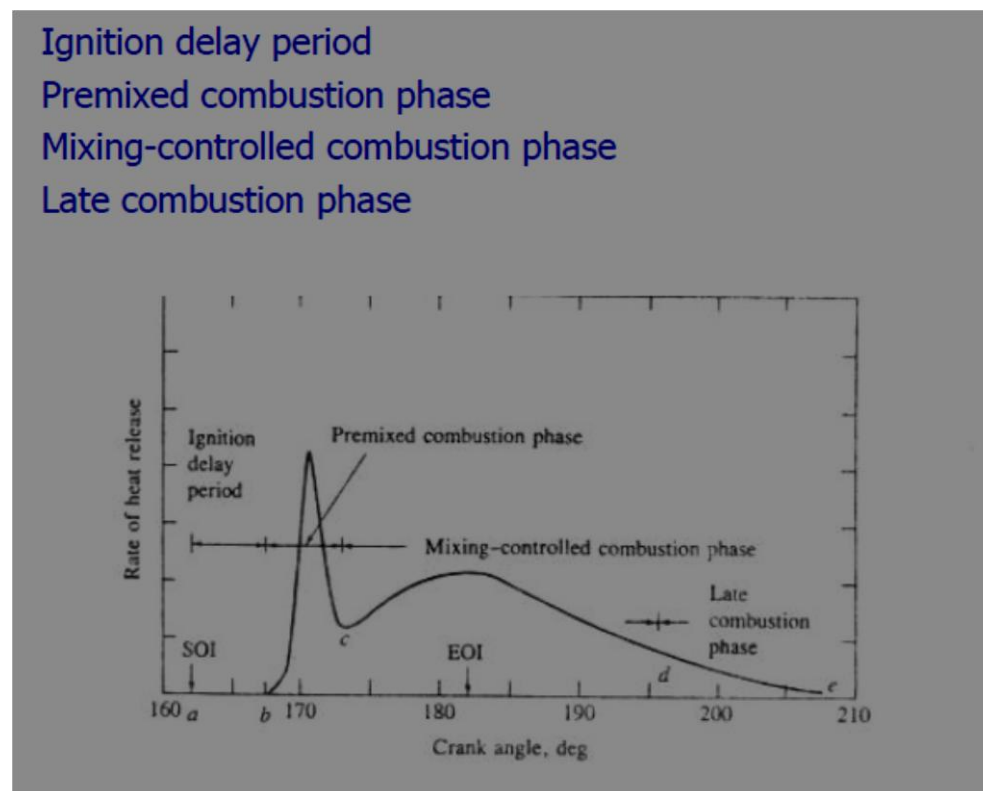


Two types of Indirect-injection diesel engine (a) swirl pre chamber (b) Turbulent pre chamber.

LECTURE – 7

Combustion chambers

In order to study about the combustion chambers we require knowing the combustion process. The below given fig graph shows the various process of combustion in diesel engines.



LECTURE – 8

TURBO CHARGERS

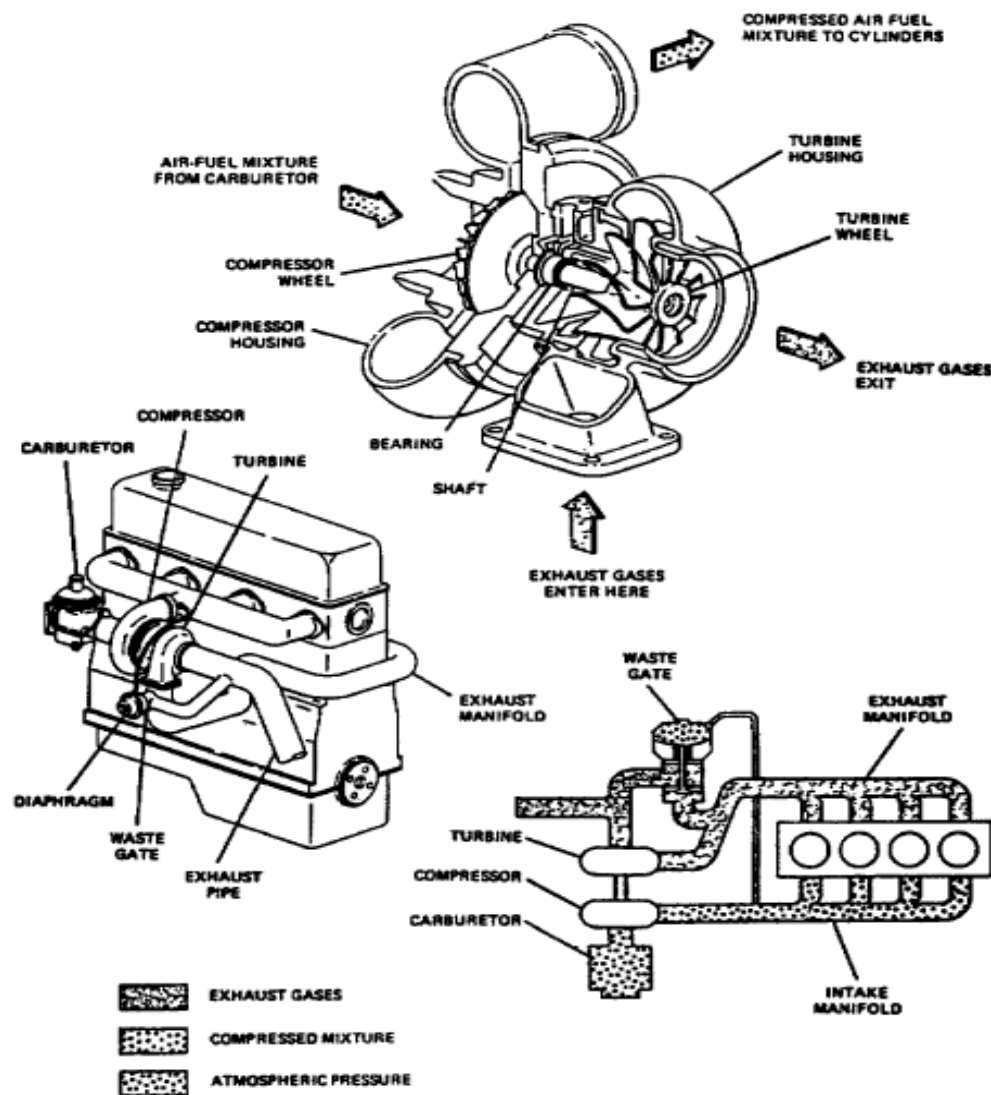
Turbocharging:

Turbocharging is a method of increasing engine volumetric efficiency by forcing the air fuel mixture into the intake rather than merely allowing the pistons to draw it in naturally.

Turbocharger:

A turbocharger uses the force of the engine exhaust stream to force the air fuel mixture into the engine. It consists of a housing containing two chambers. One chamber contains a turbine that is spun as hot exhaust gases are directed against it. The turbine shaft

drives an impeller that is located in the other chamber. The spinning impeller draws an air fuel mixture from the carburetor and forces it into the engine. Because the volume of exhaust gases increases with engine load and speed, the turbocharger speed will increase proportionally, keeping the manifold pressure fairly uniform. A device known as a waste gate is installed on turbocharged engines to control manifold pressure. It is a valve which, when open, allows engine exhaust to bypass the turbocharger turbine, effectively reducing intake pressure. The wastegate valve is operated by a diaphragm that is operated by manifold pressure. The diaphragm will open the waste gate valve whenever manifold pressure reaches the desired maximum.



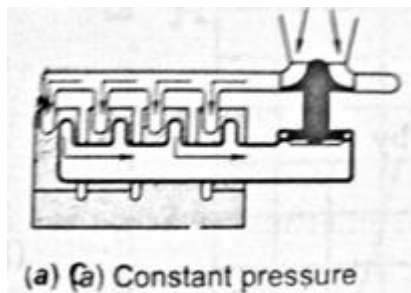
Different types of Turbo-charging:

There are different types of turbocharging, they are

1. Constant pressure turbo-charging.
2. Pulse turbo charging (Buchi-type)
3. Pulse convertor turbo-charging.
4. Complex supercharger.
5. Two stage supercharger.

Constant pressure turbo-charging:

In this system, the exhaust pressure of the engine is constant and higher than atmospheric pressure so that the turbine can operate at an optimum efficiency. The objective dictates a large exhaust manifold to absorb pressure fluctuation and therefore the kinetic energy in the exhaust blow down is dissipated and becomes a reheat factor. This arrangement is shown in figure below.



The exhaust gases from the entire cylinder are released through exhaust valve at a constant pressure in the common manifold and then to turbine. The blow-down energy in the form of internal energy is converted into work in the turbine. The recovery of blow down energy is higher if the pressure ratio of the turbine is high.

Advantages:

1. The pressure ratio is higher in turbine and compressors; the recovery of exhaust energy is efficient and is lower than other systems.
2. The turbine runs at higher efficiency because of constant pressure and temperature of the exhaust gases supplied to the turbine. This method is more efficient and effective than pulse system if the pressure ratio is 3 or higher.
3. The exhaust piping arrangement is very simple for multi cylinder engines and highly efficient turbine can be used.
4. Engine speed is no limited by the pressure waves in the exhaust pipes.

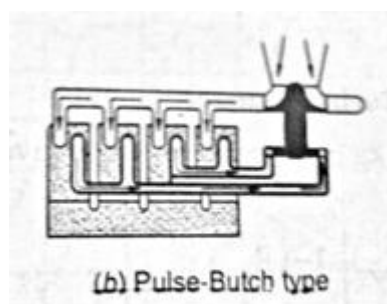
5. Exhaust manifold being large to hold pressure fluctuations with $\pm 5\%$, the pipe diameter is kept 1.4 times piston diameter. However, the insulation of exhaust manifold is essential to preserve the temperature of exhaust gases.

Disadvantages:

1. As the pressure in the exhaust manifold is maintained constant, this requires larger diameter of exhaust pipes. This effect is marked in case of small engines.
2. The response of the system to the load change is considerably poor. Because, acceleration occurs very slowly when load is suddenly increased. This is because, only a small amount of energy due to increased exhaust temperature is available to accelerate the engine at high loads.
3. For higher efficiency of the turbine, the higher pressure drop in the turbine is required. To achieve this, large pressure drop during release is required which makes scavenging a bit difficult (p_2 / p_1) is necessary and do not require acceleration and operation at part loads.
4. The part load η of the turbine is reduced due to partial admission to the turbine.
5. This system is not suitable at all suitable for two strokes engines as it is impossible to run the compressor with the help of turbo charger alone and some additional means have to be provided for supplying the air to the engine.

Pulse turbo charging (Buchi-type):

The main objective of this system is to use the kinetic energy in the blow-down process to drive the turbine without increase in exhaust pressure. To accomplish this objective, the exhaust lines must be small and grouped to receive the exhaust pressure. To accomplish this objective, the exhaust lines must be small and grouped to receive the exhaust from cylinders which are blowing down at different times. This arrangement is shown in figure below.



Advantages:

1. The recovery system of the exhaust blow-down energy is more efficient than constant pressure system.
2. Separate exhaust pipes are used so that the exhaust process of various cylinders do not interfere with one another. A common exhaust pipe can also be used for those cylinders whose exhaust cycles do not overlap significantly in terms of times.
3. The space required is less due to short and smaller diameter pipes.
4. Rapid acceleration of turbo charger to a higher speed can be fed to the turbine without delay.
5. Better scavenging can be obtained at low load due to reduced pressure in the exhaust manifold.

Disadvantages:

1. The recovery of energy is poor when the pressure ratio of turbine is high.
2. Complicated inlet and exhaust piping is required with the multi-cylinder engines.
3. The turbine efficiency becomes poor in case of one or two cylinder engines.
4. The scavenging process is disturbed if the waves have to travel long distance to reach to the turbine.

Application:

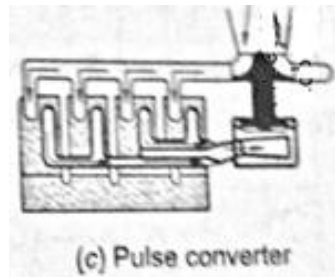
1. This system is widely used for low pressure ratio turbines and rapidly acceleration is required.
2. This is generally used when numbers of cylinder are four or more and connected to a common exhaust pipe.

Pulse converter turbo-charging:

The pulse convertor has the advantages of pulse and the constant pressure turbo-charging to be used simultaneously and avoids most of the drawbacks of both.

A constant pressure turbo-charging requires steady flow for maximum efficiency whereas pulse type turbo-charging operates relatively at lower efficiency due to partial operation. But pulse supercharging operates most efficiently at part load-condition and provides good scavenging also.

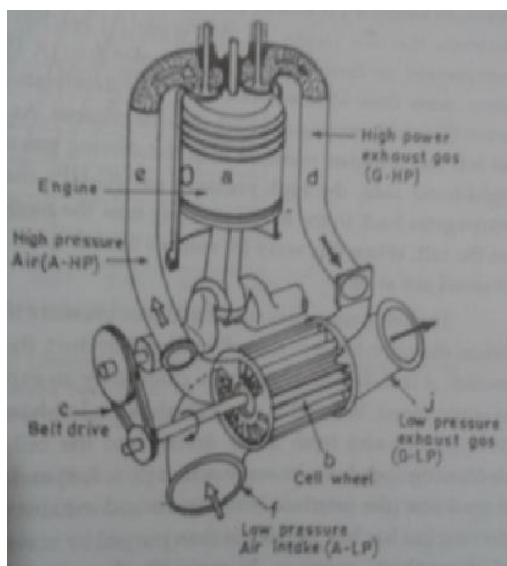
Therefore, a combination of two systems is essential for good efficiency during overall operation of the engine. This is done by connecting different branches of exhaust manifolds together in a specially designed for maximum pulse-system designed for maximum pulse utilisation is retained and the turbine run at full admission conditions to provide good efficiency.



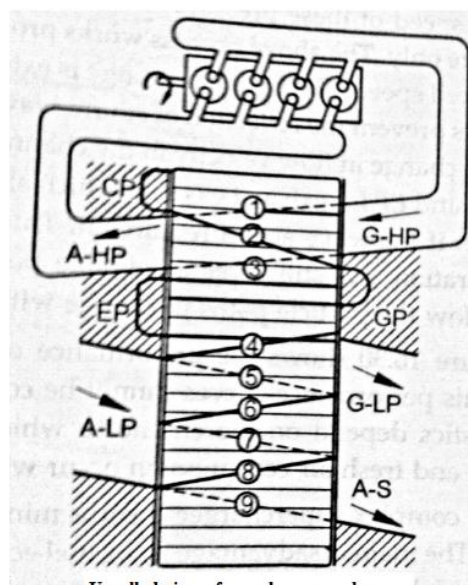
Complex Supercharger:

The characteristics of turbo-chargers are fundamentally different from those reciprocating I.C engines and leads to complex matching problems when they are combined. Supercharger has added complication of mechanical drive and the compressor efficiencies are usually such that the overall economy is reduced. However, flow characteristics are better matched and transient response is good.

The pressure wave superchargers make use of the fact of two fluids having different pressures are brought into direct contact in long narrow channels; equalization of pressure occurs faster mixing. One such device, the complex, has been developed for I.C engine super-charging which operates using this principle.



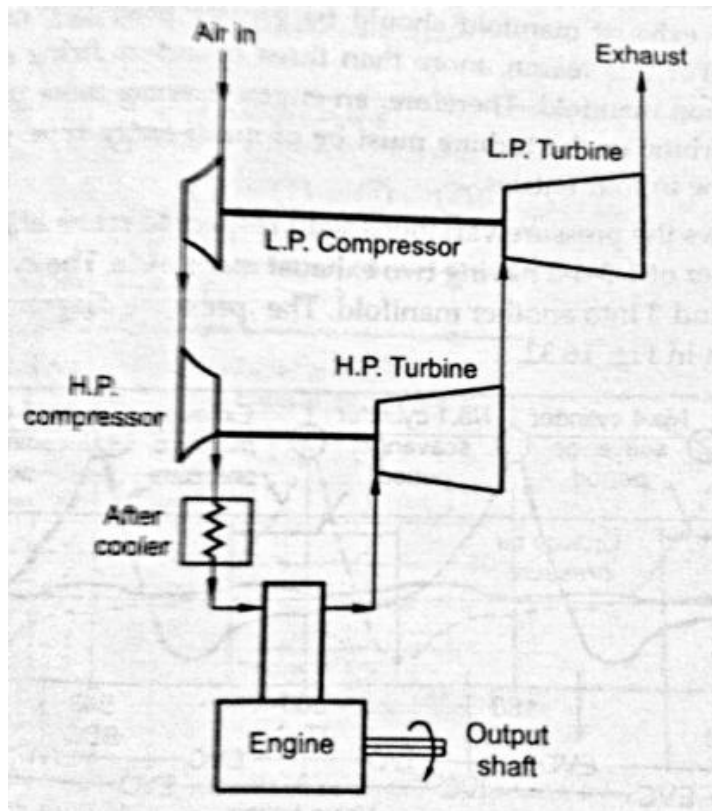
Schematic representation of complex supercharger



Unrolled view of complex supercharger

Two stage supercharger:

For diesel engines, high supercharging is required as 25 to 30 bar BMEP is expected. This cannot be obtained in a single stage turbo-charging so a two stage is used as shown in figure given below.



Advantages:

1. High pressure ratio can be obtained and provides wide range of operation.
2. The efficiency of two stages turbo-charged is higher than single because of higher boost ratio. This gain is further increased by introducing after-cooler.
3. Better matching of turbocharger to engine operating conditions is possible.
4. The transient response of two stages turbo-charged is better than single stage.

Disadvantages:

1. The space requirement is higher.
2. It is heavier and costs higher.
3. Matching of turbo-charged with engine is quite difficult.

LECTURE – 9**Thermodynamic analysis of CI engines:**

Cylinder pressure versus crank angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion. Suitable methods of analysis which yield the rate of release of the fuel's chemical energy (often called heat release), or rate of fuel burning, through the diesel engine combustion process will now be described. The method of analysis starts with the first law of thermodynamics for an open system which is quasi static (i.e., uniform in pressure and temperature). The first law for such a system

$$\frac{dQ}{dt} - p \frac{dV}{dt} + \sum_i \dot{m}_i h_i = \frac{dU}{dt}$$

where dQ/dt is the heat-transfer rate across the system boundary into the system, $p(dV/dt)$ is the rate of work transfer done by the system due to system boundary displacement, \dot{m}_i is the mass flow rate into the system across the system boundary at location i (flow out of the system would be negative), h_i is the enthalpy of flux i entering or leaving the system, and U is, the energy of the material contained inside the system boundary.

