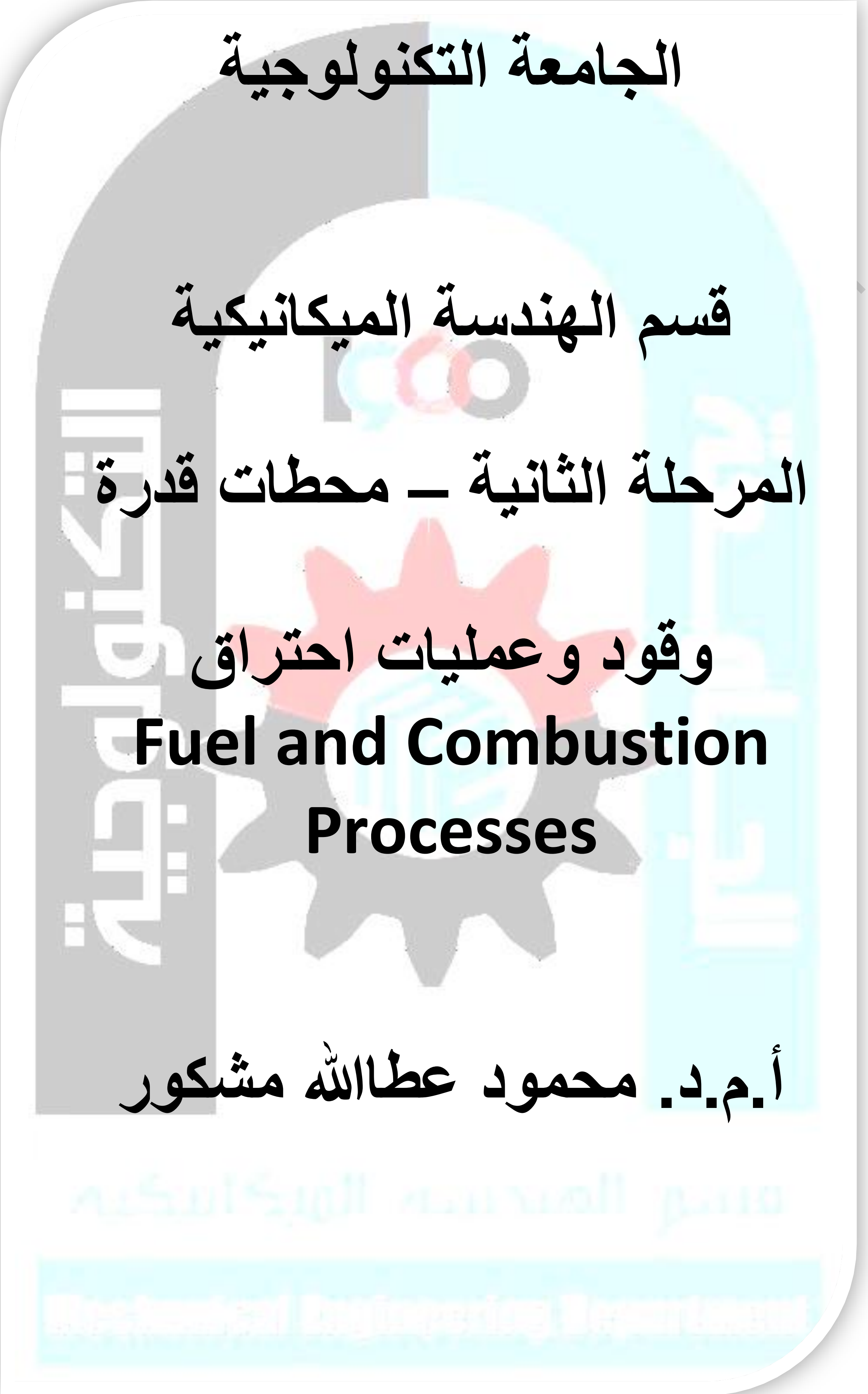

الجامعة التكنولوجية

قسم الهندسة الميكانيكية

المرحلة الثانية – محطات قدرة

وقود وعمليات احتراق
Fuel and Combustion Processes

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Fuel: is any material that will undergo an exothermic structural change. The energy stored in the chemical bonds that hold the carbon and hydrogen atoms together, releases when bonds are broken and atoms are rearranged. This energy stores in fuel by the process of photosynthesis which converts sunlight into chemical energy and storing it in the bond of sugar. Plants needs Carbon dioxide, Water and sunlight to make sugar. The overall reaction is: $6CO_2 + 6H_2O + Sunlight \rightarrow C_6H_{12}O_6 + 6O_2$. These plants when became dead, buried under the soil, as more and more soil deposited over them for thousands of years, they became compressed and then under high temperature and pressure converted into fossil fuel.

The various types of fuels (fig. 1) are available for firing in boilers, furnaces and other combustion equipments. Almost 70% of energy used in the world came from combustion sources, for example:

- Heat for homes comes directly from combustion.
- Electricity for homes generated by burning fossil fuel.
- Our transportation system relies almost entirely on combustion.
  - Aircraft are entirely powered by fuel burning.
  - Most trains are powered by diesel engine.
  - Most of the domestic vehicles use gasoline.
- Industrial process rely heavily on combustion.
  - Iron, steel, aluminum and other metal refining industries employ furnaces for producing raw products.
  - Cement industry is the heavy user of heat energy delivered by combustion.
  - Other examples of industrial combustion devices are boiler, refinery, glass melters and solid dryers etc.

Fuels are originated from marine life or vegetable life (e.g. plants):

$$CO_2 + H_2O + Solar\ energy \xrightarrow{photosynthess} plant\ material$$
$$\xrightarrow{long\ time\ geological} fuel + O_2 \xrightarrow{combustion} CO_2 + H_2O + Energy$$

From this we can see that the products are the same reactant materials at very long time like 2,000,000 years. 25% of the O_2 comes from marine life on the ocean.

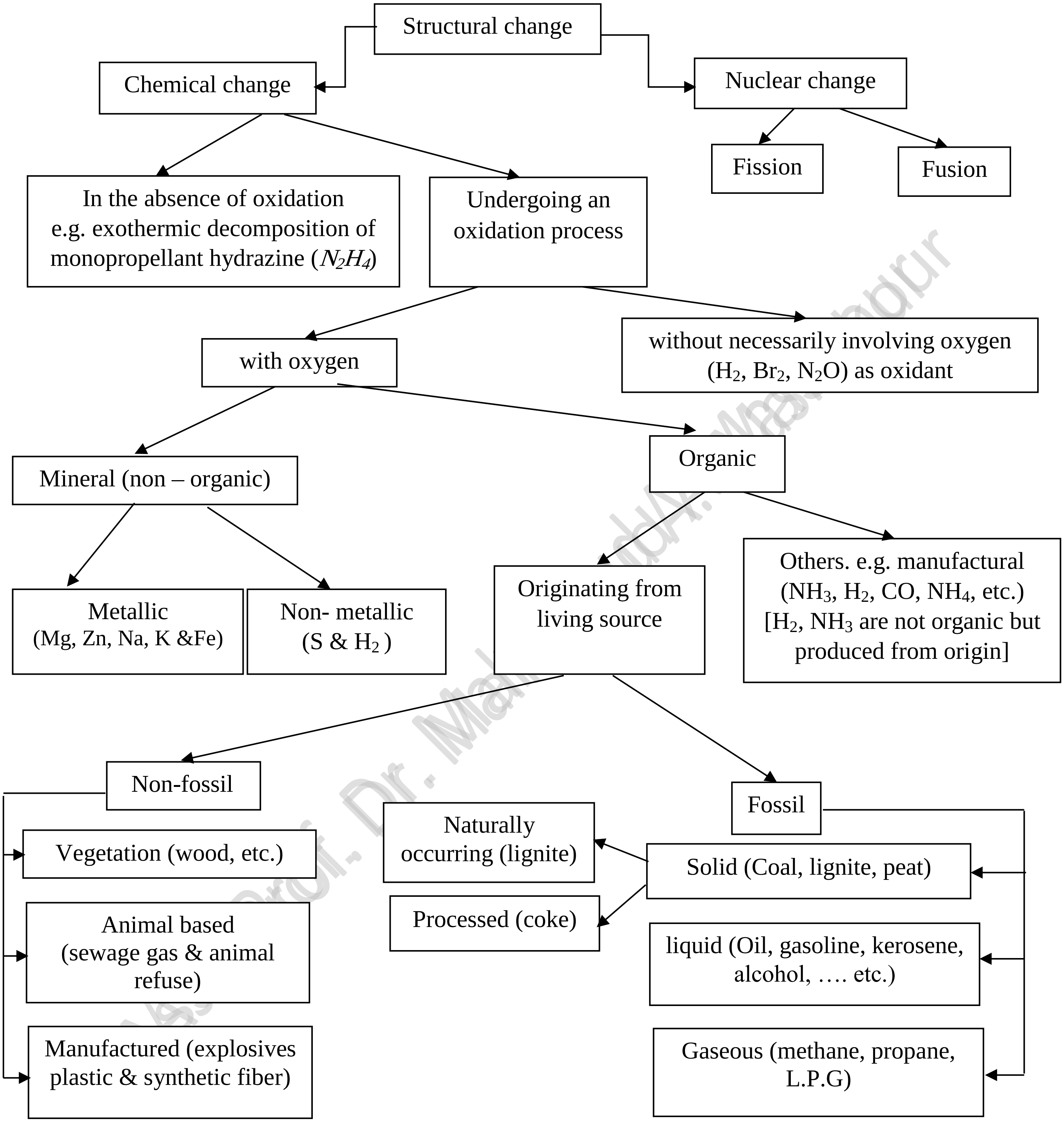

Structural change
Chemical change
Nuclear change
Fission
Fusion
In the absence of oxidation e.g. exothermic decomposition of monopropellant hydrazine (N_2H_4)
Undergoing an oxidation process
with oxygen
without necessarily involving oxygen (H_2, Br_2, N_2O) as oxidant
Mineral (non – organic)
Organic
Metallic (Mg, Zn, Na, K &Fe)
Non- metallic (S & H_2)
Originating from living source
Others. e.g. manufactural (NH_3, H_2, CO, NH_4, etc.) [H_2, NH_3 are not organic but produced from origin]
Non-fossil
Fossil
Vegetation (wood, etc.)
Animal based (sewage gas & animal refuse)
Manufactured (explosives plastic & synthetic fiber)
Naturally occurring (lignite)
Processed (coke)
Solid (Coal, lignite, peat)
liquid (Oil, gasoline, kerosene, alcohol, …. etc.)
Gaseous (methane, propane, L.P.G)

Fig. 1.

General Requirements of Fuel in combustion systems

High power output: Higher power output is affected by the following characteristics of fuel.

a. Calorific value; heat of combustion.
b. Thermal efficiency.
c. Inlet temperature.

Low consumption:

a. Calorific value.
b. Thermal Efficiency.

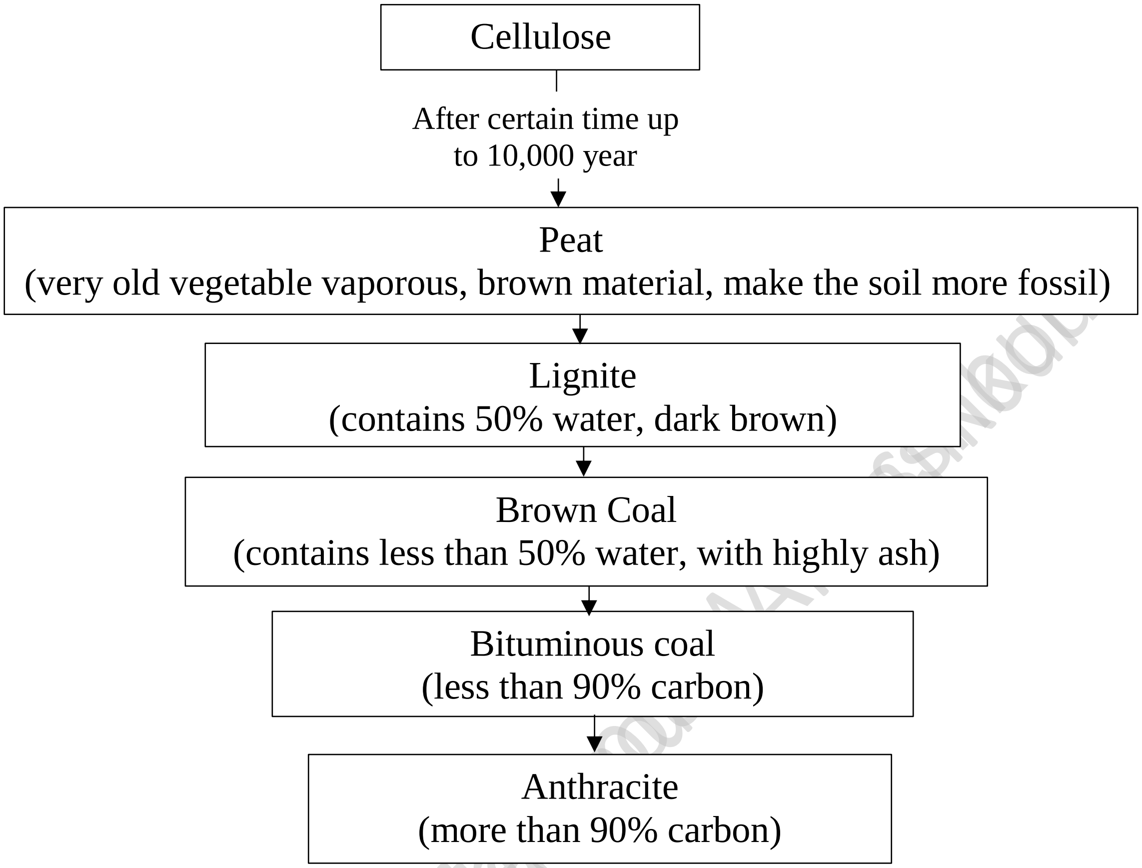
Low Pollution:

a. Inlet temperature.
b. Volatility.
c. Ignition temperature.

Besides that, the reaction characteristics of the fuel should be such that the rate of pressure and temperature rise inside the chamber due to combustion is moderate and reduce the possibility of large mechanical and thermal stresses in the components of the system. Other requirements are:

- High thermal stability.
- Low deposit forming tendencies.
- Good fire safety.
- Low toxicity.
- Easy transfer, handle & storage.
- Easy starting under various working conditions.

Solid Fuels: - Originate from plant life. Stages information of coal:



Each of these steps take thousands of years through the action of:

1.Time, 2. Heat, 3. Catalysts, 4. Pressure, 5. Bacteria.

The oxygen, Sulphur, and water are decrease while the heating value (calorific value) by mass, (C/H) ratio increased.

<u>Definitions:</u>

Peat: Peat is the first stage in the formation of coal from wood (cellulose). It found as layers. if we cut a piece and let it dry, the result will be a piece of material and even we can see through it. These pieces used as fuel in power stations. It has a high quantity of water, Sulphur, oxygen and volatile, etc.

Lignite: It is the second stage product in the formation of coal from wood. It is a solid contains a lot of water, very high volatile and it is dangerous in storage. Also, it's contain a lot of sulphur & Ash. It can be used as fuel (have to be granded) with 10% gasoline or natural gas. It's heating (colorific) value high than that of peat.

Bituminous Coal: It is the most common variety of coal. It is black and brittle which burns and ignites readily with yellow smoky flame. It must be crushed as a powder, because as we know in order to burn any fuel, it should be in the gaseous form.

Anthracite: It is most matured coal hence of highest rank. It is hard and burns without smoke with a short non-luminous flame. Very deep in the ground, needs long time, very shine, very low in volatility, very limited & expensive. The chief uses of anthracites are in boilers and metallurgical furnaces.

Graphite: Is a pure carbon, it is crystiles, and it is grows like ice crystal.

The most important things, about coal to remember are, it is vary in composition (as wet, ashy & solid). However, the bulk world resource is coal, as brown coal & lignite. There are many problems associated with solid fuels, these problems are:

a- Transportation,
b- Air pollution mainly due to sulphur,
c- A lot of ash.

Fresh wood contains (25% -50 %) moisture. (C.V.) of dry wood (18150 - 22570) kJ/kg. Wood contains a much higher percentage of volatilities than coal and with lower ash (0.3 to 0.8%). It needs much air for combustion as secondary air and special furnaces. The presence of oil in wood as in pines tends to rise (H.V.) of the wood but causes it to consumed faster.

Charcoal: is highly porous and has enormous effective surface area of contact percent unit mass.

Coke: mainly used for metallurgical applications is obtained by heating coal the absence of air producing tar & combustible gas as by product. The coke produced is usually light and porous, hence, it is a good burning fuel & used usually in blast furnaces for steel making.

Liquid fuel:

Crude oil:

Crude oils are extremely complex mixture consisting predominately of hydrocarbons & compounds containing sulfur, nitrogen, oxygen, & trace metals as minor constituents. The physical and chemical characteristics of crude oils very widely depending on the percentages of the various compounds that are present. The specific gravities cover a wide range, but most crude oils are between (0.80 to 0.97) g/ml, or gravity between 45 and 50 degrees API [American petroleum Institute]. There also is a wide variation in viscosity but most crude oils are in the range from 2.3 to 23 centistokes. The ultimate composition shows 84 to 86% carbon 10 to 14% hydrogen, and small percentage of sulfur, nitrogen and oxygen. The sulfur content is usually below 1.0%, but it may be as high as 5.0%.

The various properties of liquid fuels are given below.

1. Density

Density is defined as the ratio of the mass of the fuel to the volume of the fuel at a reference temperature of 15°C. Density is measured by an instrument called a hydrometer. The knowledge of density is useful for quantitative calculations and assessing ignition qualities.

2. Specific gravity

This is defined as the ratio of the weight of a given volume of oil to the weight of the same volume of water at a given temperature. The density of fuel, relative to water, is called specific gravity. The specific gravity of water is defined as 1. Since specific gravity is a ratio, it has no units. The measurement of specific gravity is generally made by a hydrometer. Specific gravity is used in calculations involving weights and volumes.

3. Viscosity

The viscosity of a fluid is a measure of its internal resistance to flow. Viscosity depends on the temperature and decreases as the temperature increases. Any numerical value for viscosity has no meaning unless the temperature is also specified. Viscosity is measured in Stokes / Centistokes. Each type of oil has its own temperature - viscosity relationship. The measurement of viscosity is made with an instrument called a Viscometer.

Viscosity is the most important characteristic in the storage and use of fuel oil. It influences the degree of pre- heating required for handling, storage and satisfactory atomization. If the oil is too viscous, it may become difficult to pump, hard to light the burner, and difficult to handle. Poor atomization may result in the formation of carbon deposits on the burner tips or on the walls. Therefore pre-heating is necessary for proper atomization.

4. Flash Point

The flash point of a fuel is the lowest temperature at which the fuel can be heated so that the vapour gives off flashes momentarily when an open flame is passed over it.

5. Pour Point

The pour point of a fuel is the lowest temperature at which it will pour or flow when cooled under prescribed conditions. It is a very rough indication of the lowest temperature at which fuel oil is ready to be pumped.

6. Specific Heat

Specific heat is the amount of kCals needed to raise the temperature of 1 kg of oil by 1^0C. The unit of specific heat is kcal/kg^{0}C. It varies from 0.22 to 0.28 depending on the oil specific gravity. The specific heat determines how much steam or electrical energy it takes to heat oil to a desired temperature. Light oils have a low specific heat, whereas heavier oils have a higher specific heat.

7. Calorific Value

The calorific value is the measurement of heat or energy produced, and is measured either as gross (higher) calorific value or net (lower) calorific value. The difference is determined by the latent heat of condensation of the water vapour produced during the combustion process. Gross (Higher) calorific value (GCV(HCV)) assumes all vapour produced during the combustion process is fully condensed. Net (Lower) calorific value (NCV(LCV)) assumes the water leaves with the combustion products without fully being condensed. Fuels should be compared based on the net calorific value.

The calorific value of coal varies considerably depending on the ash, moisture content and the type of coal while calorific value of fuel oils are much more consistent.

8. Sulphur

The amount of sulphur in the fuel oil depends mainly on the source of the crude oil and to a lesser extent on the refining process. The normal sulfur content for the residual fuel oil (furnace oil) is in the order of 2 - 4 %.

The main disadvantage of sulphur is the risk of corrosion by sulphuric acid formed during and after combustion, and condensation in cool parts of the chimney or stack, air pre-heater and economizer.

9. Ash Content

The ash value is related to the inorganic material or salts in the fuel oil. The ash levels in distillate fuels are negligible. Residual fuels have higher ash levels. These salts may be compounds of sodium, vanadium, calcium, magnesium, silicon, iron, aluminum, nickel, etc.

Typically, the ash value is in the range 0.03 - 0.07 %. Excessive ash in liquid fuels can cause fouling deposits in the combustion equipment. Ash has an erosive effect on the burner tips, causes damage to the refractories at high temperatures and gives rise to high temperature corrosion and fouling of equipments.

10. Carbon Residue

Carbon residue indicates the tendency of oil to deposit a carbonaceous solid residue on a hot surface, such as a burner or injection nozzle, when its vaporizable constituents evaporate. Residual oil contains carbon residue of 1 percent or more.

11. Water Content

The water content of furnace oil when it is supplied is normally very low because the product at refinery site is handled hot. An upper limit of 1% is specified as a standard.

Water may be present in free or emulsified form and can cause damage to the inside surfaces of the furnace during combustion especially if it contains dissolved salts. It can also cause spluttering of the flame at the burner tip, possibly extinguishing the flame, reducing the flame temperature or lengthening the flame.

Oil Refinery

An oil refinery or petroleum refinery is an industrial process plant where crude oil is processed and refined into more useful petroleum products, such as gasoline, diesel fuel, kerosene, and liquefied petroleum gas.

Crude oil varies greatly in appearance depending on its composition. It is usually black or dark brown (although it may be yellowish, reddish, or even greenish). Crude oil is made up almost entirely of carbon and hydrogen.

Such combination may include vary a large molecular to very light molecular components. Large molecular weight MWt components have high boiling temperature. Small MWt components have low boiling point. High boiling point components, such as tar and asphalt can be returned back to the refinery for cracking into smaller MWt. Low boiling point components are a very desirable products, such as solvent and fuels like gasoline.

Crude oil is separated into fractions by fractional distillation. The fractions at the top of the fractionating column have lower boiling points than the fractions at the bottom. The heavy bottom fractions are often cracked into lighter, more useful products. All of the fractions are processed further in other refining units. However, two specific operations are used to separates crude oil components:

1. Cracking: is the process in breaking large molecular components into more useful smaller molecular weight components.
2. Distillation: is the process used to separate the mixture into single or smaller rangers of components.

Petroleum products are usually grouped into three categories: light distillates (LPG, gasoline, naphtha), middle distillates (kerosene, diesel), heavy distillates and residuum (heavy fuel oil, lubricating oils, wax, asphalt). Therefore the products from refined crude oil can be used for many applications such as for example: Automobile gasoline; Diesel fuel; Jet fuel; Home heating fuel; Industrial fuel; Natural gas (cooking); Lubricants; Asphalt; Rubber manufacturing; Paint manufacturing; Plastic manufacturing; Explosion production; Alcohols; Wax.

Hydrocarbons families

Since crude oil comes out from the ground in various parts of the world, it has been agreed to standardize the products of HCs by means of number of different families such as:

1. Paraffins
2. Olefins
3. Diolfins
4. Acetylenes
5. Cyclo-parafins
6. Aromatics
7. Alcohols

All of which may be considered as "saturated & unsaturated"

1. Saturated hydrocarbons (alkanes or paraffins) are the simplest of the hydrocarbon species and are composed entirely of single bonds and are saturated with hydrogen or another carbon atom.

   The general formula for saturated hydrocarbons is C_nH_{2n+2} (assuming non-cyclic structures).

   Saturated hydrocarbons are the basis of petroleum fuels and are either found as liner or branched species.

2. Unsaturated hydrocarbons have one or more double or triple bonds between carbon atoms.

   - Those with double bond are called alkenes.

     i- One double bond have the formula C_nH_{2n} (olefins)
     ii- Two double bond with formula C_nH_{2n-2} (diolefins)

   - Those containing triple bonds are alkynes, with general formula C_nH_{2n-2} (acetylene)

3. Cyclo-alkanes (cyclo-paraffins)

   Are hydrocarbons containing one or more carbon rings to which hydrogen atoms are attached. The general formula for a saturated hydrocarbon containing one ring is C_nH_{2n}.

4. Aromatic hydrocarbon, also known as arenes, are hydrocarbons that have at least one aromatic ring.

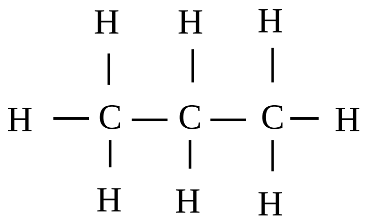
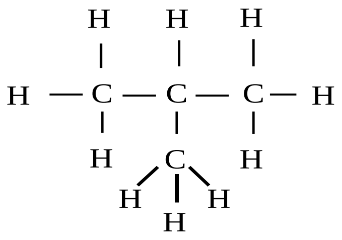
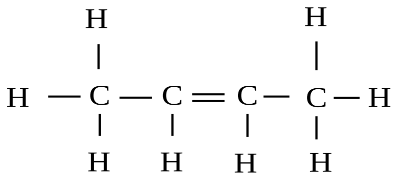
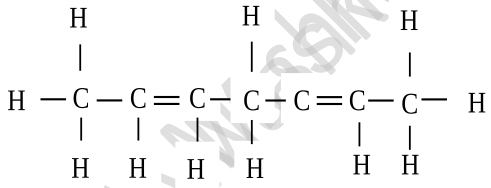
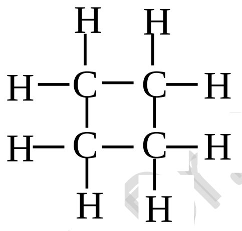
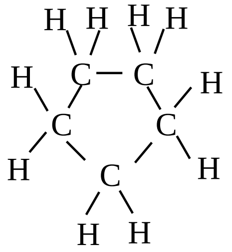
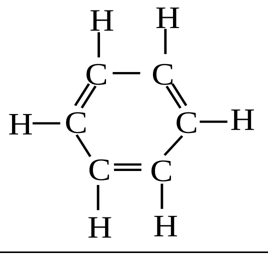
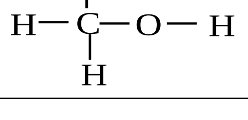
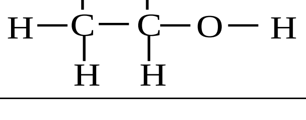
   General formula is C_nH_{2n-6}

5. Alcohols

   These are paraffins with (OH) replaces one (H)

   Examples:

   methanol CH_3OH
   ethanol C_2H_5OH

Linear chain	branch chain
Propane C_3H_5	isobutene C_4H_{10}
	
Alkenes:	
olfines	diolfines
	
Alkynes-acetylene (C_2H_2): $H-C\equiv C-H$	
Cyclo:	
cyclobutane (C_4H_8)	cyclopentane (C_5H_{10})
	
Aromatics (C_nH_{2n-6})-benzene (C_6H_6): 	
Alcohols:	
methanol (CH_3OH)	ethanol (C_2H_5OH)
	

General characteristics of Hydrocarbons

The heat of combustion of the fuel decrease as the carbon hydrogen ratio in the molecule increase. Therefore, in the refined petroleum group, Paraffins have the highest heat of combustion and aromatics have the lowest heat of combustion. Alcohols have lower heat of combustion than refined petroleum product due to the presence of OH group in the molecule.

- The self- ignition temperature of the fuel decrease as the carbon hydrogen ratio in the molecule decrease. Therefore, Paraffins have the lowest self - ignition temperature and aromatics have the highest self-ignition temperature.
- In general, as the number of atoms in the molecule increase, the boiling point rises. Thus fuels with less number of atoms in the molecule are more volatile.
- Ignition delay of fuel increase as the carbon hydrogen ratio increase. Therefor paraffin’s have low ignition delay and aromatics have comparatively high ignition delay. In the same family the more complex structure of the molecule the higher is the ignition delay. So that branched chain paraffins have higher ignition delay than straight chain paraffins.
- The anti-knock quality of fuel used in spark ignition engines improves with increase in ignition delay of the fuel. Therefore, paraffin’s offer less resistance to detonation while the aromatics offer better resistance. Also branched chain paraffins are more resistant to detonation than straight chain paraffin’s. Alcohol due to their complex structure has high anti-knock quality.
- The anti-knock quality of compression ignition engines fuel improves with decrease in ignition delay. Therefore, paraffins are better fuel and aromatics are least desirable. Within the paraffin group the straight chain paraffin’s with larger number of atoms in the molecule are considered to be best for compression ignition engines.

Gasoline

Gasoline or petrol, is a toxic translucent, yellow-tinted liquid mixture of many hydrocarbons, derived from petroleum, which is primarily used as a fuel in internal combustion engines. It consists mostly of aliphatic hydrocarbons obtained by the fractional distillation of petroleum, enhanced with isooctane or the aromatic hydrocarbons toluene and benzene to increase its octane rating. Small quantities of various additives are common, for the purposes of tuning engine performance or reducing harmful exhaust emissions. Some mixture also contain significant quantities of ethanol as a partial alternative fuel.

Gasoline is produced in oil refineries. Material that is separated from crude oil via distillation, called straight – run gasoline, does not meet the required specifications for modern engines (in particular octane rating; see below), but will from part of the blend. The bulk of a typical gasoline consists of hydrocarbons with between four and 12 carbon atoms per molecule.

Overall, a typical gasoline is predominantly a mixture of paraffins (alkanes), naphthenes (cycloalkanes), and olefins (alkenes). The actual ratio depends on:

1. The oil refinery that makes the gasoline, as not all refineries have the same set of processing units;
2. Crude oil feed used by the refinery;
3. The grade of gasoline, in particular, the octane rating.

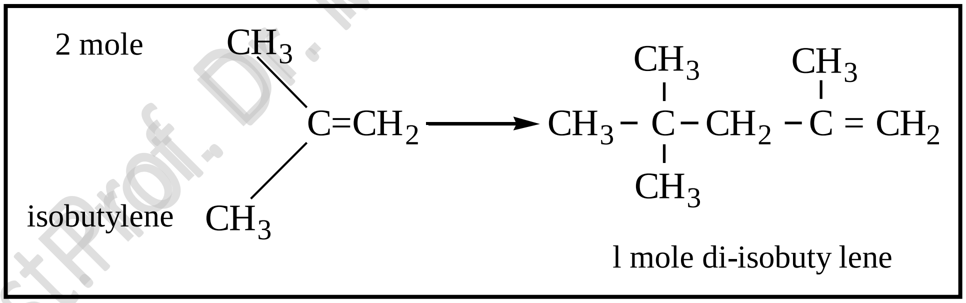
Gasoline for reasonable approximation can be considered as single component hydro-carbon fuel, such as:

1. Gasoline
   Chemical formula C_8H_{15}
   $MWt = 111$
2. iso-octane
   chemical formula C_8H_{18}
   $MWt = 114$

The product of gasoline can be increased by various conversion processes carried out at special conditions. These are:

1. **Cracking**
   Cracking consists of breaking down of heavy high boiling temperature hydrocarbons to lighter hydrocarbons of lower boiling temperature. Cracking can be brought about either- by subjecting the heavy hydrocarbons to heat and pressure only (thermal cracking) or by the addition of a catalysis (catalytic cracking). Catalytic cracking produce gasoline of higher anti- knocks quality.