The following problems make the application of this equation to diesel combustion difficult:

1. Fuel is injected into the cylinder. Liquid fuel is added to the cylinder which vaporizes and mixes with air to produce a fuel/air ratio distribution which is nonuniform and varies with time. The process is not quasi static.
2. The composition of the burned gases (also nonuniform) is not known.
3. The accuracy of available correlations for predicting heat transfer in diesel which cannot be determined exactly.
4. Crevice regions (such as the volumes between the piston, rings, and cylinder wall) constitute a few percent of the clearance volume. The gas in the regions is cooled to close to the wall temperature, increasing its density and, therefore, the relative importance of these crevices. Thus crevices increase heat transfer and contain a nonnegligible fraction of the cylinder charge at conditions that are different from the rest of the combustion chamber.

Due to difficulties in dealing with these problems, both sophisticated method of analysis and more simple methods give only approximate answers.

UNIT-III ENGINE EXHAUST EMISSION CONTROL**LECTURE – 1****Formation of NO:****Kinetics of NO formation:**

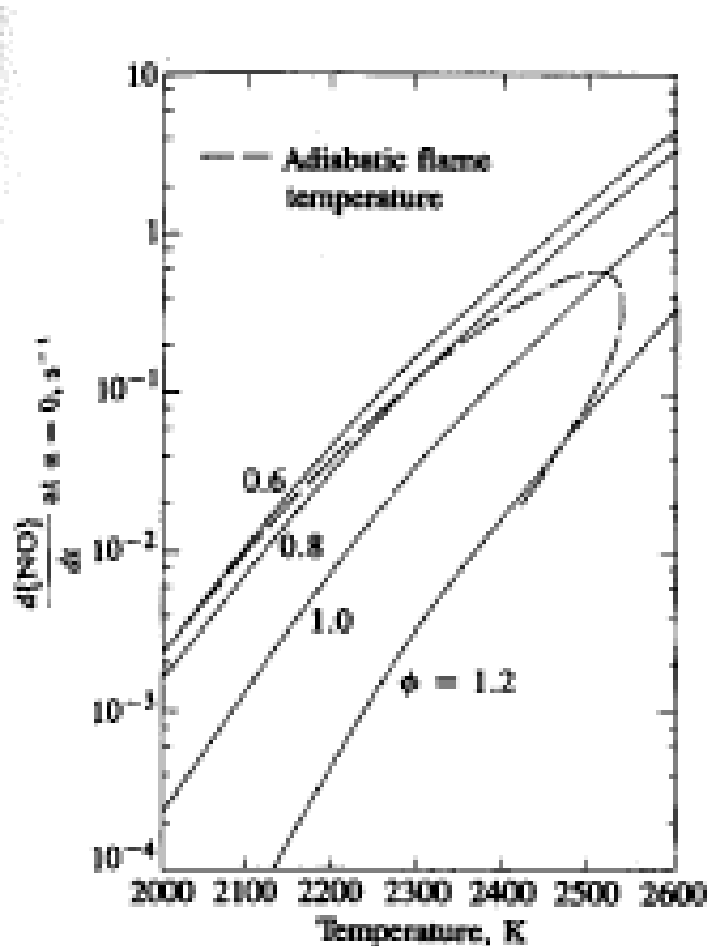
While nitric oxide (NO) and nitrogen dioxide (NO₂) are usually grouped together as NO_x emissions, nitric oxide is the predominant oxide of nitrogen produced inside the engine cylinder. The principal source of NO is the oxidation of atmospheric (molecular) nitrogen. Gasoline contain negligible amounts of nitrogen: although diesel fuels contain more nitrogen, current levels are not significant. The mechanism of nitrogen formation is given below (Zeldovich Mechanism):



NO forms in both the flame front and the postflame gases. In engines however, combustion occurs at high pressure so the flame reaction zone is extremely thin (-0.1 **mm**) and residence time within this zone is short. Also, cylinder pressure rises during most of the combustion process, so burned gases produced early in the combustion process are compressed to a higher temperature than they reached immediately after combustion. Thus, NO formation in the postflame gases almost always dominates any flame-front-produced NO. It is, therefore, appropriate to assume that the combustion and NO formation processes are decoupled and to approximate the concentrations of O, O₂, OH, H and N₂ by their equilibrium temperature.

The strong dependence of $d[\text{NO}]/dt$ on temperature in the exponential term is evident. High temperatures and high oxygen concentrations result in high NO. Figure shows the NO formation rate as a function of gas temperature and fuel/air equivalence ratio in post flame gases. Also shown is the adiabatic flame temperature attained by a fuel-air mixture initially at **700 K** at a constant pressure of 15 atm. For adiabatic constant-pressure combustion (**an**

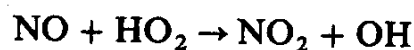
appropriate model for each element of fuel that burns in an engine), this initial NO formation rate peaks at the stoichiometric composition, and decreases rapidly as the mixture becomes leaner or richer.

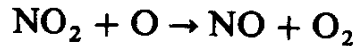


LECTURE – 2

Formation of NO_2

Chemical equilibrium considerations indicate that for burned gases at typical flame temperatures, NO_2 / NO ratios should be negligibly small. While experimental data show this is true for spark ignition engines, in diesels NO_2 can be 10 to 30 percent of the total exhaust oxides of nitrogen emissions. NO formed in the flame zone can be rapidly converted to NO_2 via reactions such as



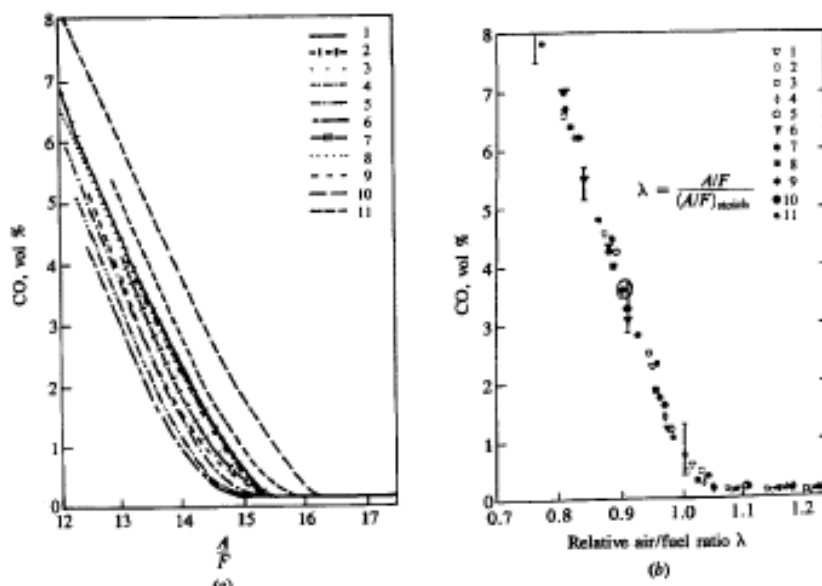


Important factors for formation of NO:

1. Equivalence ratio.
2. Burned gas formation.
3. Excess air.
4. Spark timing.

Formation of CO:

Carbon monoxide (CO) emissions from internal combustion engines are controlled primarily by the fuel/air equivalence ratio. Figure 11-20 shows CO levels in the exhaust of a conventional spark-ignition engine for several different fuel compositions.²⁷ When the data are plotted against the relative air/fuel ratio or the equivalence ratio, they are correlated by a single curve. For fuel-rich mixtures CO concentrations in the exhaust increase steadily with increasing equivalence ratio, as the amount of excess fuel increases. For fuel-lean mixtures, CO concentrations in the exhaust vary little with equivalence ratio and are of order **10⁻³** mole fraction. Since spark-ignition engines often operate close to stoichiometric at partload and fuel rich at full load, CO emissions are significant and must be controlled. Diesels, however, always operate well on the lean side of stoichiometric; CO emissions from diesels are low enough to be unimportant, therefore, and will not be discussed further.



The levels of CO observed in spark-ignition engine exhaust gases are lower than the maximum values measured within the combustion chamber, but are higher than equilibrium values for the exhaust conditions. Thus the processes which govern CO exhaust levels are kinetically controlled. In premixed hydrocarbon-air flames, the CO concentration increases rapidly in the flame zone to a maximum value, which is larger than the equilibrium value for adiabatic of the fuel-air mixture. CO formation is one of the principal reaction steps in the hydrocarbon combustion mechanism, which may be summarized by



where **R** stands for the hydrocarbon radical. The CO formed in the combustion process via this path is then oxidized to CO₂, at a slower rate. The principal CO oxidation reaction in hydrocarbon-air flames is

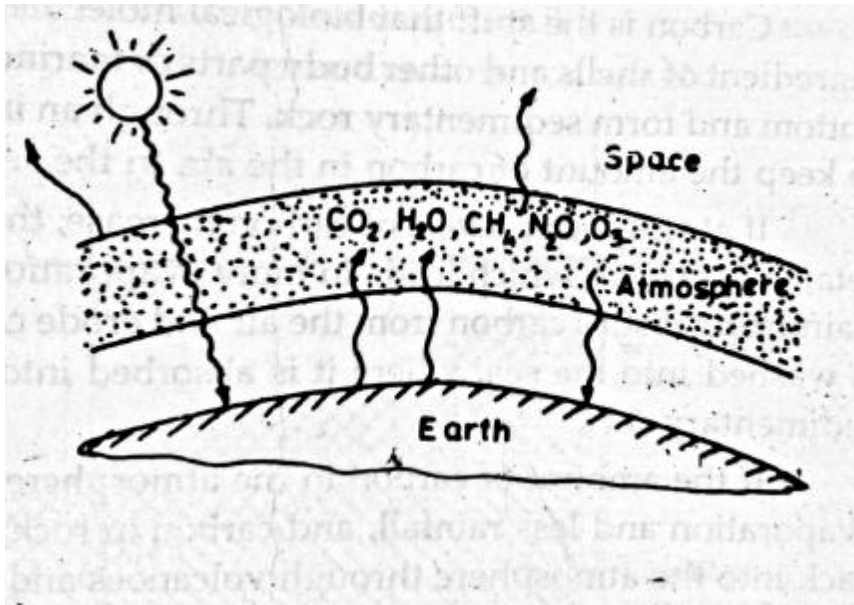


CO increases rapidly as the inlet mixture becomes richer than stoichiometric ratio. And also improved cylinder-to-cylinder fuel/air ratio distribution has become essential. In addition to this, it is necessary to enrich the fuel-air mixture when the engine is cold since CO emission is higher at engine warm up.

LECTURE – 3

Greenhouse effect (GHE)

Radiation from the sun always tries to pass through the earth. This wave length is usually shorter. This shortwave radiation when strikes the inner earth surface of green house, converts into heat long – wave radiation. This long wave radiation is again reflected back into atmosphere from the inside surfaces but it cannot go out as the atmosphere restricts the long wave going to out and traps the heat. This trapped heat (which should have happened without glass) contributes to warming of earth and provides energy for growth of plants. Therefore, this effect is known as greenhouse effect.



The presence of CO_2 in earth atmosphere gives it the warmth essential for the substance of life. This happens because CO_2 has peculiar optical properties. CO_2 absorbs a good part of this long wave radiation, thus warming the globe. This is important life sustained that poses a threat to the human life in the coming decades.

CO_2 is the chief and also CH_4 , NO_2 and certain CFCs have similar effects. Collectively these gases are responsible for greenhouse effect, threatening an average increase in earth temperature by 1.5 to 5°C by the middle of this century which will seriously affect sea level, agriculture and forestry. From the ground, earth atmosphere is nearly invisible and easy to take for granted. From space, it is perceived more readily as a thin blanket of glasses, shielding the earth from sun's UV – radiation and trapping the sun's warmth to keep Earth Rivers and oceans from freezing. The greenhouse gases emitted into the atmospheric functions significantly, degrading the UV-shielding of O_3 – layer and intensifying the heat trapping properties of the atmosphere as a whole.

Factors affecting Greenhouse effect:

Carbon is the stuff that biological molecules are made of. In the oceans, it serves as the basic ingredients of shells and other body parts of marine organisms, which eventually die and sink to the bottom and form sedimentary rock. Through an intricate feedback system, the earth has contrived to keep the amount of carbon in the air, in the sea on the land relatively constant.

If the atmospheric levels of carbon increase, resulting greenhouse effect causes the earth to retain more heat, which leads to more evaporation of water from oceans and thus more rains. Rain drops absorb carbon from the air and erode carbon – laden rock; eventually the excess carbon is washed into the sea, where it is absorbed into seashells and returned to ocean bottom as sedimentary rocks.

If the amount of carbon in the atmosphere decreases, the process is reversed. There is less evaporation and less rainfall, carbon in rock at the ocean's bottom eventually works its way back into the atmospheric through volcanoes and deep-sea vents.

The cycle operates over millions of years; however human disturbances such as the burning of fossils fuels and deforestation have outpaced this natural process, resulting in atmospheric carbon.

Steps taken to control greenhouse effect:

The greenhouse effect is overwhelming in scope and will specially impact on using departments dealing with energy. In accordance with this, some of the steps have been taken by many countries which are as follows:

1. Increased usage of Natural gases instead of fuels which highly emit CO₂.
2. Finding source and using more hydro-power.
3. Use of alternative energy.
4. Increasing the usage of nuclear power.
5. Using efficient equipment that controls CO₂.
6. Eliminating the usage of CFC from refrigeration industry and finding the alternate refrigerant.
7. Increasing forestation and stopping deforestation.
8. Implementing methodology of tax charges respect to carbon emission by industries.

LECTURE – 4

HC emission:

Hydrocarbons (HC) are the consequence of incomplete combustion of hydrocarbon fuel. The level of unburned hydrocarbons (HC) in the exhaust gases specified in

terms of total hydrocarbon concentration expressed in parts per million carbon atoms. Engine exhaust gases contain a wide variety of hydrocarbon compounds.

HC is basically divided into two classes namely methane and non-methane hydrocarbons. Below table shows classifications of hydrocarbons according to their relative reactivity which is the scale of 0 to 100 based on their NO_2 formation relatively to HC.

Reactivity of classes of hydrocarbons

Hydrocarbons	Relative reactivity†
C_1 - C_4 paraffins Acetylene Benzene	0
C_4 and higher molecular weight paraffins Monoalkyl benzenes Ortho- and para-dialkyl benzenes Cyclic paraffins	2
Ethylene Meta-dialkyl benzenes Aldehydes	5
1-Olefins (except ethylene) Diolefins Tri- and tetraalkyl benzenes	10
Internally bonded olefins	30
Internally bonded olefins with substitution at the double bond Cycloolefins	100

LECTURE – 5

Smoke

Formations of smoke and affecting factors:

Engine Exhaust smoke is the result of incomplete combustion. Smoke from exhaust is a visible indicator of the combustion process within the engine. It is generated at any volume in the engine where mixture is rich. The fuel air ratio greater than 1.5 and pressures developed in diesel engine produced soot. Once soot is formed, it can burn if it finds sufficient O_2 otherwise it comes out with exhaust. It becomes visible if it is dense. The size of the soot

particles effect the appearance of smoke. The soot particles agglomerate into bigger particles which have an objectionable darkening effect on diesel exhaust.

Measurement of smoke:

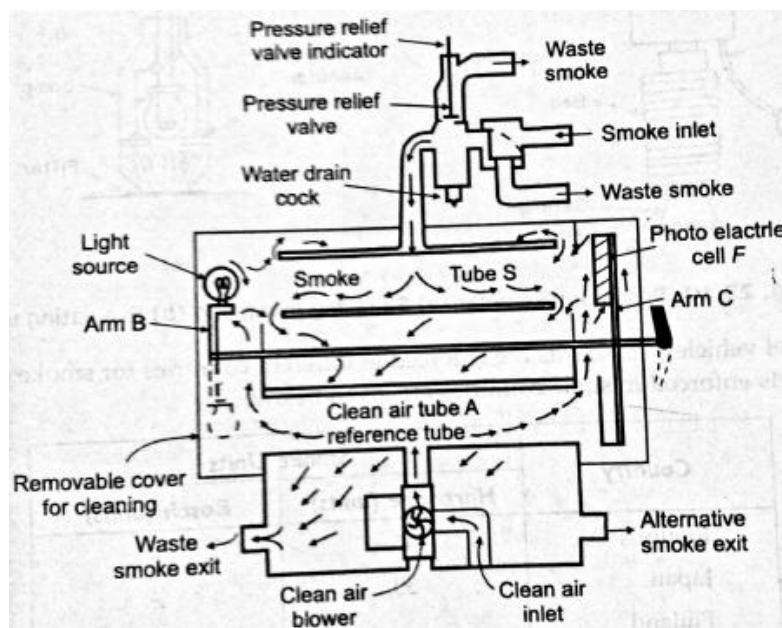
The main purpose of smoke measurement is to quantify the black smoke from diesel engine. As visibility is the main criterion in evaluating the intensity of smoke, development or principle of the smoke meter depends on the light obstruction by the smoke.

Even though there are several smoke measuring types, two basic and main types are discussed in this lecture.

1. Hartridge smoke meter.
2. Bosch smoke meter.

Hartridge smoke meter:

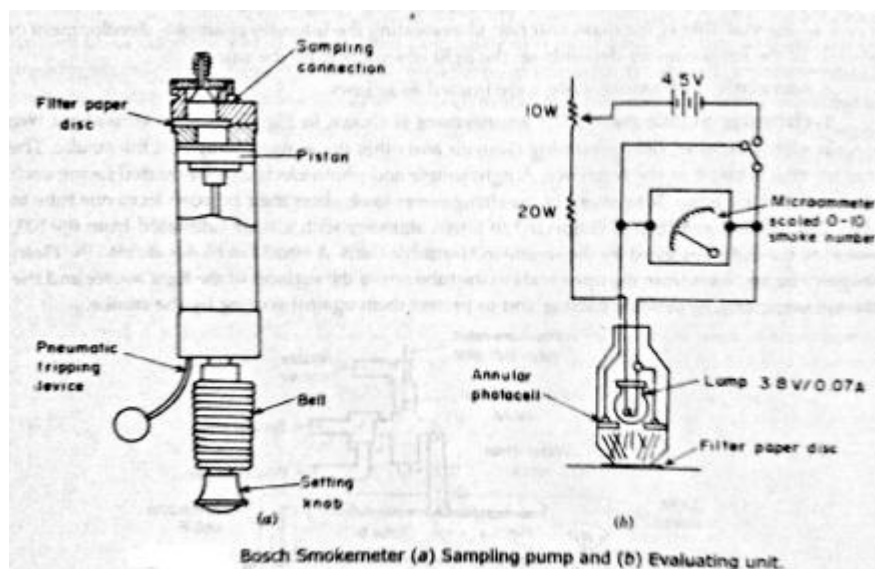
The arrangement is shown in the figure. This consists of two optically identical tubes, one containing clean air and other the moving sample of the smoke. The clean air tube is taken as reference. A light source and photo-electric cell mounted facing each other on swinging arms. Movement of the change-over knob alters their position from 0-100, indicating the light absorbed by the smoke in hartridge units. A small fan blows air into the clean air-tube. The air flow the open ends of the tube across the surfaces of the light source and the photo-electric cell, to provide cooling and to protect them against sooting by the smoke.



The sampling probe is connected either to a tapping on the exhaust pipe. The smoke meter is switched on and control lever set to bring the clean air tube between light and cell. The smoke meter dial should read zero otherwise the meter is to adjusted to read zero. The control lever valve, the meter gives continuous and direct reading of the smoke density.

Bosch smoke meter:

The bosch meter is widely accepted for measuring the diesel engine smoke. This consists of sampling pump and evaluating unit shown in the figure given below. The sampling pump is used to draw nearly 300CC of exhaust gas by means of spring operated pump and released by pneumatic operation of a diaphragm. The gas sample is drawn is through the filter paper darkening to give precise assessment of the intensity of the spot. The intensity of the spot is measured on a scale of 10 arbitrary units called arbitrary units, called Bosch smoke units for white to black.



Particulate emissions:

Spark ignition particulates:

There are three classes of spark-ignition engine particulate emissions organic particulates (including soot), and sulphates.

Significant sulphate emissions can occur with oxidation-catalyst engines. Unleaded gasoline contains 150 to 600 ppm by weight sulphur, which is oxidized within the engine cylinder to sulphur dioxide, SO_2 . This SO_2 can be oxidized by the exhaust catalyst to SO_3 , which

combines with water at ambient temperatures to form a sulphuric acid aerosol. Levels of sulphate emissions depend on the fuel sulphur content, the operating conditions of the engine, and the details of the catalyst system used. Typical average automobile sulphate emission rates are 20 mg/km or less.

The particle size distribution with leaded fuel is about 80 percent by mass below 2 μm diameter about 40 percent below 0.2 μm diameter. Most of these particles are presumed to form and grow in the exhaust system due to vapour phase condensation enhanced by coagulation. Some of the particles are emitted directly, without settling. Some of the particles either form or are deposited on the walls where agglomeration may occur. Many of these are removed when the exhaust flow rate is suddenly increased, and these particles together with rust and scale account for the increase in mass and size of particles emitted during acceleration. Only fraction (between 10 and 50 percent) of the lead consumed in the fuel is exhausted, the remainder being deposited within the engine and exhaust system.

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Diesel particulates:

Diesel particulates consist of principally of combustion generated carbonaceous material (soot) on which some organic compounds has become absorbed. Most particulate material results from incomplete combustion of fuel hydrocarbons: some is contributed by the lubricating oil. The emission rates are typically 0.2 to 0.6 g/km for light duty diesels in an automobile. In larger direct injection engines, particulate emission rates are 0.5 to 1.5 g/brakeKW.h. The composition of particulate material depends on the conditions in the engine exhaust and particulate collection system. At temperatures above 500°C, the individual particles are principally clusters of many small spheres or spherules of carbon (with a small amount of hydrogen) with individual spherule diameter of about 15 to 30 mins. As temperature decreases: below 500°C, the particles become coated with absorbed and condensed high molecular weight organic compounds which include: unburned hydrocarbons, oxygenated hydrocarbons (ketones, esters, ethers, organic acids), and polynuclear aromatic hydrocarbons. The condensed material also includes inorganic species such as sulphur dioxide, nitrogen dioxide and sulphuric acid and sulphates.

Method of controlling emissions:

To reduce atmospheric pollution, two different approaches are followed:

1. To reduce the formation of pollutants in the emission by redesigning the engine system, fuel system, cooling system and ignition system.
2. By destroying the pollutants after these have been formed.

In petrol engine, the main pollutants which are objectionable and are to be reduced are HC, CO and NO_x. These methods are

- a. Modifications in the engine design.
- b. Modifying the fuel used.
- c. Exhaust gas treatment devices.
- d. Evaporative emissive control devices.

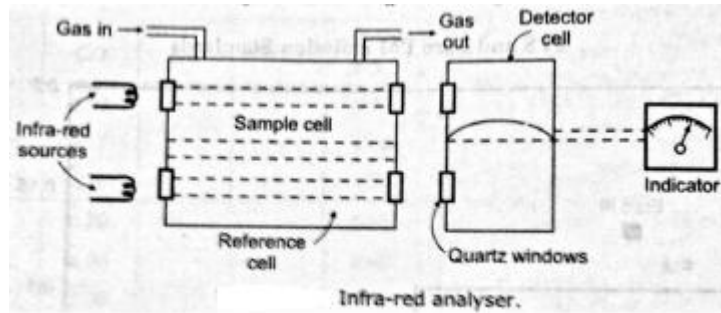
Emission measuring equipment:**Infra-red Absorption Gas analyser for measuring CO:****Principle:**

Infra-red radiation is absorbed by a wide range of gas molecules, each of which has characteristics absorption spectrum. The fraction of radiation (τ_λ) at a particular wavelength λ is given by Beer's law as

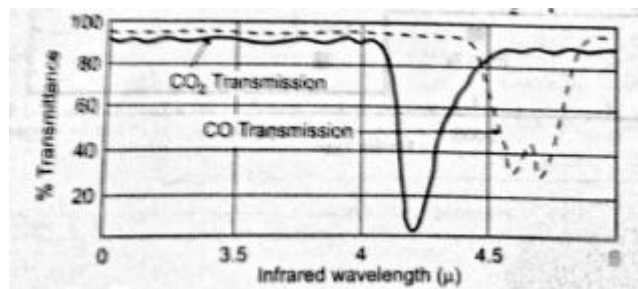
$$\tau_\lambda = (e)^{-\rho \alpha_\lambda L}$$

where ρ is gas density and α_λ is the monochromatic absorptivity and L is the path length.

Fig below shows the arrangement of this analyser. The detector cells are filled with the gas that to be measured (CO or CO₂), so that they absorb the radiation in the wave length band associated with that gas. The energy absorbed in the detector cells causes the cell pressure to rise. The reference cell is present in the sample then infra-red will be absorbed in the sample cell and less infra-red will be absorbed in the detector cell. This cell leads to a differential pressure in the detector cells which can be measured and related to the gas (CO) concentration. The calibration is carried out by passing gasses of known composition through the sample cell.



The below figure shows the absorption spectra of CO and CO₂. This shows that, infra-red radiation is absorbed by both in the region of 4.4 μ . This means that when CO₂ is present in the sample, it will affect the reading of CO and vice versa. This problem is eliminated by using a filter cell between the infra-red sources and the sample and reference cells. If the CO is to be measured, then the filter cell is filled with CO₂ and any CO₂ in the sample should not lead to any infra-red absorption. The windows of the analyser should be made of such materials (mica or quartz) which are transparent to infra-red radiation.

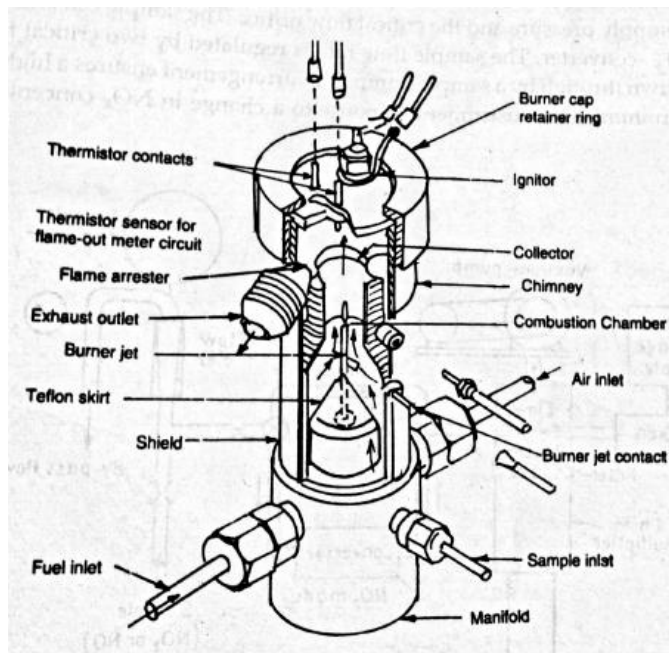


Transmittance of infra-red for CO & CO₂

Flame ionisation detector for measuring HC-emissions:

Principle: When hydro-carbons are burned, electrons and positive ions are formed. If unburned hydrocarbons are burned in the electric field, then the current flow corresponds very closely to the number of carbon atoms present.

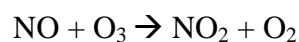
Flame ionisation detector is shown in figure given below. The sample is mixed with the fuel and burned in air. The fuel should not cause any ionisation so a hydrogen-helium mixture is used. The air should be of high purity for reducing the risk of introducing hydrocarbons. The fuel and sample flows are to be regulated as the response of the instrument is directly proportional to the flow rate of sample as it influences the burner temperature. The flow is regulated by maintaining fixed pressure difference across the device.



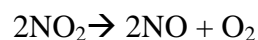
The burner jet and annular collector from the electrodes and a potential of about 100V is applied between them. The signals are amplified and calibration is achieved by zeroing instrument with a sample containing pure N_2 .

Chemiluminescence of measuring NO:

Principle: Chemiluminescence technique depends on the emission of light. NO (nitric oxide) reacts with O_3 to produce NO_2 in activated NO_2 which emits in due course as it converts to normal state (NO_2).



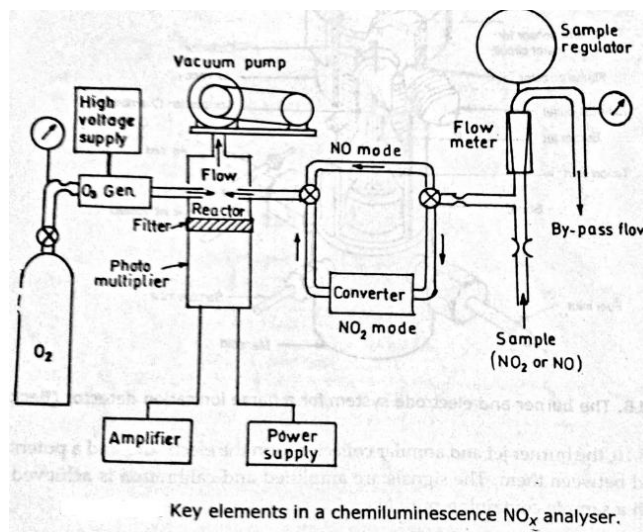
The photon light emitted is proportional to the concentration of NO in the sample stream. Both NO and NO_2 exists in the engine exhaust NO_2 can be measured by passing the sample over a catalyst that converts



By switching the convertor in and out of the sample line the concentration of NO and ($NO + NO_2$) can be measured in the exhaust sample.

Working

The below figure represents the key element of NO_x , ($\text{NO} + \text{NO}_2$) analyser. The vacuum pump controls the pressure in the reaction chamber and responsible for drawing in O_3 and exhaust sample. The O_3 is generated by an electric discharge in O_2 at low pressure and flow of O_3 is controlled by O_2 supply pressure and the critical flow orifice. The sample can be either by-pass or flow through the NO_2 -converter. The sample flow rate is regulated by two critical flow rate orifices. The bypass flow is drawn through by a sample pump. This arrangement ensures a high flow rate of sample gas, so as to minimise the instrument response to change in NO_x concentration in the sample.



The flow of the sample into the reactor is controlled by the pressure differential across the critical flow orifice upstream of the NO_x converter. This pressure differential is controlled by a differential pressure regulator. The light emission in the reactor is measured by a photo multiplier and then amplified.

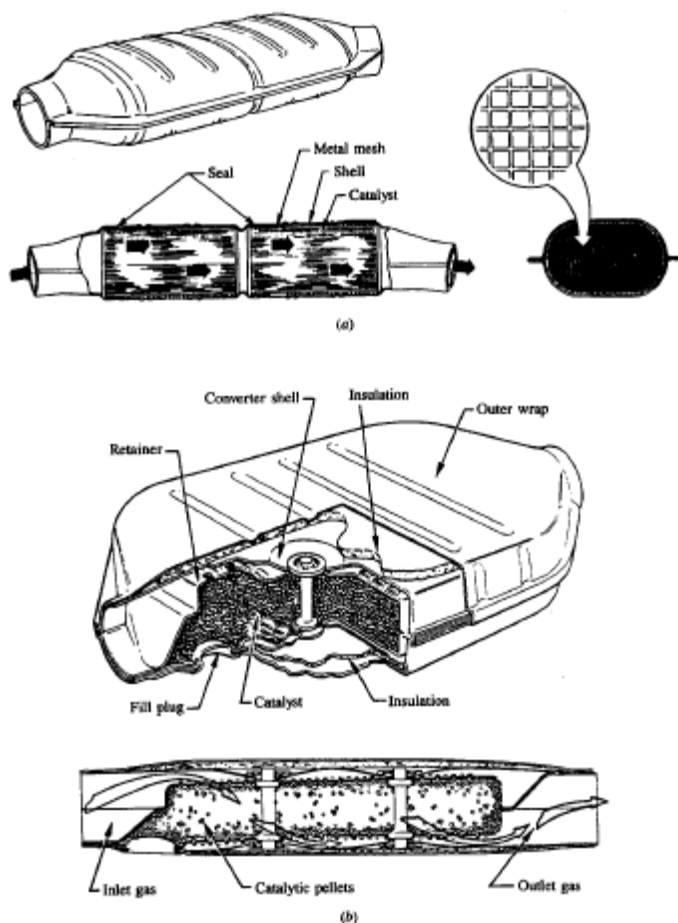
LECTURE – 6

Catalytic convertor:

The catalytic converters used in spark-ignition engines consist of an active catalytic material in a specially designed metal casing which directs the exhaust gas flow through the catalyst bed. The active material employed for CO and HC oxidation or NO reduction (normally noble metals, though base metal oxides can be used) must be distributed over a large surface area so that the mass transfer characteristics between the gas phase and the active catalyst surface are sufficient to allow close to 100 percent conversion with high catalytic activity. The two configurations commonly used are shown in Fig. 11-53. One

system employs a ceramic honeycomb structure or monolith held in a metal can in the exhaust stream. The active (noble metal) catalyst material is impregnated into a highly porous alumina washcoat about 20 μm thick that is applied to the passageway walls. The typical monolith has square-cross-section passageways with inside dimensions of $\approx 1\text{ mm}$ separated by thin (0.15 to 0.3 mm) porous walls.

The number of passageways per square centimetre varies between about 30 and 60. The washcoat, 5 to 15 percent of the weight of the monolith, has a surface area of 100 to 200 m^2/g . The other converter design uses a bed of spherical ceramic pellets to provide a large surface area in contact with the flow. With pellet catalysts, the noble metal catalyst is impregnated into the highly porous surface of the spherical alumina pellets (typically 3 mm diameter) to a depth of about 250 μm . The pellet material is chosen to have good crush and abrasion resistance after exposure to temperatures of order 1000°C . The gas flow is directed down through the bed as shown to provide a large flow area and low pressure drop. The gas flow is turbu



lent which results in high mass-transfer rates; in the monolith catalyst passageways, it is laminar.

LECTURE - 7

Particulate traps:

An exhaust treatment technology that substantially reduces diesel engine particulate emissions is the trap oxidizer. A temperature-tolerant filter or trap removes the particulate material from the exhaust gas; the filter is then "cleaned off" by oxidizing the accumulated particulates. This technology is difficult to implement because: (1) the filter, even when clean, increases the pressure in the exhaust system; (2) this pressure increase steadily rises as the filter collects particulate matter; (3) under normal diesel engine operating conditions the collected particulate matter will not ignite and oxidize; (4) once ignition of the particulate occurs, the burnup process must be carefully controlled to prevent excessively high temperatures and trap damage or destruction. Trap oxidizers have been put into production for light-duty automobile diesel engines. Their use with heavy-duty diesel engines poses more difficult problems due to higher particulate loading and lower exhaust temperatures.

Types of particulate filters include: ceramic monoliths, alumina-coated wire mesh, ceramic foam, ceramic fiber mat, woven silica-fiber rope wound on a porous tube. Each of these has different inherent pressure loss and filtering efficiency. Regeneration of the trap by burning up the filtered particulate material can be accomplished by raising its temperature to the ignition point while providing oxygen-containing exhaust gas to support combustion and carry away the heat released. Diesel particulate matter ignites at about **500 to 600°C**. This is above the normal temperature of diesel exhaust so either the exhaust gas flowing through the trap during regeneration must be heated (positive regeneration) or ignition must be made to occur at a lower temperature with catalytic materials on the trap or added to the fuel (catalytic regeneration). Catalytic coatings on the trap reduce the ignition temperature by up to 200°C.

LECTURE – 8

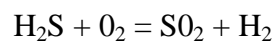
Chemical methods to reduce emissions

Development work has been done on large stationary engines using cyanuric acid to reduce NO_x emissions. Cyanuric acid is a low-cost solid material that sublimates in the exhaust flow. The gas dissociates, producing isocyanide that reacts with NO_x to form N₂, H₂O, and CO₂. Operating temperature is about 500°C. Up to 95% NO_x reduction has been achieved

with no loss of engine performance. At present, this system is not practical for vehicle engines because of its size, weight, and complexity. Research is being done using zeolite molecular sieves to reduce NO_x emissions. These are materials that absorb selected molecular compounds and catalyse chemical reactions. Using both SI and CI engines, the efficiency of NO_x reduction is being determined over a range of operating variables, including AF, temperature, flow velocity, and zeolite structure. At present, durability is a serious limitation with this method.

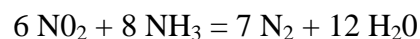
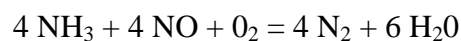
Various chemical absorbers, molecular sieves, and traps are being tested to reduce HC emissions. HC is collected during engine startup time, when the catalytic converter is cold, and then later released back into the exhaust flow when the converter is hot. The converter then efficiently burns the HC to H₂O and CO₂. A 35% reduction of cold-start HC has been achieved.

H₂S emissions occur under rich operating conditions. Chemical systems are being developed that trap and store H₂S when an engine operates rich and then convert this to SO₂ when operation is lean and excess oxygen exists. The reaction equation is



Ammonia Injection Systems

Some large ship engines and some stationary engines reduce NO_x emissions with an injection system that sprays NH₃ into the exhaust flow. In the presence of a catalyst, the following reactions occur:



Careful control must be adhered to, as NH₃ itself is an undesirable emission. Emissions from large ships were not restricted for many years, even after strict laws were enforced on other engines. It was reasoned that ships operated away from land masses most of the time and the exhaust gases could be absorbed by the atmosphere without affecting human habitat. However, most seaports are in large cities, where emission problems are most critical, and polluting from all engines is now restricted, including ship engines.

Ammonia injection systems are not practical in automobiles or on other smaller engines. This is because of the needed NH_3 storage and fairly complex injection and control system.