2. **Polymerization**
   Large quantities of very low boiling point gaseous hydrocarbons are formed during cracking. These by- products gases (Oleffins) can be combined together in the presence of a catalyst to form a higher molecule weight hydrocarbons within the range of gasoline (catalyst polymerization). The typical reaction is:



$$2\,(CH_3)_2C{=}CH_2 \longrightarrow CH_3-C(CH_3)_2-CH_2-C(CH_3){=}CH_2$$

3. **Alkylation**
   Alkylation combines an olefin with an isoparaffin to produce a branched chain isoparaffin of molecular weight equal to the sum of reactants, in the presence of catalyst.
   This gives a fuel of very high octane number such as iso-octane or iso-pentane.

$$\underset{Butylene}{C_4H_8} + \underset{iso-butane}{C_4H_{10}} \xrightarrow{Alkylation} \underset{iso-octance}{C_8H_{18}}$$

4. **Isomerisation**

   Isomerisation is process by which the atoms of carbon and hydrogen in normal hydrocarbons are re- arranged to produce a more complex structure (iso- structure) of higher antiknock value. Isomerisation is used to:

   Convert n- butane into iso-butane for alkylation.

   Convert n- pentane and n- hexane into iso- paraffins to improve knock rating of highly volatile gasoline.

5. **Reforming**

   Reforming is a process for convert low- octane gasoline or naphtha by reaction with hydrogen so it can be called hydro-forming.

6. **Super-fractionation**

   Super-fractionation is a method for separation of branched chain alkanes from their isomers the normal a lkanes, which are poor in antiknock properties, the separated branched chain alkanes are used for blending to produce high octane number gasoline.

7. **Blending**

   Blending is the process of obtaining a product of desired quality by mixing certain products in some suitable proportions.

Gasoline properties:

1. Density:

   The specific gravity (or relative density) of gasoline ranges from 0.71-0.77 (719.7 kg/m^3), gasoline floats on water; water cannot generally be used to extinguish a gasoline fire, unless used in a fine mist.

2. Volatility:

   Is the tendency of a substance to vaporize. Volatility is directly related to a substance's vapor pressure. At a given temperature, a substance with higher vapor pressure vaporizes more readily than a substance with a lower vapor pressure.

As mentioned previously, low boiling point HC components is:

i. Is very useful products

ii. It's low boiling temperature means it has high volatility.

iii. High volatility components can vaporize at low temperature, which is useful for cold starting of an engine, as fuel must vaporize before it can be burned.

iv. However, too high volatility is undesirable, because :

  a. Reduces volumetric efficiency, as fuel vapor replaces air during induction stroke.

  b. It can cause vapor lock which prevent fuel flow through carburetor lines.

3. Self-ignition temperature SIT

   If the temperature of air-fuel AF mixture is raised high enough, the mixture will ignite itself without the need of a spark or external igniter. The temperature above which this occurs is called the "self-ignition temperature" and will be denoted "SIT".

SIT is usually obtained when having high compression ratio CR. In CI engines, CR is usually high and SIT is obtained, this what really makes the principles of ignition in CI engines. In SI engines SIT may be referred to as "auto-ignition", or "per-ignition". And this is not desirable at all in SI engine operation. As ignition start by the spark of a spark plug at pre-set time of the cycle, this is why low CR is imposed on the design of SI engines. If SIT is reached in SI engines, then pressure pulses are generated, often in audible frequency, which may cause serious damage to the engine. This phenomena is called "knocking or pinging or detonation".

Octane rating:

Internal combustion engines are designed to burn gasoline in a controlled process called deflagration. Deflagration is a technical term describing subsonic combustion that usually propagates through thermal conductivity (hot burning material heats the next layer of cold material and ignites it). But in some cases, gasoline can also combust abnormally by detonation, which wastes energy and can damage the engine. (Detonation (knock, spark knock, pinging or pinking) in SI engines occurs when combustion of the air/fuel mixture in the cylinder starts off correctly in response to ignition by the spark plug, but one or more pockets of air/fuel mixture explode outside the envelope of the normal combustion front). One way to reduce detonation is to increase the gasoline's resistance to auto-ignition, which is expressed by its octane rating. (Auto-ignition the auto ignition temperature of a substance is the lowest temperature at which it will spontaneously ignite in a normal atmosphere without an external source of ignition, such as a flame or spark).

Octane rating is measured relative to a mixture of 2,2,4-trimethylpentane and n-heptane. There are different conventions for expressing octane ratings, so a fuel may have several different octane ratings based on the measure used. A higher octane rating simulates a higher compression ratio, and thus higher temperature and pressure, which translate to higher power output. For example; Petrol with the same knocking characteristics as a mixture of 90% iso-octane and 10% heptanes would have an octane rating of 90. This does not mean that the petrol contains just iso-octane and heptanes in these proportions, but that it has the same detonation resistance properties. Because some fuels are more knock-resistant than iso-octane, the definition has been extended to allow for octane numbers higher than 100.

Octane number

Octane number is a fuel properties that describes "how well the fuel will or will not self-ignite". Or; Octane number is a measure of the fuel resistance to spontaneous (self) ignition during compression stroke prior to the desired position of the piston in the cylinder set by valve and ignition timing. The higher the ON the higher it's resistance to self-ignition. Therefore, high ON improves engine performance and prevents knocking, and therefore, prevent possible engine damage, thus improving engine service life.

Measuring ON:

Two kind of fuels are used as standard or primary reference fuels (PRF):

1. Iso-octane (2, 2, 4 tri-methylpentane) having low boiling point and has very slight tendency to knock, and therefore it's ON = 100.
2. n-heptane, which is characterized by high tendency to detonate, therefore it's ON = 0.

To find the (ON) of a fuel, the following test is carried using special test engine:

1. The fuel (**x**) is used, CR is adjusted at specified operating conditions until a standard level of knock is experienced.
2. Fuel (**x**) is replaced by mixture of the (PRF) through special designed intake system to have any blend of mixture, until same level of knock is obtained.
3. At that instant, the (%) of iso-octane in the mixture represent the (ON) of fuel (**x**).

Measuring methods:

1. Research method RON [Research Octane Number]

The most common type of octane rating worldwide is the (RON), determined by running the fuel in a test engine with a variable compression ratio under controlled conditions, and comparing the results with those for mixture of iso-octane and n-heptane.

2. Motoring method MON [Motor Octane Number]

(MON), is a better measure of how the fuel behaves when under load, as it is determined at 900 rpm engine speed, instead of the 600 rpm for RON. MON testing uses a similar test engine to that used in RON testing, but with a preheated fuel mixture, higher engine speed, and variable ignition timing to further stress the fuel's knock resistance.

3. Aviation method AON

"octane booster" gasoline additives was commonly used for ON improvement, but since the 1970s, its use in most of the industrialized world has been restricted, and its use is currently limited mostly to aviation gasoline.

4. Road Octane Number (R_dON)

The final type of octane rating, called Observed Road Octane Number (R_dON), is derived from testing gasoline in real world multi-cylinder engines, normally at wide open throttle. The actual testing was done in cars on the road but as technology developed the testing was moved to chassis dynamometers with environmental controls to improve consistency.

(ON) characteristics:

1. Anti-knock index AKI

$$\mathrm{AKI} = 1/2\,(\mathrm{MON} + \mathrm{RON})$$

2. Fuel sensitivity FS it shows how sensitive a fuel knock characteristics to engine geometry.

$$\mathrm{FS} = \mathrm{RON} - \mathrm{MON}$$

0 ← FS → 10
better ← | → poorer
performance | performance

3. Performance number PN:

$$\mathrm{PN} = \frac{\text{knock limit imep of fuel}_x}{\text{knock limit imep of iso} - \text{octane}}$$

4. Fuel mixture ON:

$$\mathrm{ON}]_{\mathrm{mix}} = \sum [(\text{mass\% i})(\mathrm{ON})\mathrm{i}]$$

ON extended scale:

As stated earlier that "some fuels are more knock-resistant than iso-octane", and so this may lead to an extended ON rating for a value higher than 100. In this case, how can such fuels be compared when they are better than iso-octane in anti-knock characteristics. The answer to this problem is through the "Wiese method" of extended ON scale by relating [ON to PN] as follow:

$$\mathrm{ON} = 100 + \frac{\mathrm{PN} - 100}{3}$$

Where

$$\mathrm{PN} = \frac{\text{knock limit imep of fuel}_{\mathrm{x}}}{\text{knock limit imep of iso} - \text{octane}}$$

Another method of extended ON rating is by relating ON to anti-knock agent called tetraethyl lead (TEL) as follows:

$$\mathrm{ON} = 100 + \frac{28.28\ \mathrm{X}}{1.0 + 0.736\mathrm{X} + \mathrm{Y}}$$

$$\mathrm{X} = \text{ml TEL/gallon gasoline}$$

$$\mathrm{Y} = \sqrt{1.0 + 1.472\ \mathrm{X} - 0.035216\ \mathrm{X}^2}$$

Variable effecting ON:

1. Combustion chamber geometry.
2. Gas turbulence
3. Induction swirl
4. Temperature
5. Inert gas addition
6. Flame speed

Engine knock:

Engine knock, is a sound that is made when the fuel ignites too early in the compression stroke. Severe knock causes severe engine damage, such as:

1. broken connecting rods,
2. melted pistons,
3. melted or broken valves and other components.

The Octane rating is measure of how likely a gasoline or liquid petroleum fuel is to self-ignite. The higher the ON, the less likely an engine is to pre-ignite and suffer damage.

Figure below Self-ignition characteristics of fuels. If the temperature of a fuel is raised above the self-ignition temperature (SIT), the fuel will spontaneously ignite after a short ignition delay (ID) time. The higher above SIT which the fuel is heated, the short will be ID. Ignition delay is generally on the order of thousandths of a second.

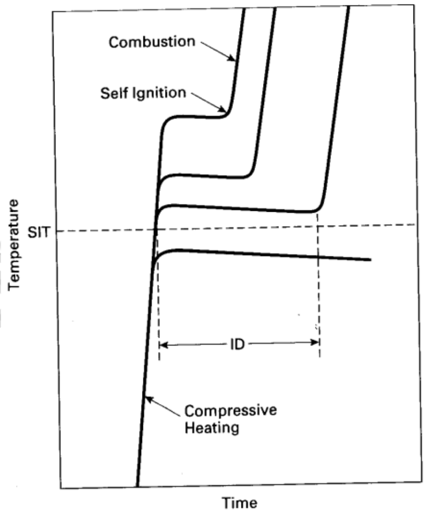


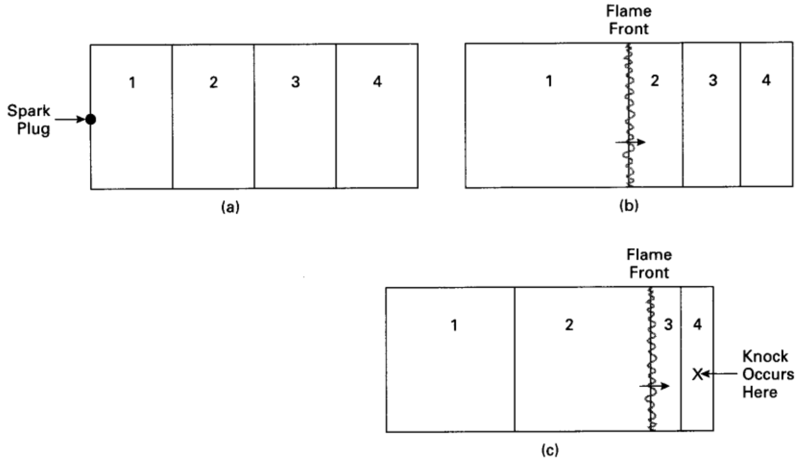
Figure below SI engine combustion chamber schematically visualized as long hollow cylinder with the spark plug located at left end.

(a) Mass of air-fuel is equally distributed as spark plug is fired to start combustion.

(b) As flame front moves across chamber, unburned mixture in front of flame is compressed into smaller volume.

(c) Flame front continues to compress unburned mixture into smaller volume, which increases its temperature and pressure.

If compression raises temperature of end gas above SIT, self-ignition and knock can occur.



How knock occurs:

In SIE, during normal combustion a flame front is produced by the spark, spreading uniformly across the combustion chamber with a smooth pressure rise. The heat released due to combustion increases temperature and pressure of the burned gases. The burned gases expand compressing the unburned gases, thereby increasing their (T &P), and thus the flame front proceeds throughout the combustion chamber. If the temperature of unburned mixture exceeds SIT of the fuel, then spontaneous ignition or uncontrolled auto-ignition occurs at various location in the combustion chamber. This phenomenon is called "knock" or "detonation" in SI engines.

Improving (ON)

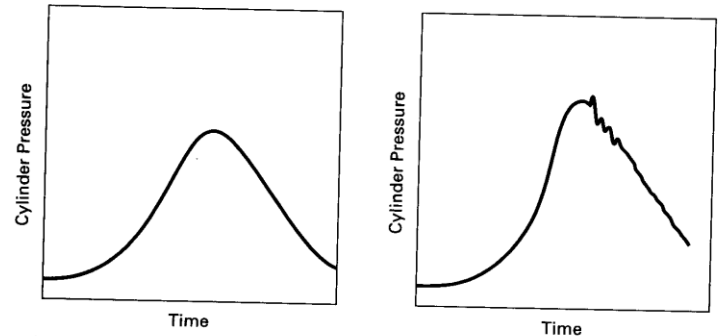
An antiknock agent is a gasoline additive used to reduce engine knocking and increase the fuel's octane rating. The fuel mixture (gasoline), when used in high compression internal combustion engines, has a tendency to ignite early (pre-ignition or detonation) causing a damaging "engine knocking" (also called "pinging" or "pinking") noise. The typical octane booster or antiknock agents in use are: TEL – Tetra – ethyl lead (phased out); MMT –Methylcyclopentadienyl manganese tricarbonyl; MTBE –

Methyl-tert-butyl ether; ETBE – ethyl-tert-butyl ether; Ferrocene; Iron pentacarbonyl; Toluene; Isooctane.

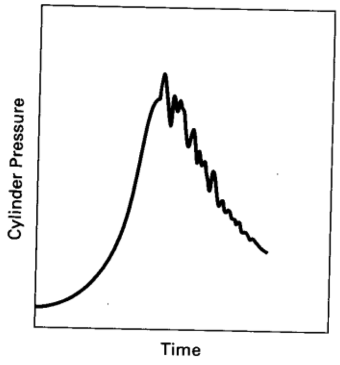
Tetraethyllead (TEL) it's chemical formula is [pb$(C_2H_5)_4$]. It is knock inhibitor which increases the gasoline fuel knock resistance when small quantity is added to the fuel [(0.5 – 1.5 ml) per liter gasoline]. It raises the ON $>$ 100.

Cylinder pressure as a function of time in a typical SI engine combustion chamber showing:

a- normal combustion
b- combustion with high knock
c- combustion with heavy knock



(a) Normal Combustion with No Knock (b) Combustion with Light Knock



(c) Combustion with Heavy Knock

Diesel fuel

Diesel fuel is also known as diesel oil or fuel oil. It is petroleum fraction that lies between kerosene and lubricating oil. Two types usually used in IC engines:

1- Light diesel fuel $\mathbf{C_{12}H_{22}}$ of $\mathbf{MWt = 170}$ less viscous, easy to pump, and cost more.
2- Heavy diesel fuel $\mathbf{C_{14}H_{24}}$ of $\mathbf{MWt = 200}$.

In order to burn well, diesel fuel should:

1. Atomize properly;
2. Ignite quickly;
3. Burns completely.

Diesel fuel properties that determine the ignition quality are:

1- Specific gravity:

This is related to the energy content of the fuel and greatly affects the (specific & volumetric) fuel consumption of a diesel engine.

2- Volatility:

It is the tendency to evaporate under given set of conditions. Evaporation and mixing with air is essential for good ignition quality and complete combustion, as this will effects the exhaust contents (HCs, NOx, smoke density, odour). It is measured by "distillation temp. vs % evaporated" according to ASTM distillation curve.

3. Ignition quality:

i- Smooth spontaneous ignition at low temperature is essential for good quality of ignition.
ii- Ignition delay (ID) which is the time period between start of injection and start of combustion.

Influence the quality of fuel optimization. Too long (ID) causes rapid rate of pressure rise at start of ignition which causes diesel knock. Too short provides insufficient time for good mixing which causes smoking. Effected by design parameters such as "CR, injection rate & timing, inlet air temperature".

4. Cetane number (CN):
   - It is a measure of the fuel ability to auto-ignite quickly when injected into the hot compressed air.
   - It is a numerical measure of the influence the diesel fuel has in determining the (ID).

$\text{CN}\uparrow\ \rightarrow\ \text{diesel knock}\downarrow;\ \text{exh. emission}\downarrow;\ \text{noise}\downarrow;\ \text{delay period}\downarrow.$

$\text{CN}\uparrow\uparrow\ \rightarrow\ \text{pre}-\text{ignition occurs.}$

Measuring CN :

A special test engine is used for the purpose of measuring (CN) of fuel (x).

- The test procedure :
  i- Fuel injection begins at (13 $^{\circ}bTDC$)
  ii-(CR) to be adjusted for ignition to start at TDC
- Reference fuel
  1- n-cetane [$C_{16}H_{34}$]
  CN = 100
  2- heptomethylnonane HMN [$C_{12}H_{34}$]
  CN = 15

Ref. fuel mixture variation according to test procedure until ignition start at TDC. Accordingly, the mixture blend is used to calculate (CN) of fuel (x) as:

$$CN\,]_x = \%\,n-cetane + (\,0.15\,)\%\,HMN$$

Methods of diesel fuel rating:

1. Cetane index (CI)

$$\mathrm{CI} = -420.34 + 0.016\mathrm{G}^2 + 0.192\mathrm{G}\,(\log \mathrm{Tmp}) + 65.01(\log \mathrm{Tmp})^2 - 0.0001809\,\mathrm{Tmp}^2$$

Where

G – API gravity (American Petroleum Institute) {APIg}

$$\mathrm{G} = \left(\frac{141.5}{\mathrm{sp.grv}}\right) - 131.5$$

sp.grv :- specific gravity

sp.grv:- [gravity of fuel / gravity of water] at 15 ℃

Tmp:- mid point boiling temperature in (°F), which is the temperature at which (50 %) of mixture will be evaporated.

2. Diesel index (DI):

$$(DI = 0.01\,[aniline\ point\ (°\mathrm{F}) \times \mathrm{APIg\ at\ 60°F}]$$

Aniline point :– is the lowest temperature at which the oil is completely miscible with an equal volume of aniline. [good quality diesel oil → APIg = 70 °F].

Differences between ON & CN

1. For gasoline, (ON) is a measure of the resistance to spontaneous ignition.
2. For diesel fuel, (CN) is a measure of the ease of spontaneous ignition.
3. (ON) & (CN) are therefore the inverse measurement of the same property (spontaneous ignition).
4. Approximate simple relationship between ON & CN:

$$CN = (104 - ON)\,/\,2.75$$

$$ON = 104 - 2.75\,CN$$

5. This means that "a good diesel engine fuel is bed gasoline engine fuel"
6. Increasing (ON) is not harmful but a cost waste. Increasing (CN) is harmful as it might produce pre-ignition.

Liquefied Petroleum Gas (L.P.G)

LPG is a general term for hydrocarbons that are gases at ordinary atmospheric temperature and pressure but can be readily liquefied by compressing or cooling. By liquefaction, the volume of LPG reduces to 1/250 of the gaseous phase, making it convent to store and handle. LPG has a simple composition as it consists essentially of propane and butane which have carbon number of 3-4.

LPG is used in the spark ignition engines as a substitute for motor gasoline.

The fuel system of LPG engine is partly different from that of gasoline engine. It consists of a fuel cylinder, a solenoid valve a regulator (vaporizer), and carburetor. The solenoid valve opens only when the engine ignition switch is turned on to allow LPG to be supplied from the cylinder to the regulator wherein it is heated in air or hot water and vaporize. Completely vaporized LPG is then mixed uniformly with air in the carburetor. After that the mixture is induced to the combustion chamber.

Thus LPG combustion implementation occurs with a nearly uniform fuel-air mixture that reduces deposit, such as soot, on the combustion chamber walls. Further LPG does not cause fuel dilution of engine oil.

The fuel consumption of LPG is found to be 10% less than that of gasoline. The reason for this is that LPG generates the maximum engine torque at a higher air-fuel ratio, a leaner mixture condition which is a result of LPG air-fuel mixture being more uniformly distributed in the combustion chamber.

The LPG fuel has a comparative disadvantage relative to gasoline. The power output is less. In case of gasoline engine, the evaporation of the fuel caused the fuel-air mixture to cool and increase its density, while in LPG engines the fuel is preheated and vaporized in a heat exchange-pressure regulator before it mixed with air in the carburetor, resulting in a relatively high mixture temperature and a low mixture density. Consequently, for LPG the weight of fuel supplied to the engine is less than gasoline. further, its calorific value (885 kcal/m^3) is less than gasoline (928 kcal/m^3). Low power output is the result.

The calorific value by weight for LPG is 10750-11080 kcal/kg, while for gasoline is 10500 kcal/kg.

Alternative fuels

The importance of reduce pollution, scarcity of existence of petroleum in some countries, fears of fuel shortage, energy crises, and cost increase of the petroleum fuels have generated intense interest in alternative fuels, especially non-petroleum fuels, for engines.

Types of alternative fuels:

1. Alcohol
2. Methanol (CH_3OH)
3. Ethanol (C_2H_5OH)
4. Hydrogen
5. Natural gas – methane
6. Propane
7. Reformulated gasoline
8. Coal – water slurry
9. Other

- Why alternative fuel needed
    - *i-* Crude oil is vanishing sources.
    - *ii-* Emission problems from HCs
    - *iii-* Increasing crude oil transport cost
- Why alternative fuels still out of use
    - *i.* Engine modification
    - *ii.* High cost
    - *iii.* Service stations construction

Methanol CH_3OH:

Methanol is a good alternate fuel for internal combustion and other engines, either in combustion with gasoline or directly ("pure"). Mixture of methanol and gasoline is usually designated by (MX), for example:

M15 → 15% methanol + 85% gasoline
M100 → pure methanol
M 0 → pure gasoline

Ethanol C_2H_5OH :

It is also as methanol a good alternative fuel for automobiles. Made much of it from corn and sugar, although the manufacturing processes are costly. Most common mixture used are :

1. M 85
2. M 10

Combustion

The term combustion refers to the fairly rapid reaction, usually accompanied by a flame, which accurse between a fuel and an oxygen carrier such as air. The molecules of fuel and air have a certain amount of energy stored in the bonds between their constituent atoms. In the new molecules formed by a reaction, this (chemical) energy is at a lower level and the energy release can be transferred to the surrounding, in the form of heat. Combustion is there for said to be an (exothermic) reaction.

Fuels: - Most common fuels consist mainly of hydrogen and carbon whether the fuel is solid (e.g. coal), liquid (e.g. petroleum) or gaseous (e.g. natural gas). For solid and liquid fuels the analysis is usually quoted as a percentage by mass of each chemical element in the fuel. The analysis of a gaseous fuel is usually given in terms of the percentage by volume of each gas.

Chemical Equations of Combustion

In any purely chemical reaction the process is simply based on the arrangement of atoms to form a new molecule, and the total number of atoms of each element is unchanged. A chemical equation expresses the principle of the conservation of mass in terms of the conservation of atoms.

* **Consider the reaction**

$$C + O_2 \rightarrow CO_2$$
$$12kg\ C + 32\ kgO_2 \rightarrow 44\ kg\ CO_2$$
$$1kmol\ C + 1\ kmol\ O_2 \rightarrow 1\ kmol\ CO_2$$
$$1\ vol\ C + 1\ vol\ O_2 \rightarrow 1\ vol\ CO_2$$

$$2H_2 + O_2 \rightarrow 2H_2O$$
$$4kg\ H_2 + 32kg\ O_2 \rightarrow 36\ kg\ H_2O$$
$$2\ kmol\ H_2 + 1\ kmol\ O_2 \rightarrow 2\ kmol\ H_2O$$
$$2\ vol\ H_2 + 1vol\ O_2 \rightarrow 2\ vol\ H_2O$$

If the gases are considered ideal, then according to Avogadro's hypothesis:"All gases contain the same number of molecules per unit volume at the same pressure and temperature". This implies that:

1 kmol of any gaseous substance occupies the same volume. (22.4 m^3 at STD. conditions)

For example: 44 kg of CO_2, 32 kg of O_2, and 16kg of CH_4 will occupy a volume of 22.4 m^3 at STD.conditions.

It should be clearly understood that in any reaction the mass is conserved but the number of moles or volume may or may not be conserved, for example:

$$2CO + O_2 \rightarrow 2CO_2$$
$$2vol. + 1vol. \rightarrow 2\ vol..$$
$$56kg + 32kg \rightarrow 88\ kg$$

Combustion Stoichiometry: -

Stoichiometry means complete combustion of chemically correct mixture. The principle reaction of combustion for any fuel is

$$C + O_2 \dashrightarrow CO_2 \quad Called\ Stoichiometric\ equation\ of\ combustion$$
$$C + 0.5\, O_2 \rightarrow CO$$
$$CO + 0.5\, O_2 \rightarrow CO_2$$
$$H_2 + 0.5\, O_2 \rightarrow H_2O$$
$$S + O_2 \rightarrow SO_2$$
$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

Since the most common oxidizer is air which is a mixture of 21% O_2 & 79% N_2 (on volume or mole bases) i.e. 1 kmol of O_2 is accompanied by 79/21 = 3.76 kmol of N_2

.: the above equation become: -

$$C + O_2 + 3.76\, N_2 \rightarrow CO_2 + 3.76\, N_2$$
$$\dots\dots\dots..and\ so\ other\ equations$$

The minimum amount of air required for complete combustion of a fuel is known as **(theoretical air).**

The fuel rich mixture, or mixture with less than Stoichiometric air give incomplete combustion, that result m some quantity of undesirable carbon monoxide (CO) in the exhaust gases and also some loss of heat energy.

*** Theoretical air required for complete combustion.**

If the fuel composition is known the requirement of O_2 or air can be calculated as follows: -

1- On mass bases.

$$12\ kg\ of\ C\ require\ 32\ kg\ of\ O_2\ to\ form\ 44\ kg\ of\ CO_2\ or:$$
$$1kg\ C\ +\ 32/12\ kg\ O_2 \rightarrow 44/12\ kgCO_2\ or$$
$$1kg\ C\ +\ 8/3\ kg\ O_2 \rightarrow 11/3\ kg\ CO_2 \quad Similarly:$$
$$4kg\ 2\ H_2\ +\ 32\ kgO_2 \rightarrow 36kg\ H_2O\quad or$$
$$1kg\ H_2\ +\ 8\ kg\ O_2 \rightarrow 9kg\ H_2O$$

<u>In general</u>

$$1kg\ CmHn\ +\ [[\,(m+n/4)\ \times 32]\,/(12m+n)\,]\ kgO_2 \rightarrow [44m/(12m+n)\,]\ kg\ CO_2\ +\ [\,9n\ /\ (12m+n)\,]\ kgH_2O$$

2- On mole bases:-

$$1kmol\ C + 1kmol\ O_2 \rightarrow\ 1kmol\ CO_2$$
$$1kmol\ H_2\ +\ 0.5\ kmol\ O_2 \rightarrow 1kmol\ H_2O$$

<u>In general</u>

$$1\ kmol\ CmHn\ + [\,(m+n/4)]\ kmol\ O_2\ +\ 3.76[\,(m+n/4)\,]\ kmol\ N_2 \rightarrow m\ kmol\ CO_2\ +\ [n/2]\ \ kmol\ H_2O\ +\ 3.76[\,(\,m+n/4)\,]\ \ kmol\ N_2\,.$$

Note: - The theoretical amount of air required for complete combustion can be calculated using the following formula.

$$(m_{\,theo.})_{\,air}\ =\ [(4.31\ *32)/\ M_f]\ *\ [(m+(n/4)\,]\ \ kg\ air/kg\ of\ fuel.$$

Where M_f is the molecular weight of fuel, m is the number of moles of carbon, and n is the number of mole of hydrogen.

Notes: - To find the Stoichiometric amount of air for complete combustion of a fuel: -

<u>1-</u> If the analysis of a fuel is given by mass, then

a- Determine the **mass** of oxygen required for each constituent, from this; find the total mass of oxygen by adding all the separate masses required.
b- Subtract any oxygen which may be in the fuel since this does not have to be supplied.
c- Stoichiometric mass of air = O_2 required / 0.233 kg

2- If the analysis of a fuel is given by volume or mole, then

a- Determine the **volume** of oxygen required for each constituent, from this; find the total volume of oxygen by adding all the separate volumes required.
b- Subtract any oxygen which may be in the fuel since this does not have to be supplied.
c- Stoichiometric volume of air = O_2 required / 0.21 mole or vol.

Non – Stoichiometric Mixture: -

The term weak and rich are used where, respectively, O_2 & fuel are available in excess of their Stoichiometric proportions. It is possible to have complete combustion to CO_2 & H_2O with a weak mixture the excess O_2 appearing in the products side of chemical equation, it is however, impossible to have complete combustion of a rich mixture.

An example of incomplete combustion is:

$$C_8H_{18} + A\,(O_2 + 3.76\,N_2) \rightarrow a\,CO_2 + b\,CO + d\,H_2O + e\,H_2 + f\,N_2$$

a) The percentage of excess air:

$$Px = (actual\ A/F - Stoichiometric\ A/F) / (Stoichiometric\ A/F)$$

b) Equivalence ratio (Ø): is the ratio of the theoretical air –fuel ratio (A/F) $_{theo.}$ to the actual air-fuel ratio (A/F) $_{act.}$

$$Ø = (A/F)_{theo.} / (A/F)_{act.}$$

Sometimes this ratio is given in term of F/A ratio; and sometimes called mixture strength.