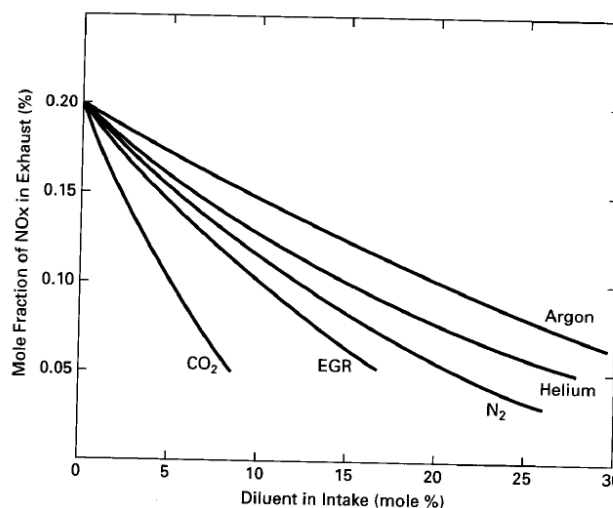
LECTURE - 9

EXHAUST GAS RECYCLE-EGR

The most effective way of reducing NO_x emissions is to hold combustion chamber temperatures down. Although practical, this is a very unfortunate method in that it also reduces the thermal efficiency of the engine. We have been taught since infancy in our first thermodynamics course that for maximum engine thermal efficiency, Q_{in} should be at the highest temperature possible.

Probably the simplest practical method of reducing maximum flame temperature is to dilute the air-fuel mixture with a non-reacting parasite gas. This gas absorbs energy during combustion without contributing any energy input. The net result is a lower flame temperature. Any non-reacting gas would work as a diluent, as shown in Fig. Those gases with larger specific heats would absorb the most energy per unit mass and would therefore require the least amount; thus less CO_2 would be required than argon for the same maximum temperature. However, neither CO_2 nor argon is readily available for use in an engine. Air is available as a diluent but is not totally non-reacting. Adding air changes the AF and combustion characteristics. The one

Non-reacting gas that is available to use in an engine is exhaust gas, and this is used in all modern automobile and other medium-size and large engines.



Exhaust gas recycle (EGR) is done by ducting some of the exhaust flow back into the intake system, usually immediately after the throttle. The amount of flow can be as high as 30% of the total intake. EGR gas combines with the exhaust residual left in the cylinder from the previous cycle to effectively reduce the maximum combustion temperature.

Not only does EGR reduce the maximum temperature in the combustion chamber, but it also lowers the overall combustion efficiency. Above Fig shows that as EGR are increased, the percent of inefficient *slow-burn* cycles increases. Further increase in EGR results in some cycle *partial burns* and, in the extreme, total misfires. Thus, by using EGR to reduce NO_x emissions, a costly price of increased HC emissions and lower thermal efficiency must be paid.

The amount of EGR is controlled by the EMS. By sensing inlet and exhaust conditions the flow is controlled, ranging from 0 up to 15-30%. Lowest NO_x emissions with relatively good fuel economy occur at about stoichiometric combustion, with as much EGR as possible without adversely affecting combustion. No EGR is used during WOT, when maximum power is desired. No EGR is used at idle and very little at low speeds. Under these conditions, there is already maximum exhaust residual and greater combustion inefficiency. Engines with fast-burn combustion chambers can tolerate a greater amount of EGR.

A problem unique to CI engines when using EGR is the solid carbon soot in the exhaust. The soot acts as an abrasive and breaks down the lubricant. Greater wear on the piston rings and valve train results.

NON EXHAUST EMISSIONS:

Engines and fuel supply systems also have sources of emissions other than exhaust flow. Historically, these were considered minor and were just released to the surrounding air.

A major source of HC emissions was the crankcase breather tube that was vented to the air in older automobiles. Blowby flow past the pistons ended up in the crankcase, and due to the higher pressure it created, it was then pushed out the breather vent tube. Blowby gas is very high in HCs, especially in 81 engines. Also, in older engines with greater clearance between the piston and cylinder wall, blowby flow was much higher. As much as 1% of the fuel was vented to the atmosphere through the crankcase breather in some automobiles. This accounted for up to 20% of total emissions. A simple solution to this problem, which is used on all modern engines, is to vent the crankcase breather back into the intake system. This not only reduces emissions but also increases fuel economy.

To keep the pressure at one atmosphere in the fuel tank and in the fuel reservoir of a carburetor, these systems are vented to the surroundings. Historically, these vents were an additional source of HC emissions when fuel evaporated from these fuel reservoirs. To eliminate these emissions, fuel vents now include some form of filter or absorption system which stops the HC vapour from escaping. One such system absorbs the HCs onto the surface of a carbon filter element. Then, when the engine is operating, the element is back flushed and the HC is desorbed off the surface. The recovered HC is ducted into the engine intake with no resulting emissions.

Many modern gasoline pumps and other fuel-dispensing systems are equipped with vapour-collecting nozzles that reduce HC vapour lost to the atmosphere during refuelling.

UNIT-IV ALTERNATE FUELS**LECTURE – 1**

Sometime during the 21st century, crude oil and petroleum products will become very scarce and costly to find and produce. At the same time, there will likely be an increase in the number of automobiles and other IC engines. Although fuel economy of engines is greatly improved from the past and will probably continue to be improved, numbers alone dictate that there will be a great demand for fuel in the coming decades. Gasoline will become scarce and costly. Alternate fuel technology, availability, and use must and will become more common in the coming decades. Although there have always been some IC engines fuelled with non-gasoline or diesel oil fuels, their numbers have been relatively small. Because of the high cost of petroleum products, some third-world countries have for many years been using manufactured alcohol as their main vehicle fuel. Many pumping stations on natural gas pipelines use the pipeline gas to fuel the engines driving the pumps. This solves an otherwise complicated problem of delivering fuel to the pumping stations, many of which are in very isolated regions. Some large displacement engines have been manufactured especially for pipeline work. These consist of a bank of engine cylinders and a bank of compressor cylinders connected to the same crankshaft and contained in a single engine block similar to a V-style engine.

Alcohols

Alcohols are an attractive alternate fuel because they can be obtained from a number of sources, both natural and manufactured. Methanol (methyl alcohol) and ethanol (ethyl alcohol) are two kinds of alcohol that seem most promising and have had the most development as engine fuel.

The advantages of alcohol as a fuel include:

1. Can be obtained from a number of sources, both natural and manufactured.
2. Is high octane fuel with anti-knock index numbers (octane number on fuel pump) of over 100. High octane numbers result, at least in part, from the high flame speed of alcohol. Engines using high-octane fuel can run more efficiently by using higher compression ratios.

3. Generally less overall emissions when compared with gasoline.
4. When burned, it forms more moles of exhaust, which gives higher pressure and more power in the expansion stroke.
5. Has high evaporative cooling (*hfg*) which results in a cooler intake process and compression stroke. This raises the volumetric efficiency of the engine and reduces the required work input in the compression stroke.
6. Low sulphur content in the fuel.

The disadvantages of alcohol fuels include:

1. Almost twice as much alcohol as gasoline must be burned to give the same energy input to the engine. With equal thermal efficiency and similar engine output usage, twice as much fuel would have to be purchased, and the distance which could be driven with a given fuel tank volume would be cut in half. The same amount of automobile use would require twice as much storage capacity in the distribution system, twice the number of storage facilities, and twice the volume of storage at the service station, twice as many tank trucks and pipelines, etc. Even with the lower energy content of alcohol, engine power for a given displacement would be about the same. This is because of the lower air-fuel ratio needed by alcohol. Alcohol contains oxygen and thus requires less air for stoichiometric combustion. More fuel can be burned with the same amount of air.
2. More aldehydes in the exhaust. If as much alcohol fuel was consumed as gasoline, aldehyde emissions would be a serious exhaust pollution problem.
3. Alcohol is much more corrosive than gasoline on copper, brass, aluminium, rubber, and many plastics. This puts some restrictions on the design and manufacturing of engines to be used with this fuel. This should also be considered when alcohol fuels are used in engine systems designed to be used with gasoline. Fuel lines and tanks, gaskets, and even metal engine parts can deteriorate with long-term alcohol use (resulting in cracked fuel lines, the need for special fuel tank, etc). Methanol is very corrosive on metals.
4. Poor cold weather starting characteristics due to low vapour pressure and evaporation. Alcohol-fuelled engines generally have difficulty starting at temperatures below 10°C. Often a small amount of gasoline is added to alcohol fuel, which greatly improves cold-weather starting. The need to do this, however, greatly reduces the attractiveness of any alternate fuel.
5. Poor ignition characteristics in general.

6. Alcohols have almost invisible flames, which are considered dangerous when handling fuel. Again, a small amount of gasoline removes this danger.
7. Danger of storage tank flammability due to low vapor pressure. Air can leak into storage tanks and create a combustible mixture.
8. Low flame temperatures generate less NO_x, but the resulting lower exhaust temperatures take longer to heat the catalytic converter to an efficient operating temperature.
9. Many people find the strong odor of alcohol very offensive. Headaches and dizziness have been experienced when refuelling an automobile.
10. Vapour lock in fuel delivery systems.

LECTURE – 2

Further Alcohols can be classified as two major types, they are:

1. **Methanol**
2. **Ethanol**

Methanol:

Of all the fuels being considered as an alternate to gasoline, methanol is one of the more promising and has experienced major research and development. Pure methanol and mixtures of methanol and gasoline in various percentages have been extensively tested in engines and vehicles for a number of years [88, 130]. The most common mixtures are M85 (85% methanol and 15% gasoline) and M10 (10% methanol and 90% gasoline). The data of these tests which include performance and emission levels are compared to pure gasoline (MO) and pure methanol (M100). Some smart **flexible-fuel** (or **variable-fuel**) engines are capable of using any random mixture combination of methanol and gasoline ranging from pure methanol to pure gasoline. Two fuel tanks are used and various flow rates of the two fuels can be pumped to the engine, passing through a mixing chamber. Using information from sensors in the intake and exhaust, the EMS adjusts to the proper air-fuel ratio, ignition timing, injection timing, and valve timing (where possible) for the fuel mixture being used. Fast, abrupt changes in fuel mixture combinations must be avoided to allow for these adjustments to occur smoothly.

One problem with gasoline-alcohol mixtures as a fuel is the tendency for alcohol to combine with any water present. When this happens the alcohol separates locally from the

gasoline, resulting in a non-homogeneous mixture. This causes the engine to run erratically due to the large AF differences between the two fuels. At least one automobile company has been experimenting with a three-fuel vehicle which can use any combination of gasoline-methanol-ethanol. Methanol can be obtained from many sources, both fossil and renewable. These include coal, petroleum, natural gas, biomass, wood, landfills, and even the ocean. However, any source that requires extensive manufacturing or processing raises the price of the fuel and requires an energy input back into the overall environmental picture, both unattractive.

Emissions from an engine using MIO fuel are about the same as those using gasoline. The advantage (and disadvantage) of using this fuel is mainly the 10% decrease in gasoline use. With M85 fuel there is a measurable decrease in HC and CO exhaust emissions. However, there is an increase in NO_x and a large (= 500%) increase in formaldehyde formation.

Methanol is used in some dual-fuel CI engines. Methanol by itself is not a good CI fuel because of its high octane number, but if a small amount of diesel oil is used for ignition, it can be used with good results. This is very attractive for third-world countries, where methanol can often be obtained much cheaper than diesel oil.

Older CI bus engines have been converted to operate on methanol in tests conducted in California. This resulted in an overall reduction of harmful emissions compared with worn engines operating with diesel fuel.

Ethanol

Ethanol has been used as automobile fuel for many years in various regions of the world. Brazil is probably the leading user, where in the early 1990s, 4.5 million vehicles operated on fuels that were 93% ethanol. For a number of years **gasohol** has been available at service stations in the United States, mostly in the Midwest corn-producing states. Gasohol is a mixture of 90% gasoline and 10% ethanol. As with methanol, the development of systems using mixtures of gasoline and ethanol continues. Two mixture combinations that are important are E85 (85% ethanol) and E10 (gasohol). E85 is basically an alcohol fuel with 15% gasoline added to eliminate some of the problems of pure alcohol (i.e., cold starting, tank flammability, etc.). E10 reduces the use of gasoline with no modification needed to the

automobile engine. Flexible-fuel engines are being tested which can operate on any ratio of ethanol-gasoline.

Ethanol can be made from ethylene or from fermentation of grains and sugar. Much of it is made from corn, sugar beets, sugar cane, and even cellulose (wood and paper). In the United States, corn is the major source. The present cost of ethanol is high due to the manufacturing and processing required. This would be reduced if larger amounts of this fuel were used. However, very high production would create a food-fuel competition, with resulting higher costs for both. Some studies show that at present in the United States, crops grown for the production of ethanol consume more energy in ploughing, planting, harvesting, fermenting, and delivery than what is in the final product. This defeats one major reason for using an alternate fuel. Ethanol has less HC emissions than gasoline but more than methanol.

LECTURE – 3

Hydrogen as alternate fuels

Hydrogen is a long-term renewable and less-polluting fuel. In addition hydrogen is clean burning characteristics and better performance drives more interest in hydrogen fuel. When it is burnt in an internal combustion engine, the primary combustion product is water with no CO₂. Although NO_x emissions are formed when hydrogen is used.

Combustive Properties of Hydrogen

Wide range of flammability

Compared to nearly all other fuels, hydrogen has a wide flammability range (4-75% versus 1.4-7.6% volume in air for gasoline). This first leads to obvious concerns over the safe handling of hydrogen. But, it also implies that a wide range of fuel-air mixtures, including a lean mix of fuel to air, or, in other words, a fuel-air mix in which the amount of fuel is less than the stoichiometric, or chemically ideal, amount.

Small quenching distance

Hydrogen has a small quenching distance (0.6 mm for hydrogen versus 2.0 mm for gasoline), which refers to the distance from the internal cylinder wall where the combustion flame extinguishes. This implies that it is more difficult to quench a hydrogen flame than the flame of most other fuels, which can increase backfire since the flame from a hydrogen-air mixture more readily passes a nearly closed intake valve, than a hydrocarbon-air flame

Flame velocity and adiabatic flame

Hydrogen burns with a high flame speed, allowing for hydrogen engines to more closely approach the thermodynamically ideal engine cycle (most efficient fuel power ratio) when the stoichiometric fuel mix is used. However, when the engine is running lean to improve fuel economy, flame speed slows significantly.

Flame velocity and adiabatic flame temperature are important properties for engine operation and control, in particular thermal efficiency, combustion stability and emissions.

Minimum ignition source energy

The minimum ignition source energy is the minimum energy required to ignite a fuel-air mix by an ignition source such as a spark discharge. For a hydrogen and air mix it is about an order of magnitude lower than that of a petrol-air mix 0.02 mJ as compared to 0.24 mJ for petrol - and is approximately constant over the range of flammability.

The low minimum ignition energy of the hydrogen-air mix means that a much lower energy spark is required for spark ignition. This means that combustion can be initiated with a simple glow plug or resistance hot-wire. It also ensures prompt ignition of the charge in the combustion chamber.

High diffusivity

Hydrogen has very high diffusivity. This ability to disperse into air is considerably greater than gasoline and is advantageous for two main reasons. Firstly, it facilitates the formation of a uniform mixture of fuel and air. Secondly, if a hydrogen leak develops, the hydrogen disperses rapidly. Thus, unsafe conditions can either be avoided or minimized

Low density

The most important implication of hydrogen's low density is that without significant compression or conversion of hydrogen to a liquid, a very large volume may be necessary to store enough hydrogen to provide an adequate driving range. Low density also implies that the fuel-air mixture has low energy density, which tends to reduce the power output of the engine. Thus when a hydrogen engine is run lean, issues with inadequate power may arise.

High auto-ignition temperature

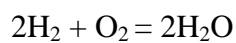
The auto ignition temperature is the minimum temperature required to initiate self-sustained combustion in a combustible fuel mixture in the absence of an external ignition. For hydrogen, the auto ignition temperature is relatively high 585°C. This makes it difficult to ignite a hydrogen-air mixture on the basis of heat alone without some additional ignition source. This temperature has important implications when a hydrogen-air mixture is compressed. In fact, the auto ignition temperature is an important factor in determining what

maximum compression ratio an engine can use, since the temperature rise during compression is related to the compression ratio.

The temperature may not exceed hydrogen's auto ignition temperature without causing premature ignition. Thus, the absolute final temperature limits the compression ratio. The high auto ignition temperature of hydrogen allows larger compression ratios to be used in a hydrogen engine than in a hydrocarbon engine.

Advantages of hydrogen as alternate fuel:

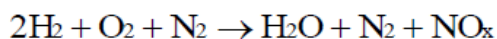
1. Hydrogen produces only water after combustion.



2. When hydrogen is burned, hydrogen combustion does not produce toxic products such as hydrocarbons, carbon monoxide, and oxide of sulfur, organic acids or carbon dioxides
3. Hydrogen has some peculiar features compared to hydrocarbon fuels, the most significant being the absence of carbon.
4. The burning velocity is so high that very rapid combustion can be achieved.
5. The density of hydrogen is 0.0838 kg/m³, which is lighter than air that it can disperse into the atmosphere easily.
6. Hydrogen has the highest energy to weight ratio of all fuels.

Disadvantages of hydrogen as alternate fuels:

1. NO_x is formed as emission.



2. Storage of hydrogen is more difficult as it leads to crack.
3. It is not possible to achieve ignition of hydrogen by compression alone. Some sources of ignition have to be created inside the combustion chamber to ensure ignition.

Formation of NO_x depends on the factors like

- The air/fuel ratio
- The engine compression ratio

- The engine speed
- The ignition timing
- Whether thermal dilution is utilized

LECTURE – 4

Biogases as alternate fuels

Biogas is the product of fermentation of man and animals' biological activity waste products when bacteria degrade biological material in the absence of oxygen, in a process known as anaerobic digestion. Since biogas is a mixture of methane (also known as marsh gas or natural gas) and carbon dioxide it is a renewable fuel produced from waste treatment. Biogas contains 50% to 70% of CH₄, 2 % of H₂ and up to 30 % of CO₂. After being cleaned of carbon dioxide, this gas becomes a fairly homogeneous fuel containing up to 80 % of methane with the calorific capacity of over 25 MJ/m³. The most important component of biogas, from the calorific point of view, is methane, CH₄. The other components are not involved in combustion process, and rather absorb energy from combustion of CH₄ as they leave the process at higher temperature than the one they had before the process.

The actual calorific value of biogas is function of the CH₄ percentage, the temperature and the absolute pressure, all of which differ from case to case.

Economical and operational considerations:

a) Biogas availability or potential

- ☐ A biogas plant already exists and the gas yield is larger than what is already consumed in other equipment or the yield could be increased.
- ☐ Organic matter is available and otherwise wasted; the boundary conditions allow for anaerobic digestion.
- ☐ Environmental laws enforce anaerobic treatment of organic waste from municipalities, food industries, distilleries, etc.

b) Demand for mechanical power

- ☐ Other fuels are practically not available.

- ☐ Other sources of energy or fuels are more expensive or their supply is unreliable.
 - ☐ Having a fuel at one's own disposal is of specific advantage.
- c) Possible revenue through selling mechanical power, electric power or related services to other customers (e.g. the public electricity supply company).

LECTURE – 5

Natural gas as Alternate fuel:

Natural gas is produced from gas wells or tied in with crude oil production. Natural gas (NG) is made up primarily of methane (CH_4) but frequently contains trace amounts of ethane, propane, nitrogen, helium, carbon dioxide, hydrogen sulphide, and water vapour. Methane is the principal component of natural gas. Normally more than 90% of natural gas is methane.