Combustion Analysis: -

One of the primary objects in combustion analysis is the determination of the amount of air required to burn a fuel and to determine the amount of the products of combustion which will have been formed. Some analysis made by mass, some made by volume, whiles others admit of analysis both by mass and by volume. An analysis which includes the steam in the exhaust is called a wet analysis and that without steam is called dry analysis. The following examples will give an indication of the above cases.

Example(1) :- A fuel consists of 72% carbon , 20% hydrogen and 8% oxygen by mass , determine the stoichiometric mass of air required to completely burn 1kg of the fuel

Solution:-

$$C + O_2 \rightarrow CO_2$$
$$[12kg\ C + 32kg\ O_2 \rightarrow 44kg\ CO_2]/12$$
$$1kg + 32/12\ kg \rightarrow 44/12\ kg$$
$$1kg\ C\ required\ 8/3\ O_2 \rightarrow .:\ 0.72\ kgC\ require$$
$$0.72 * 8/3 = 1.92\ kg\ O_2$$
$$H_2 + 0.5\ O_2 \rightarrow H_2O\ or\ 2H_2 + O_2 \rightarrow 2H_2O$$
$$[2kg\ H_2 + 16kg\ O_2 \rightarrow 18kg\ H_2O]/2$$
$$1kg\ H_2 + 8kg\ O_2 \rightarrow 9kg\ H_2O \rightarrow .:\ 0.2kg\ H_2\ need$$
$$0.2 * 8 = 1.6\ kg\ O_2$$

The total amount of O_2 required to complete combustion of C & H is then 1.92 + 1.6 = 3.52 kg of O_2 but there is 0.08 kg of O_2 available in the fuel so that this amount of O_2 must subtract from the total amount of air required. The amount of O_2 required becomes 3.52 – 0.08 = 3.44 kg O_2.

The amount of air by mass associated with this amount of oxygen is (3.44* 100)/23.3 = 14.8 kg air /kg of fuel

***** For the same example calculate A/F if the fuel contain only 0.80 carbon & 0.20 hydrogen

Example (2):- Determine the stoichiometric mass of air required to completely burn 1kg of heptane (C_7H_{16})

Solution:- The chemical equation is

$$C_7H_{16} + 11O_2 \rightarrow 7CO_2 + 8H_2O$$
$$(12*7+1*16) + 11*32 \rightarrow 7(12 + 16*2) + 8(2+16)$$
$$(100+352 \rightarrow 308 + 144)\ /\ 100$$
$$1 + 3.52 \rightarrow 3.08 + 1.44$$

.: 1 kg of C_7H_{16} requires 3.52 kg of O_2 which is associated with the amount of air equal to 7.52/0.233 = 15.17 kg air /kg fuel.

Example (3):- The analysis of a supply of coal gar is H_2 49.4% , CO 18% , CH_4 20% , C_4H_8 2% , O_2 0.4 % ,N_2 6.2% and CO_2 4% . Calculate the stoichiometric A/F ratio. Find also the wet and dry analysis of the products of combustion if the actual mixture is 20% weak.

Solution: - The example can be solved by a tabular method, for CH_4

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$
$$1mole\ CH_4 + 2mole\ O_2 \rightarrow 1mole\ CO_2 + 2mole\ H_2O$$

1mole of CH_4 require 2mole of O_2 for complete combustion

.: 0.2 mole of CH_4 require

0.2 * 2 = 0.4 mole of oxygen. The equation become →

$$0.2\ CH_4 + (0.2*2)\ O_2 \rightarrow 0.2\ CO_2 + (0.2*2)\ H_2O$$

and so on for other components

Composition	Percentage	Equation	O2 moles/mole of fuel	CO2	H2O
H_2	0.494	$2H_2 + O_2 \rightarrow 2H_2O$	0.247	---	0.494
CO	0.18	$2CO + O_2 \rightarrow 2CO_2$	0.09	0.18	---
CH_4	0.2	$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$	0.4	0.2	0.4
C_4H_8	0.02	$C_4H_8 + 6O_2 \rightarrow 4CO_2 + 4H_2O$	0.12	0.08	0.08
O_2	0.004	----	-0.004	---	---
N_2	0.062	-----	---	---	---
CO_2	0.04	-----	---	0.04	---

Total 0.853 0.5 0.974

.: The amount of air required is 0.853/0.21 = 4.06 mole air / mole of fuel

For a mixture which is 20% weak the A/F is

Actual A/F ratio = 4.06 + 20/100 * 4.06 = 4.872/1

The amount of nitrogen associated with this amount of air is

4.872 * 0.79 = 3.85 moles/ mole fuel
Excess oxygen = 4.872 * 0.21 – 0.853 = 0.171 mole oxygen / mole fuel
The total N_2 in products is 3.85 + 0.062 = 3.912 mole nitrogen / mole of fuel
.: The analysis by volume of the wet and dry products is:

Products	mole/mole of fuel	% by vol . (dry)	% by vol.(wet)
CO_2	0.5	10.9	9.0
H_2O	0.974	----	17.5
O_2	0.171	3.72	3.08
N_2	3.912	85.4	70.4
	Total wet 5.557- H_2O 0.974 **Total dry 4.583**	100	100

The Orsat apparatus: -

The orsat apparatus is a device commonly used in the laboratory to determine the volumetric composition of products of combustion. It consists essentially of three absorption vessels filled with different reagents. These are: -

a- Potassium hydroxide (KOH) for the absorption of carbon dioxide (CO_2).
b- Pyrogallic acid in potassium hydroxide (or caustic soda) for the absorption of oxygen (O_2).
c- Cuprous chloride solution for the absorption of carbon monoxide (CO).

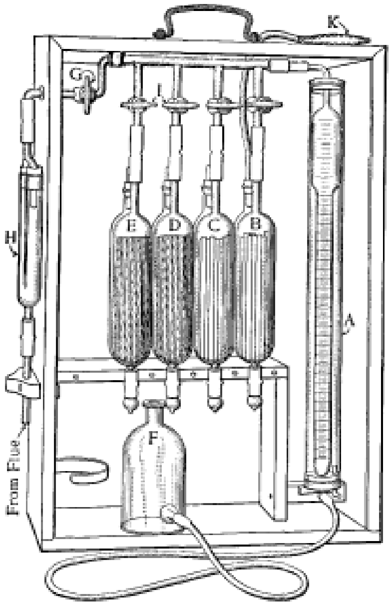
The gases must be absorbed in the above order, since the pyrogallic acid solution will absorb CO_2 as well as O_2. Also cuprous chloride will absorb CO_2. O_2 and CO.

The sample of exhaust gas is drawn into measuring burette by lowering the leveling bottle containing water. This known volume of sample at atmospheric temperature and pressure is forced in succession into each of the reagent bottles which contain the absorbents.

The volume is measured after each absorption process by returning the sample to the burette and bringing it to atmospheric pressure with the aid of the leveling bottle.

The change in volume after a particular constituents has been absorbed is then the partial volume of that constituents in the original sample. The gas which remains is assumed to be nitrogen.

The apparatus gives an analysis of the dry products of combustion in order to ensure this, a U-tube containing calcium chloride or some other drying agent is sometimes fitted in the beginning of the sample intake tube.



Example (4):-

An orsat analysis of the dry exhaust from an internal combustion engine gave 12% CO_2, 2% CO, 4% CH_4, 1% H_2, 4.5 % O_2 and 76.5% N_2. Calculate the proportions by mass of carbon to hydrogen in the fuel.

Solution:-

Mass of N_2 in dry exhaust gas = 0.765 * 28 = 21.42 kg/mole of dry exhaust gases

Mass of O_2 associated with this amount of nitrogen is (0.233*21.42) / 0.767 = 6.5 kg/mole

Mass of O_2 accounted for in the dry exhaust gas = 32 [0.32 + (0.02/2) + 0.045] = 5.6 kg / mole

.: Mass of O_2 burned to H_2O = 6.5 – 5.6 = 0.9 kg/mole

$$H_2 + 0.5O_2 \rightarrow H_2O$$

$$[2kg + 16kg \rightarrow 18kg\ H_2O] / 2$$

$$1kg + 8 \rightarrow 9kg\ H_2O$$

.: Mass of H_2 burned to H_2O = 0.9/8 = 0.1125 kg / mole

Mass of H_2 accounted for in the dry exhaust gas = 1 [(0.04*4) + (0.01 *2)] = 0.18 kg / mole

.: Mass of H_2 in the fuel 0.18 + 0.1125 = 0.2925 kg / mole

Mass of carbon in the fuel = 12[0.12+0.02+0.04] = 2.16 kg / mole

.: Ratio of C to H_2 in the final = 2.16 / 0.292 = 7.38 /1

Flame properties

Formation of Flame

If the combustible substances produce vapor during burning process, a flame is produced. Flame is a luminous zone of the rapid exothermic reaction in combustion of vapor with the formation of light and heat energy. A non-luminous region is appeared just after the flame where the temperature is slightly reduced. A flame is bounded between the ignition zone and a non-luminous gaseous zone. The combustion of gaseous fuels in a flame need the contact of fuels with an oxidant, either oxygen or air prior to the reaction. The ranges of flammability and the point at which the mixture spontaneously ignites must be known. They must be heated to the combustion temperature and the flame produced will be at a high temperature. Then the reaction take place in within a narrow zone or region in the flame. This combustion zone is called the flame front.

Types of Flames

Flames may be of different types depending on the extent of mixing of fuel and oxidizer or how the mixture reach the reaction zone. The flow patterns in the reaction vessel, such as well mixed and plug flows are the major tools to classify the flames in different types. The flame may be turbulent and laminar types depending on the flow behavior of the combustion gases.

In a premixed flame the fuel and the oxidant are molecularly mixed before the combustion process takes place.

The flames are mainly classified as:

i) Non -Premixed or Diffusion Flames and ii) Premixed flames. These flames may be both laminar or turbulent types.

Non-Premixed Flames

In many combustion processes, the fuel and air are often initially not mixed. The fuel and oxidizer are kept on either side of the reaction zone and moved to the reaction zone. The resultant flame is termed as the diffusion or non-premixed flame. In some cases, the gas and air are injected in a coaxial parallel tubes and ignited. The flow behavior of the non-premixed flame is laminar type. Molecular or turbulent diffusion is responsible for the mixing of the gases in non-premixed flames.

In a laminar flow region, the reach the reaction front by diffusion, before the reaction takes place. The products of combustion also diffuse to come out from the reaction zone. The flame is also called Laminar Non-Premixed Flames or Diffusion Flame. The diffusion flame occurs at the interface of the gaseous fuel and air. With the progress of time as the flame propagates, the thickness of the reaction zone increases. The mechanism of this flame is very complicated. The fuel approaches gradually towards the flame zone, there is a deficiency of oxygen and it is pyrolysed to smaller molecules or radicals. Then, there will be the formation of carbon soot and fuel burns with a bright luminous yellow flame. As the flame propagates, the oxygen concentration in the zone increases and the product of pyrolyzed products are further reacted. Gradually the reaction occurs under the stoichiometric proportion of oxygen and combustible. The non-premixed flames at stoichiometric conditions optimize the flame temperature with a definite fuel-air ratio.

Premixed Flames

In this type, the fuel and oxidant gas are mixed together at ambient condition before being delivered to the flame zone. A reaction takes place before reaching the flame front. Gradually the mixture is sufficiently heated at the reaction front and the chemical reaction takes place. If the fuel and the oxidant gas are thoroughly mixed prior to reaching the flame front, the location of the flame front does not depend on the diffusion of reactants. The flame speed or the velocity of the reactants in the reaction zone is important. In another case, the fuel is injected into the oxidant gas flow in the upstream side of the flame front but the mixing is improper.

The turbulent pre-mixed flame plays an important role in the various practical applications because, it increases the ignition of fuel with the reduction in the emission of gases. Both the laminar premixed or laminar non-premixed flames (diffusion flame) are slow processes and not economic. Turbulence actually results in a reduction in flame length. The turbulence can be increased by recirculation of the fuel-air mixture.

Flame Structure

The example of the laminar premixed flame is the flame in a Bunsen Burner as shown in Fig.1. In the premixed type, the laminar flame is the most simple type. The structure of the flame may be analyzed by a flame in the burner. The flame consists of four distinct regions as shown in Fig 1.

1. Zone containing unburnt gases,
2. Reaction zone,
3. Incomplete combustion zone, and
4. Complete combustion zone.

The idealized shape of the reaction zone of a laminar premixed flame is a cone. The height of the cone represents the flame length, and depends on the velocity at the burner outlet.

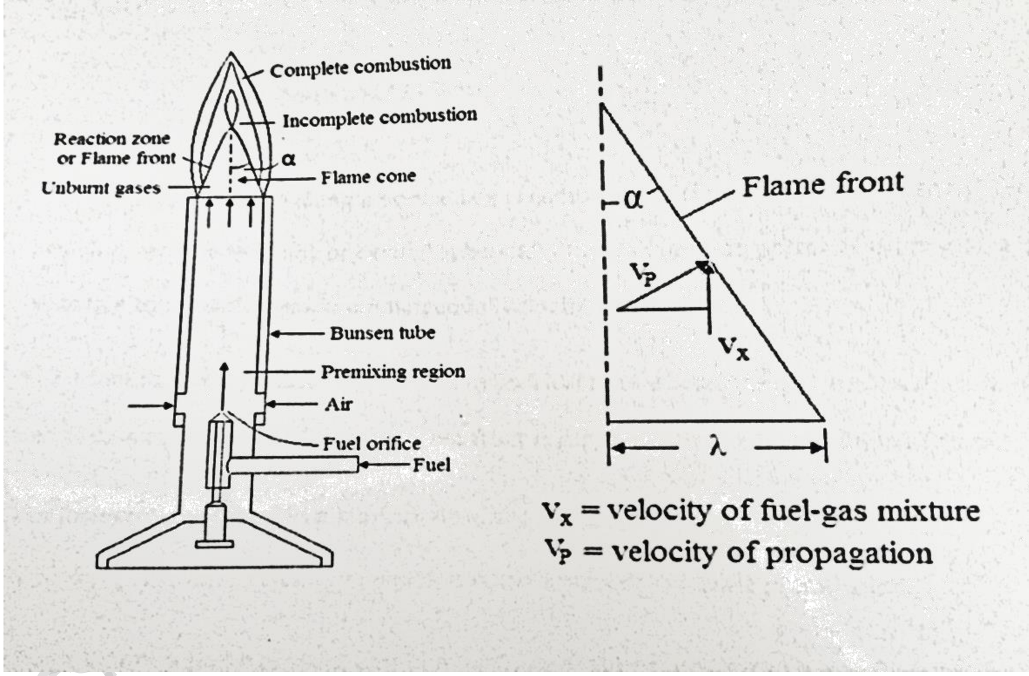


Fig.1 Laminar premixed flame in a Bunsen Burner.

Flame Propagation

Since the flame front is stationary, the velocity of flame propagation V_p , with respect to the unburnt mixture will be equal to the flow velocity of the unburnt mixture normal to the flame front. This is also called burning velocity.

$$\text{Velocity of flame propagation, } V_p = V_x \sin\alpha$$

Where, α = flame cone angle and V_x is the velocity air fuel mixture.

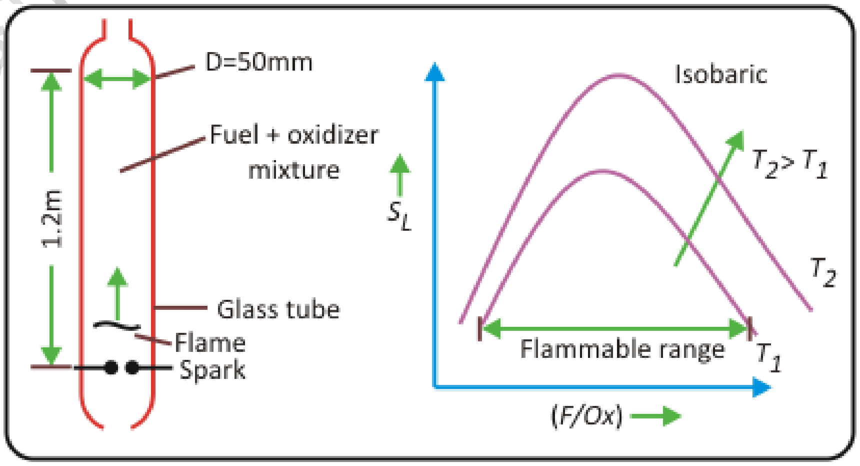
Velocity of flame propagation depends on the properties of the fuel-air mixture, pressure and temperature of the process. A premixed flame of a particular fuel-air combination is characterized by three main parameters, the burning velocity, flame temperature and flammability limit, which are also determined by the pressure, temperature and, air-flue ratio. Dilution of the burning mixture with an inert gas, such as helium or nitrogen, lowers the temperature and, consequently, the reaction rate. Higher amounts of inert gas extinguish the flame.

Flammability limit

A spontaneous or explosive reaction of fuel may be possible when ignited in presence of either air or oxygen for a particular range of gas composition at fixed temperature and pressure. Flammability limits describe the range of fuel concentration in terms of volume % and this reaction may occur even when the mixture is cold. For example, the lower flammability limit for aviation kerosene is about 0.7% by volume and the higher flammability limit is about 5% by volume. For hydrogen gas, lower limit 4%, and higher is 75%.

Flammability limits also are much wider in oxygen than in air. So it highly depends on the burning atmosphere, pressure, temperature, and on the ignition source.

Instrument to determine flammability limit: Vertical glass tube of 1.2 m length and 50 mm ID.



Flammability limit is determined by upward propagation of flame from ignition source. Mixture is said to be flammable only when it travels above half of the tube. Large diameter is preferred as it provides consistent results and is free from quenching. Direction of flame propagation affects the flammability limit. Flammability limit varies linearly with temperature.

Flammability limits are the minimum (lower) and the maximum (upper) contents of fuel in the fuel/air mixture, between which flame propagation in this mixture is possible (the mixture is flammable). Flammability limits mean the same as ignition limits or explosion limits.

The flammability limit of the combustible mixture can be calculated from the Chatelier formula knowing the explosion limits of the components of the mixture:

$$L = \frac{100}{\frac{a_1}{L_1} + \frac{a_2}{L_2} + \frac{a_3}{L_3} + \cdots \ldots + \frac{a_n}{L_n}}$$

where: L – lower or upper limit of ignition, %;

a_i – concentration of a single, combustible component of the mixture ($\sum a_i = 100\%$).

Table: Flammability limits of gases

Gas	Flammability limits			
	% gas in air		% gas in oxygen	
	lower	upper	lower	upper
hydrogen H_2	4,1	74,2	4,0	94,0
carbon monoxide CO	12,5	74,2	15,5	94,0
methane CH_4	5,0	15,0	5,1	61,0
ethane C_2H_6	3,2	12,5	3,0	66,0
propane C_3H_8	2,4	9,5	2,3	55,0
n-buthane C_4H_{10}	1,9	8,4	1,8	48,0
n-pentane C_5H_{12}	1,4	7,8	–	–
n-hexane C_6H_{14}	1,25	6,9	–	–
ethylene C_4H_{10}	3,75	29,6	2,9	79,9
acetylene C_2H_2	2,5	80,0	2,5	89,4
benzene C_6H_6	1,41	6,75	2,6	30,0
methanol CH_3OH	6,72	36,5	–	–
ethanol C_2H_6OH	3,28	18,95	–	–

Ex.1. Calculate the lower and upper flammability limit of a gas of a given composition: CO=28%, H_2=4%, CH_4=1%, CO2=8%.

Sol.

- the lower flammability limit:

$$L=\frac{100}{\frac{a_1}{L_1}+\frac{a_2}{L_2}+\frac{a_3}{L_3}+\cdots\ldots+\frac{a_n}{L_n}}=\frac{100}{\frac{28}{12.5}+\frac{4}{5}+\frac{1}{5}}=10.2\%$$

- the upper flammability limit:

$$L=\frac{100}{\frac{a_1}{L_1}+\frac{a_2}{L_2}+\frac{a_3}{L_3}+\cdots\ldots+\frac{a_n}{L_n}}=\frac{100}{\frac{28}{74.2}+\frac{4}{74.2}+\frac{1}{15}}=66.3\%$$

Ex.2. The mixture of fuel containing 80% Methane; 15% of Ethane and 5% of CO. Find the lower and higher flammability limits.

Sol.

- the lower flammability limit:

$$L=\frac{100}{\frac{a_1}{L_1}+\frac{a_2}{L_2}+\frac{a_3}{L_3}+\cdots\ldots+\frac{a_n}{L_n}}=\frac{100}{\frac{80}{5}+\frac{15}{3.12}+\frac{5}{12.5}}=4.715\%$$

- the upper flammability limit:

$$L=\frac{100}{\frac{a_1}{L_1}+\frac{a_2}{L_2}+\frac{a_3}{L_3}+\cdots\ldots+\frac{a_n}{L_n}}=\frac{100}{\frac{80}{15}+\frac{15}{14.75}+\frac{5}{74.2}}=15.61\%$$

Combustion Burners

Types of Burners

The basic combustion theory and laws are applied for the design of burning equipments depending on the type of fuels, such as gaseous or liquid fuels. These burning equipments are called burners. There are different liquid fuel and gaseous fuel burners. There are some similarities in their design but it depends on the type and properties of fuel in addition to their type of applications. Although many designs are available, but they are common in one aspect of requiring proper mixing of air and fuel prior to their combustion. These oil or gas burners are used in furnaces.

Gas Burners

Gas burners are usually classified based on their operating gas pressure. They are operated both atmospheric and high-pressure conditions. The gases are supplied in different ways depending on the pressure. In low pressure burner, the gas pressure varies from 1 to 4 kPa. Whereas, high pressure burners the pressure varies from 7 to 70 kPa.

Oil Burners

The oldest burning process of fuel oil is the pot type burner in which the oil is first vaporized by applying heat. After vaporization, the oil vapors are mixed with excess air and then burned. As oil is vaporized at a slower rate than the rate of combustion reaction, the process is modified by using atomizer to form tiny droplets of oil before its ignition. The droplets can be vaporized easily. The atomization is carried out by either a mechanical device or applying a fuel jet. The nozzles are designed in such a way so that, it can be used to atomize oil at an elevated oil pressure at about 100 psi. The process can produce oil droplets in the range from 0.0002 inch to 0.010 inch. Quick ignition of relatively smaller droplets results in the rapid flame formation. Proper designing of nozzle is required to deliver droplets at a uniform rate for a particular burner. Oil is also atomized by using a rotating cup, rotating disc and swirling method. The examples of some oil burners are described in this section.

Swirl Oil Burner

The liquid oil is first pressurized at about 100-150 psi by a pump. In the swirl oil burner, the pressurized liquid fuel enters tangentially through the slots at a high velocity in the oil swirl chamber. It flows in the form of a vortex and escapes through a nozzle at the other end of the chamber. The sketch of a swirl type burner is shown in Fig.1. The centrifugal force is exerted on to the oil and it moves forward in the shape of a hollow tube. Air enters to the annular space and moves forward. Then the fine droplets emerge at the exit of the chamber in the form of a spray.

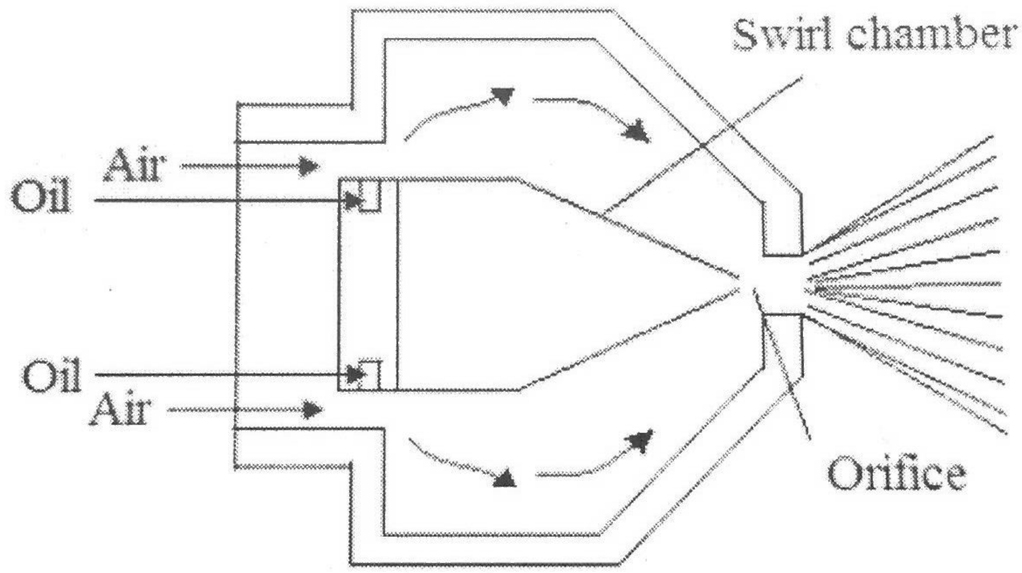


Fig. 1 Swirl oil burner

Rotary Cup Burner

The rotary cup burners are usually used in industry as well as for domestic purpose. A sketch of the rotary cup burner is shown in Fig.2. The capacity of fuel varies in wide range. The device consists of a rotating cup and a fan. Both the cup and the fan are rotated by an electric motor (not shown in figure) with a shaft. Oil is fed to the oil distributer to throw at the inner surface of the cup. The cup is rotated at a speed of about 3600 rpm. Then oil flows in a swirling motion and is thrown as fine droplets at the other end. The primary air is supplied by a fan or blower in a whirling motion opposite to the oil motion. This helps to further disintegrate the oil particles and finally to reach the combustion chamber.

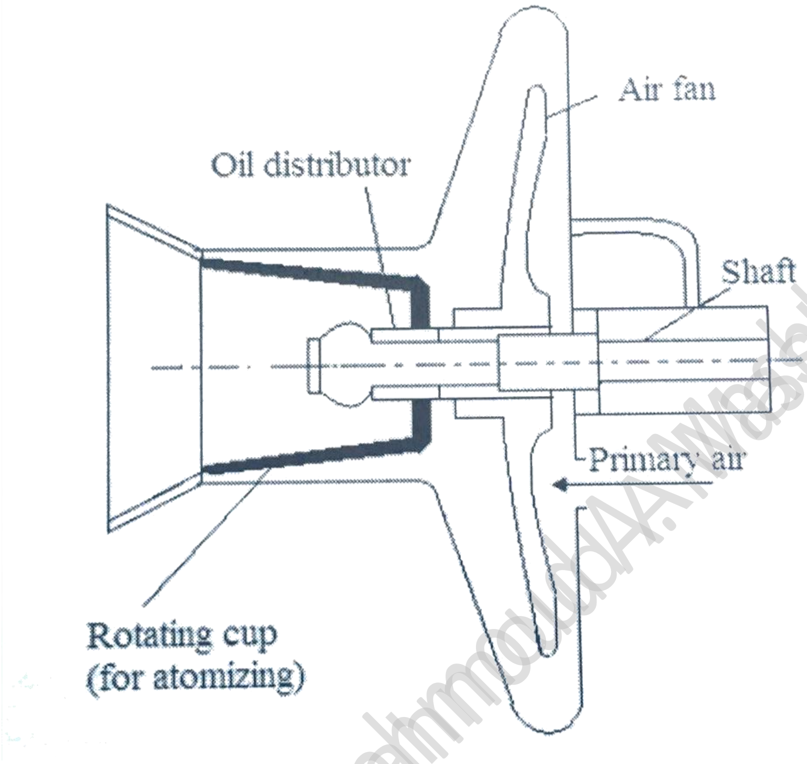


Fig.2 Rotary cup burner

Gas Burners

Gaseous fuel burners are classified based on the air and fuel mixing phenomena. They are either premixed type or diffusion type. If the fuel and gas are premixed before passing through the burner nozzle it is called premixed type. In diffusion type, a small quantity of air is mixed in the flow of gas and there will be diffusion between them. Industrial burners for gaseous fuel are diffusive type. In diffusion burner air and gaseous fuel are supplied separately in the furnace.

These burners are also used as for domestic purposes. The mixing rate between air and fuel controls the combustion process. The burners are also categorized based on the applied gas pressure. The gas burners may be either atmospheric or high pressure type. The construction of an atmospheric burner is described in Fig.3. The design and operation is similar to the Bunsen burner.

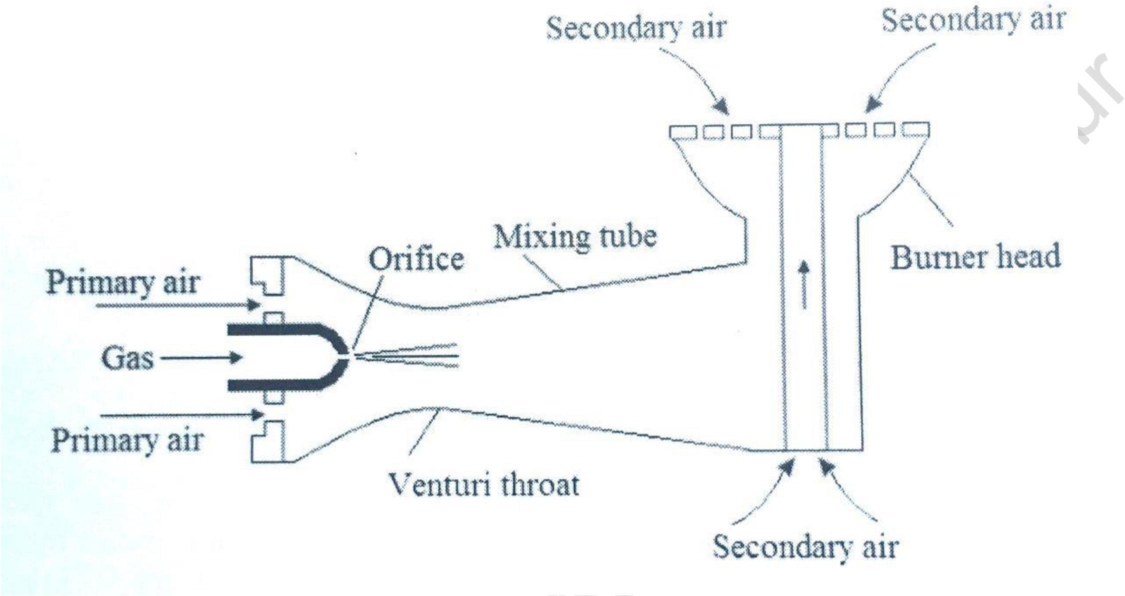


Fig.3 Atmospheric gas burner

Air for combustion is supplied in the furnace chamber. In these burners small portion of air is mixed with fuel as primary air and the rest amount, known as secondary air, is supplied above the burner port. Figure 3 shows an atmospheric gas burner where, the mass of air is about 10 times than that of fuel. Some fraction of total air is mixed with the fuel and this air supplied with the fuel is called the primary air. The secondary air is further added in the combustion point in the furnace. Mixing and combustion take place simultaneously. A free jet is produced in the downstream side of the burner during the discharge of fuel. The gas velocity at the orifice will be proportional to the square root of the differential pressure across the orifice. The venture tube is provided for better mixing. When a mixture of air and gaseous fuel passes through the nozzle of the burner, they mixed thoroughly in the divergent section of the venture. In this section pressure increases gradually as the velocity head is converted to the pressure head. The secondary air is supplied to the flame to complete combustion of the fuel.

Emissions and Air Pollution

This chapter explores the undesirable emissions generated in the combustion process of automobile and other IC engines. These emissions pollute the environment and contribute to global warming, acid rain, smog, odors, and respiratory and other health problems. The major causes of these emissions are non-stoichiometric combustion, dissociation of nitrogen, and impurities in the fuel and air. The emissions of concern are hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NOx), sulfur, and solid carbon particulates. Ideally, engines and fuels could be developed such that very few harmful emissions are generated, and these could be exhausted to the surroundings without a major impact on the environment. With present technology this is not possible, and aftertreatment of the exhaust gases to reduce emissions is very important. This consists mainly of the use of thermal or catalytic converters and particulate traps.

AIR POLLUTION

Until the middle of the 20th century the number of IC engines in the world was small enough that the pollution they emitted was tolerable, and the environment, with the help of sunlight, stayed relatively clean. As world population grew, power plants, factories, and an ever-increasing number of automobiles began to pollute the air to the extent that it was no longer acceptable. During the 1940s, air pollution as a problem was first recognized in the Los Angeles basin in California. Two causes of this were the large population density and the natural weather conditions of the area. The large population created many factories and power plants, as well as one of the largest automobile densities in the world. Smoke and other pollutants from the many factories and automobiles combined with fog that was common in this ocean area, and *smog* resulted. During the 1950s, the smog problem increased along with the increase in population density and automobile density. It was recognized that the automobile was one of the major contributors to the problem, and by the 1960s emission standards were beginning to be enforced in California. During the next decades, emission standards were adopted in the rest of the United States and in Europe and Japan. By making engines more fuel efficient, and with the use of exhaust aftertreatment, emissions per vehicle of HC, CO, and NOx were reduced by about 95% during the 1970s and 1980s. Lead, one of the major air pollutants, was phased out as a fuel additive during the 1980s. More fuel-efficient engines were developed, and by the 1990s the average automobile consumed less than half the fuel used in 1970. However, during this time the number of automobiles greatly increased, resulting in no overall decrease in fuel usage. Additional reduction will be difficult and costly. As world population grows, emission standards become more stringent out of necessity. The strictest laws are generally initiated in California, with the rest of the United States and world following. Although air pollution is a global problem, some regions of the world still have no emission standards or laws.

HYDROCARBONS (HC)

Exhaust gases leaving the combustion chamber of an SI engine contain up to 6000 ppm of hydrocarbon components, the equivalent of 1-1.5 % of the fuel. About 40% of this is unburned gasoline fuel components. The other 60% consists of partially reacted components that were not present in the original fuel. These consist of small non-equilibrium molecules which are formed when large fuel molecules break up (thermal cracking) during the combustion reaction.

The makeup of HC emissions will be different for each gasoline blend, depending on the original fuel components. Combustion chamber geometry and engine operating parameters also influence the HC component spectrum. When hydrocarbon emissions get into the atmosphere, they act as irritants and odorants; some are carcinogenic. All components except CH4 react with atmospheric gases to form photochemical smog.

Causes of HC Emissions

Nonstoichiometric Air-Fuel Ratio. Figure 1 shows that HC emission levels are a strong function of AF. With a fuel-rich mixture there is not enough oxygen to react with all the carbon, resulting in high levels of HC and CO in the exhaust products. This is particularly true in engine startup, when the air-fuel mixture is purposely made very rich. It is also true to a lesser extent during rapid acceleration under load. If AF is too lean poorer combustion occurs, again resulting in HC emissions. The extreme of poor combustion for a cycle is total misfire. This occurs more often as AF is made more lean. One misfire out of 1000 cycles gives exhaust emissions of 1 gm/kg of fuel used.

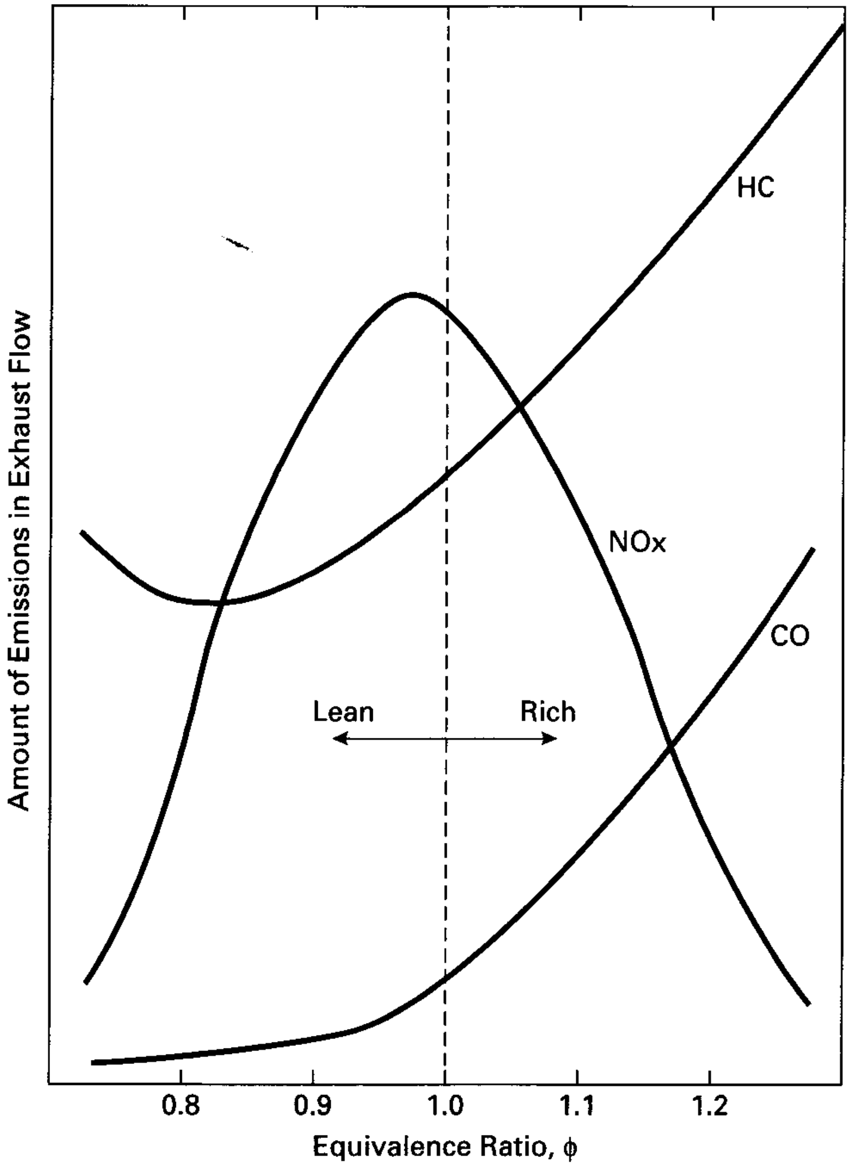


Figure 1 Emissions from an SI engine as a function of equivalence ratio. A fuel rich air-fuel ratio does not have enough oxygen to react with all the carbon and hydrogen, and both HC and CO emissions increase. HC emissions also increase at very lean mixtures due to poor combustion and misfires. The generation of nitrogen oxide emissions is a function of the combustion temperature, being greatest near stoichiometric conditions when temperatures are the highest. Peak NOx emissions occur at slightly lean conditions, where the combustion temperature is high and there is an excess of oxygen to react with the nitrogen.

Incomplete Combustion. Even when the fuel and air entering an engine are at the ideal stoichiometric mixture, perfect combustion does not occur and some HC ends up in the exhaust. There are several causes of this. Incomplete mixing of the air and fuel results in some fuel particles not finding oxygen to react with. Flame quenching at the walls leaves a small volume of unreacted air-and-fuel mixture. The thickness of this unburned layer is on the order of tenths of a mm. Some of this mixture, near the wall that does not originally get burned as the flame front passes, will burn later in the combustion process as additional mixing occurs due to swirl and turbulence.

Another cause of flame quenching is the expansion which occurs during combustion and power stroke. As the piston moves away from TDC, expansion of the gases lowers both temperature and pressure within the cylinder. This slows combustion and finally quenches the flame somewhere late in the power stroke. This leaves some fuel particles unreacted.

High exhaust residual causes poor combustion and a greater likelihood of expansion quenching. This is experienced at low load and idle conditions. High levels of EGR will also cause this.

It has been found that HC emissions can be reduced if a second spark plug is added to an engine combustion chamber. By starting combustion at two points, the flame travel distance and total reaction time are both reduced and less expansion quenching results.

Crevice Volumes. During the compression stroke and early part of the combustion process, air and fuel are compressed into the crevice volume of the combustion chamber at high pressure. As much as 3% of the fuel in the chamber can be forced into this crevice volume. Later in the cycle during the expansion stroke, pressure in the cylinder is reduced below crevice volume pressure, and reverse blowby occurs. Fuel and air flow back into the combustion chamber, where most of the mixture is consumed in the flame reaction. However, by the time the last elements of reverse blowby flow occur, flame reaction has been quenched and unreacted fuel particles remain in the exhaust. Location of the spark plug relative to the top compression ring gap will affect the amount of HC in engine exhaust, the ring gap being a large percent of crevice volume. The farther the spark plug is from the ring gap, the greater is the HC in the exhaust. This is because more fuel will be forced into the gap before the flame front passes.

Crevice volume around the piston rings is greatest when the engine is cold, due to the differences in thermal expansion of the various materials. Up to 80% of all HC emissions can come from this source.

Leak Past the Exhaust Valve. As pressure increases during compression and combustion, some air-fuel is forced into the crevice volume around the edges of the exhaust valve and between the valve and valve seat. A small amount even leaks past the valve into the exhaust manifold. When the exhaust valve opens, the air-fuel which is still in this crevice volume gets carried into the exhaust manifold, and there is a momentary peak in HC concentration at the start of blowdown.

Valve Overlap. During valve overlap, both the exhaust and intake valves are open, creating a path where air-fuel intake can flow directly into the exhaust. A well-designed engine minimizes this flow, but a small amount can occur. The worst condition for this is at idle and low speed, when real time of overlap is greatest.

Deposits on Combustion Chamber Walls. Gas particles, including those of fuel vapor, are absorbed by the deposits on the walls of the combustion chamber. The amount of absorption is a function of gas pressure, so the maximum occurs during compression and combustion. Later in the cycle, when the exhaust valve opens and cylinder pressure is reduced, absorption capacity of the deposit material is lowered and gas particles are desorbed back into the cylinder. These particles, including some HC, are then expelled from the cylinder during the exhaust stroke. This problem is greater in engines with higher compression ratios due to the higher pressure these engines generate. More gas absorption occurs as pressure goes up. Clean combustion chamber walls with minimum deposits will reduce HC emissions in the exhaust. Most gasoline blends include additives to reduce deposit buildup in engines.

Older engines will typically have a greater amount of wall deposit buildup and a corresponding increase of HC emissions. This is due both to age and to less swirl that was generally found in earlier engine design. High swirl helps to keep wall deposits to a minimum. When lead was eliminated as a gasoline additive, HC emissions from wall deposits became more severe. When leaded gasoline is burned the lead treats the metal wall surfaces, making them harder and less porous to gas absorption.

Oil on Combustion Chamber Walls. A very thin layer of oil is deposited on the cylinder walls of an engine to provide lubrication between them and the moving piston. During the intake and compression strokes, the incoming air and fuel comes in contact with this oil film. In much the same way as wall deposits, this oil film absorbs and desorbs gas particles, depending on gas pressure. During compression and combustion, when cylinder pressure is high, gas particles, including fuel vapor, are absorbed into the oil film. When pressure is later reduced during expansion and blowdown, the absorption capability of the oil is reduced and fuel particles are desorbed back into the cylinder. Some of this fuel ends up in the exhaust.

Propane is not soluble in oil, so in propane-fueled engines the absorption-desorption mechanism adds very little to HC emissions.

As an engine ages, the clearance between piston rings and cylinder walls becomes greater, and a thicker film of oil is left on the walls. Some of this oil film is scraped off the walls during the compression stroke and ends up being burned during combustion. Oil is a high-molecular-weight hydrocarbon compound that does not burn as readily as gasoline. Some of it ends up as HC emissions. This happens at a very slow rate with a new engine but increases with engine age and wear.

Oil consumption also increases as the piston rings and cylinder walls wear. In older engines, oil being burned in the combustion chamber is a major source of HC emissions. Figure 2 shows how HC emissions go up as oil consumption increases.

In addition to oil consumption going up as piston rings wear, blowby and reverse blowby also increase. The increase in HC emissions is therefore both from combustion of oil and from the added crevice volume flow.

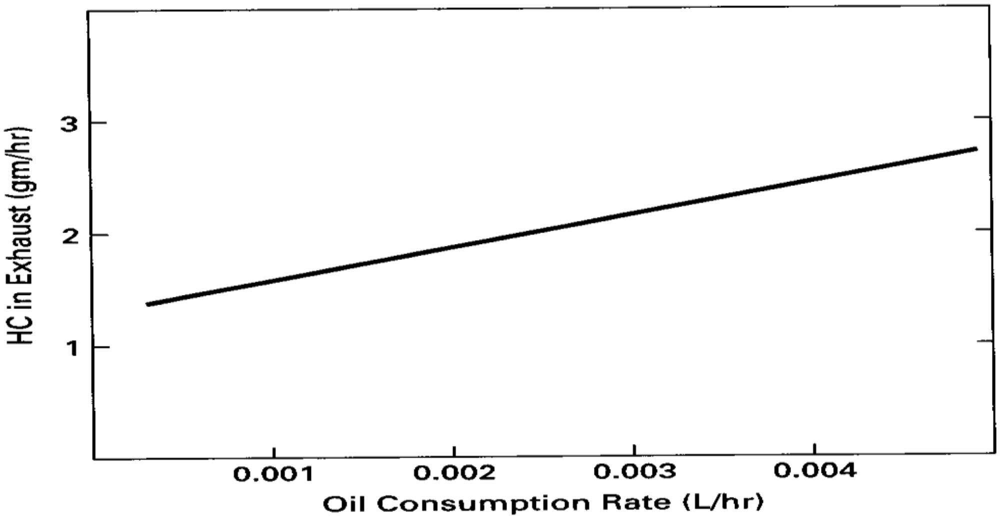


Figure 2 HC exhaust emissions as a function of engine oil consumption. Often as an engine, ages clearance between the pistons and cylinder walls increases due to wear. This increases oil consumption and contributes to an increase in HC emissions in three ways: There is added crevice volume, there is added absorption-desorption of fuel in the thicker oil film on cylinder walls, and there is more oil burned in the combustion process.

Two-Stroke Cycle Engines. Older two-stroke cycle SI engines and many modem small two-stroke cycle SI engines add HC emissions to the exhaust during the scavenging process. The air-fuel intake mixture is used to push exhaust residual out the open exhaust port. When this is done, some of the air and fuel mix with the exhaust gases and are expelled out of the cylinder before the exhaust port closes. This can be a major source of HC in the exhaust and is one of the major reasons why there have been no modem two-stroke cycle automobile engines. They could not pass anti-pollution requirements. Some experimental automobile two-stroke cycle engines and just about all small engines use crankcase compression, and this is a second source of hydrocarbon emissions. The crankcase area and pistons of these engines are lubricated by adding oil to the inlet air-fuel. The oil is vaporized with the fuel and lubricates the surfaces which come in contact with the air-fuel-oil mixture. Some of the oil vapor is carried into the combustion chamber and burned with the air-fuel mixture. Lubricating oil is mostly hydrocarbon components and acts like additional fuel. However, due to the high molecular weight of its components, oil does not fully combust as readily as fuel, and this adds to HC emissions in the exhaust.

Modem experimental two-stroke cycle automobile engines do not add fuel to the intake air, but scavenge the cylinders with pure air, avoiding putting HC into the exhaust. After the exhaust port closes, fuel is added by fuel injection directly into the cylinder. This creates a need for very fast and efficient vaporization and mixing of the air and fuel, but it eliminates a major source of HC emissions. Some automobile engines use superchargers instead of crankcase compression, and this eliminates HC pollution from that source.

Until recently most small engines, such as those used on lawn mowers and boats, were not regulated for pollution control. Many of these engines are still being manufactured with uncontrolled scavenging and oil vapor lubrication, contributing to serious HC (and other) pollution. This problem is starting to be addressed, and in some parts of the world (starting in California) emission laws and standards are starting to be applied to lawn mowers, boats, and other small engines. This will probably phase out, or at least greatly reduce, the number of small two-stroke cycle engines. Low cost is a major requirement for small engines, and fuel injection systems are much more costly than the very simple carburetors found on older engines. Many small engines now operate on a cleaner four-stroke cycle, but still use a less costly carburetor for fuel input.

In the early 1990s, there were an estimated 83 million lawn mowers in the United States producing as much air pollution as 3.5 million automobiles. Government studies of equipment using small engines give the following pollution comparison to that of automobiles (the numbers represent one hour of operation compared to miles traveled in an average automobile):

Riding mower-20 miles
Garden tiller-30 miles
Lawn mower-50 miles
String trimmer-70 miles
Chain saw-200 miles
Forklift-250 miles
Agricultural tractor-500 miles
Outboard motor-800 miles

CI Engines. Because they operate with an overall fuel-lean equivalence ratio, CI engines have only about one-fifth the HC emissions of an SI engine.

The components in diesel fuel have higher molecular weights on average than those in a gasoline blend, and this results in higher boiling and condensing temperatures. This allows some HC particles to condense onto the surface of the solid carbon soot that is generated during combustion.

Most of this is burned as mixing continues and the combustion process proceeds. Only a small percent of the original carbon soot that is formed is exhausted out of the cylinder. The HC components condensed on the surface of the carbon particles, in addition to the solid carbon particles themselves, contribute to the HC emissions of the engine.

In general, a CI engine has about a 98% combustion efficiency, with only about 2% of the HC fuel being emissions (Fig. a). Some local spots in the combustion chamber will be too lean to combust properly, and other spots will be too rich, with not enough oxygen to consume all the fuel. Less than total combustion can be caused by undermixing or overmixing. Unlike the homogeneous air-fuel mixture of an SI engine that essentially has one flame front, the air-fuel mixture in a CI engine is very much nonhomogeneous, with fuel still being added during combustion. Local spots range from very rich to very lean, and many flame fronts exist at the same time. With undermixing, some fuel particles in fuel-rich zones never find oxygen to react with. In fuel-lean zones, combustion is limited and some fuel does not get burned. With overmixing, some fuel particles will be mixed with already burned gas and will therefore not combust totally.

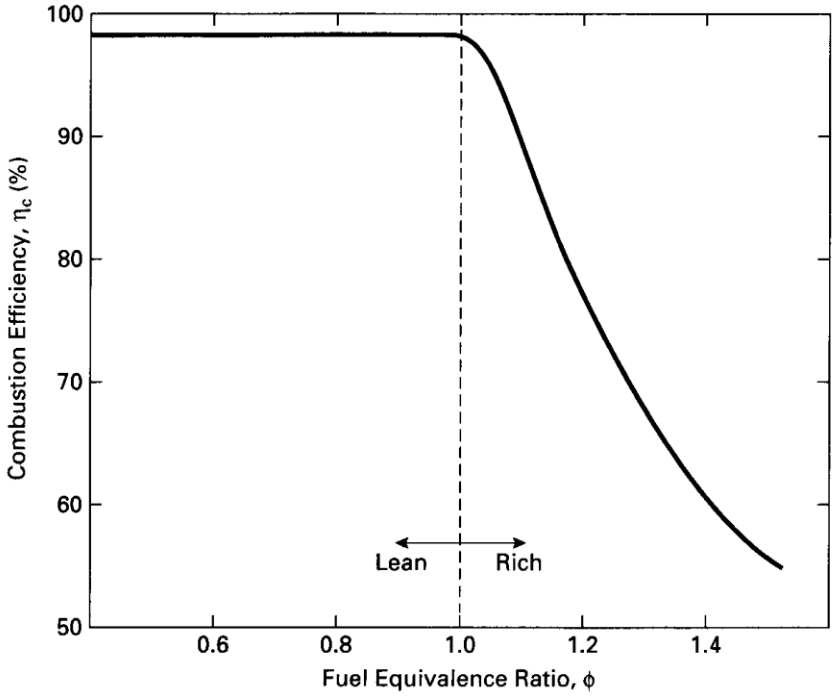
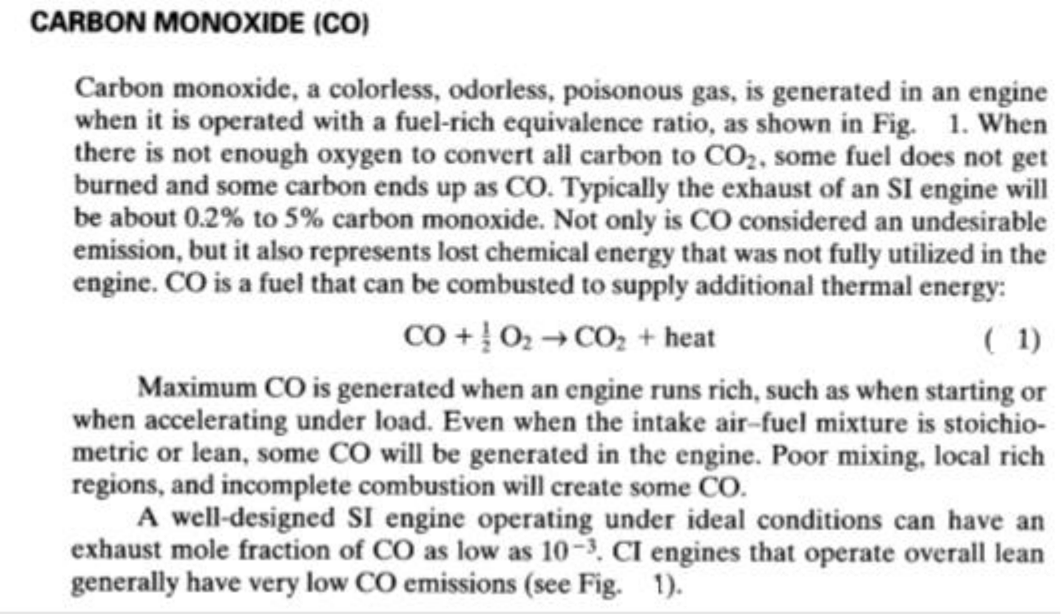


Figure a. Combustion efficiency as a function of fuel equivalence ratio. Efficiency for engines operating lean is generally on the order of 98%. When an engine operates fuel rich, there is not enough oxygen to react with all the fuel, and combustion efficiency decreases. CI engines operate lean and typically have high combustion efficiency.

It is important that injectors be constructed such that when injection stops there is no dribble from the nozzle. A small amount of liquid fuel will be trapped on the tip of the nozzle, however. This very small volume of fuel is called sac volume, its size depending on the nozzle design. This sac volume of liquid fuel evaporates very slowly because it is surrounded by a fuel-rich environment and, once the injector nozzle closes, there is no pressure pushing it into the cylinder. Some of this fuel does not evaporate until combustion has stopped, and this results in added HC particles in the exhaust. CI engines also have HC emissions for some of the same reasons as SI engines do (i.e., wall deposit absorption, oil film absorption, crevice volume, etc.).



CARBON MONOXIDE (CO)

Carbon monoxide, a colorless, odorless, poisonous gas, is generated in an engine when it is operated with a fuel-rich equivalence ratio, as shown in Fig. 1. When there is not enough oxygen to convert all carbon to CO_2, some fuel does not get burned and some carbon ends up as CO. Typically the exhaust of an SI engine will be about 0.2% to 5% carbon monoxide. Not only is CO considered an undesirable emission, but it also represents lost chemical energy that was not fully utilized in the engine. CO is a fuel that can be combusted to supply additional thermal energy:

$$CO + \tfrac{1}{2} O_2 \rightarrow CO_2 + \text{heat} \qquad (1)$$

Maximum CO is generated when an engine runs rich, such as when starting or when accelerating under load. Even when the intake air–fuel mixture is stoichiometric or lean, some CO will be generated in the engine. Poor mixing, local rich regions, and incomplete combustion will create some CO.

A well-designed SI engine operating under ideal conditions can have an exhaust mole fraction of CO as low as 10^{-3}. CI engines that operate overall lean generally have very low CO emissions (see Fig. 1).

OXIDES OF NITROGEN (NOx)

Exhaust gases of an engine can have up to 2000 ppm of oxides of nitrogen. Most of this will be nitrogen oxide (NO), with a small amount of nitrogen dioxide (NO_2), and traces of other nitrogen-oxygen combinations. These are all grouped together as NOx, with *x* representing some suitable number. NOx is a very undesirable emission, and regulations that restrict the allowable amount continue to become more stringent. Released NOx reacts in the atmosphere to form ozone and is one of the major causes of photochemical smog.

NOx is created mostly from nitrogen in the air. Nitrogen can also be found in fuel blends, which may contain trace amounts of NH_3, NC, and HCN, but this would contribute only to a minor degree. There are a number of possible reactions that form NO, all of which are probably occurring during the combustion process and immediately after. These include but are not limited to:

$$O + N_2 \rightarrow NO + N \qquad 2)$$

$$N + O_2 \rightarrow NO + O \qquad 3)$$

$$N + OH \rightarrow NO + H \qquad 4)$$

NO, in turn, can then further react to form NO_2 by various means, including

$$NO + H_2O \rightarrow NO_2 + H_2 \qquad 5)$$

$$NO + O_2 \rightarrow NO_2 + O \qquad 6)$$

Atmospheric nitrogen exists as a stable diatomic molecule at low temperatures, and only very small trace amounts of oxides of nitrogen are found. However, at the very high temperatures that occur in the combustion chamber of an engine, some diatomic nitrogen (N_2) breaks down to monatomic nitrogen (N) which is reactive:

$$N_2 \rightarrow 2\,N \qquad 7)$$

Table A-3 shows that the chemical equilibrium constant for Eq. 7) is highly dependent on temperature, with a much more significant amount of N generated in the 2500-3000 K temperature range that can exist in an engine. Other gases that are stable at low temperatures but become reactive and contribute to the formation of NOx at high temperatures include oxygen and water vapor, which break down as follows:

$$O_2 \rightarrow 2\,O \qquad 8)$$

$$H_2O \rightarrow OH + \frac{1}{2}\,H_2 \qquad 9)$$

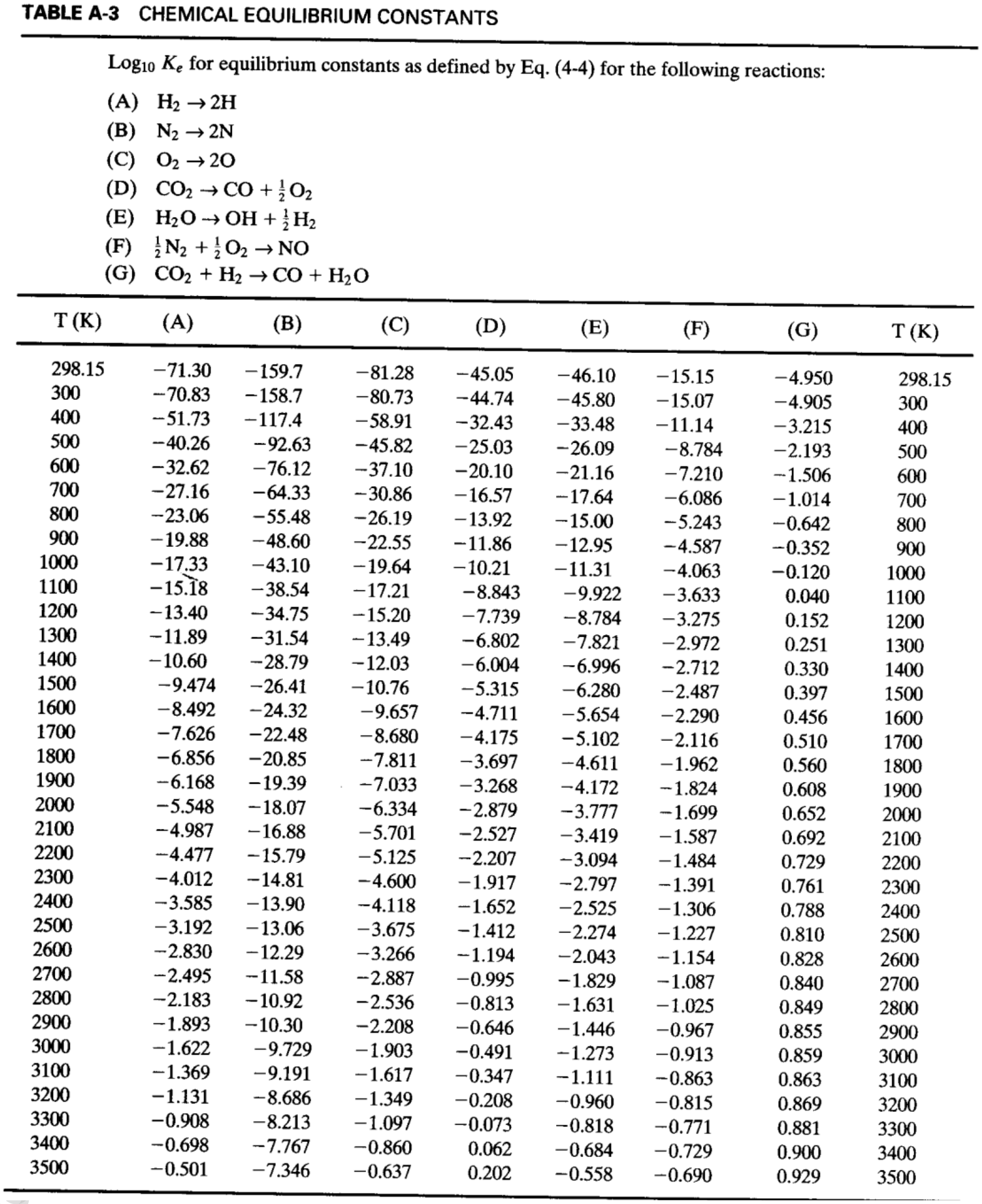


TABLE A-3 CHEMICAL EQUILIBRIUM CONSTANTS

$\log_{10} K_e$ for equilibrium constants as defined by Eq. (4-4) for the following reactions:

(A) $H_2 \rightarrow 2H$
(B) $N_2 \rightarrow 2N$
(C) $O_2 \rightarrow 2O$
(D) $CO_2 \rightarrow CO + \frac{1}{2}O_2$
(E) $H_2O \rightarrow OH + \frac{1}{2}H_2$
(F) $\frac{1}{2}N_2 + \frac{1}{2}O_2 \rightarrow NO$
(G) $CO_2 + H_2 \rightarrow CO + H_2O$

T (K)	(A)	(B)	(C)	(D)	(E)	(F)	(G)	T (K)
298.15	−71.30	−159.7	−81.28	−45.05	−46.10	−15.15	−4.950	298.15
300	−70.83	−158.7	−80.73	−44.74	−45.80	−15.07	−4.905	300
400	−51.73	−117.4	−58.91	−32.43	−33.48	−11.14	−3.215	400
500	−40.26	−92.63	−45.82	−25.03	−26.09	−8.784	−2.193	500
600	−32.62	−76.12	−37.10	−20.10	−21.16	−7.210	−1.506	600
700	−27.16	−64.33	−30.86	−16.57	−17.64	−6.086	−1.014	700
800	−23.06	−55.48	−26.19	−13.92	−15.00	−5.243	−0.642	800
900	−19.88	−48.60	−22.55	−11.86	−12.95	−4.587	−0.352	900
1000	−17.33	−43.10	−19.64	−10.21	−11.31	−4.063	−0.120	1000
1100	−15.18	−38.54	−17.21	−8.843	−9.922	−3.633	0.040	1100
1200	−13.40	−34.75	−15.20	−7.739	−8.784	−3.275	0.152	1200
1300	−11.89	−31.54	−13.49	−6.802	−7.821	−2.972	0.251	1300
1400	−10.60	−28.79	−12.03	−6.004	−6.996	−2.712	0.330	1400
1500	−9.474	−26.41	−10.76	−5.315	−6.280	−2.487	0.397	1500
1600	−8.492	−24.32	−9.657	−4.711	−5.654	−2.290	0.456	1600
1700	−7.626	−22.48	−8.680	−4.175	−5.102	−2.116	0.510	1700
1800	−6.856	−20.85	−7.811	−3.697	−4.611	−1.962	0.560	1800
1900	−6.168	−19.39	−7.033	−3.268	−4.172	−1.824	0.608	1900
2000	−5.548	−18.07	−6.334	−2.879	−3.777	−1.699	0.652	2000
2100	−4.987	−16.88	−5.701	−2.527	−3.419	−1.587	0.692	2100
2200	−4.477	−15.79	−5.125	−2.207	−3.094	−1.484	0.729	2200
2300	−4.012	−14.81	−4.600	−1.917	−2.797	−1.391	0.761	2300
2400	−3.585	−13.90	−4.118	−1.652	−2.525	−1.306	0.788	2400
2500	−3.192	−13.06	−3.675	−1.412	−2.274	−1.227	0.810	2500
2600	−2.830	−12.29	−3.266	−1.194	−2.043	−1.154	0.828	2600
2700	−2.495	−11.58	−2.887	−0.995	−1.829	−1.087	0.840	2700
2800	−2.183	−10.92	−2.536	−0.813	−1.631	−1.025	0.849	2800
2900	−1.893	−10.30	−2.208	−0.646	−1.446	−0.967	0.855	2900
3000	−1.622	−9.729	−1.903	−0.491	−1.273	−0.913	0.859	3000
3100	−1.369	−9.191	−1.617	−0.347	−1.111	−0.863	0.863	3100
3200	−1.131	−8.686	−1.349	−0.208	−0.960	−0.815	0.869	3200
3300	−0.908	−8.213	−1.097	−0.073	−0.818	−0.771	0.881	3300
3400	−0.698	−7.767	−0.860	0.062	−0.684	−0.729	0.900	3400
3500	−0.501	−7.346	−0.637	0.202	−0.558	−0.690	0.929	3500

Examination of Table A-3 and more elaborate chemical equilibrium constant tables found in chemistry handbooks show that chemical Eqs. 7- 9 all react much further to the right as high combustion chamber temperatures are reached. The higher the combustion reaction temperature, the more diatomic nitrogen, N_2, will dissociate to monatomic nitrogen, N, and the more NOx will be formed. At low temperatures very little NOx is created.