Natural gas can be compressed, so it can be stored and used as compressed natural gas (CNG).

CNG requires a much larger volume to store the same mass of natural gas and the use of very high pressure on about 200 bar or 2,900 psi.

Advantages of CNG:

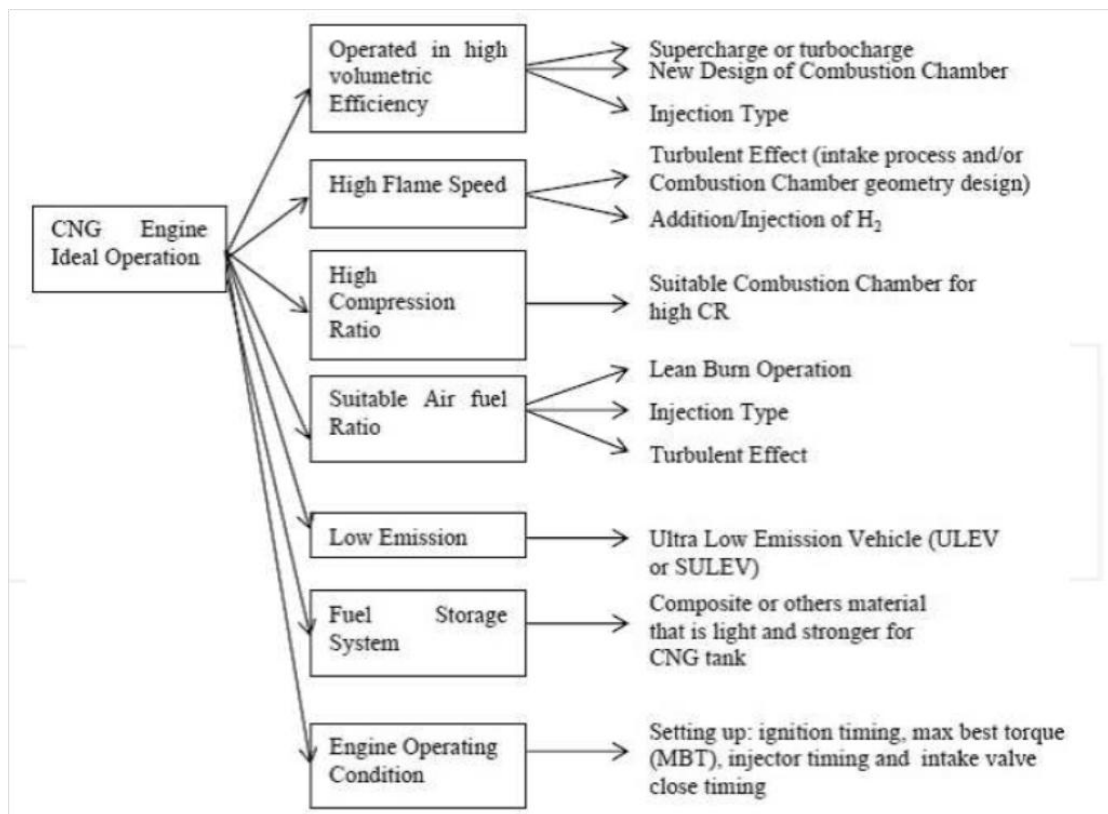
1. Natural gas is safer than gasoline in many respects.
2. The ignition temperature for natural gas is higher than gasoline and diesel fuel.
3. Natural gas is lighter than air and will dissipate upward rapidly if a rupture occurs.
4. Gasoline and diesel will pool on the ground, increasing the danger of fire.
5. Compressed natural gas is non-toxic and will not contaminate groundwater if spilled.
6. Compressed natural gas is a largely available form of fossil energy and therefore non-renewable.
7. It is a cleaner fuel than either gasoline or diesel as far as emissions are concerned.

Fuel Characteristics of CNG:

The octane rating of natural gas is about 130, meaning that engines could operate at compression ratio of up to 16:1 without “knock” or detonation. Many of the automotive makers already built transportation with a natural gas fuelling system and consumer does not

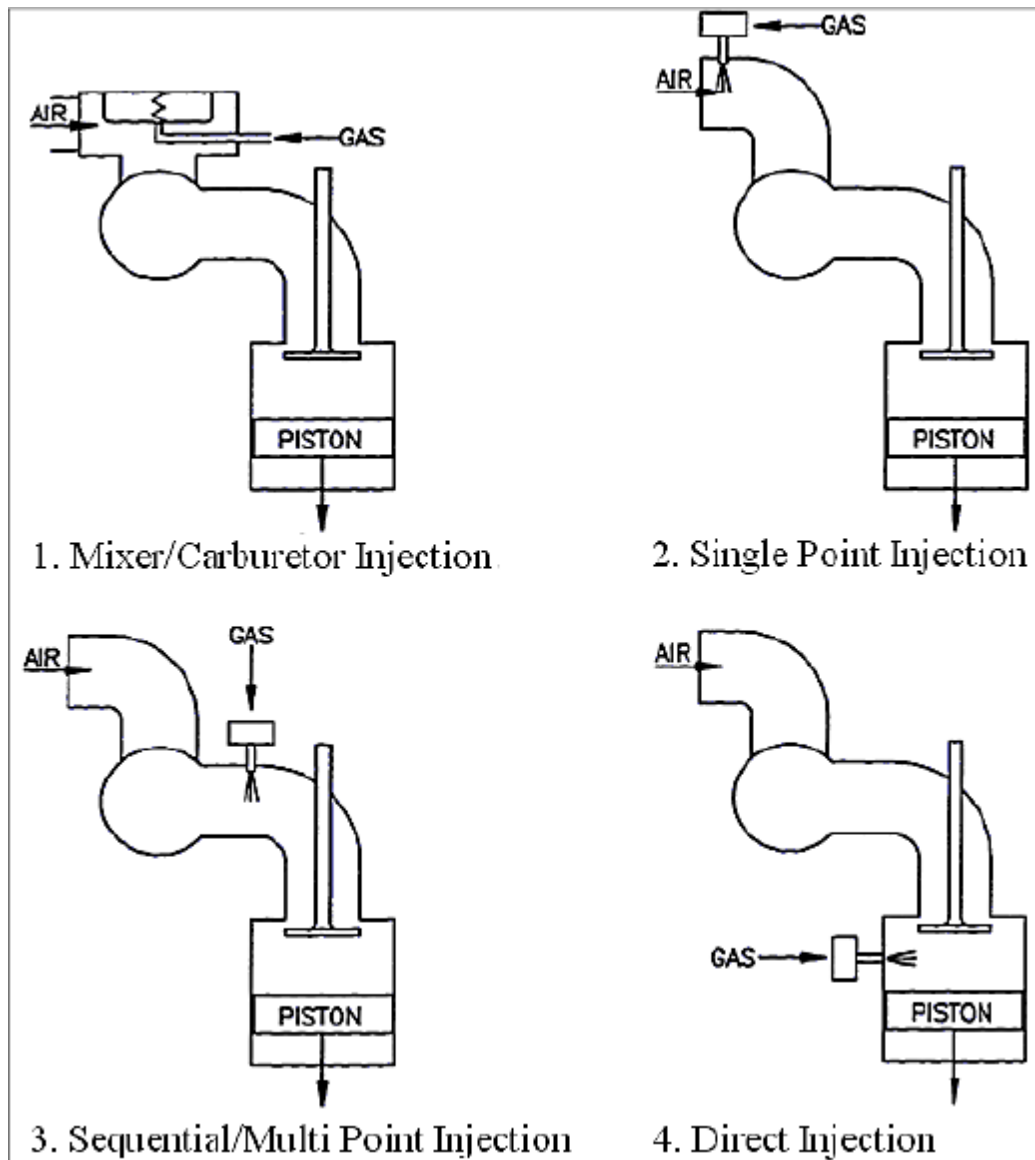
have to pay for the cost of conversion kits and required accessories. Most importantly, natural gas significantly reduces CO₂ emissions by 20-25% compared to gasoline because simple chemical structures of natural gas (primarily methane – CH₄) contain one Carbon compared to diesel (C₁₅H₃₂) and gasoline (C₈H₁₈). Like methane and hydrogen is a lighter than air type of gas and can be blended to reduce vehicle emission by an extra 50%. Natural gas composition varies considerably over time and from location to location. Methane content is typically 70-90% with the remainder primarily ethane, propane and carbon dioxide. At atmospheric pressure and temperature, natural gas exists as a gas and has low density. Since the volumetric energy density (joules/m³) is so low, natural gas is often stored in a compressed state (CNG) at high pressure stored in pressure vessels.

CNG Ideal operation:



Injection methods of Natural Gas engine

There are four methods to inject the NG into the engine cylinder. First type is gas mixer / carburetor injection, second type is the single point injection, third type is multi point injection and fourth type is direct injection. The illustration of the four methods of NG injection is shown in Fig.



LECTURE - 6

Vegetable oil as Alternate Fuel:

Vegetable oils used as DF as well as neat methyl esters prepared from vegetable oils or animal fats and blends of conventional diesel fuel with vegetable oils or methyl esters. Most vegetable oils are triglycerides (TGs; triglyceride = TG). Chemically, TGs are the triacylglycerol esters of various fatty acids with glycerol. The published engineering literature strongly indicates that the use of SVO will lead to reduced engine life. This reduced engine life is caused by the buildup of carbon deposits inside the engine, as well as negative impacts of SVO on the engine lubricant. Both carbon deposits and excessive build-up of SVO in the lubricant are caused by the very high boiling point and viscosity of SVO relative to the

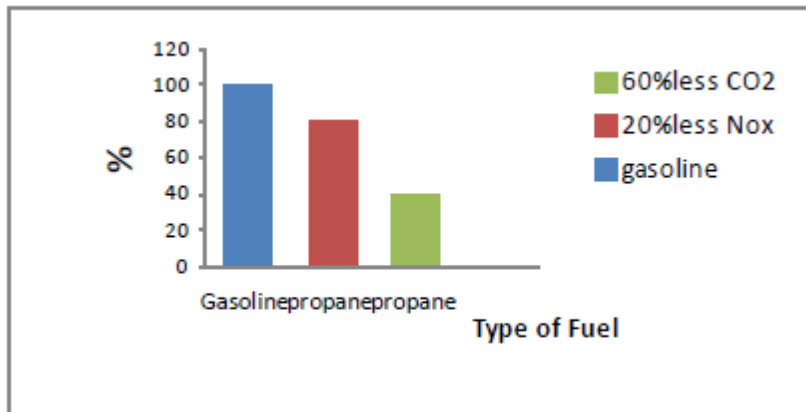
required boiling range for diesel fuel. The carbon buildup doesn't necessarily happen quickly but instead over a longer period. These conclusions are consistent across a significant body of technical information in multiple articles and reports. Biodiesel is an alternative fuel that can be made from SVO in a chemical process called transesterification that involves a reaction with methanol using caustic soda (sodium hydroxide) as a catalyst. Biodiesel has substantially different properties than SVO and results in better engine performance. In particular, biodiesel has a lower boiling point and viscosity than SVO. Because of its improved qualities, vehicle and engine manufacturers generally approve the use of biodiesel blends in their products, though not all approve blend levels as high as 20%.

LECTURE – 7

LPG as Alternate fuels:

LPG is typically a mixture of several gases in varying proportions. Major constituent gases are propane (C_3H_8) and butane (C_4H_{10}), with minor quantities of propane (C_3H_6), various butanes (C_4H_8), iso-butane, and small amounts of ethane (C_2H_6). The composition of commercial LPG is quite variable. About 55% of the LPG processed from natural gas purification. The other 45% comes from crude oil refining. LPG is derived from petroleum; LPG does less to relieve the country's dependency on foreign oil than some other alternative fuels. The gaseous nature of the fuel/air mixture in an LPG vehicle's combustion chambers eliminates the cold-start problems associated with liquid fuels. LPG defuses in air fuel mixing at lower inlet temperature than is possible with either gasoline or diesel. This leads to easier starting, more reliable idling, smoother acceleration and more complete and efficient burning with less unburned hydrocarbons present in the exhaust. In contrast to gasoline engines, which produce high emission levels while running cold, LPG engine emissions remain similar whether the engine is cold or hot. Also, because LPG enters an engine's combustion chambers as a vapor, it does not strip oil from cylinder walls or dilute the oil when the engine is cold. This helps to have a longer service life and reduced maintenance costs of engine. Also helping in this regard is the fuel's high hydrogen-to-carbon ratio (C_3H_8), which enables propane-powered vehicles to have less carbon build-up than gasoline and diesel-powered vehicles. LPG delivers roughly the same power, acceleration, and cruising speed characteristics as gasoline. Its high octane rating means engine's power output and fuel efficiency can be increased beyond what would be possible with a gasoline engine without causing *Destructive Knocking*. Such fine-tuning can help compensate for the fuel's lower

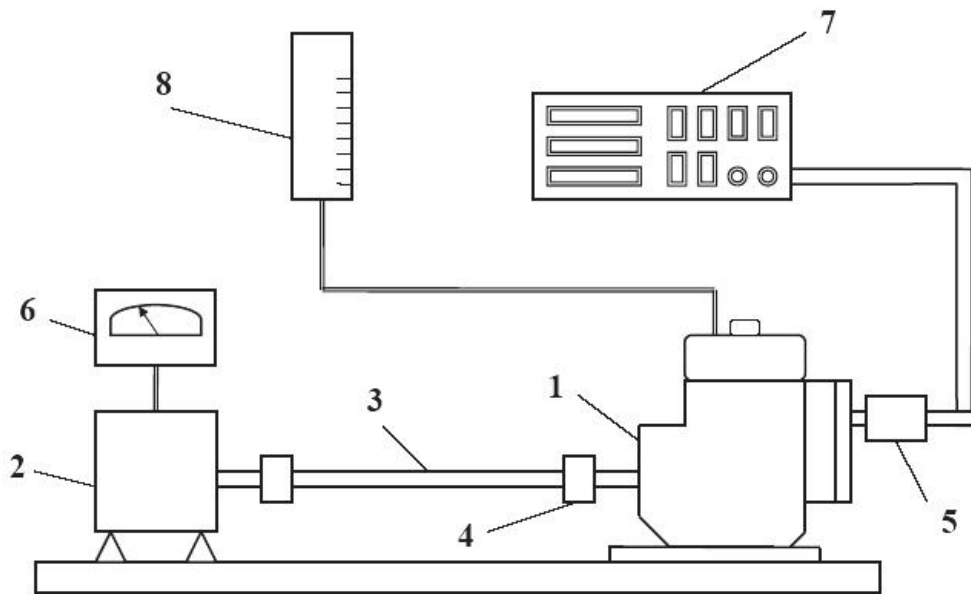
energy density. The higher ignition temperature of gas compared with petroleum based fuel leads to reduced auto ignition delays, less hazardous than any other petroleum based fuel and expected to produce less CO, NO_x emissions and may cause less ozone formation than gasoline and diesel engines.



LPG engines similar to petrol engines, and deliver nearly similar performance and good in combustion characteristics than Gasoline. In the short term, LPG as a alternative fuels reviewed could displace 10 per cent of current usage of oil, or bring significant reductions in CO, CO₂ emissions and help to reduce harmful greenhouse gas emissions. In the next five to ten years, LPG will be more widely available and gaining market share across vehicle ranges.

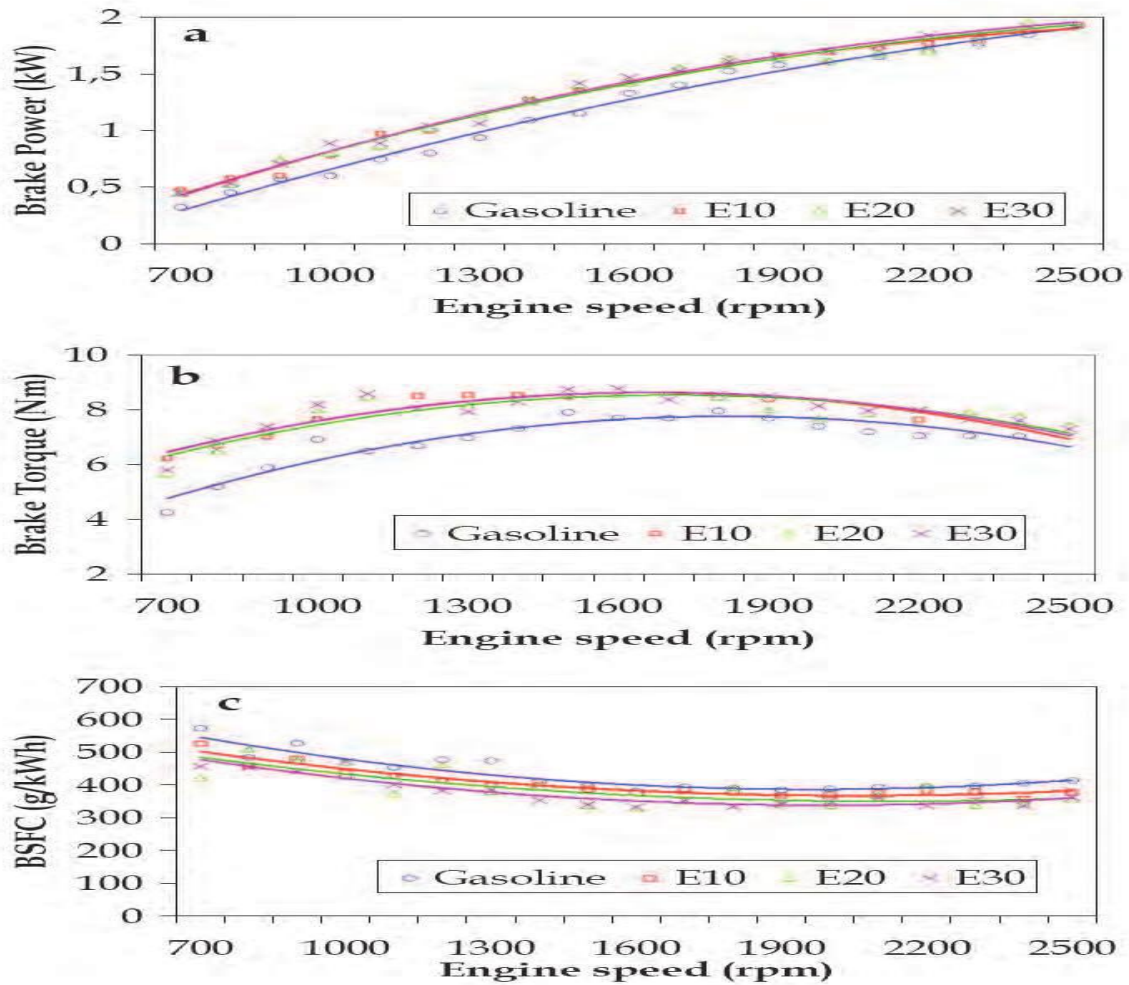
LECTURE – 8**Performance, combustion and emission characteristics of SI and CI engines**

Testing procedure

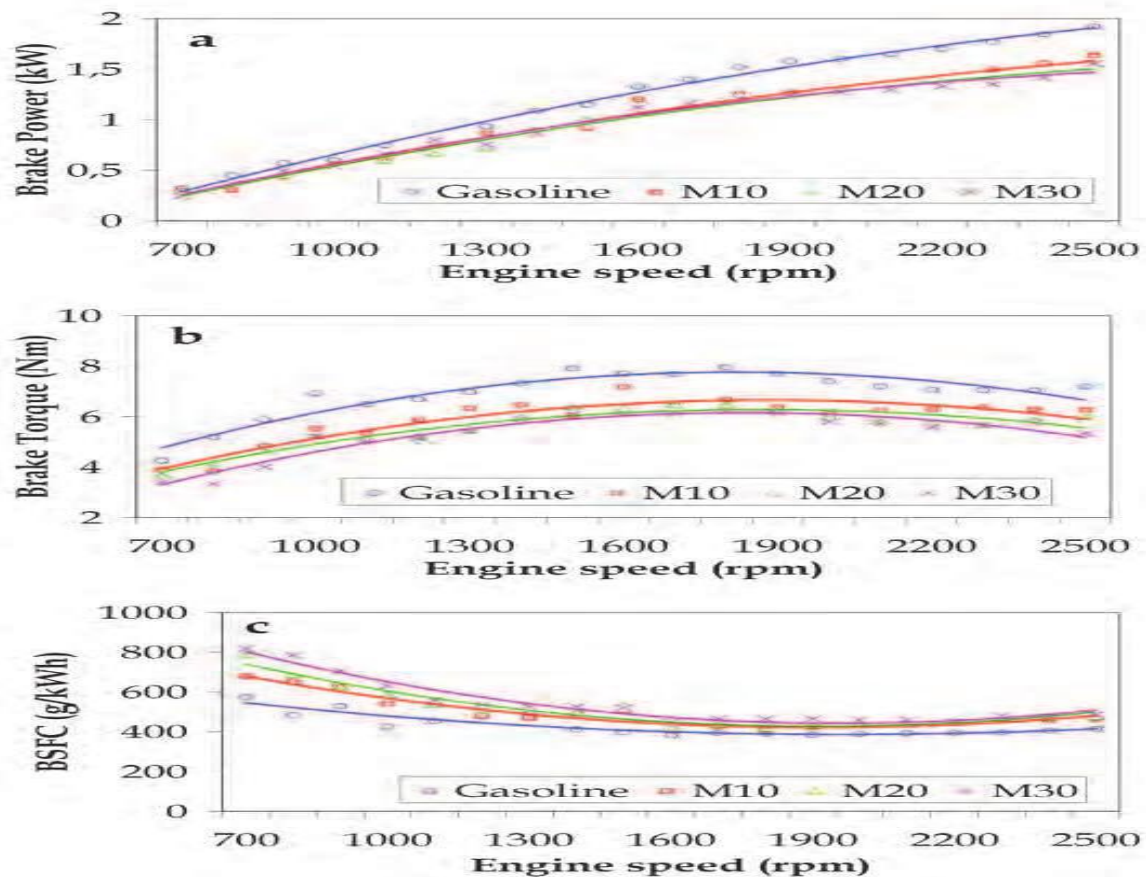


The schematic diagram of the experimental set-up. (1) Engine, (2) Dynamometer, (3) Shaft, (4) Flywheel, (5) Exhaust pipe, (6) Dynamometer control unit, (7) Gas analyzer, and (8) Fuel measurement system.

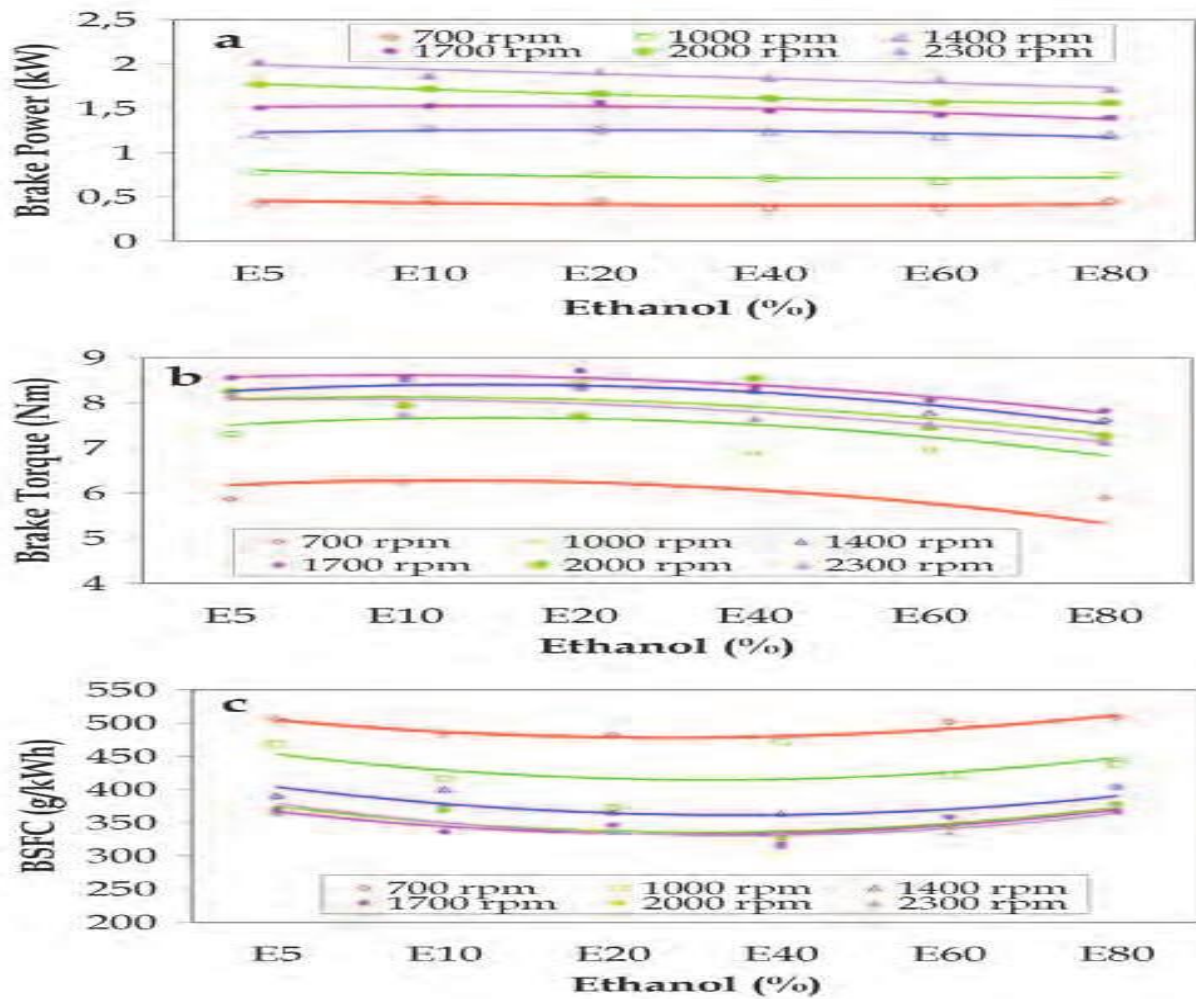
Experimental results of engine performance characteristics using different ethanol gasoline blended fuels under various engine speeds. (a) Brake power, (b) Brake torque, and (c) Brake specific fuel consumption.



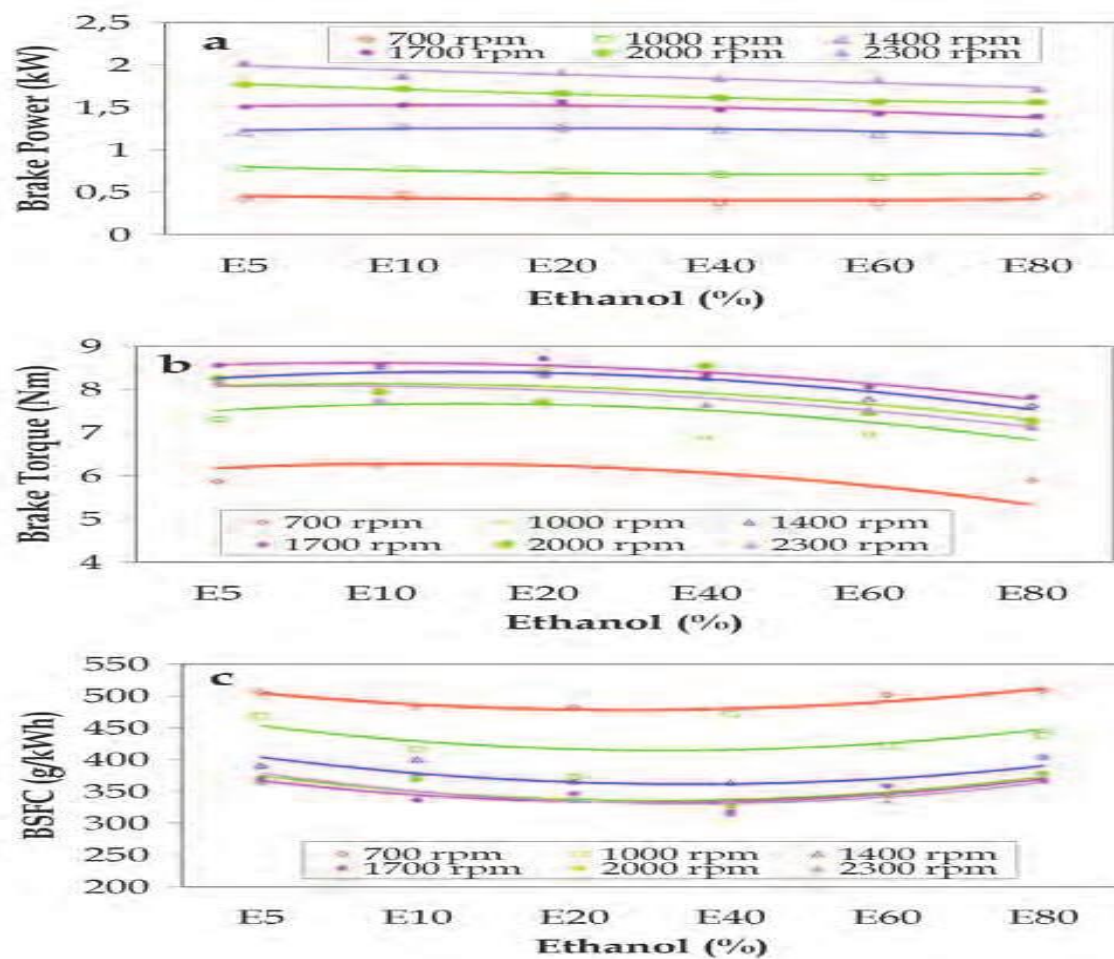
Experimental results of engine performance characteristics using different methanol gasoline blended fuels under various engine speeds. (a) Brake power, (b) Brake torque, and (c) Brake specific fuel consumption.



The influence of ethanol addition on the engine performance characteristics. (a) Brake power, (b) Brake torque, and (c) Brake specific fuel consumption.

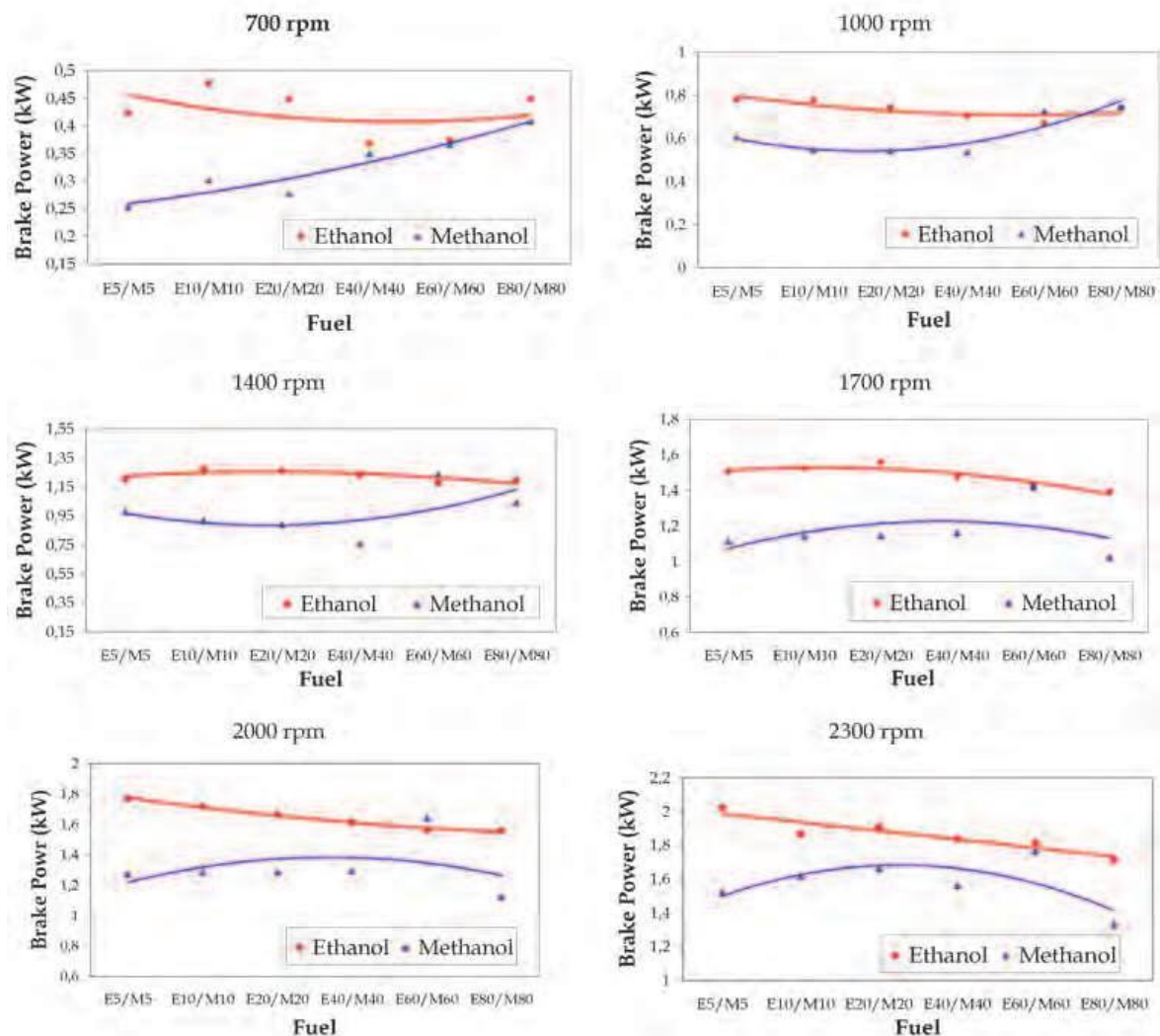


The influence of methanol addition on the engine performance characteristics. (a) Brake power, (b) Brake torque, and (c) Brake specific fuel consumption.

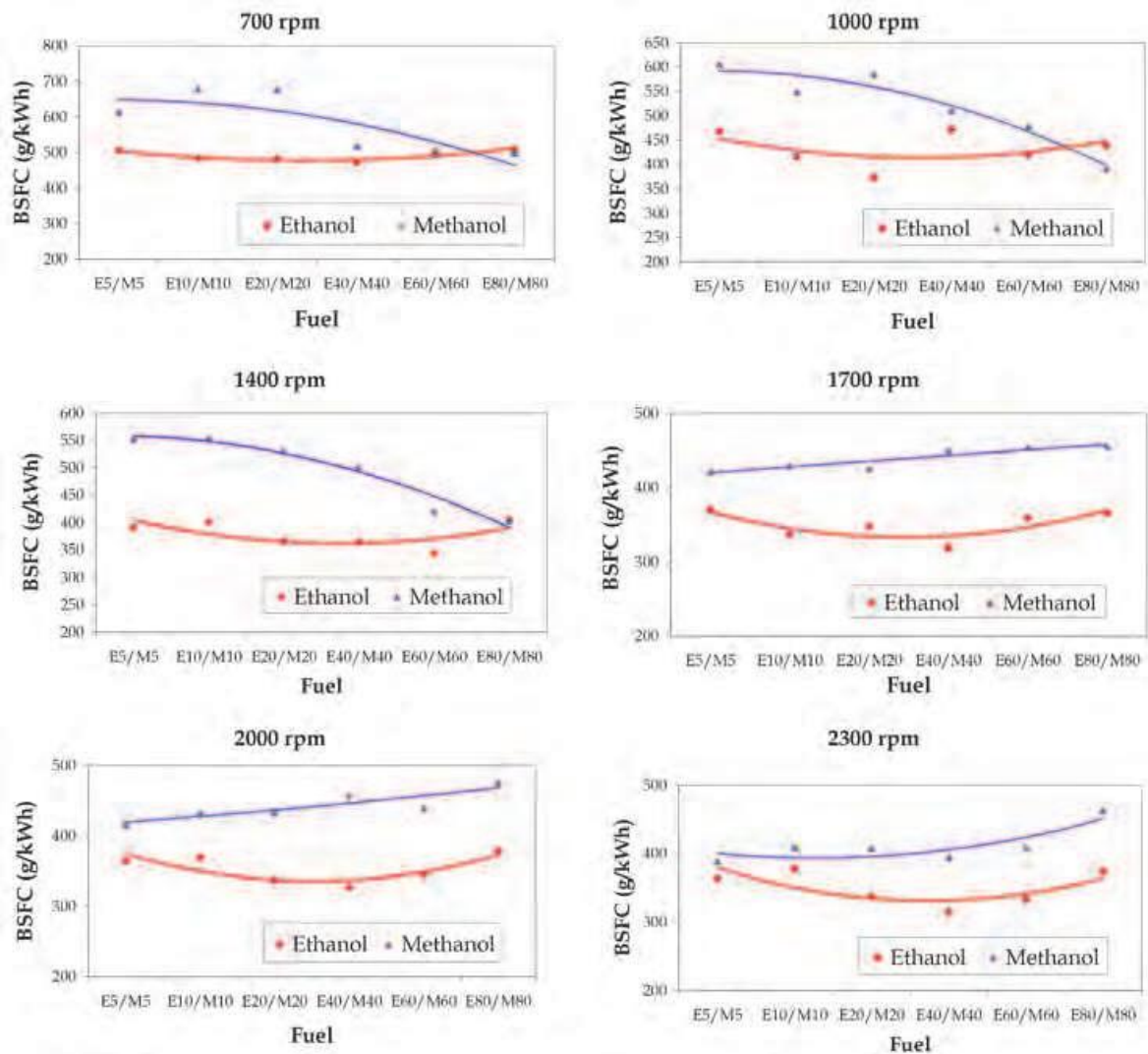


LECTURE – 9

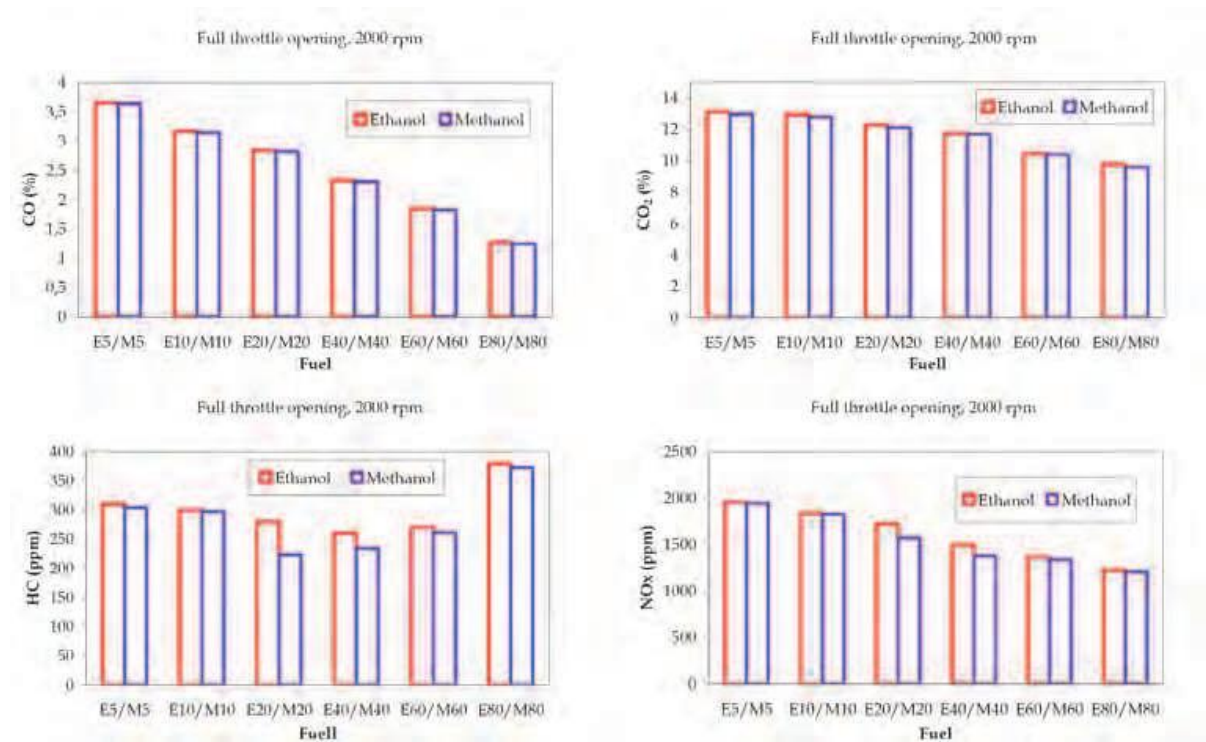
Comparison of brake power characteristics using different ethanol and methanolgasoline blended fuels.