Although maximum flame temperature will occur at a stoichiometric air-fuel ratio ($\phi = 1$), Fig. 1 shows that maximum NOx is formed at a slightly lean equivalence ratio of about ϕ = 0.95. At this condition flame temperature is still very high, and in addition, there is an excess of oxygen that can combine with the nitrogen to form various oxides.

In addition to temperature, the formation of NOx depends on pressure, air-fuel ratio, and combustion time within the cylinder, chemical reactions not being instantaneous. Figure 3 shows the NOx-versus-time relationship and supports the fact that NOx is reduced in modern engines with fast-burn combustion chambers. The amount of NOx generated also depends on the location within the combustion chamber. The highest concentration is formed around the spark plug, where the highest temperatures occur. Because they generally have higher compression ratios and higher temperatures and pressure, CI engines with divided combustion chambers and indirect injection (IDI) tend to generate higher levels of NOx.

Figure 4 shows how NOx can be correlated with ignition timing. If ignition spark is advanced, the cylinder temperature will be increased and more NOx will be created.

Photochemical Smog. NOx is one of the primary causes of photochemical smog, which has become a major problem in many large cities of the world. Smog is formed by the photochemical reaction of automobile exhaust and atmospheric air in the presence of sunlight. NO2 decomposes into NO and monatomic oxygen:

$$NO_2 + energy\ from\ sunlight \rightarrow NO + O + smog -----10$$

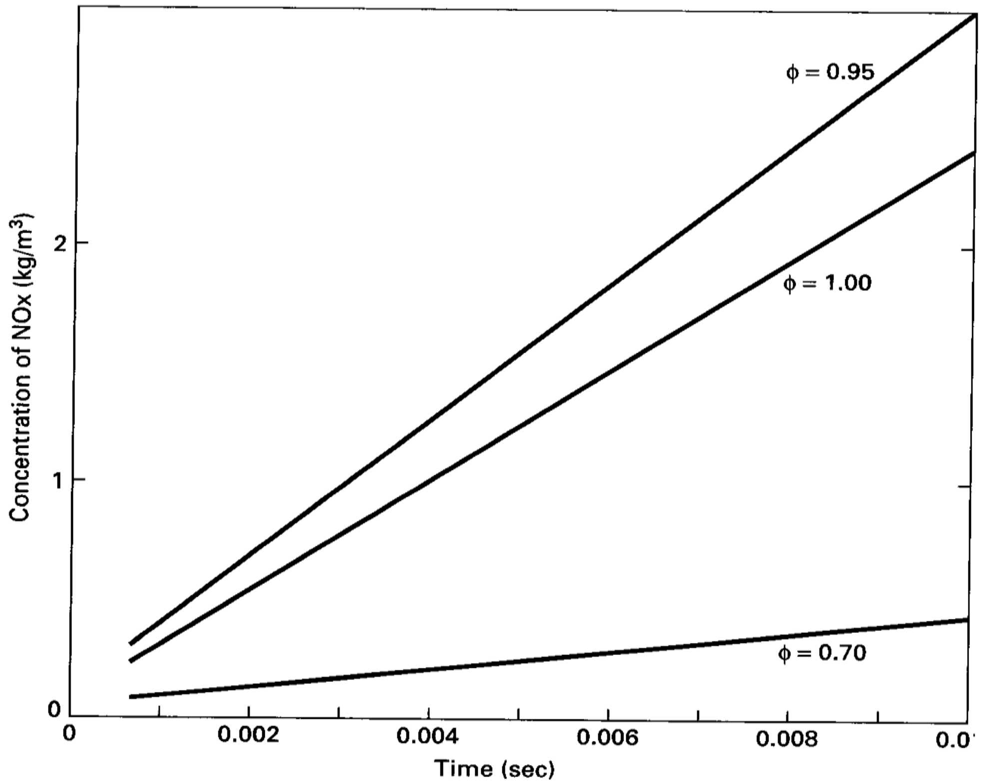


Figure 3 Generation of NOx in an engine as a function of combustion time. Many modern engines produce lower NOx emissions due to fast-burn combustion chamber design.

Monatomic oxygen is highly reactive and initiates a number of different reactions, one of which is the formation of ozone:

$$O + O_2 \rightarrow O_3 \qquad -11$$

Ground-level ozone is harmful to lungs and other biological tissue. It is harmful to trees and causes billions of dollars of crop loss each year. Damage is also caused through reaction with rubber, plastics, and other materials. Ozone also results from atmospheric reactions with other engine emissions such as HC, aldehydes, and other oxides of nitrogen.

PARTICULATES

The exhaust of CI engines contains solid carbon *soot* particles that are generated in the fuel-rich zones within the cylinder during combustion. These are seen as exhaust smoke and are an undesirable odorous pollution. Maximum density of particulate emissions occurs when the engine is under load at WOT. At this condition maximum fuel is injected to supply maximum power, resulting in a rich mixture and poor fuel economy. This can be seen in the heavy exhaust smoke emitted when a truck or railroad locomotive accelerates up a hill or from a stop.

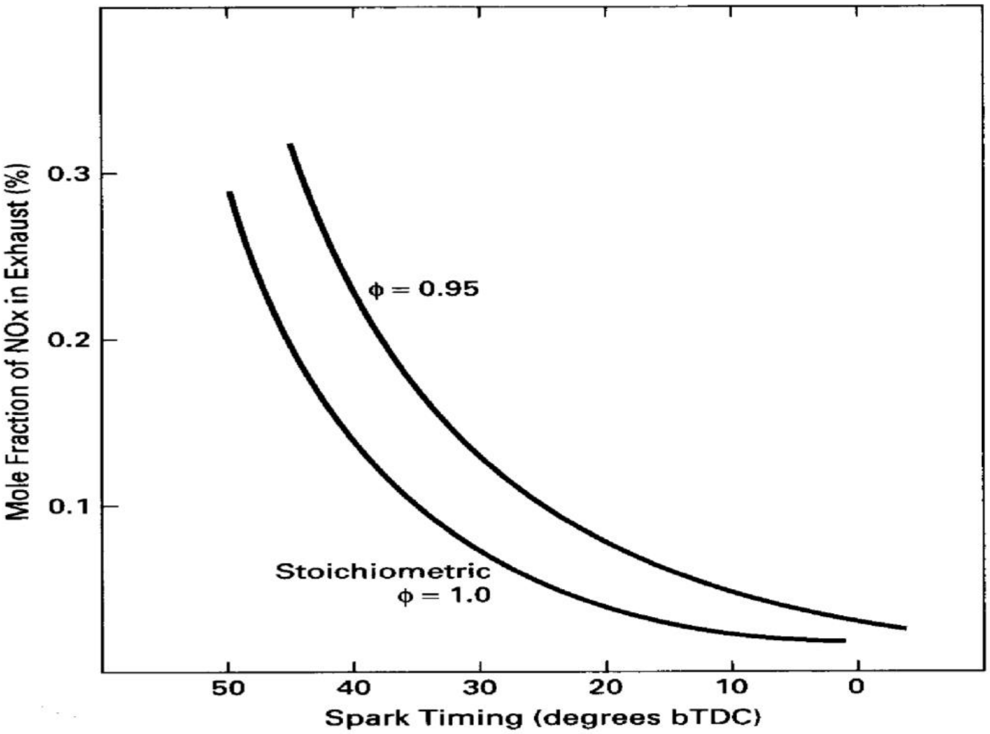


Figure -4 Generation of NOx in an SI engine as a function of spark timing. Earlier spark ignition creates a higher combustion temperature, which generates higher levels of NOx.

Soot particles are clusters of solid carbon spheres. These spheres have diameters from 10 nm to 80 nm (1 nm = 10-9 m), with most within the range of 15-30 nm. The spheres are solid carbon with HC and traces of other components absorbed on the surface. A single soot particle will contain up to 4000 carbon spheres.

Carbon spheres are generated in the combustion chamber in the fuel-rich zones where there is not enough oxygen to convert all carbon to CO2:

$$CxHy + z\,02 \rightarrow a\,CO2 + bH20 + cCO + dC(s) \qquad (12)$$

Then, as turbulence and mass motion continue to mix the components in the combustion chamber, most of these carbon particles find sufficient oxygen to further react and are consumed to CO2:

$$C(s) + O2 \sim CO2 \qquad (13)$$

Over 90% of carbon particles originally generated within an engine are thus consumed and never get exhausted. If CI engines would operate with an overall stoichiometric air-fuel mixture, instead of overall lean as they do, particulate emissions in the exhaust would far exceed acceptable levels.

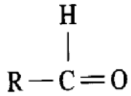
Up to about 25% of the carbon in soot comes from lubricating oil components which vaporize and then react during combustion. The rest comes from the fuel and amounts to 0.2-0.5% of the fuel. Because of the high compression ratios of CI engines, a large expansion occurs during the power stroke, and the gases within the cylinder are cooled by expansion cooling to a relatively low temperature. This causes the remaining high-boiling-point components found in the fuel and lubricating oil to condense on the surface of the carbon soot particles. This absorbed portion of the soot particles is called the **soluble organic fraction** (SOF), and the amount is highly dependent on cylinder temperature. At light loads, cylinder temperatures are reduced and can drop to as low as 200°C during final expansion and exhaust blowdown. Under these conditions, SOF can be as high as 50% of the total mass of soot. Under other operating conditions when temperatures are not so low, very little condensing occurs and SOF can be as low as 3% of total soot mass. SOF consists mostly of hydrocarbon components with some hydrogen, S02, NO, N02, and trace amounts of sulfur, zinc, phosphorus, calcium, iron, silicon, and chromium. Diesel fuel contains sulfur, calcium, iron, silicon, and chromium, while lubricating oil additives contain zinc, phosphorus, and calcium.

Particulate generation can be reduced by engine design and control of operating conditions, but quite often this will create other adverse results. If the combustion time is extended by combustion chamber design and timing control, particulate amounts in the exhaust can be reduced. Soot particles originally generated will have a greater time to be mixed with oxygen and combusted to CO2. However, a longer combustion time means a high cylinder temperature and more NOx generated. Dilution with EGR lowers NOx emissions but increases particulates and HC emissions. Higher injection pressure gives a finer droplet size, which reduces HC and particulate emissions but increases cylinder temperature and NOx emissions. Engine management systems are programmed to minimize NOx, HC, CO, and particulate emissions by controlling ignition timing, injection pressure, injection timing, and/or valve timing. Obviously, compromising is necessary. In most engines, exhaust particulate amounts cannot be reduced to acceptable levels solely by engine design and control.

OTHER EMISSIONS

Aldehydes

A major emission problem when alcohol fuel is used is the generation of aldehydes, an eye and respiratory irritant. These have the chemical formula of:



$$R-\overset{\overset{\displaystyle H}{|}}{C}=O$$

where R denotes various chemical radicals.

This is a product of incomplete combustion and would be a major problem if as much alcohol fuel were used as presently is gasoline.

Sulfur

Many fuels used in CI engines contain small amounts of sulfur which, when exhausted, contribute to the acid rain problem of the world. Unleaded gasoline generally contains 150–600 ppm sulfur by weight. Some diesel fuels contain up to 5000 ppm by weight, but in the United States and some other countries this is restricted by law to a tenth of this value or less.

At high temperatures, sulfur combines with hydrogen to form H_2S and with oxygen to form SO_2:

$$H_2 + S \rightarrow H_2S \qquad \text{-14)}$$

$$O_2 + S \rightarrow SO_2 \qquad \text{-15)}$$

Engine exhaust can contain up to 20 ppm of SO_2. SO_2 then combines with oxygen in the air to form SO_3:

$$2\,SO_2 + O_2 \rightarrow 2\,SO_3 \qquad \text{-16)}$$

These combine with water vapor in the atmosphere to form sulfuric acid (H_2SO_4) and sulfurous acid (H_2SO_3), which are ingredients in acid rain:

$$S03 + H2O \rightarrow H2S04 \qquad 17)$$

$$SO2 + H2O \rightarrow HzS03 \qquad 18)$$

Many countries have laws restricting the amount of sulfur allowed in fuel, and these are continuously being made more stringent. During the 1990s, the United States reduced acceptable levels in diesel fuel from 0.05% by weight to 0.01%.

The amount of sulfur in natural gas can range from little (sweet) to large (sour) amounts. This can be a major emissions problem when this fuel is used in a IC engine or any other combustion system.

Lead

Lead was a major gasoline additive from its introduction in 1923to when it was phased out in the 1980s. The additive TEL (tetraethyllead) was effectively used to increase gasoline octane number, which allowed higher compression ratios and more efficient engines. However, the resulting lead in the engine exhaust was a highly poisonous pollutant. During the first half of the 1900s, due to the lower number of automobiles and other engines, the atmosphere was able to absorb these emissions of lead without notice-able problems. As population and automobile density increased, the awareness of air pollution and its danger also increased. The danger of lead emissions was recognized, and a phase-out occurred during the 1970s and 1980s.

The use of lead could not be stopped immediately but had to be phased out over a number of years. First, low-lead gasoline was introduced, and then, years later no-lead gasoline. Lead was still the major additive to raise the octane number of gasoline, and alternate octane raisers had to be developed as lead was phased out. Millions of modern high-compression engines could not use low-octane fuel. Metals used in engines also had to be changed as lead in gasoline was phased out. When leaded fuel is burned, it hardens the surfaces in the combustion chamber and on the valves and valve seats. Engines designed to use leaded fuel had softer metal surfaces to start and relied on surface hardening effects that occurred in use. If these engines are used with unleaded fuel, surface hardening is not realized and serious wear is quickly experienced. Catastrophic failures of valve seats or piston faces are common in a short period of time (i.e., 10,000-20,000 miles in an automobile). Harder metals and added surface treatments are used for engines designed to use unleaded fuel. It was necessary to phase out leaded gasoline over a period of time as older automobiles wore out and were taken out of operation.

Leaded gasoline contains about 0.15 gm/liter of lead in the fuel. Between 10% and 50% of this gets exhausted out with the other combustion products. The remaining lead gets deposited on the walls of the engine and exhaust system. The hardened combustion chamber surfaces which resulted from the burning of leaded gasoline were quite impervious to the absorption of gases such as fuel vapor. HC emissions were also, therefore, slightly reduced in these engines.

AFTERTREATMENT

After the combustion process stops, those components in the cylinder gas mixture that have not fully burned continue to react during the expansion stroke, during exhaust blowdown, and into the exhaust process. Up to 90% of the HC remaining after combustion reacts during this time either in the cylinder, near the exhaust port, or in the upstream part of the exhaust manifold. CO and small component hydrocarbons react with oxygen to form CO2 and H2O and reduce undesirable emissions. The higher the exhaust temperature, the more these secondary reactions occur and the lower the engine emissions. Higher exhaust temperature can be caused by stoichiometric air-fuel combustion, high engine speed, retarded spark, and/or a low expansion ratio.

Thermal Converters

Secondary reactions occur much more readily and completely if the temperature is high, so some engines are equipped with thermal converters as a means of lowering emissions. Thermal converters are high-temperature chambers through which the exhaust gas flows. They promote oxidation of the CO and HC which remain in the exhaust.

$$CO + \frac{1}{2}\,O2 \rightarrow CO2 \qquad 19)$$

For this reaction to occur at a useful rate, the temperature must be held above 700°C.

$$CxHy + Z\,O2 \rightarrow x\,CO2 + \frac{1}{2}y\,H2O \qquad 20)$$

where $Z = x + \frac{1}{2}y$.

This reaction needs a temperature above 600°C for at least 50 msec to substantially reduce HC. It is therefore necessary for a thermal converter not only to operate at a high temperature but to be large enough to provide adequate dwell time to promote the occurrence of these secondary reactions. Most thermal converters are essentially an enlarged exhaust manifold connected to the engine immediately outside the exhaust ports. This is necessary to minimize heat losses and keep the exhaust gases from cooling to non-reacting temperatures. However, in automobiles this creates two very serious problems for the engine compartment. In modern, low-profile, aerodynamic automobiles, space in the engine compartment is very limited, and fitting in a large, usually insulated thermal converter chamber is almost impossible. Secondly, because the converter must operate above 700°C to be efficient, even if it is insulated the heat losses create a serious temperature problem in the engine compartment.

Some thermal converter systems include an air intake which provides additional oxygen to react with the CO and HC. This increases the complexity, cost, and size of the system. Flow rate of air is controlled by the EMS as needed. Air addition is especially necessary during rich operating conditions such as startup. Because exhaust from engines is often at a lower temperature than is needed for efficient operation of a thermal converter, it is necessary to sustain the high temperatures by the reactions within the system. Adding outside air, which is at a lower temperature, compounds this problem of maintaining the necessary operating temperature.

NOx emissions cannot be reduced with a thermal converter alone.

CATALYTIC CONVERTERS

The most effective aftertreatment system for reducing engine emissions is the catalytic converter found on most automobiles and other modern engines of medium or large size. HC and CO can be oxidized to H2O and CO2 in exhaust systems and thermal converters if the temperature is held at 600°-700°C. If certain catalysts are present, the temperature needed to sustain these oxidation processes is reduced to 250°-300°C, making for a much more attractive system. A catalyst is a substance that accelerates a chemical reaction by lowering the energy needed for it to proceed. The catalyst is not consumed in the reaction and so functions indefinitely unless degraded by heat, age, contaminants, or other factors. Catalytic converters are chambers mounted in the flow system through which the exhaust gases flow. These chambers contain catalytic material, which promotes the oxidation of the emissions contained in the exhaust flow.

Generally, catalytic converters are called three-way converters because they promote the reduction of CO, HC, and NOx. Most consist of a stainless steel container mounted somewhere along the exhaust pipe of the engine. Inside the container is a porous ceramic structure through which the gas flows. In most converters, the ceramic is a single honeycomb structure with many flow passages (see Figure 5). Some converters use loose granular ceramic with the gas passing between the packed spheres. Volume of the ceramic structure of a converter is generally about half the displacement volume of the engine. This results in a volumetric flow rate of exhaust gas such that there are 5 to 30 changeovers of gas each second, through the converter. Catalytic converters for CI engines need larger flow passages because of the solid soot in the exhaust gases.

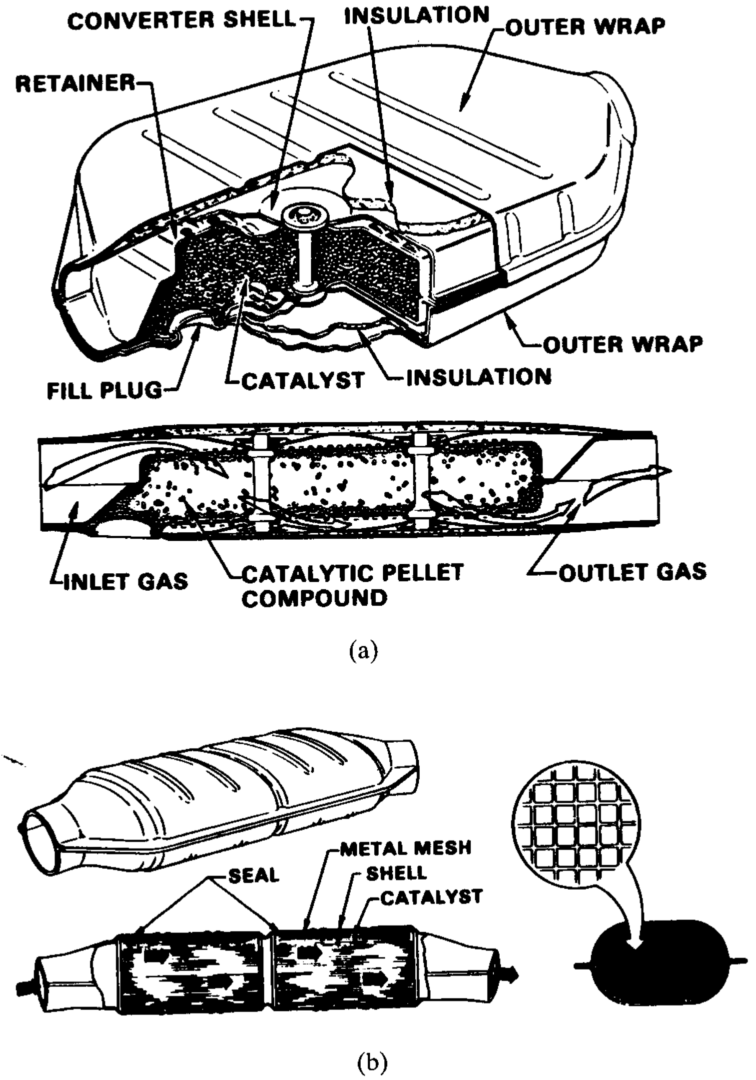


Figure 5 Catalytic converters for SI engines: (a) packed spheres; (b) honeycomb structure.

The surface of the ceramic passages contains small embedded particles of catalytic material that promote the oxidation reactions in the exhaust gas as it passes. Aluminum oxide (alumina) is the base ceramic material used for most catalytic converters. Alumina can withstand the high temperatures, it remains chemically neutral, it has very low thermal expansion, and it does not thermally degrade with age. The catalyst materials most commonly used are platinum, palladium, and rhodium.

Palladium and platinum promote the oxidation of CO and HC as in Eqs. -19) and (-20), with platinum especially active in the hydrocarbon reaction. Rhodium promotes the reaction of NOx in one or more of the following reactions:

$$NO + CO \rightarrow \tfrac{1}{2} N_2 + CO_2 \qquad \text{-21)}$$

$$2\,NO + 5\,CO + 3\,H_2O \rightarrow 2\,NH_3 + 5\,CO_2 \qquad \text{-22)}$$

$$2\,NO + CO \rightarrow N_2O + CO_2 \qquad \text{-23)}$$

$$NO + H_2 \rightarrow \tfrac{1}{2} N_2 + H_2O \qquad \text{-24)}$$

$$2\,NO + 5\,H_2 \rightarrow 2\,NH_3 + 2\,H_2O \qquad \text{-25)}$$

$$2\,NO + H_2 \rightarrow N_2O + H_2O \qquad \text{-26)}$$

Also often used is cerium oxide, which promotes the so-called water gas shift:

$$CO + H_2O \rightarrow CO_2 + H_2 \qquad \text{-27)}$$

This reduces CO by using water vapor as an oxidant instead of O_2, which is very important when the engine is running rich.

Figure 6 shows that the efficiency of a catalytic converter is very dependent on temperature. When a converter in good working order is operating at a fully warmed temperature of 400°C or above, it will remove 98–99% of CO, 95% of NOx, and more than 95% of HC from exhaust flow emissions. Figure 7 shows that it is also necessary to be operating at the proper equivalence ratio to get high converter efficiency. Effective control of HC and CO occurs with stoichiometric or lean mix-

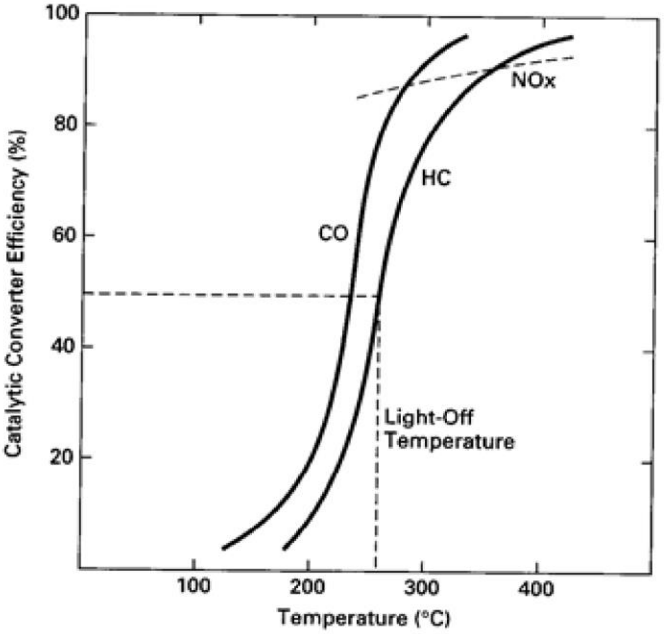


Figure 6 Conversion efficiency of catalytic converters as a function of converter temperature. A converter in good condition will generally reduce emissions by over 90% if it is at normal operating temperature. When cold, a converter is very inefficient. The temperature at which a converter becomes 50% efficient is often called light-off temperature.

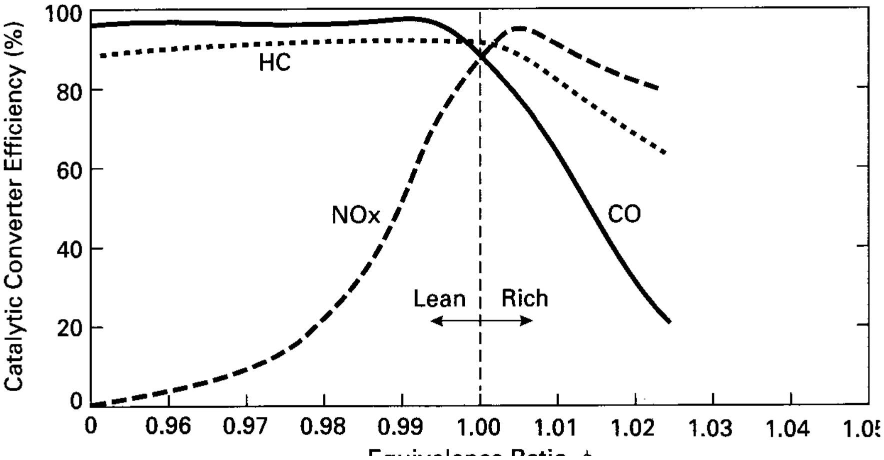


Figure 7 Conversion efficiency of catalytic converters as a function of fuel equivalence ratio. Greatest efficiency occurs when engines operate near stoichiometric conditions. Converters are very inefficient for NOx conversion when an engine operates lean. This creates a greater problem for modern CI engines and stratified charge SI engines, which generally operate very lean overall.

tures, while control of NOx requires near stoichiometric conditions. Very poor NOx control occurs with lean mixtures.

Because'1m engine has a number of cyclic variations occurring-including AF-the exhaust flow will also show variation. It has been found that this cyclic variation lowers the peak efficiency of a catalytic converter but spreads the width of the equivalence ratio envelope of operation in which there is acceptable emissions reduction.

It is important that a catalytic converter be operated hot to be efficient, but no hotter. Engine malfunctions can cause poor efficiency and overheating of converters. A poorly tuned engine can have misfires and periods of too lean and/or too rich conditions. These cause the converter to be inefficient at the very time emissions are very high and maximum converter efficiency is needed. A turbocharger lowers the exhaust temperature by removing energy, and this can make a catalytic converter less efficient.

It is desirable that catalytic converters have an effective lifetime equal to that of the automobile or at least 200,000 km. Converters lose their efficiency with age due to thermal degradation and poisoning of the active catalyst material. At high temperature the metal catalyst material can sinter and migrate together, forming larger active sites which are, overall, less efficient. Serious thermal degrading occurs in the temperature range of 500^o-900°C. A number of different impurities contained in fuel, lubricating oil, and air find their way into the engine exhaust and poison the catalyst material. These include lead and sulfur from fuels, and zinc, phosphorus, antimony, calcium, and magnesium from oil additives. Figure 8 shows how just a small amount of lead on a catalyst site can reduce HC reduction by a factor of two or three. Small amounts of lead impurities are found in some fuels, and 10-30% of this ends up in the catalytic converter. Up until the early 1990s leaded gasoline was quite common, and it was imperative that it not be used in engines equipped with catalytic converters; it was unlawful. Two fuel tanks of leaded gasoline would completely poison a converter and make it totally useless.

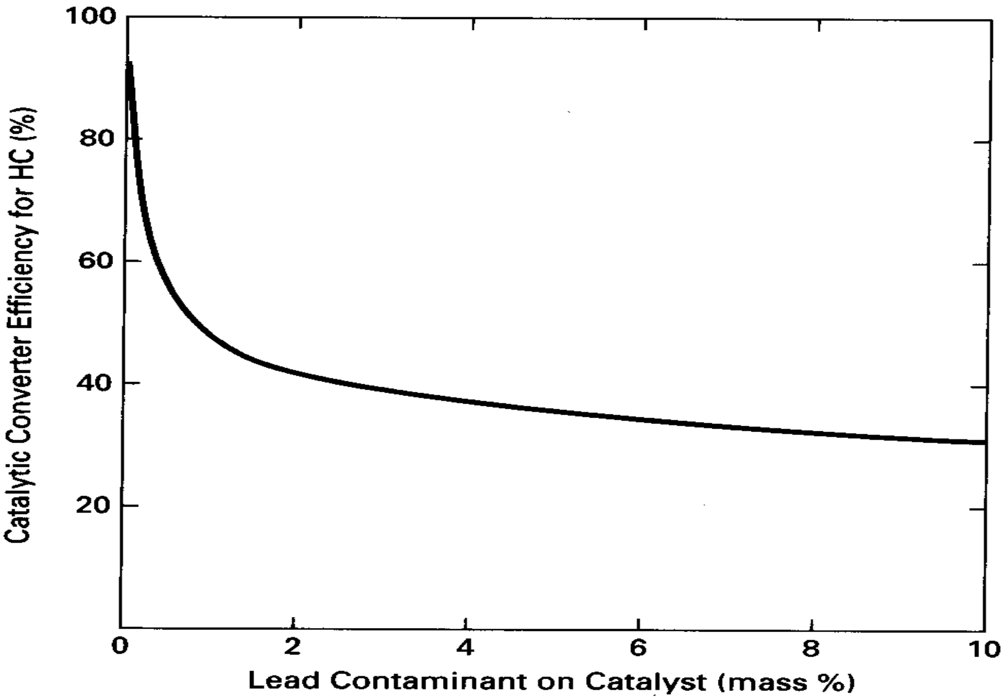


Figure 8 Reduction of catalytic converter efficiency due to contamination by lead. Some of the lead contained in fuels gets deposited on the catalyst material in a converter, greatly reducing converter efficiency. It is imperative (and unlawful) that leaded gasoline not be used in automobiles equipped with catalytic converters. To reduce the chances of accidently using leaded gasoline with a catalytic converter, the fuel pump nozzle size and the diameter of the fuel tank inlet are made smaller for nonleaded gasoline.

Sulfur

Sulfur offers unique problems for catalytic converters. Some catalysts promote the conversion of SO2 to SO3, which eventually gets converted to sulfuric acid. This degrades the catalytic converter and contributes to acid rain. New catalysts are being developed that promote the oxidation of HC and CO but do not change SO2 to SO3. Some of these create almost no S03 if the temperature of the converter is kept at 400°C or lower.

Cold Start-Ups

Figure 6 shows that catalytic converters are very inefficient when they are cold. When an engine is started after not being operated for several hours, it takes several minutes for the converter to reach an efficient operating temperature. The temperature at which a converter becomes 50% efficient is defined as the **light-off temperature,** and this is in the range of about 250°-300°C. A large percent of automobile travel is for short distances where the catalytic converter never reaches efficient operating temperature, and therefore emissions are high. Some studies suggest that half of the fuel used by automobiles in the United States is on trips of less than 10 miles distance. Unfortunately, most short trips occur in cities where high emissions are more harmful. Add to this the fact that most engines use a rich mixture when starting and it can be seen that cold start-ups pose a major problem. It is estimated that cold start-ups are the source of 70--90% of all HC emissions. A major reduction in emissions is therefore possible if catalytic converters could be preheated, at least to light-off temperature, before engine startup. Preheating to full steady-state operating temperature would be even better. Several methods of preheating have been tried with varying success. Because of the time involved and amount of energy needed, most of these methods preheat only a small portion of the total converter volume. This small section is large enough to treat the low exhaust flow rate which usually occurs at startup and immediately following. By the time higher engine speeds are used, more of the catalytic converter has been heated by the hot exhaust gas, and the higher flow rates are fully treated. Methods of catalytic converter preheating include the following.

Locate Converter Close to the Engine: One method used to heat a converter as quickly as possible is to locate it in the engine compartment very close to the exhaust ports. This method does not actually preheat the converter but does heat it as quickly as possible after the engine is started. It eliminates the large heat loss from the exhaust pipe that occurs between the engine and a converter in more common systems where the converter is located away from the engine. These converters can also be insulated to reduce early heat loss. This method does reduce overall emissions by quick heat-up of the converter, but there is still a short period of time before light-off temperature is reached. In addition, the same problems described for thermal converters mounted in the engine compartment are encountered with this type of converter. Adequate cooling of the engine compartment because of the high temperatures and restricted flow rate of air caused by the converter is a serious problem. If located in the hot engine compartment, a catalytic converter will also have a higher steady-state temperature, and this will cause a greater long-term thermal degrading problem.

Some automobiles use a small secondary catalytic converter mounted in the engine compartment close to the engine. Because of its small size and location, it heats up very quickly and is sufficient to oxidize the emissions in the low flow rates at engine startup. There is also a normal full-size catalytic converter mounted away from the engine compartment which supplies the catalytic action for the larger flow rates of normal operation. This converter is heated by the first exhaust flow and ideally reaches efficient operating temperature before the engine is speeded up and higher flow rates are experienced. These small pre-converters restrict flow in the exhaust manifold and add some back pressure to the engine. This results in a slight reduction in engine power output.

Superinsulation: Some systems have been developed which have superinsulated catalytic converters. These do not actually preheat the converter on first engine startup, but they do accelerate the temperature rise to steady-state conditions. They also keep the converter at an elevated temperature for up to a day after the engine has been turned off. The converter is therefore preheated for subsequent engine starts.

The converter is double walled with a vacuum between the walls. This gives the super-insulation characteristics much like that of a vacuum bottle. When the engine is cold and/or not running, the vacuum is sustained. When the engine is running and the converter is at operating temperature, the vacuum is eliminated and the space between the walls is filled with a gas. This allows for normal heat losses during operation and keeps the catalytic converter from overheating.

Electric Heating: A number of systems have been tested using electric preheating, usually by resistance heating. Heating resistors are embedded in the preheat zone of the converter, and an electrical discharge is initiated before the engine is started. The preheat zone can be a separate, small pre converter, or it can be the front end of the normal catalytic converter. Some systems replace the ceramic honeycomb solid in the preheat zone with a metal structure of multiple flow passages. This allows a much quicker heating of the flow passage walls by means of thermal conduction, metal having much higher thermal conductivity than ceramic. Electrical energy for this type of system usually comes from a battery that is recharged when the engine is running. Typical values for preheating are 24 volts and 500--700amps.

There is some time delay between heating the electrical elements and reaching light-off temperature, due to the conduction needed. The most serious problem, however, is the inability of a normal-sized battery to deliver the amount of energy needed for such a system.

Flame Heating: A catalytic converter can be heated with a flame from a burner nozzle mounted within the structure of the converter [57]. Before the engine is started (for instance, when the ignition key is inserted), a flame is initiated in the burner using fuel and air pumped from external sources. Concern must be given to what emissions this flame would contribute to the overall air pollution problem. A fuel like propane burned with the correct amount of air would create very little pollution. However, this would require an axillary propane fuel tank on the automobile, something which would be undesirable unless the automobile engine was also fueled with propane. In a gasoline-fueled engine, it would be logical to use gasoline in the converter preheater. However, it would be more difficult to get clean burning with gasoline. Cost, complexity, and some time delay are disadvantages of this type of system.

A variation of this system used by at least one major automobile manufacturer is an *afterburner* mounted directly before the catalytic converter. A very rich air-fuel mixture is used at startup, which leaves excess fuel in the first exhaust flow. Air is added to this exhaust by an electric pump, and the resulting mixture is combusted in the afterburner, preheating the catalytic converter.

Thermal Battery: Energy from a thermal storage system can be used to preheat a catalytic converter if the engine is started within about three days of last being used. With present technology only partial preheating to a temperature around 60°C is possible, which is still below light-off temperature and well below normal operating temperatures. In addition, the limited amount of energy available in a thermal battery is often distributed between preheating the engine, warming the passenger compartment, and preheating the catalytic converter.

Chemical Reaction Preheating: A possible method for preheating a catalytic converter has been suggested that uses the heat liberated from an exothermic chemical reaction. When the ignition key is inserted, a small amount of water is introduced into the converter from an injector mounted through the side of the converter housing. The water spray reacts with a salt imbedded in the surface of the ceramic honeycomb. This exothermic reaction releases enough energy to heat the surrounding ceramic structure to a temperature above light-off temperature, and the converter is ready for efficient use in a matter of seconds. When the engine is then started, the hot exhaust gases dry the imbedded salt by evaporating away the water. The water vapor is carried away with the exhaust gas and the system is ready for the next cold start. One major problem with this method is degradation of the salt with age. Also, there is a freezing problem with the water reservoir in cold climates. No practical systeffi..1lsingthis method has so far appeared on the market.

Dual-Fuel Engines

Some engines are made to run on a combination of gasoline and methanol, with the percent volume of methanol ranging from 0% to 85%. The engine control systems on these engines are capable of adjusting the air and fuel flow to give optimum combustion and minimum emissions with any combination of these fuels. However, this creates a unique problem for a catalytic converter. Each of these fuels requires separate catalysts. Incomplete combustion of methanol produces formaldehyde, which must be removed from the exhaust. To effectively reduce the formaldehyde and any remaining methanol, a catalytic converter must be operated above 300°C. Preheating of the converter on these systems is very important.

Lean-Burn Engines

A number of automobiles on the market obtain high fuel efficiency by the use of **lean-burn** engines. By using a stratified charge, these engines obtain efficient combustion with overall air-fuel ratios of 20 or 21 ($\phi = 0.7$). Figure 7 shows that normal catalytic converters will work in reducing HC and CO at lean conditions but are very inefficient at reducing NOx. Special converters, which use platinum and rhodium combined with alkaline rare earths, have been developed for lean-burn engines. Combustion temperatures must be limited in these engines so that NOx production is kept within manageable limits.

Two-Stroke Cycle Engines

Modem two-stroke cycle engines that use fuel injectors have cooler exhaust because of their high efficiency and lean operation. Both the lower exhaust temperature and lean operation make the typical catalytic converter less efficient and create a more difficult emissions problem with these engines.

HISTORIC-HYBRID POWERED AUTOMOBILES

One method of reducing emissions in cities is to use hybrid vehicles powered by both an electric motor and a small internal combustion engine. The vehicle uses the electric motor for normal operation and can be classified as a zero emissions vehicle (ZEV) under these conditions. The IC engine is mostly used to recharge the batteries for the electric motor, and only occasionally used to power the vehicle when an extended range is needed. The engine is clean running with very low emissions, being designed for low power and constant speed, an ultra-low-emissions vehicle (ULEV).

CI ENGINES

Catalytic converters are used with CI engines but are not efficient at reducing NOx due to their overall lean operation. HC and CO can be adequately reduced, although there is greater difficulty because of the cooler exhaust gases of a CI engine (because of the larger expansion ratio). This is counter-balanced by the fact that less HC and CO are generated in the lean bum of the CI engine. NOx is reduced in a CI engine by the use of EGR, which keeps the maximum temperature down. EGR and lower combustion temperatures, however, contribute to an increase in solid soot.

Platinum and palladium are two main catalyst materials used for converters on CI engines. They promote the removal of 30-80% of the gaseous HC and 40--90% of the CO in the exhaust. The catalysts have little effect on solid carbon soot but do remove 30-60% of the total particulate mass by oxidizing a large percent of the HC absorbed on the carbon particles. Diesel fuel contains sulfur impurities, and this leads to poisoning of the catalyst materials. However, this problem is being reduced as legal levels of sulfur in diesel fuels continue to be lowered.

Particulate Traps

Compression ignition engine systems are equipped with particulate traps in their exhaust flow to reduce the amount of particulates released to the atmosphere. Traps are filter-like systems often made of ceramic in the form of a monolith or mat, or else made of metal wire mesh. Traps typically remove 60-90% of particulates in the exhaust flow. As traps catch the soot particles, they slowly fill up with the particulates. This restricts exhaust gas flow and raises the back pressure of the engine. Higher back pressure causes the engine to run hotter, the exhaust temperature to rise, and fuel consumption to increase. To reduce this flow restriction, particulate traps are regenerated when they begin to become saturated. Regeneration consists of combusting the particulates in the excess oxygen contained in the exhaust of the lean-operating CI engine.

Carbon soot ignites at about 550°-650°C, while CI engine exhaust is 150°-350°C at normal operating conditions. As the particulate trap fills with soot and restricts flow, the exhaust temperature rises but is still not high enough to ignite the soot and regenerate the trap. In some systems, automatic flame igniters are used which start combustion in the carbon when the pressure drop across the trap reaches a predetermined value. These igniters can be electric heaters or flame nozzles that use diesel fuel. If catalyst material is installed in the traps, the temperature needed to ignite the carbon soot is reduced to the 350°-450°C range. Some such traps can automatically regenerate by self-igniting when the exhaust temperature rises from increased back pressure. Other catalyst systems use flame igniters.

Another way of lowering the ignition temperature of the carbon soot and promoting self-regeneration in traps is to use catalyst additives in the diesel fuel. These additives generally consist of copper compounds or iron compounds, with about 7 grams of additive in 1000 liters of fuel being normal.

To keep the temperatures high enough to self-regenerate in a catalytic system, traps can be mounted as close to the engine as possible, even before the turbocharger.

On some larger stationary engines and on some construction equipment and large trucks, the particulate trap is replaced when it becomes close to filled. The removed trap is then regenerated externally, with the carbon being burned off in a furnace. The regenerated trap can then be used again.

Vanous methods are used to determine when soot buildup becomes excessive and regeneration is necessary. The most common method is to measure pressure drop in the exhaust flow as it passes through the trap. When a predetermined Δp is reached, regeneration is initiated. Pressure drop is also a function of exhaust flow rate, and this must be programmed into the regeneration controls. Another method used to sense soot buildup is to transmit radio frequency waves through the trap and determine the percent that is absorbed. Carbon soot absorbs radio waves while the ceramic structure does not. The amount of soot buildup can therefore be determined by the percent decrease in radio signal. This method does not readily detect soluble organic fraction (SOF).

Modern particulate traps are not totally satisfactory, especially for automobiles. They are costly and complex when equipped for regeneration, and long-term durability does not exist. An ideal catalytic trap would be simple, economical, and reliable; it would be self-regenerating; and it would impose a minimum increase in fuel consumption.

Modern Diesel Engines

Carbon soot particulate generation has been greatly reduced in modern CI engines by advanced design technology in fuel injectors and combustion chamber geometry. With greatly increased mixing efficiency and speeds, large regions of fuel-rich mixtures can be avoided when combustion starts. These are the regions where carbon soot is generated, and by reducing their volume, far less soot is generated. Increased mixing speeds are obtained by a combination of indirect injection, better combustion chamber geometry, better injector design and higher pressures, heated spray targets, and air-assisted injectors. Indirect injection into a secondary chamber that promotes high turbulence and swirl greatly speeds the air-fuel mixing process. Better nozzle design and higher injection pressures create finer fuel droplets which evaporate and mix quicker. Injection against a hot surface speeds evaporation, as do air-assisted injectors.

Some modern, top-of-the-line CI automobile engines (e.g., Mercedes) have reduced particulate generation enough that they meet stringent standards without the neea for particulate traps.

CHEMICAL METHODS TO REDUCE EMISSIONS

Development work has been done on large stationary engines using cyanuric acid to reduce NOx emissions. Cyanuric acid is a low-cost solid material that sublimes in the exhaust flow. The gas dissociates, producing isocyanide that reacts with NOx to form N2, H20, and CO2. Operating temperature is about 500°C. Up to 95% NOx 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 NOx emissions. These are materials that absorb selected molecular compounds and catalyze chemical reactions. Using both SI and CI engines, the efficiency of NOx 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 H20 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 S02 when operation is lean and excess oxygen exists. The reaction equation is:

$$H2S + 02 \sim S02 + H2 \qquad 28)$$

Ammonia Injection Systems

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

$$4\,NH3 + 4\,NO + 02 \sim 4\,N2 + 6\,H20 \qquad 29)$$

$$6\,N02 + 8\,NH3 \sim 7\,N2 + 12\,H20 \qquad 30)$$

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, inclMdingship engines.

Ammonia injection systems are not practical in automobiles or on other smaller engines. This is because of the needed NH3 storage and fairly complex injection and control system.

EXHAUST GAS RECYCLE-EGR

The most effective way of reducing NOx 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 nonreacting gas would work as a diluent, as shown in Fig. 9. Those gases with larger specific heats would absorb the most energy per unit mass and would therefore require the least amount; thus less C02 would be required than argon for the same maximum temperature. However, neither C02 nor argon is readily available for use in an engine. Air is available as a diluent but is not totally nonreacting. Adding air changes the AF and combustion characteristics. The one nonreacting 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.

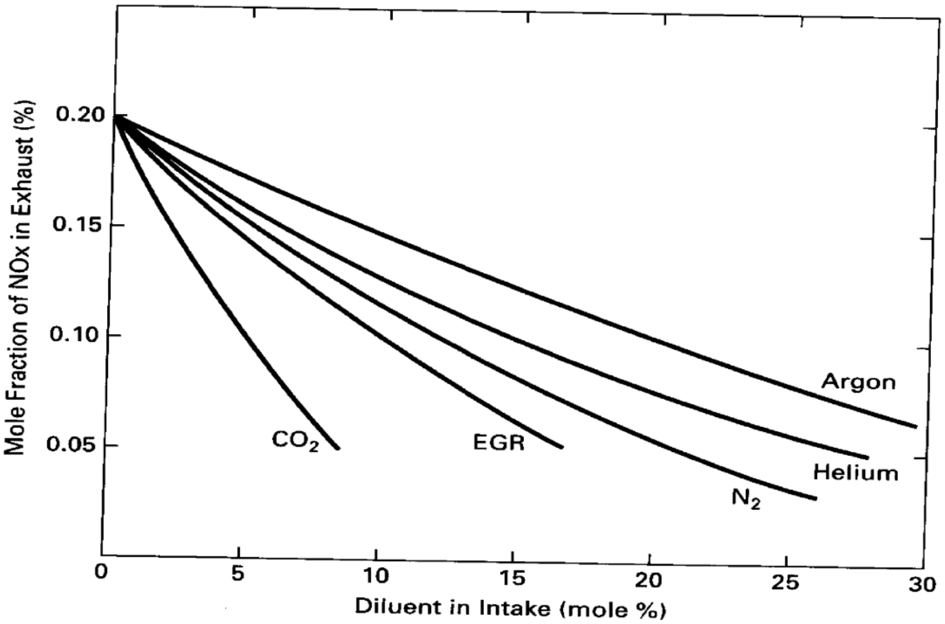


Figure 9 Reduction of NOx generation by adding non combustible diluent gas to intake mixture. Adding any non reacting neutral gas to the inlet air–fuel mixture reduces flame temperature and NOx generation. Exhaust gas (EGR) is the one gas that is readily available for engine use.

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 residualleft in the cylinder from the previous cycle to effectively reduce the maximum combustion temperature. The flow rate of

EGR is controlled by the EMS. EGR is defined as a mass percent of the total intake flow:

$$EGR = [mEGR/mJ\,(100) \qquad 31)$$

where *m;* is the total mass flow into the cylinders.

After EGR combines with the exhaust residual left from the previous cycle, the total fraction of exhaust in the cylinder during the compression stroke is;

$$Xex = (EGR/lOO)\,(1 - xr) + Xr \qquad 32)$$

where *Xr* is the exhaust residual from previous cycle.

Not only does EGR reduce the maximum temperature in the combustion chamber, but it also lowers the overall combustion efficiency. Figure 10 shows that as EGR is increased, the percent of inefficient *slow-burn* cycles increases. Further increase in EGR results in some cycle *partial burns* and, in the extreme, total mis306 fires. Thus, by using EGR to reduce NOx 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 NOx 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 a 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.

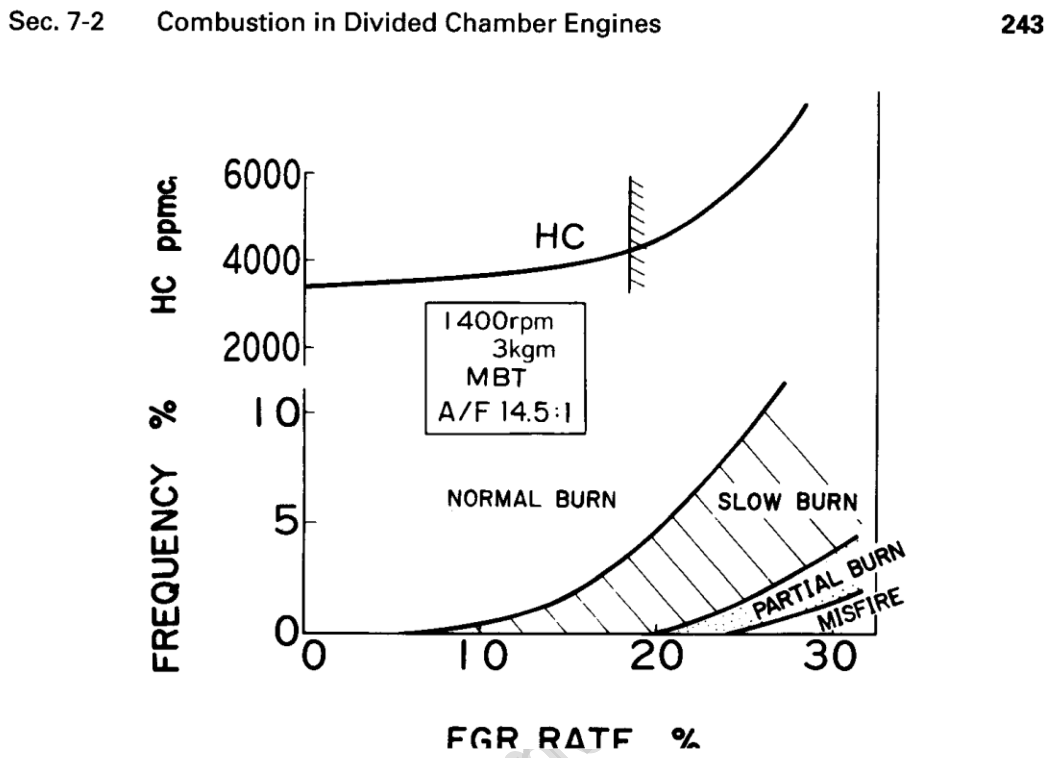


Figure 10 Effect of EGR on quality of combustion in an SI engine and hydrocarbon emissions in the exhaust. With no EGR, most cycles will have normal burn time. As the percent of EGR is increased, there will be an increase in cycles with slow burn or partial burn. With too much EGR, combustion in some cycles will die out, resulting in a misfire.

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 vapor 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 vapor-collecting nozzles that reduce HC vapor lost to the atmosphere during refueling.

INTERNAL COMBUSTION ENGINES

Syllabus

1. Engine classification and engine components.
2. Air standard cycles.
3. Cycles having the Carnot cycle efficiency.
4. Comparison of Otto, Diesel and dual cycles.
5. Fuel- Air cycles; variation of specific heats.
6. Fuel- Air cycles; effect of engine variables.
7. Internal combustion engines fuels.
8. Combustion; basic chemistry.
9. Combustion; stoichiometry.
10. Combustion; exhausts gas analysis.
11. Combustion; Dissociation.
12. Combustion; internal energy, enthalpy of combustion and enthalpy of formation, and calorific value of fuels.
13. Real cycles.
14. Spark ignition engine and its components.
15. Compression ignition engine and its components.
16. Four stroke engine.
17. Two stroke engine
18. Scavenging in two - stroke engines.
19. Rotary engines; Gas turbine.
20. Rotary engines; Wankel engine.
21. Engine testing and performance; performance parameters.
22. Performance characteristics.
23. Testing and basic measurement of I.C. engines.

24. Supercharging and its effect on engine performance.
25. Turbo - charging.
26. I.C.E. fuel systems, carburetion, fuel injection.
27. Valves (ports) timing.
28. Firing order.

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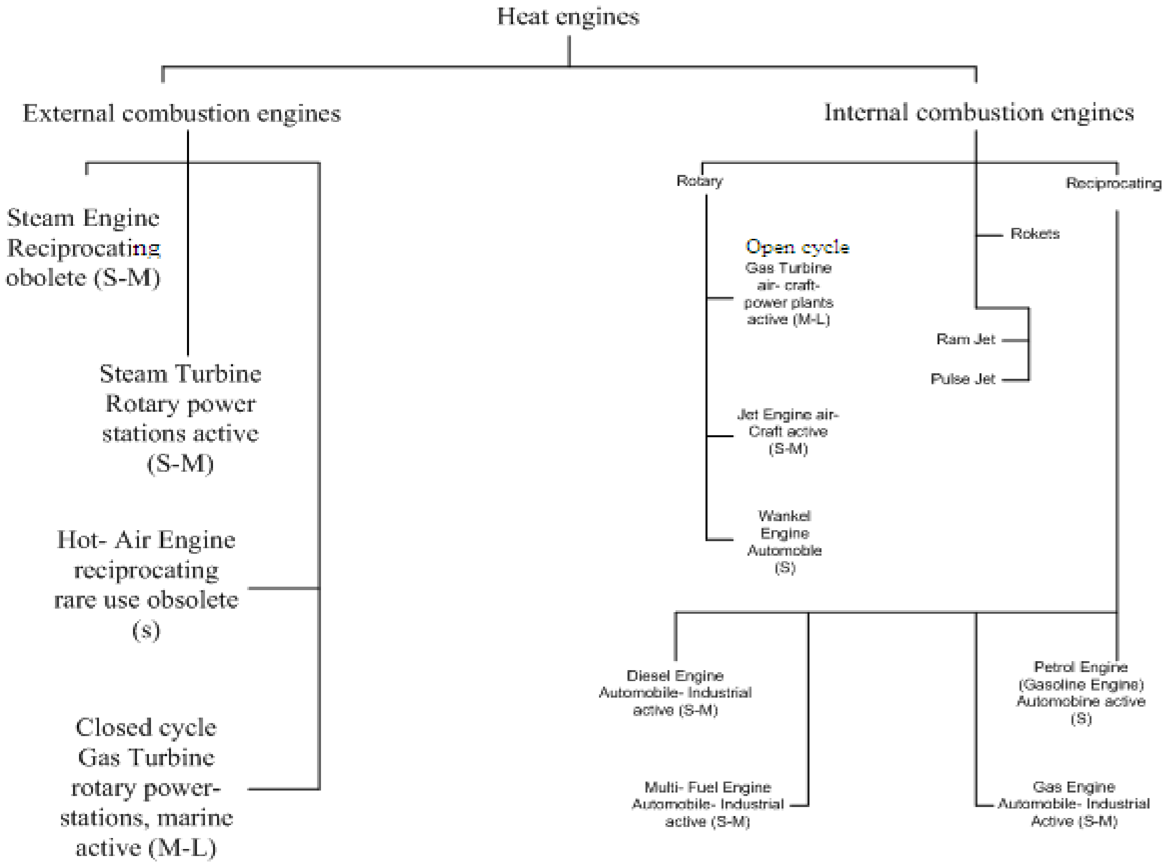
CHAPTER (1)

ENGINE CLASSIFICATION

Classification of Heat Engines:



L: Large over 10000 KW
M: Medium between 1000- 10000 KW
S: Small under 1000 KW

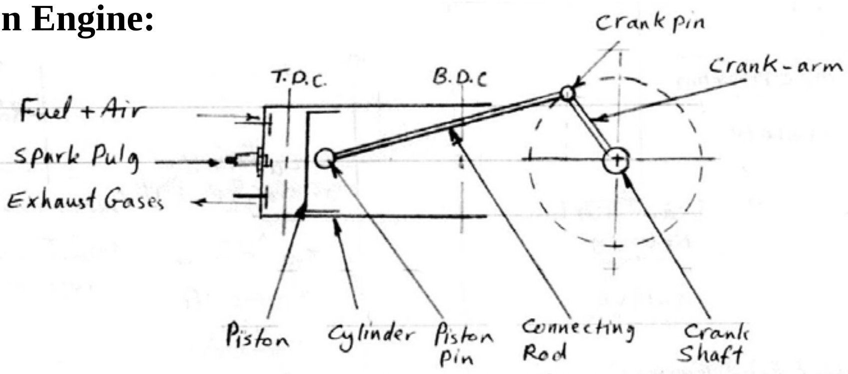
Figure (1-1): Engine Classification

Heat engines can be classified as in figure (1-1); external combustion type in which the working fluid is entirely separated from the fuel- air mixture (ECE), and the internal - combustion (ICE) type, in which the working fluid consists of the products of combustion of the fuel- air mixture itself.