Comparison of brake specific fuel consumption using different ethanol and methanol gasoline blended fuels.



The effect of various ethanol/methanol gasoline blend fuels on CO, CO₂, HC and NO_x emissions.



UNIT – V IN PART – II FILE

Course material Compiled from:

- “A course in internal combustion engines” by V.M. Domkundwar.*
- John B Heywood,” Internal Combustion Engine Fundamentals”*
- Gupta H.N, “Fundamentals of Internal Combustion Engines”*
- “Engineering fundamentals of Internal combustion engines” by Willard W. Pulkrabek.*
- “Advanced internal combustion engine research” Peter Van Blarigan Sandia, National Laboratories, Livermore, CA 94550.*
- And many more publications with the help of internet published materials and sites.*

UNIT – V RECENT TRENDS

LECTURE – 1

Homogeneous Charge Compression Ignition (HCCI)

- HCCI is an alternative piston-engine combustion process that can provide efficiencies as high as compression-ignition, direct-injection (CIDI) engines (an advanced version of the commonly known diesel engine) while, unlike CIDI engines, producing ultra-low oxides of nitrogen (NO_x) and particulate matter (PM) emissions.

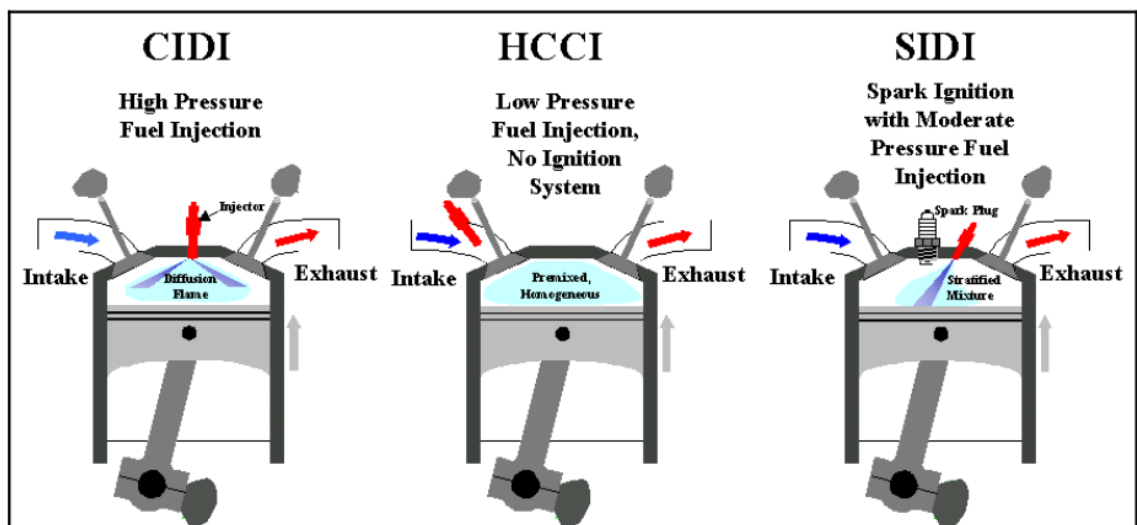


Figure 1. HCCI (as most-typically envisioned) would use low-pressure fuel injection outside the cylinder, and no ignition system. If charge stratification is desired, it may be necessary to use in-cylinder injection.

WORKING PRINCIPLE OF HCCI

Homogenous charge is drawn in to the cylinder during suction and compress to high enough temperature to achieve spontaneous ignition of the charge.

After Combustion initiation the temperature rapidly increases and whole fuel burn simultaneously.

As whole mixture burns simultaneously and no flame propagation, combustion temp can be controlled less than 700° Centigrade and thus NO_x formation is avoided.

Advantages of HCCI:

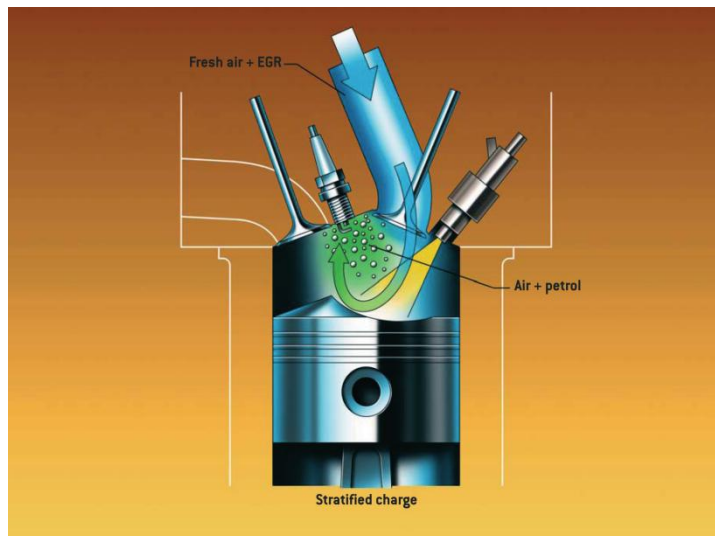
- HCCI engines are more efficient thus eliminating of throttling losses, the use of high compression ratios, and a shorter combustion duration.
- HCCI engines also have lower engine-out NO_x than SI engines.
- Lower emissions of PM and Nox
- Because flame propagation is not required, dilution levels can be much higher than the levels tolerated by either SI or CIDI engines.
- The combustion duration in HCCI engines is much shorter than in CIDI engines since it is not limited by the rate of fuel/air mixing. This leads to higher efficiency.
- HCCI engines may be lower cost than CIDI engines since they would likely use lower-pressure fuel-injection equipment.

Challenges of HCCI:

- Controlling Ignition Timing over a Range of Speeds and Loads.
- Extending the Operating Range to High Loads.
- Cold-Start Capability.
- Hydrocarbon and Carbon Monoxide Emissions.

LECTURE – 2**STRATIFIED CHARGE ENGINE**

- The principle of stratified charge applies to direct injection petrol engines.
- It involves concentrating spraying of the fuel close to the spark plug rather than throughout the whole of the combustion chamber.
- This method of operation delivers a reduction in fuel consumption that can reach 40% when the engine is running at very low charge.



Basic principle:

The principle of the stratified charge engine is to deliver a mixture that is sufficiently rich for combustion in the immediate vicinity of the spark plug and in the remainder of the cylinder, a very lean mixture that is so low in fuel that it could not be used in a traditional engine. On an engine with stratified charge, the delivered power is no longer controlled by the quantity of admitted air, but by the quantity of petrol injected, as with a diesel engine.

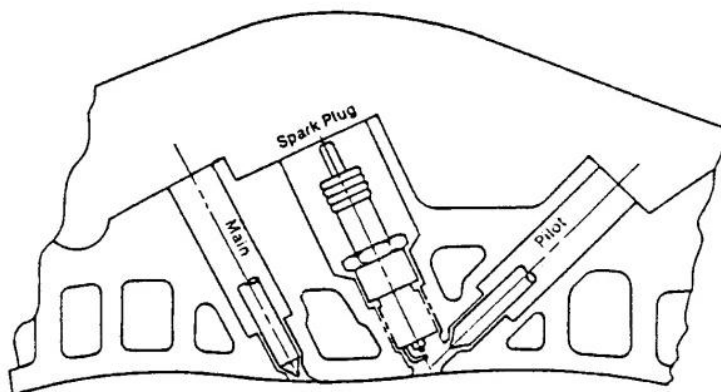
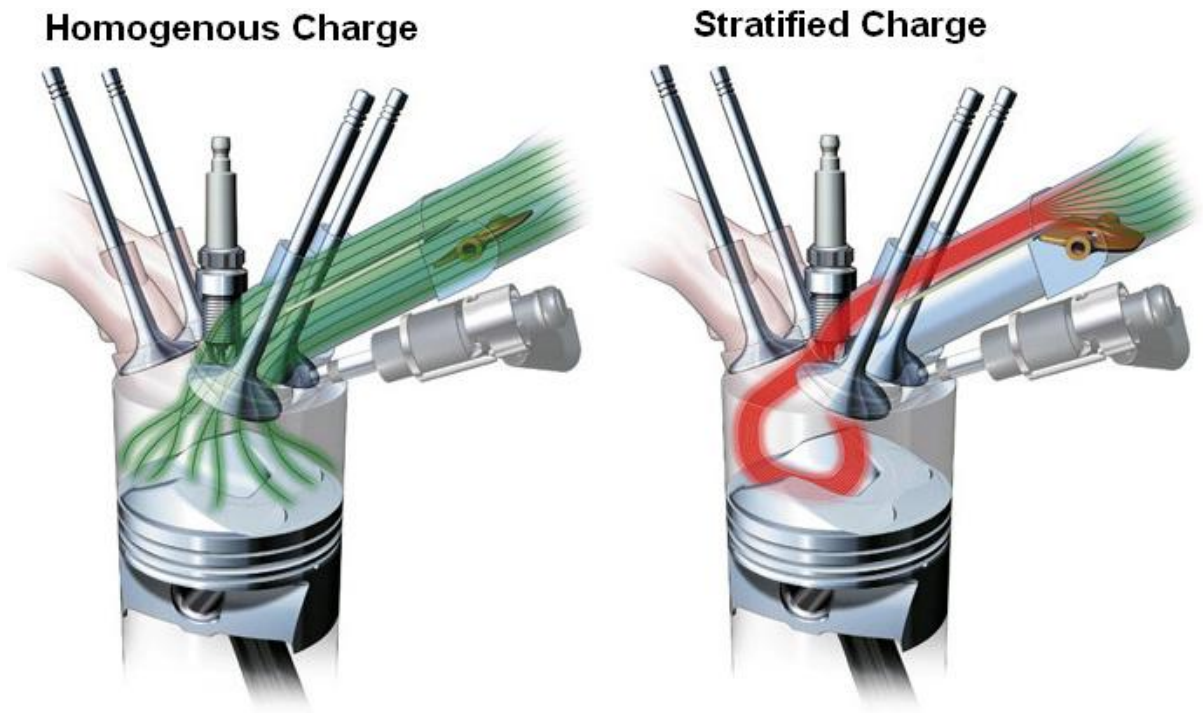


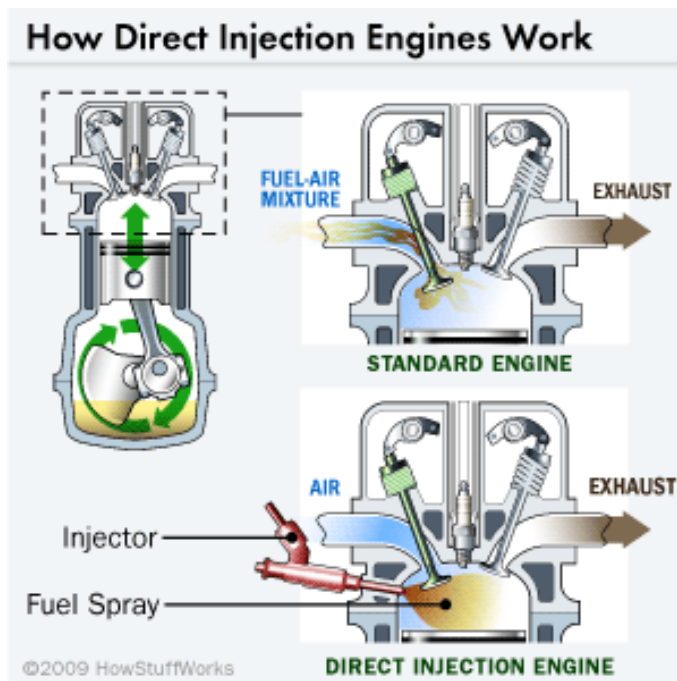
Fig. 4 dual nozzle stratified charged configuration.

This concept involves a single orifice 'pilot' nozzle which essentially injects a constant quantity of fuel optimized for producing an ignitable mixture in the vicinity of the spark plug. A second nozzle which incorporates multiple orifices, serves as the main fuel source for the combustion process, and is designed to



LECTURE – 3

Gasoline Direct Ignition Engine



- First, the fuel travels via pump from the fuel tank, through the fuel line and into fuel injectors that are mounted into the engine.

- The injectors spray gasoline into the air intake manifold, where fuel and air mix together into a fine mist. At precisely timed intervals, intake valves open, corresponding to the different cylinders of the engine.
- As a cylinder's intake valve opens, a piston in that cylinder descends, sucking the fuel-air mist from the air manifold above into the chamber below. As the piston ascends once more, it squeezes (compresses) the fuel-air mix until it is nearly nine times as dense as it was to begin with.
- Then, that cylinder's designated spark plug fires, igniting the chamber into a high-pressure, high-energy explosion.
- This little bang pushes the piston back down with tremendous force, causing it to turn the crankshaft and ultimately send power to the wheels.
- The ratio of air to fuel as it burns in an engine will have certain, predictable effects on engine performance, emissions of pollutants and fuel efficiency.
- When the amount of air in the mixture is high, compared to the amount of fuel, it's known as a "lean" mixture. When the reverse is the case, it's called a "rich" fuel mixture.
- Direct injection engines use a mixture of 40 or more parts air to one part fuel, written as 40:1.
- That compares to a normal gasoline engine's mix of 14.7:1. A leaner mixture allows fuel to be burned much more conservatively.
- A second efficiency plus for direct injection engines is that they can burn their fuel more completely.
- The fuel can be squirted directly where the combustion chamber is hottest -- in a gasoline engine that means it ends up close to the spark.
- With a traditional gasoline engine, the fuel air mixture disperses widely within the chamber, leaving a substantial amount unburned and therefore ineffective.

LECTURE - 4

Components of a Direct Injection gasoline System

Common Rail System -- A long metal cylinder called a fuel rail distributes fuel to the injectors under extremely high pressure.

Distributor and Inline Pump System -- Either a rotary wheel distributor or plunger-style pump is used to push pressurized fuel to the injectors.

Unit Direct System -- In this setup, the injector and a fuel pump just for that injector are integrated into a single unit and positioned over each cylinder

Advantages over indirect injection:

Better MPG

Leaner fuel mixtures

High power output

Accurately controlled emissions levels

More aggressive ignition timing curves

Precise control over amount of fuel and injection timings

No throttling losses in engines without a throttle plate

Disadvantages:

Dramatic efficiency losses due to deposits on the piston surface.

More deposits on the intake ports and valves.

Low mileage misfire codes.

LECTURE – 5

Lean-burn engine

In response to this interest, manufacturers have introduced natural gas engine-powered generator sets that feature “lean-burn” technology. The combustion is considered “lean” when excess air is introduced into the engine along with the fuel.

Def:

Lean-burn means pretty much what it says. It is a lean amount of fuel supplied to and burned in an engine's combustion chamber. Normal air-to-fuel ratio is on the order of 15:1 (15 parts air to 1 part fuel). True lean-burn can go as high as 23:1.

Combustion process

Reduces the occurrence of “knocking” or detonation. To prevent either knocking or misfiring, the combustion process must be controlled within a narrow operating window. Charge air temperatures and volume, together with air to fuel ratio, are constantly monitored. The microprocessor-based engine controller regulates the fuel flow and air/gas mixture and ignition timing.

Design of Engine

The design of the lean-burn engine incorporates a simple open combustion chamber housed in the piston crown. The shape of the piston crown intro-

duces turbulence in the incoming air/fuel mixture that promotes more complete combustion by thoroughly exposing it to the advancing flame front. The flame

plate of the cylinder head is regular (flat) and the spark plug is centrally located. The air and gas fuel are correctly mixed under the control of the engine management system.