Comparison between the Different Kinds:

1. The Reciprocation Piston Engine:

Figure (1-2): Diagrammatic representation of reciprocating piston engine



2. Open Cycle Gas Turbine:

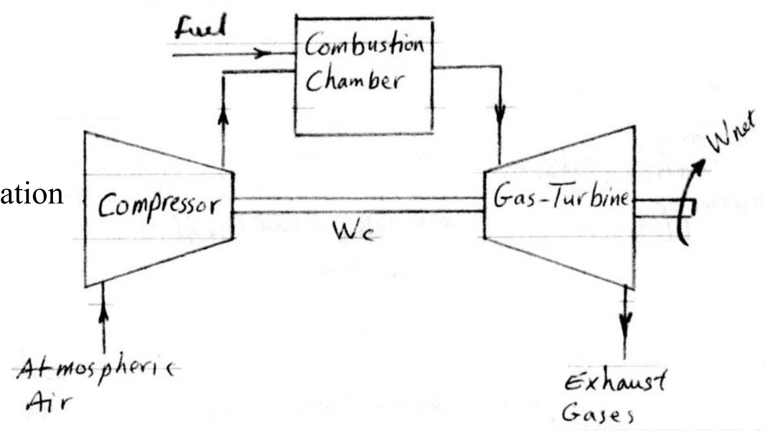


Figure (1-3): Diagrammatic representation of gas turbine

3. The Wankel Engine:

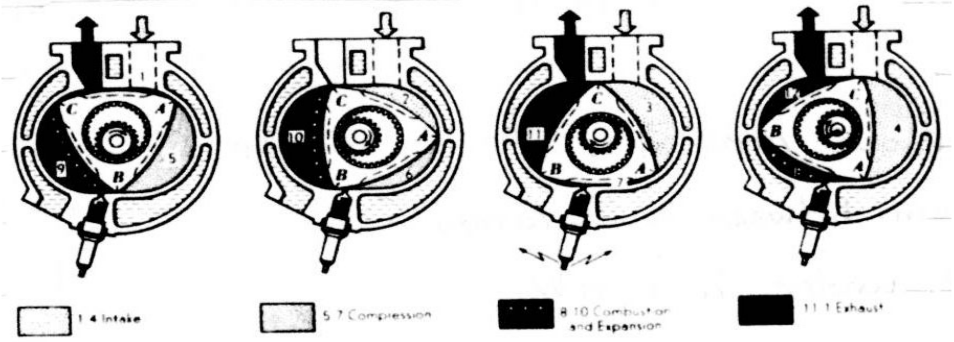


Figure (1-4): *Wankel four- process cycle*

4. Steam Power Plant:

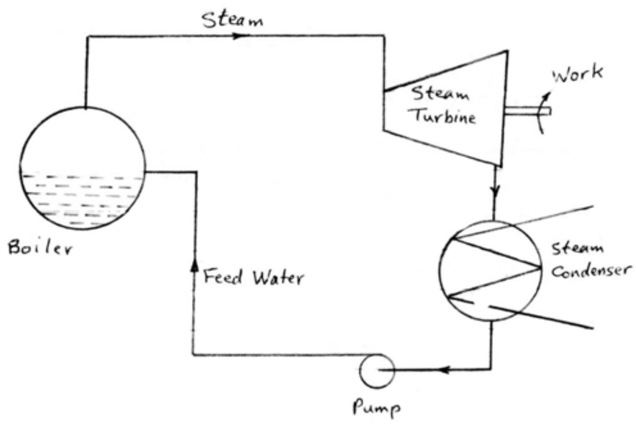


Figure (1-5): *Diagrammatic representation of steam power plant*

The Reciprocating I.C.E. is one unit and does not need other devices, the efficiency of the engine is relatively high, and the fuel used is relatively expensive.

The gas turbine group needs a compressors, its weight is smaller than reciprocating I.C.E. of the same power, its efficiency is lower, the fuel relatively cheap, and it is suitable for air craft.

Rotary engine is a substitute for the reciprocating I.C.E. Wankel engine has a three lobe rotor which is driven eccentrically in a casing in such a way that there are three separate volumes trapped between the rotor and the casing. These volumes perform induction, compression, combustion, expansion and exhaust process in sequence. This design has a good power/volume ratio. Seal wear and heat transfer, were some of the initial development problems of the Wankel engine. These problems have now been largely solved.

The steam turbine is suitable for very large powers, its efficiency is reasonable; the fuel used in the boiler is cheap. The steam turbine needs a boiler, condenser and a continuous supply of water.

Modern Developments of I.C.E.:

The modern I.C.E. is a product of research and developments extending over a long period of time many engines were proposed and tested, these include:

1. Stratified charge engine.
2. Dual Fuel and Multi- Fuel engines.
3. Sterling engine.
4. Free Piston engine.
5. Variable compression Ratio engine.
6. Combination of reciprocating engine with gas turbine.

Advantages of I.C.E. over E.C.E.:

1. More mechanical simplicity and lower weight/power ratio.
2. They do not need auxiliary equipment, such as boiler & condenser.
3. They could be started and stopped in a short time.
4. Their thermal efficiency is higher than other heat engines.
5. Their initial cost is low.

These advantages make I.C.E. more suitable in the transport sector; motor cars, small ships, submarines, and small aircrafts.

Types of Internal Combustion Engines:

I.C.E. can be divided into several groups according to different features as characteristics: operating cycles, method of charging the cylinder, fuel used, general design (position and number of cylinders, method of ignition, rotating speed, etc.), and method of cooling the engine.

Fuel Used:

1. **Liquid Fuels**
   a. Volatile liquid fuels: petrol, Alcohol, benzene. Fuel /Air mixture is usually ignited by a spark; Spark ignition.

b. Viscous liquid fuels: fuel oil, heavy and light diesel oil, gas-oil, bio-fuels. Usually combustion of fuel takes place due to its contact with high temperature compressed air (self-ignition); Compression ignition.

2. **Gaseous fuels:** Liquid Petroleum Gas (LPG), Natural gas (NG), Town gas, Blast Furnace gas; ignition usually by a spark.

3. **Dual and Multi-Fuels:**

Dual fuel engines are operated with two types of fuels, either separately or mixed together. Multi-fuel engines could be operated by a mixture of more than two fuels, gaseous; such as: Hydrogen, methane, L.P.G. etc., combined with one or more of liquid fuels, such as alcohol, ethers, esters, gasoline, diesel etc...

Method of Charging the Engine:

1. Naturally aspirated engine: Admission of charge at near atmospheric pressure.
2. Supercharged engine: Admission of charge at a pressure above atmospheric.

Number of Piston Stroke to Complete the Thermodynamic Cycle:

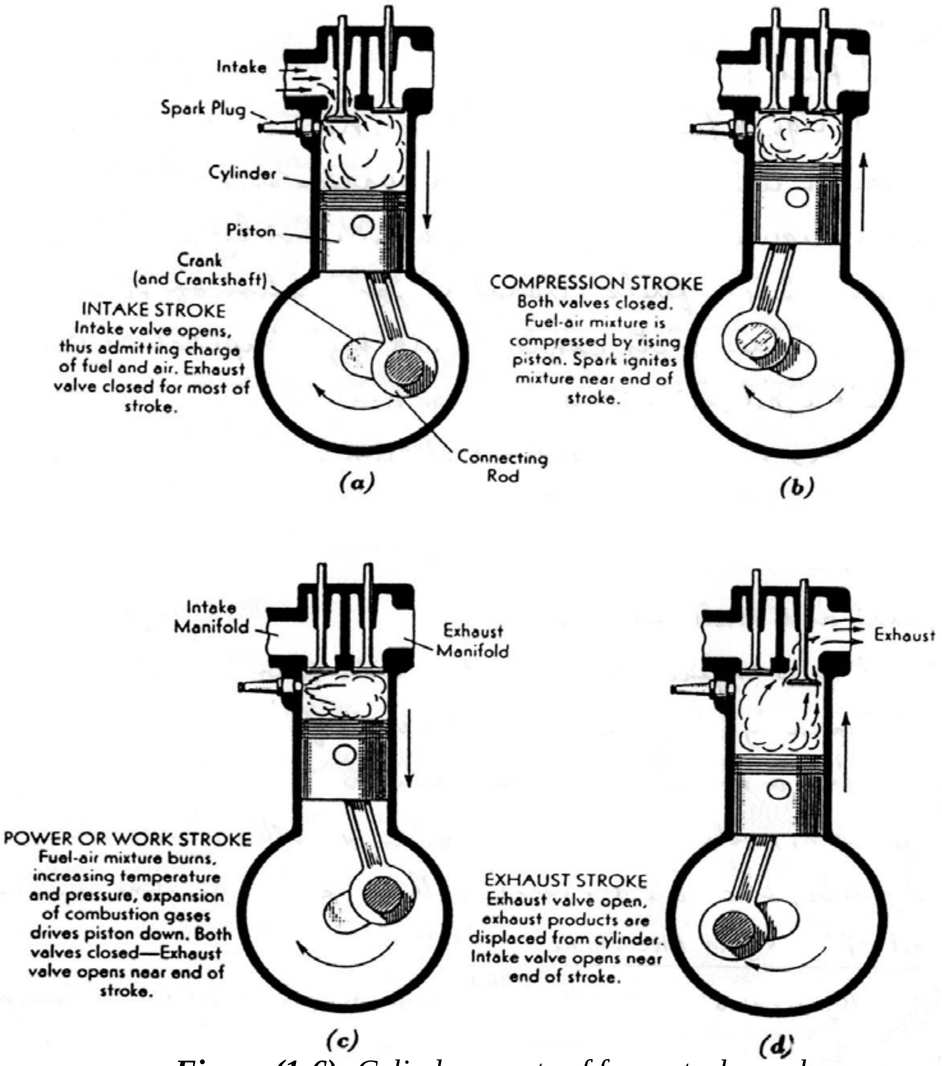


***Figure (1-6):** Cylinder events of four- stroke cycle*

Four stroke engines: the cycle of operation is completed in four strokes of the piston or two revolution of the crank shaft. Each stroke consists of 180° of crank shaft rotation. The series of operations are shown in figure (1 – 6):

1. Two strokes engine: In two-stroke engine the cycle is completed in two strokes, i.e. one revolution of the crank shaft. The difference between two-stroke and four-stroke engines is in the method of filling the cylinder with the fresh charge and removing the products of combustion. The sequence of events shown in figure **(1 – 7)**:

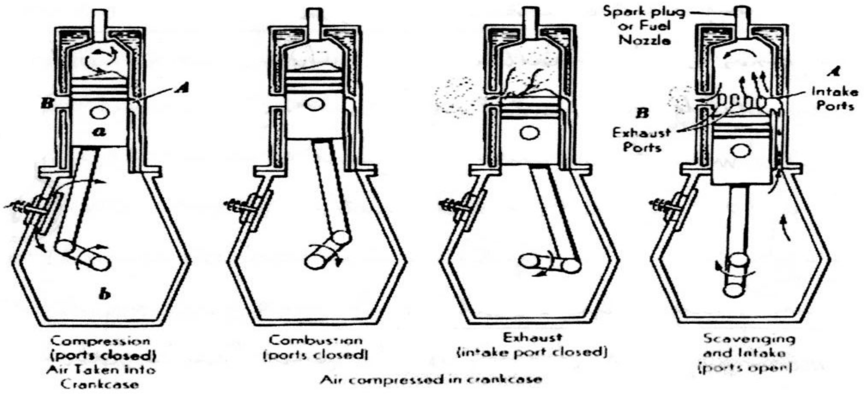
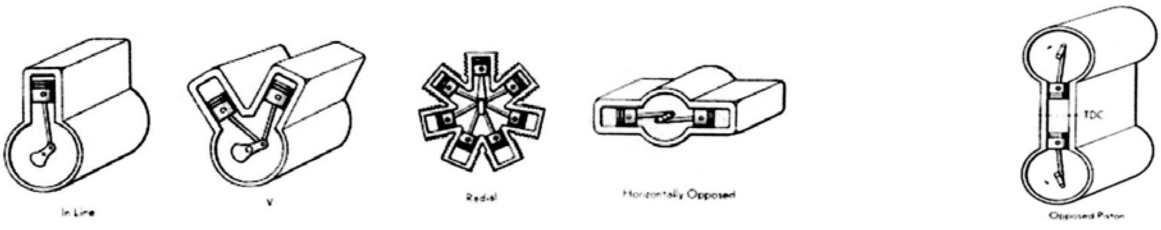


Figure (1-7): *Cylinder events of two- stroke cycle*

Cylinders Arrangement:

1. In- line engines: all cylinders are arranged linearly figure (1 – 8a).
2. "V" engines: cylinders are in two banks inclined at an angle to each other and with one crank-shaft figure (1 – 8b).
3. Radial engine: the radial engine is an engine with more than two cylinders in each row equally spaced around the crank shaft. Normally it is been used in air-crafts figure (1-8c).
4. Opposed cylinder banks located in the same plane on opposite sides of the crank-shaft figure (1 – 8d).
5. Opposed piston engine: when a single cylinder houses two pistons, each of which drives a separate crank shaft figure (1 – 8e).



a b c d e

Figure (1-8): *Classification of engines by cylinder arrangement*

6. Double- acting engine: the fuel burns on both sides of the piston figure.
7. Cylinders may be vertical or horizontal, vertical engines needs smaller area, when area is available horizontal engines may be used, this would make all engine parts accessible and the operation and maintenance are easier.

Classification by valves Location:

The most popular design is the overhead-valve design, there are also an under head valve engines and a combination of the two designs is also used see figure **(1 – 9).**

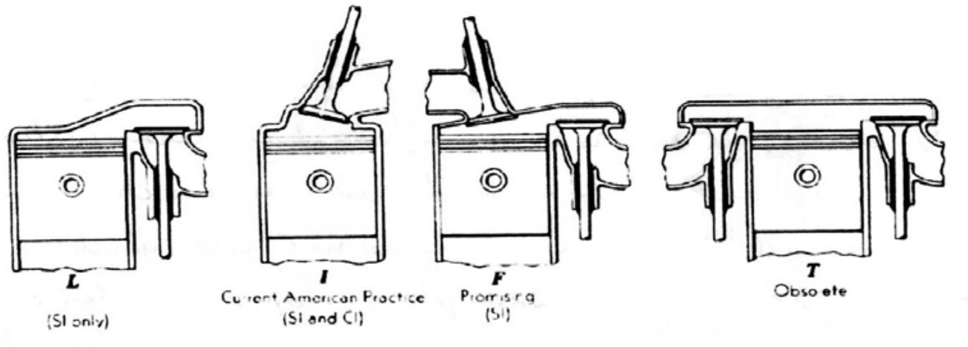


Figure (1-9): *Classification of engines by valve location*

Use of the Engine:

1. Marine engine: for propulsion of ships at sea.
2. Industrial engine: for power generation on land.
3. Automotive engine: for transport.

Method of Cooling the Engine:

1. Water- cooled engines.
2. Air- cooled engines.

The Continuous- Combustion Gas Turbine:

The main components of the gas turbine are; a compressor, a turbine and a combustion chamber, see figure (1 – 10). In operation, air is drawn into the compressor, compressed, and then passed, in part, through the combustion chamber. The high- temperature gases leaving the combustion chamber mix with the main body

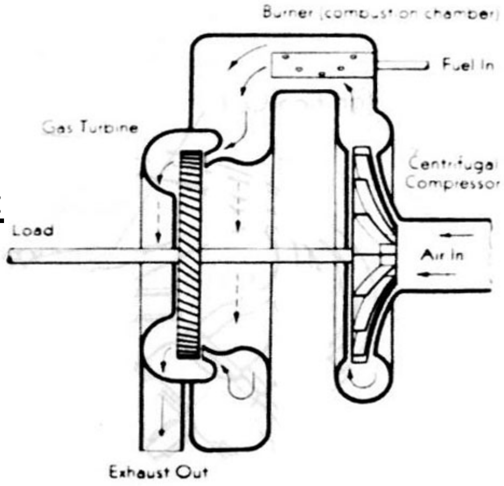
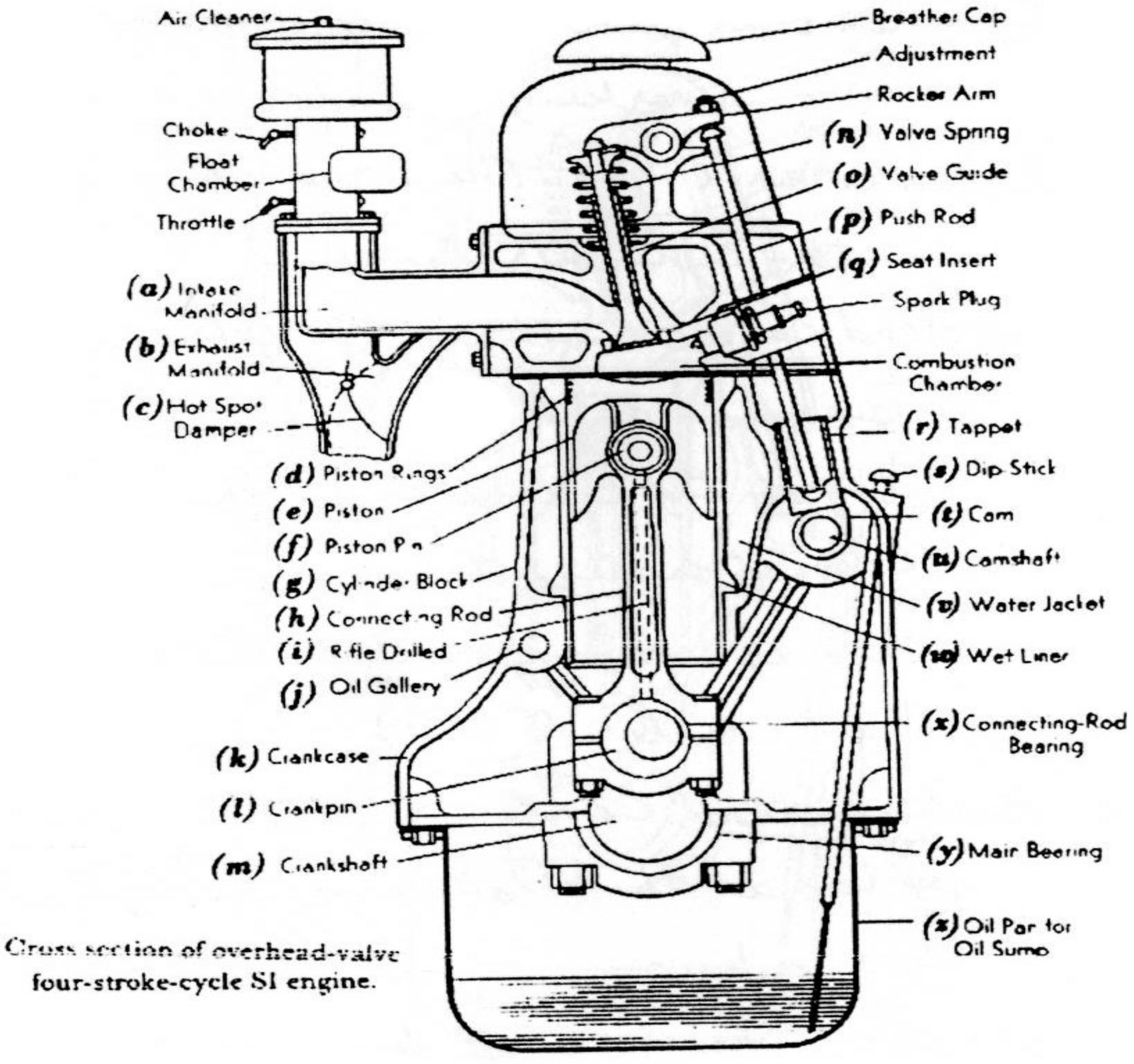


Figure (1-10): Diagrammatic sketch of gas turbine of air flowing around the combustor. This hot gas, with greatly increased volume, is led to a nozzle ring where the pressure is decreased and therefore the velocity is increased. The high- velocity gas is directed against the turbine wheel and the K.E. of the gas is utilized in turning the drive shaft, which also drives the air compressor.

I.C.Engine Parts and Details:

The main components of the reciprocating internal combustion engine are shown in Figure (1-11). Engine parts are made of various materials and perform certain functions, some of which will be explained: cylinder block (g) it is integral with crank case (m), both are made of cast iron. The piston (e) reciprocates inside the cylinder, which include the combustion chamber.



Cross section of overhead-valve four-stroke-cycle SI engine.

Figure (1-11): S.I.engine parts and details (over-head valve)

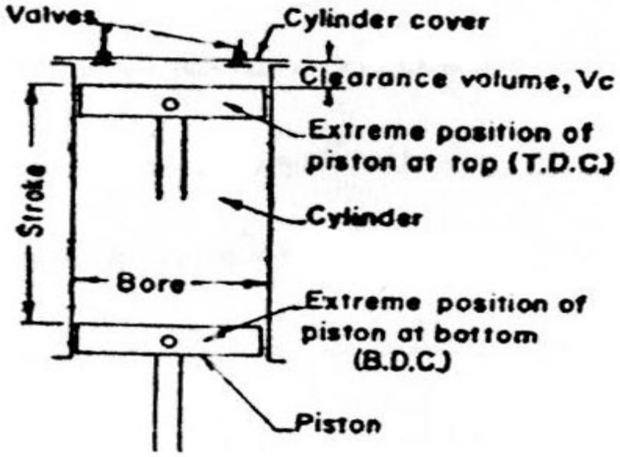
The piston is connected to the connecting rod (h) by piston pin (f). This end of the connecting rod is known as small end. The other end of the connecting rod called the big end is connected to the crank arm by crank pin (l).

Camshaft (u) makes the cam (t) to rotate and move up and down the valve rod through the tappet (r). Mainly each cylinder has two valves; one is admission or suction valve and the other is exhaust valve.

The ignition system consists of a battery, an ignition coil, a distributor with cam and breaker points, and spark plug for each cylinder. In diesel engines there is an injection system instead of ignition system.

Internal Combustion Engines Terminology:

1. Cylinder bore (B): The nominal inner diameter of the working cylinder.
2. Piston area (A): the area of a circle diameter equal to the cylinder bore.
3. Top Dead Center (T.D.C.): the extreme position of the piston at the top of the cylinder. In the case of the horizontal engines this is known as the outer dead center (O.D.C.).



4. Bottom Dead Center (B.D.C.): the extreme position of the piston at the bottom of the cylinder. In horizontal engine this is known as the Inner Dead Center (I.D.C.).
5. Stroke: the distance between TDC and BDC is called the stroke length and is equal to double the crank radius (l).
6. Swept volume: the volume swept through by the piston in moving between TDC and is denoted by Vs:

$$Vs = \frac{\pi}{4} d^2 \times l$$

Where d is the cylinder bore and l the stroke.

7. Clearance volume: the space above the piston head at the TDC, and is denoted by Vc:

$$\text{Volume of the cylinder:} \quad V = Vc + Vs$$

8. Compression ratio: it is the ratio of the total volume of the cylinder to the clearance volume, and is denoted by (r)

$$r = \frac{V}{Vc} = \frac{Vc + Vs}{Vc}$$

9. Mean piston speed: the distance traveled by the piston per unit of time:

$$V_m = \frac{2lN}{60} m/s$$

Where l is the stroke in (m) and N the number of crankshaft revolution per minute (rpm).

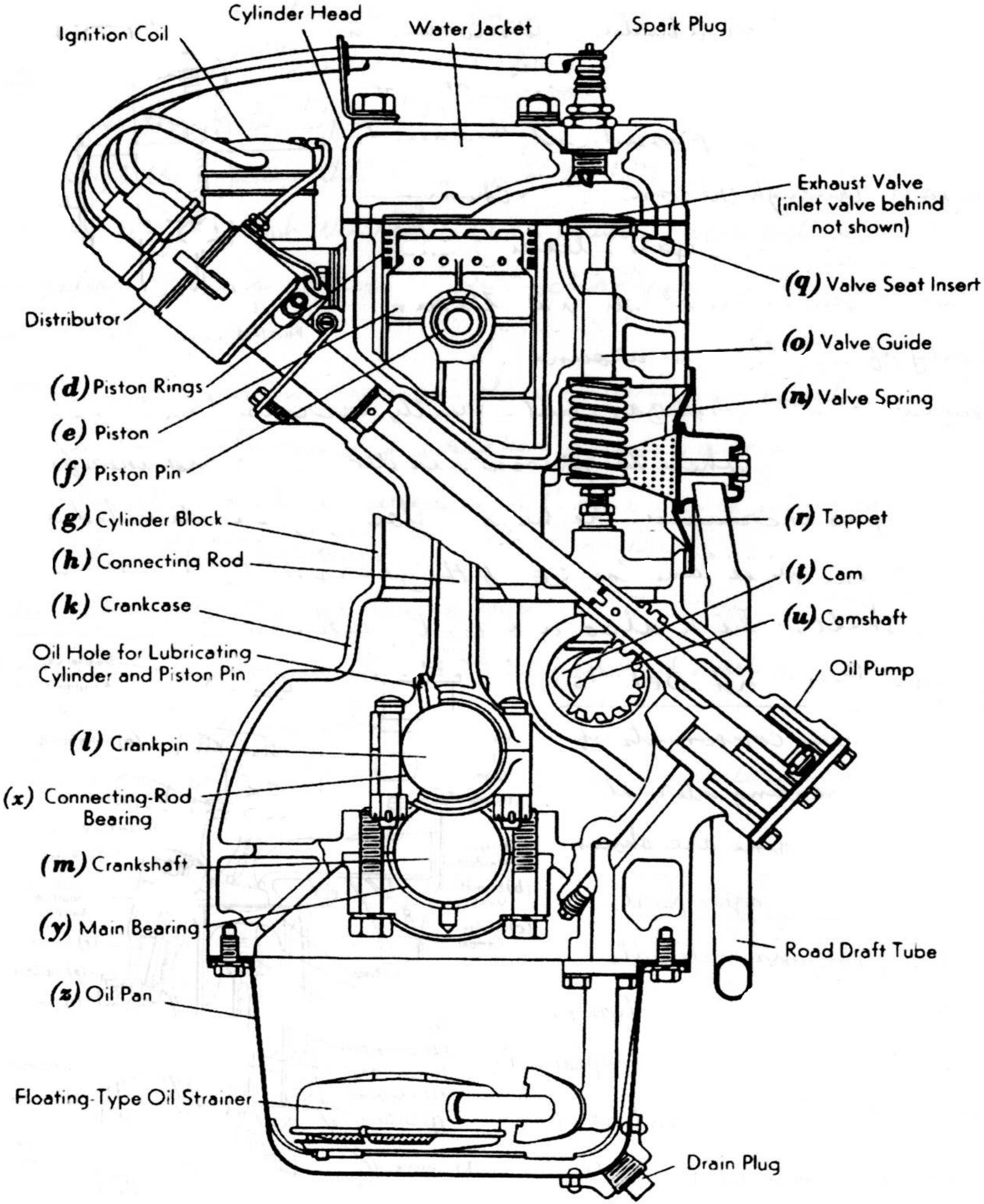


Figure (1-12): *Piston and cylinder geometry (under- head valve)*

CHAPTER (2)

AIR - STANDARD CYCLES

During every engine cycle, the medium changes sometimes it is a mixture of fuel and air or products of combustion, the specific heats and other properties of the medium change with temperature and composition.

The accurate study and analysis of I.C.E. processes is very complicated. To simplify the theoretical study "Standard Air Cycles" are introduced, these cycles are similar to the open cycles, but some simplifying assumptions are made:

1. Cylinder contains constant amount of air and it is treated as ideal gas.
2. The specific heats and other physical and chemical properties remain unchanged during the cycle.
3. Instead of heat generation by combustion, heat is transformed from external heat source.
4. The process of heat removal in the exhaust gases is represented by heat transfer from the cycle to external heat sink.
5. There is neither friction nor turbulence; all processes are assumed to be reversible.
6. No heat loss from the working fluid to the surroundings.
7. Cycles can be presented on any diagram of properties.

The Otto Cycle (constant Volume):

These cycles is applied in petrol (or gasoline) engine, gas engine, and high speed diesel (oil) engine. The cycle is shown in Figure (1 – 13), and consists of the following processes:

1. Process 1 to 2 is isentropic compression;

$$Q_{12} = 0.$$

2. process 2 to 3 is reversible heat addition at constant volume

$$Q_{23} = C_v (T_3 - T_2) \text{ kJ/kg}$$

3. process 3 to 4 is isentropic expression;

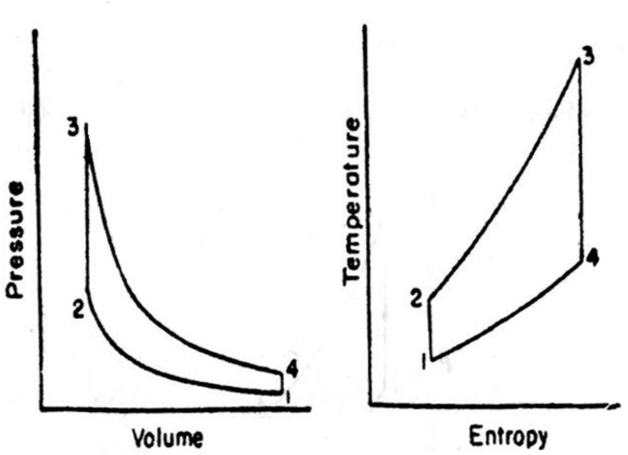


Figure (1-13)

$Q_{34} = 0$

4. process 4 to 5 is reversible constant volume cooling

$Q_{41} = C_v (T_4 - T_1)$ kj/ kg

This cycle is applied in 4- stroke and 2- stroke engines.

Work of Otto cycle = $W_{34} - W_{12}$

$$\eta = \frac{Q^{\uparrow} - Q^{\downarrow}}{Q^{\downarrow}} = 1 - \frac{Q^{\uparrow}}{Q^{\downarrow}}$$

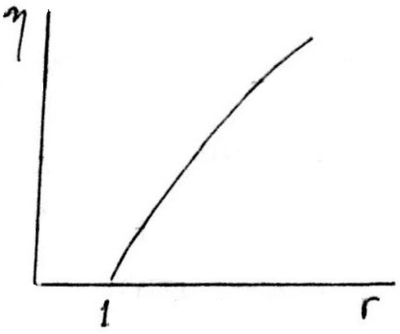
The thermal efficiency of the Otto cycle:

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

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

$T_3 = T_4 = r^{\gamma-1}$ and $T_2 = T_1 r^{\gamma-1}$

$$\therefore \eta = 1 - \frac{T_4 - T_1}{(T_4 - T_1) r^{\gamma-1}} = 1 - \frac{1}{r^{\gamma-1}}$$



η increased by increasing r

η increased by increasing γ

η independent on the heat added or load.

In modern petrol engines (r) reaches a value of 12.

To make use of that part of the energy in the exhaust gases, they may be expanded to atmospheric pressure in an exhaust gas turbine; the work of the cycle will be increased by the aria $14\bar{4}1$ as shown in figure (1-14). The new cycle $(1234\bar{4}1)$ is called **Atkinson cycle**, this cycle is applied in a combination of petrol engine and gas turbine, the turbine in usually used to drive a compressor to supercharge the engine.