Reduced Emissions

One of the results of this technology is significantly reduced emissions in the exhaust. Cummins’ new lean-burn gas engine generators have NO_x emissions

as low as .85 grams/BHP-hr, and produce low amounts of hydrocarbons (HC), carbon monoxide (CO) and particulate matter (PM). This allows the generator

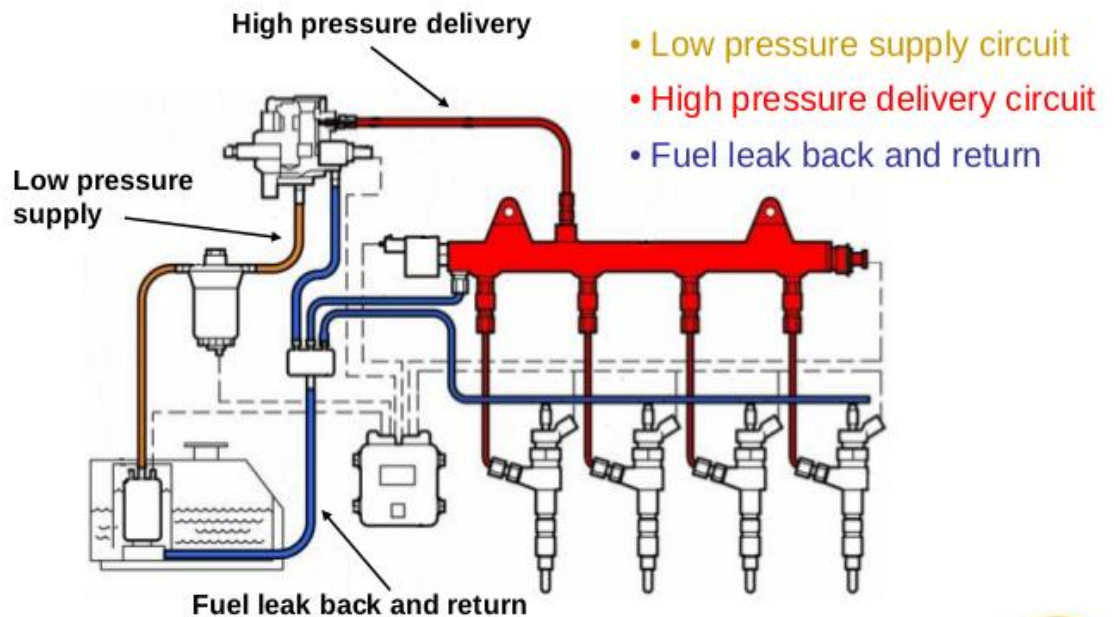
sets to meet the most stringent air quality regulations without after-treatment devices in the exhaust stream.

LECTURE -6, 7 & 8

Common Rail Diesel Fuel Systems

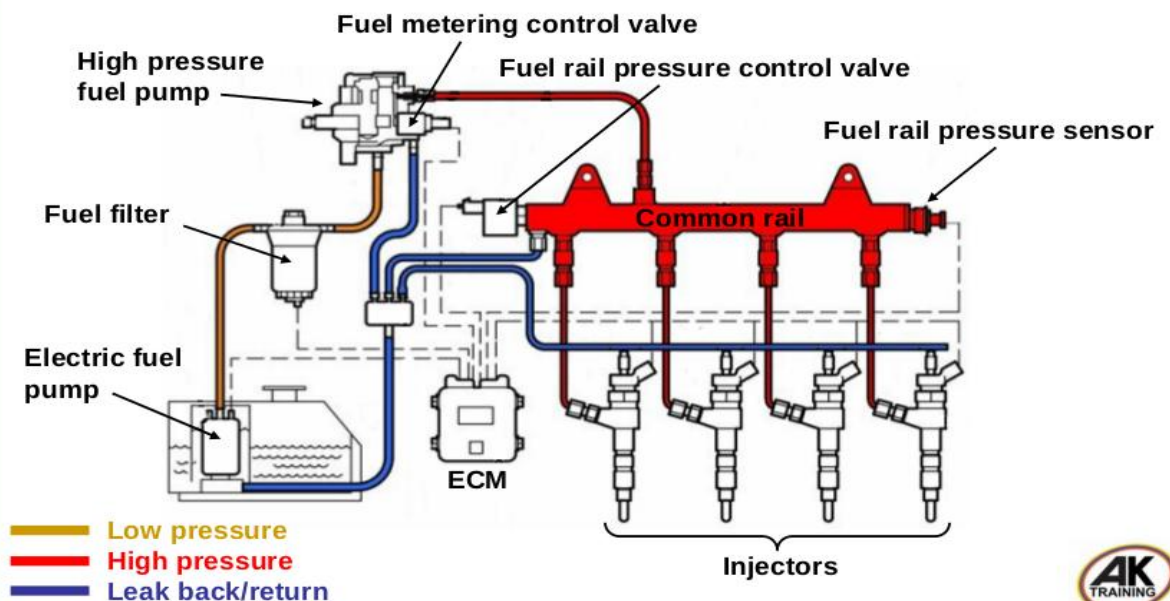
Common Rail Diesel Fuel Systems

The fuel system can be divided into three basic circuits



Common Rail Diesel Fuel Systems

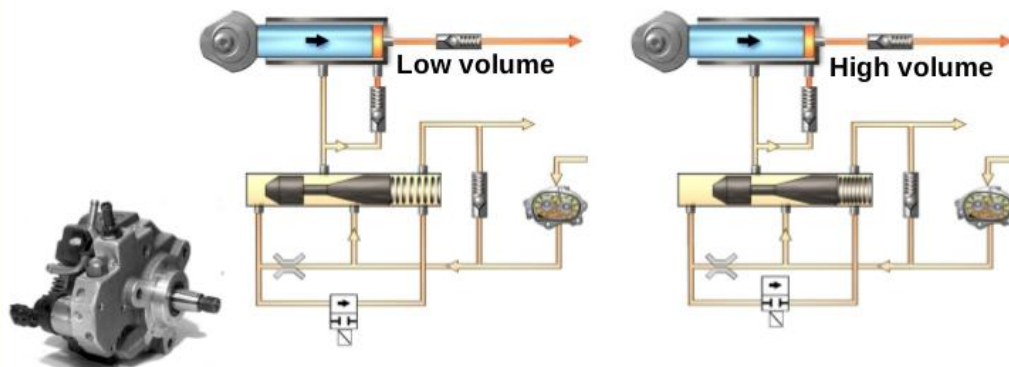
Components overview (example: Bosch EDC 16)



Common Rail Diesel Fuel Systems

Advantages of fuel intake volume regulation:

- Only the required volume of fuel is supplied to the common rail from the high pressure pump.....
- Reduced fuel flow around system results in lower fuel return flow temperature.....
- Reduced parasitic load on engine from high pressure pump contributes towards further reductions in exhaust emissions.



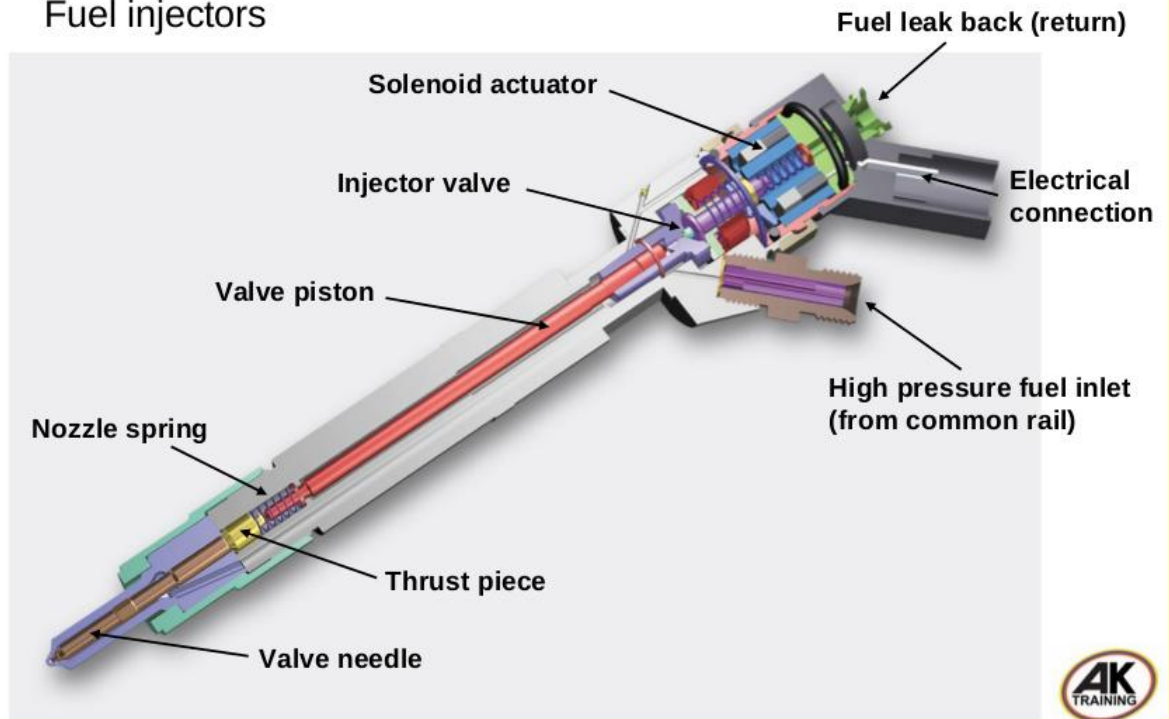
Common Rail Diesel Fuel Systems

- High pressure accumulator (common rail)
- Typical fuel rail pressure with engine idling and at running temperature:
 - approximately between 300 – 400 Bar (4410 – 5880 psi)
- Health and safety
 - Due to the extremely high working fuel pressures in the common rail fuel system, NEVER slacken fuel or injector pipes or try to disconnect components of the fuel system whilst the engine is running and high pressure is present in the system!
- Typical maximum possible fuel rail pressure:

- approximately between 1600 – 2000 Bar (23520 – 28400 psi)

Common Rail Diesel Fuel Systems

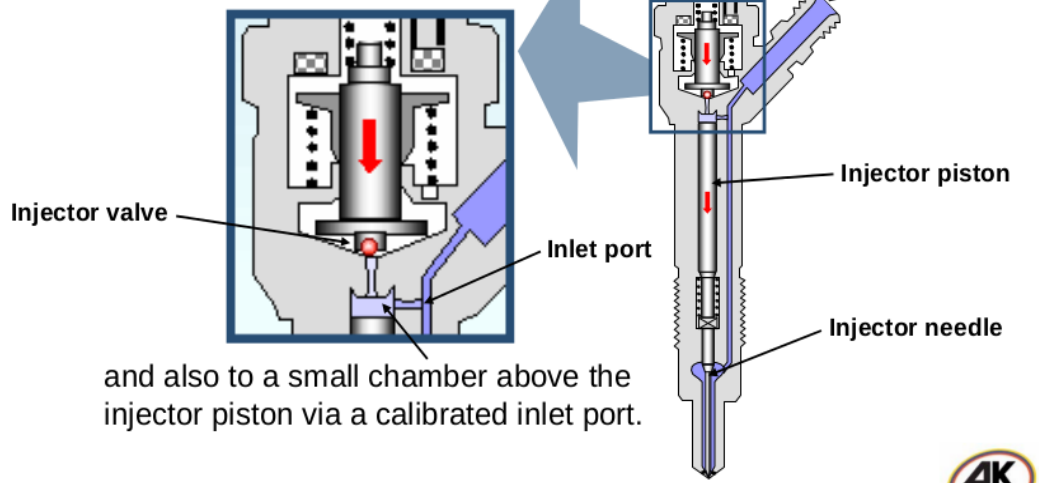
Fuel injectors



Common Rail Diesel Fuel Systems

Operation of fuel injectors

Fuel pressure is supplied to the injector needle seat area.....

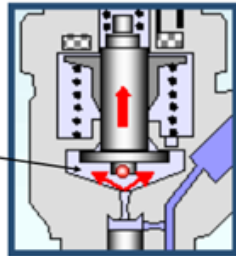


Common Rail Diesel Fuel Systems

Operation of fuel injectors

When the solenoid is energized, the injector valve opens.

Injector valve opens



Fuel pressure is relieved above the injector piston and returns to the fuel tank via the injector leak back (return) ports.

Fuel leak back (return) port

High pressure fuel inlet

Injector piston

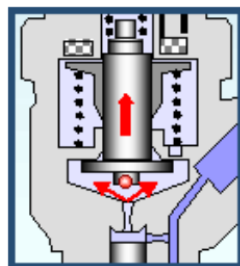
Injector needle



Common Rail Diesel Fuel Systems

Operation of fuel injectors

This creates a pressure difference above and below the injector piston.



Fuel pressure below the injector needle lifts the needle.

Fuel leak back (return) port

High pressure fuel inlet

Injector piston

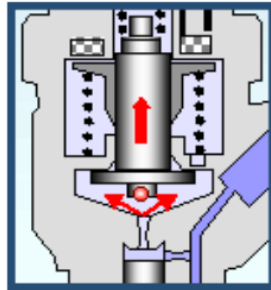
Injector needle



Common Rail Diesel Fuel Systems

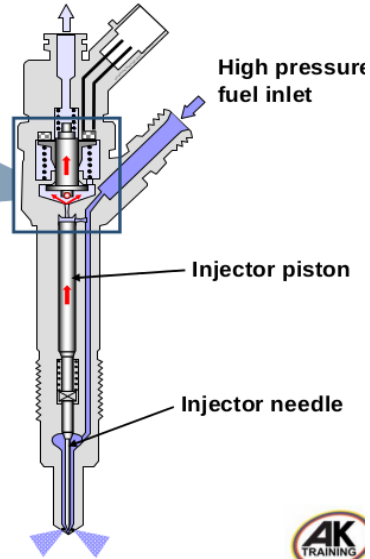
Operation of fuel injectors

Fuel is now injected into the cylinder.



Maximum stroke of solenoid valve:
approximately 50 micrometers (0.05 mm).

Fuel leak back (return) port



Common Rail Diesel Fuel Systems

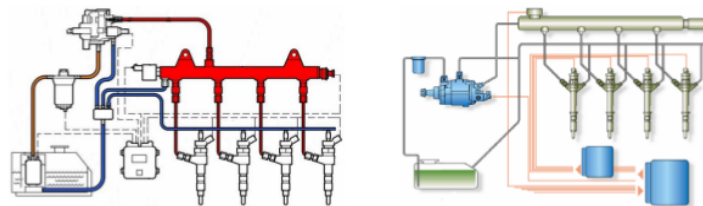
Fuel system diagnosis

Common rail diesel fuel systems operate on a closed loop basis.

The system carries out a great many complex calculations to precisely control fuel quantity and injection timing.

A range of tools and test equipment is commercially available to assist with diagnosis of the system.

The following is a brief overview to highlight some of the basic tests that can be carried out to diagnose faults with the system.



Advantages of common rail:

- Fuel pressure available on demand.....
- Higher injection pressures and finer atomization of fuel.....
- Injection pressure created independent of engine speed.....
- Multiple injections per cylinder combustion are possible.

- Reduction of overall exhaust emissions.....
- Reduction of particulate emissions.....
- Reduction of noise emissions.....
- Improved fuel efficiency.....
- Higher performance.

LECTURE – 9

Electronic Engine Management

Purpose of EEM:

- ☞ Controlling of Engine operation electronically by electronic components.
- ☞ Benefits to the motorist, more power, better mileage, a smoother idle and reduced operation expenses.
- ☞ Cost of components come down.
- ☞ Engine become smaller in size.

Basic components:

1. Electronic Control Module
2. Fuel Delivery System
3. Ignition System
4. Sensors

Electronic Control Module:

- ☞ It is a extremely reliable piece of hardware.
- ☞ It process information hundreds of times per second.
- ☞ It is actually a microprocessor.

- ☞ It is programmed by the Manufacturer.
- ☞ It controls the fuel delivery and ignition timing by receiving information from sensors.

Fuel Delivery system

- ☞ The fuel line passes through which feeds each injector and it passes through a pressure regulator.
- ☞ The surplus fuel heading back to the tank in the return line. The air is taken from the atmosphere. It is mixed with fuel just before the inlet valve by the fuel injector.
- ☞ This fuel delivery system is controlled by ECU to provide lean and rich mixtures depending on operating conditions.

Ignition system:

- ☞ To maximize the Engine output, spark should be at the precise moment.
- ☞ Maximum combustion chamber pressure can be attained.
- ☞ A mechanical advance distributor is used for this.
- ☞ A spark advance map is developed and stored in the ECU.
- ☞ As the speed increases, the spark should be advanced further.

Sensors:

1 Throttle Position Sensor

- ☞ The ECU senses how wide the throttle is opened.
- ☞ The ECU controls fuel delivery and spark timing.
- ☞ It consists of a wiper arm and resistor.
- ☞ At idle, resistance is high, Voltage is .6-.9v.
- ☞ As throttle presses, resistance decreases, voltage is 3.5-4.7v.

2. Exhaust gas oxygen sensor:

- ☞ It is placed in Engine's exhaust system.
- ☞ Consists of Zirconium Dioxide cell, provide precise indication of stoichiometric A/F ratio.(14.7:1).
- ☞ Zirconium Dioxide is the outer cell exposed to exhaust gas and inner electrode is exposed to ambient air.
- ☞ The output varies from 0v-1v.
- ☞ At stoichiometric point it produces .45v
- ☞ It provides precise output voltage measurements.

MAP sensor:

- ☞ It senses the degree of vacuum in the engine intake manifold.
- ☞ Vacuum decreases when throttle is opened/engine are under load.
- ☞ A silicon chip (piezoelectric) is provided with a reference pressure one side.
- ☞ Pressure to be measured is applied on the other side.
- ☞ Variation in pressure causes change in resistance of the silicon chip.
- ☞ It covers the amount of air drawn into voltage.
- ☞ ECU senses it and calculates the engine load.
- ☞ It is located at the intake air stream, between the air cleaner and throttle body.

Coolant temperature sensor:

- ☞ ECU Senses the temperature and decides whether to activate/deactivate the cooling fans in water cooled engines.
- ☞ It helps to enrich the mixture for cold starting, and provide lean mixtures for fuel economy.
- ☞ It directly gives the engine operating temperature.
- ☞ It is located on the coolant passage.

- ☞ It Consists of a fixed value resistor, supplied with 5V.
- ☞ ECU Senses the change in voltage between the temperature sensor and the fixed value resistor.
- ☞ When the sensor is cold, resistance is high and voltage is increases.
- ☞ As the temperature increases, the sensor resistance and the voltage decreases.

Knock Sensor:

- ☞ It detects the engine knock occurs during combustion process, it is located in the engine block.
- ☞ Engine knocks within a specific frequency range.
- ☞ It is a piezoelectric element, generates a voltage when a pressure or vibration is applied.

Throttle and timing sensor:

- ☞ It provides the information about how fast the engine is running or where the crankshaft is in its rotation.
- ☞ ECU senses this information and adjusts the fuel injection and the spark timing so that the engine speed does not exceed the safe operating limits.
- ☞ These sensors use a target wheel with a missing or odd-shaped gear tooth to provide the reference position.

Course material Compiled from:

- a) *“A course in internal combustion engines” by V.M. Domkundwar.*
- b) *John B Heywood, “Internal Combustion Engine Fundamentals”*
- c) *Gupta H.N, “Fundamentals of Internal Combustion Engines”*
- d) *“Engineering fundamentals of Internal combustion engines” by Willard W. Pulkrabek.*
- e) *“Advanced internal combustion engine research” Peter Van Blarigan Sandia, National Laboratories, Livermore, CA 94550.*
- f) *And many more publications with the help of internet published materials and sites.*