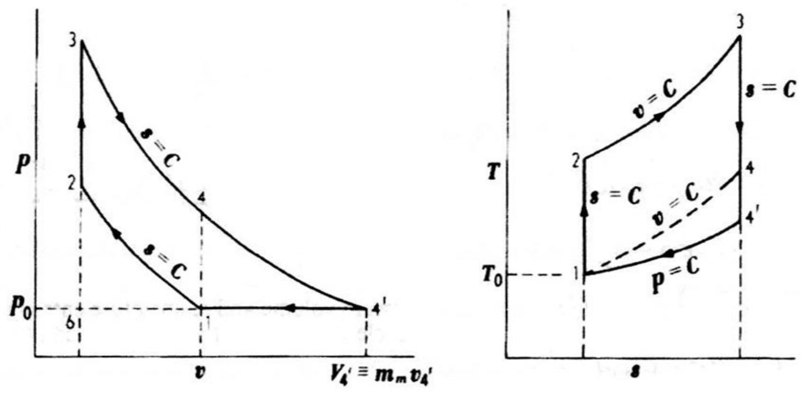


Figure (1-15): *Air- standard complete- expansion cycle*

The Diesel (or Constant Pressure) Cycle:

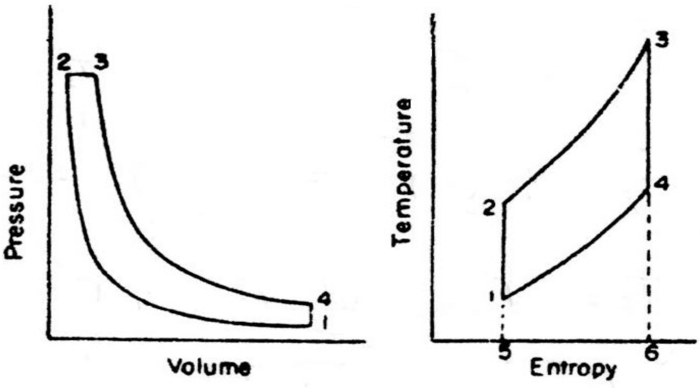


***Figure (1-16):** pressure volume and temperature entropy diagram of a cycle with constant pressure heat addition*

This cycle is the theoretical cycle for compression-ignition or diesel engine. For this cycle:

$$Q_1 = C_p\,(T_3 - T_2)$$
$$Q_2 = C_v\,(T_4 - T_1)$$

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{1}{\gamma}\left(\frac{T_4 - T_1}{T_3 - T_2}\right) = 1 - \frac{T_1}{\gamma T_2}\left(\frac{\frac{T_4}{T_2} - 1}{\frac{T_3}{T_2} - 1}\right)$$

For isentropic compression and expansion:

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1} \quad \text{and,} \quad \frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{\gamma-1}$$

For constant Pressure heat addition 2-3: $\frac{T_3}{T_2} = \frac{v_3}{v_2}$

Thus:

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}\left(\frac{v_3/v_2}{v_2/v_1}\right)^{\gamma-1} = \frac{v_3}{v_2}\left(\frac{v_3}{v_2}\right)^{\gamma-1} = \left(\frac{v_3}{v_2}\right)^{\gamma} \quad , \qquad v_4 = v_1$$

By substitution:

$$\eta = 1 - \frac{1}{\gamma\left(\frac{v_1}{v_2}\right)^{\gamma-1}}\left(\frac{\left(\frac{v_3}{v_2}\right)^{\gamma} - 1}{\left(\frac{v_3}{v_2}\right) - 1}\right)$$

$\frac{v_1}{v_2} = r$ and $\frac{v_3}{v_2} = \beta$, then:

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

This equation shows that thermal efficiency depends not only on r but also on the cut-off ratio $\left(\beta = \frac{v_3}{v_2}\right)$ and the working medium properties γ. As β increase the work done per cycle increase but η decreases. When r increase more than 22, the increase in η is small, on the other hand, maximum pressure increases much and mass of the engine increases.

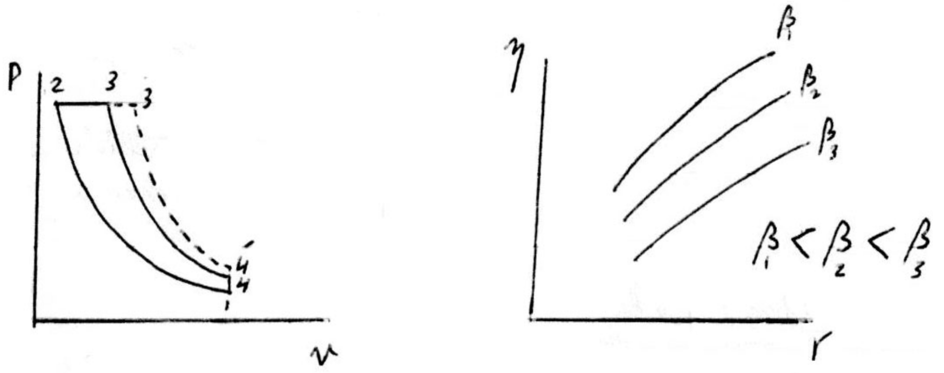


Figure (1-17)

The Dual Cycle:

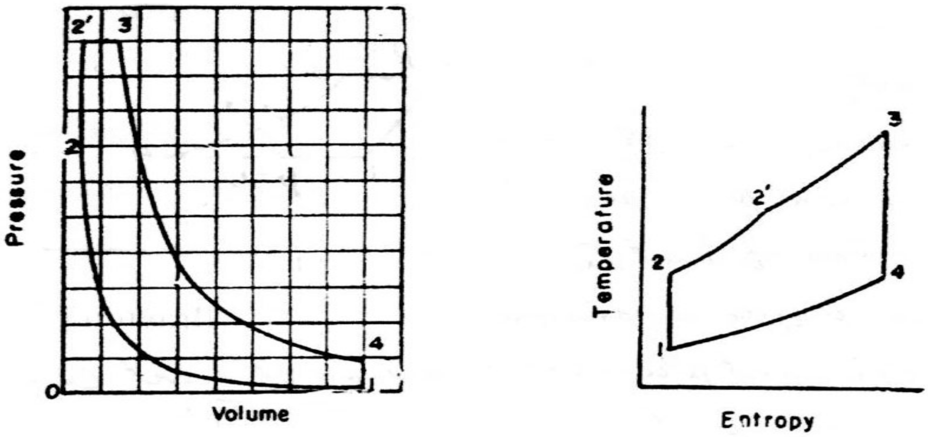


Figure (1-18): *Pressure-Volume and Temperature-Entropy diagram of dual cycle*

The cycle is applied in medium speed and high speed diesel engines. The engine may be 4 or 2 strokes

$$Q_1 = Q_1' + Q_1''$$

$$= C_v\left(T_2' - T_2\right) + C_p\left(T_3 - T_{2'}\right)$$

$$Q_2 = C_v(T_4 - T_1)$$

$$\eta = 1 - \frac{Q_2}{Q_1}$$

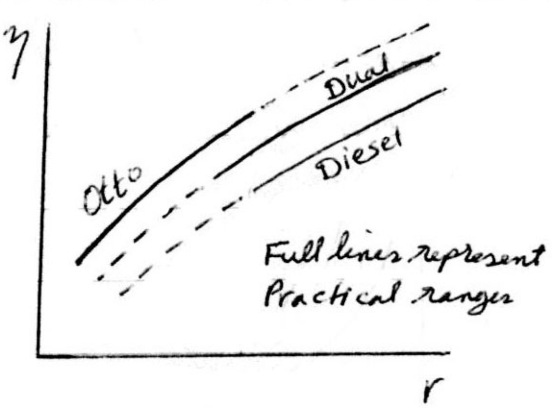
It can be proved that:

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

$$k = \frac{p_{2'}}{p_2} \text{ and } \beta = \frac{V_3}{V_{2'}}$$

When k = 1, then $p_{2'} = p_2$, we obtain diesel cycle. When β = 1, then $V_3 = V_{2'}$, we obtain Otto cycle. The indicated thermal efficiency of this cycle lies between that of the Otto and diesel.

Figure (1-19)



Joule (or Brayton) Cycle:

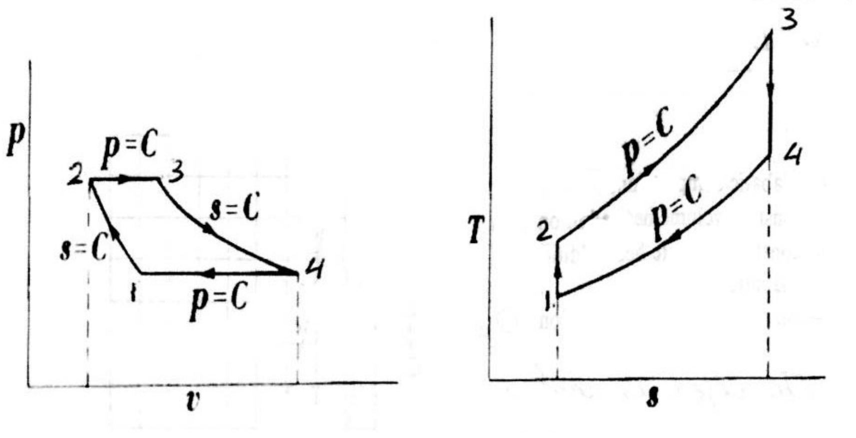


Figure (1-20): *Brayton (Joule) air- standard cycle*

This cycle is used in gas turbine, it consists of:

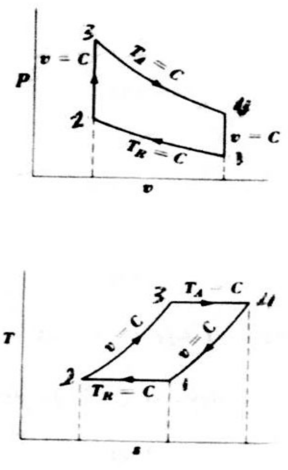
ab: isentropic compression.
bc: constant pressure addition of heat Q_1.
cd: isentropic expansion.
da: constant pressure rejection of heat Q_2.

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{Cp(T_4 - T_1)}{Cp(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

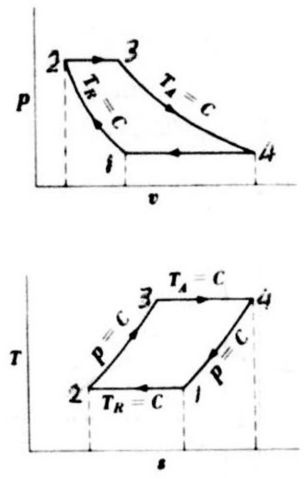
$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \frac{1}{r^{\frac{\gamma-1}{\gamma}}}$$

Regenerative Cycles:

1. Sterling Cycle: It consists of two isothermal; 12 & 34 and two constant volume 2-3 & 41. Heat is added in the constant 2 – 3 and also in the isothermal process 3 – 4. Not that the temperature of initial heat rejection is T_4 is higher than the initial temperature of heat addition T_2, therefore, it is possible to use a heat exchanger to utilize part of the rejected heat to the heat addition process. If the Stirling cycle could be perfectly regenerated, no heat would be necessary for process 23.



Sterling Cycle



Ericsson Cycle

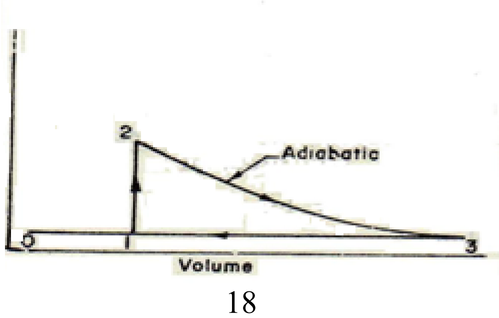
2. Ericsson Cycle: Same comments can be made for Ericsson cycle. Heat is added in the expansion at constant pressure 23 and at constant temperature 34. Heat is rejected in the compression process at constant pressure 41 and at constant temperature 12. Since 23 and 41 are parallel, then gas can be heated from 2 to 3 by cooling from 4 to 1.

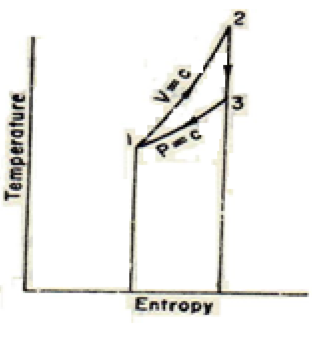
Lenoir Cycle:

$Q_{add} = C_v (T_2 – T_1)$

$Q_{rej} = C_p (T_3 – T_1)$

$\eta = 1 - \gamma\frac{(T3 - T1)}{(T2 - T1)}$



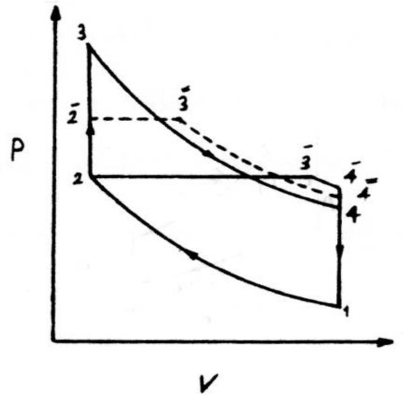


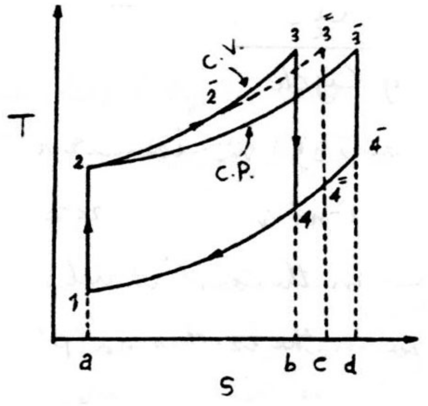
Comparison of Otto, diesel and Dual Cycles:

The important parameters in cycle analysis are compression ratio, maximum pressure, maximum temperature, head input, work output, etc.

1. *Equal Compression Ratio and Equal Heat added:*

   The three cycles start from the same point (1); compression process is the same (12) for all cycles, equal heat added means:





$$area\ a23ba = area\ a2\bar{3}da = area\ a2\bar{2}\bar{\bar{3}}ca$$

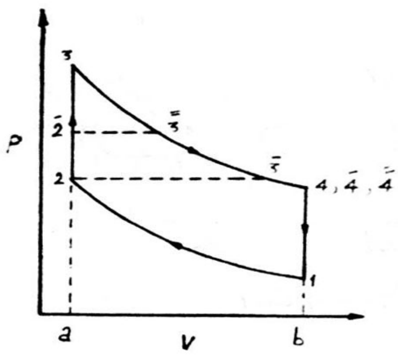
areas representing heat rejected are:

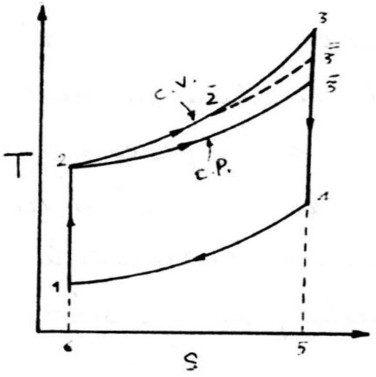
$$area\ a14ba < area\ a1\bar{\bar{4}}ca < area\ a1\bar{4}da$$

$$\eta = \frac{heat\,aded - heat\,rejected}{heat\,added}$$

$$\therefore \eta_{Otto} > \eta_{dual} > \eta_{diesel}$$

2. *Equal Compression Ratio and Heat rejected:*





Processes (12) and (41) are the same in all cycles.

Areas representing heat added are:

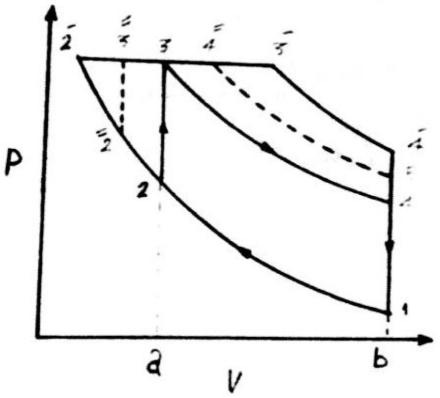
$area\,6235 > area\,62\bar{2}\bar{\bar{3}}5 > area\,62\bar{3}5$

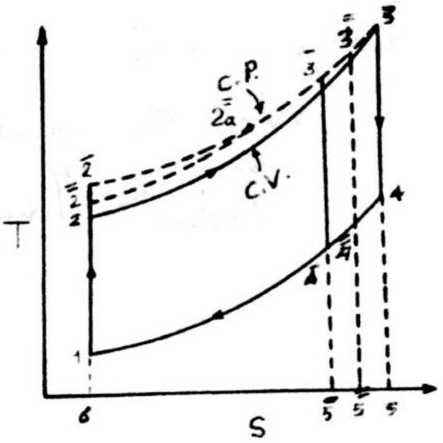
Heat rejected area is the same for the three cycles

$\therefore$ $W_{Otto} > W_{dual} > W_{diesel}$, and

$\eta_{Otto} > \eta_{dual} > \eta_{diesel}$

3. Equal Maximum Pressure and Equal Heat added:





Point (1) is common in the three cycles.

Maximum pressure is the same, therefore, compression ratio are different as indicated in the diagram above.

Areas representing heat added: $6235 = 6\bar{2}\bar{\bar{2}}a\bar{\bar{3}}\bar{\bar{5}} = 6\bar{2}\bar{3}\bar{5}$

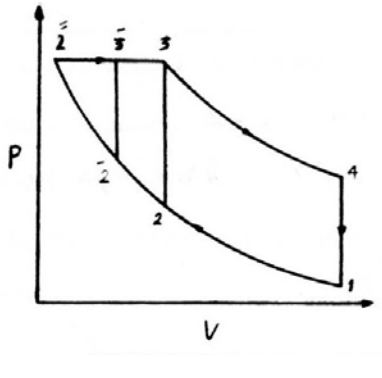
Areas representing heat rejected: $6145 > 61\bar{\bar{4}}\bar{\bar{5}} > 61\bar{4}\bar{5}$

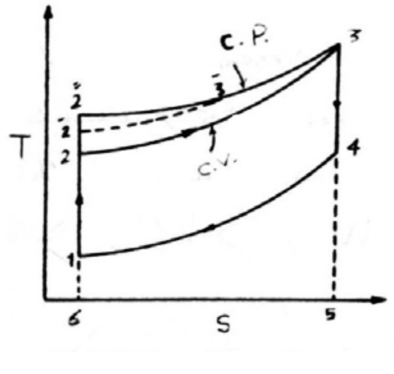
$\therefore$ $W_{Otto} < W_{dual} < W_{diesel}$

i.e.

$\eta_{Otto} < \eta_{dual} < \eta_{diesel}$

4. Equal Maximum Pressure and Maximum Temperature:





Points 1, 3, 4 are common in the three cycles

Heat rejected is the same in the three cycles

Heat added is highest in diesel cycle:

Area $6235 < 6\bar{2}\bar{3}35 < 6\bar{\bar{2}}35$

$\therefore W_{Otto} < W_{dual} < W_{diesel}$

i.e.

$\eta_{Otto} < \eta_{dual} < \eta_{diesel}$

CHAPTER (3)

FUEL AND COMBUSTION

Fuels:

Fuel is any material when burnt will produce heat energy. Various fuels commonly used are as follows:

1. Solid Fuels:

Coal is the most important solid fuel; there are other types of solid fuels, such as: nuclear fuels, solid wastes (wood, sugar-cane). Coal is divided into groups according to their chemical and physical properties. An accurate chemical analysis for the fuel by mass is called the "ultimate analysis", which gives the percentage of the important elements present in the fuel. Another approximate analysis of fuel called the "proximate analysis" gives the percentage of moisture, volatile matter, and combustible solid (fixed carbon) and ash.

Ultimate Analysis of Coals

Coal	C	H	O	N + S	Ash
Anthracite	90.27	3.00	2.32	1.44	2.97
Bituminous	74.00	5.98	13.01	2.26	4.75
Lignite	56.52	5.72	31.89	1.62	4.25

2. Liquid Fuels:

Liquid fuels are widely used for I.C.E. Practically all liquid fuels have two basic combustible elements; carbon and hydrogen, present separately or in a combination called hydrocarbons, there are principal commercial types of liquid fuels:

i. Petroleum and its Derivatives

They are easy to handle, store, and burn and have nearly constant heating values. They are usually a mixture of hydrocarbons that may be represented by the molecular formula of the form C_nH_m, where m is a function of n that depends upon the family of the hydrocarbon.

Family	**Formula**	**Prefix/suffix**	**structure**
Paraffin (alkanes)	C_nH_{2n+2}	ane	Chain, saturated
Olefin	C_nH_{2n}	ene,ylene	Chain, Unsaturated, one double- bond
Diolefin	C_nH_{2n-2}	diene	Chain, Unsaturated, two double- bond
Aromatic			
Benzene	C_nH_{2n-6}		Ring, Unsaturated
Naphthalene	C_nH_{2n-12}		Double ring, Unsaturated

ii. *Synthetic fuels*

Usually called synfuels, which are liquid and gases mainly produced from coal, oil shale, tar sands and also from various wastes and biomass.

iii. *Alcohols*

The alcohols are a partial oxidation product of petroleum, and are not found to any extent in the crude oil. The compounds are saturated, with a chain structure with the general formula R.OH. Here the radical R is the paraffin group attached to the hydroxyl radical OH. Alcohols are designated by the name of the radical:

CH_3OH: Methyl alcohol or Methanol.
C_2H_5OH: Ethyl alcohol or Ethanol.
C_3H_7OH: Propyl alcohol or Propanol.
C_4H_9OH: Butyl alcohol or Butanol.

3. Gaseous Fuels:

Gaseous fuels are may be either natural or manufactured; natural gas: is a mixture of components, consisting mainly of methane (60-98%) with small amount of other hydrocarbons. In addition it consists; N_2, CO_2, H_2 and traces of other gases. Its sulfur content ranges from very little (sweet) to larger amounts (sour). It is classified as associated or unassociated depending on whether it is associated with oil or not. It is stored as compressed natural gas (CNG) or as liquid (LNG) at pressure ranging between 70 to 210 kPa.

Manufactured Gases Include:

1. Liquid petroleum gas (LPG): It is mixture of propane and butane and some other light hydrocarbons. Propane and butane are the main constituent of LPG; their quantity varies with time of the year and place.
2. Coal gas: it is a by-product obtained during the destructive distillation of coal. Its main approximate composition is CH_4 = 25%, H = 53%, CO = 9%, N = 6%, CO_2 = 2% and other hydrocarbons ≈ 1%.
3. Coke oven Gas: It is produced during the manufacture of coke from new coal in a coke oven where the volatile matter is distilled off and the coke-oven separated from liquids and solids in the volatile matter by cooling and extraction. It consists about 50% H_2, about 30% methane(CH_4), and the remainder of various other gases. Its heating value ranges between 14200 to21300 kJ/m^3.
4. Blast-furnace gas: it is produced as by-product from blast furnaces used in iron reduction process, has about 30%, CO, 2% H_2, 11% CO_2 and about 60% N_2.

There are other gaseous fuels of lesser importance like producer gas, water gas, and town gas.

Advantages of gaseous fuels:

1. Low emissions.
2. Mixing with air is uniform, so they need less excess air.
3. Economical compared to other types of fuels.

Combustion of Fuels:

Combustion of fuel is accomplished by mixing fuel with air at elevated temperature:

Fuel + Air →Products of Combustion + Heat

The oxygen contained in the air unites chemically with carbon, hydrogen and other elements in fuel to produce heat. The amount of heat liberated during the combustion process depends on the amount of oxidation of the constituent of fuel and the nature of fuel.

In order that the combustion of fuel may take place with high efficiency, the following conditions must be fulfilled:

1. The amount of air supplied should be sufficient.
2. The air and fuel should be thoroughly mixed.
3. The temperature of the reactants should be high enough to ignite the mixture.
4. Sufficient time should be available to burn fuel completely.

Combustion Chemistry:

1. Chemical Equation:

The chemical equation shows how the atoms of the reactants are arranged to form products. Before the chemical equation can be written it is necessary to know the number of atoms of elements in the molecules of the reactants and products. During combustion process the atoms are rearranged to form new molecules, and the total number of atoms of each element is unchanged. A chemical equation expresses the principle of the conservation of mass in terms of the conservation of atoms

i- Combustion of Carbon

$C + O_2 \rightarrow CO_2$

$1\ \text{kmol}\ C + 1\ \text{kmol}\ O_2 \rightarrow 1\ \text{kmol}\ CO_2$

$12\text{ kg C} + 32\text{ kg } O_2 \rightarrow 44\text{ kg } CO_2$

If any of the reactants or products is in solid or liquid phase, the volume occupied by them can be neglected. It follows that:

$0\text{ vol. C} + 1\text{vol } O_2 \rightarrow 1\text{vol. } CO_2$

If *insufficient* oxygen is present for all carbon to burn to carbon dioxide, some will burn to carbon monoxide, then:

$C + 1/2\ O_2 \rightarrow CO$

ii- Combustion of Hydrogen

$H_2 + 1/2\ O_2 \rightarrow H_2O$

$1\text{ kmol } H_2 + 1/2\text{ kmol } O_2 \rightarrow 1\text{ kmol } H_2O$

$2\text{kg } H_2 + 16\text{ kg } O_2 \rightarrow 18\text{kg } H_2O$

$1\text{ vol. } H_2 + 1/2\text{ vol. } O_2 \rightarrow 1\text{ vol. } H_2O$ (vapour)

In most engineering combustion systems the necessary oxygen is obtained by mixing the fuel with air (except rockets) and it is necessary to use accurate and consistent analysis of air by mass and by volume. It is usual in combustion calculations to take air as 23.3% O_2, 76.7% N_2 by mass, and 21% O_2, 79% N_2 by volume. The small traces of other gases in dry air are included in the nitrogen, which is sometimes called "*atmospheric nitrogen*".

The moisture or humidity in atmospheric air varies over wide limits, depending on meteorological conditions. Its presence in most cases simply implies an additional amount of inert material.

The molar mass of O_2 can be taken as 32 kg/kmol, and that of N_2 as 28 kg/kmol and air 29 kg/kmol.

Since oxygen is accompanied by nitrogen when air is supplied for combustion, then this nitrogen should be included in the combustion equation, it will appear on both sides of the equation. With one mole of O_2 there are $\frac{79}{21} = 3.762$ moles of N_2, Hence:

$$C + O_2 + \frac{79}{21} N_2 \rightarrow CO_2 + \frac{79}{21} N_2$$

Also

$$H_2 + \frac{1}{2}(O_2 + \frac{79}{21} N_2) \rightarrow H_2O + \frac{1}{2} \times \frac{79}{21} N_2$$

A frequently used quantity in the analysis of combustion process is the *air-fuelratio* A/F. it is defined as the ratio of the mass of air to the mass of fuel for a combustion process.

$$A/F=\frac{m_a}{m_f}=\frac{mass\,of\;air}{mass\,of\;\;fuel}$$

The mass m of a substance is related to the number of moles n through the relation: m = nM, where M is the *molar mass*. The reciprocal of A/F ratio is called the *fuel-air ratio*.

The minimum amount of air needed for the complete combustion of a fuel is called the stoichiometric or theoretical air. In actual combustion processes, it is common practice to use more air than the stoichiometric amount. The amount of extra air than the stoichiometric is called (***excess air***). Amount of air less than stoichiometric amount is called (***deficiency of air***). ***Equivalence ratio*** is the ratio of the actual fuel- air ratio to the stoichiometric fuel-air ratio. Sometimes this ratio is given in term of A/F ratio and called ***mixture strength***.

$$Mixture\,strength=\frac{Stoichiometric\;A/F\;Ratio}{Actual\;A/F\;Ratio}$$

$$Equivalence\,ratio=\frac{Actual\,(F/A)\,ratio}{Stoich.\;(F/A)\,ratio}=\phi$$

$$\phi=\frac{(\mathrm{F/A})\,\mathrm{actual}}{(\mathrm{F/A})\,\mathrm{stoich}}=\frac{(\mathrm{A/F})\,\mathrm{stoich}}{(\mathrm{A/F})\,\mathrm{actual}}$$

Where:

ϕ = 1: stoichiometric

ϕ< 1: lean (week) mixture- excess of air.

ϕ> 1: rich mixture- deficiency of air.

A general reaction equation of a hydrocarbon fuel for stoichiometric condition with air is given by:

$$C_nH_m+\left(n+\frac{m}{4}\right)O_2+3.762\left(n+\frac{m}{4}\right)N_2\rightarrow nCO_2+\frac{m}{2}H_2O+3.762\left(n+\frac{m}{4}\right)N_2$$

The composition of a hydrocarbon fuel C_nH_m are carbon and hydrogen, n and m can be determined for 1 kg of fuel as follows:

$$\frac{Weight\,of\;C\;in\;fuel}{Weight\,of\;\;fuel}=\frac{C}{1}=\frac{12n}{12n+2.02\frac{m}{2}}=\frac{12n}{12n+m}$$

<u>Exhaust and Flue Gas Analysis</u>:

The products of combustion are mainly gaseous. When a sample is taken for analysis it is usually cooled down to a temperature which is below the saturation temperature of the steam present. The steam content is therefore not included in the analysis, which is then quoted as the as the analysis of *dry products*. Since the products are gaseous, it is usual to quote the analysis by volume. An analysis which includes the steam in the exhaust is called a *wet analysis*.

Example (3 – 1):

Find the stoichiometric A/F ratio for the combustion of ethyl-alcohol (C_2H_5OH) in a petrol engine. Calculate the A/F ratios for 0.9 & 1.2 equivalence ratios(ϕ). Determine the wet and dry analyses by volume of the exhaust gas for each equivalence ratio.

Solution:

Combustion equation of ethyl-alcohol is:

$$C_2H_5OH + 3O_2 + 3\times\frac{79}{21}N_2 \rightarrow 2CO_2 + 3H_2O + 3\times\frac{79}{21}N_2$$

One mole of fuel has a mass of (2×12+16+6) =46 kg

Mass of air required for complete burning of one mole of fuel is:

$$\left(3\times 32 + 3\frac{79}{21}\times 28\right) = 412.0\,kg$$

$$\therefore \text{Stoichiometric A/F ratio} = \frac{412}{46} = 8.96/1$$

$$Equivalence\ ratio(\phi) = \frac{(A/F)stoich.}{(A/F)actual}$$

$$0.9 = \frac{8.96}{(A/F)_{actual}}$$

$(A/F)_{actual} = 8.96/0.9 = 9.95$

Volumetric A/F ratio = 3 × (1+3.762) =14.3

For ϕ = 0.9; air supplied is $= \frac{1}{0.9} = 1.11$ times as much air supplied for complete combustion, then: combustion equation becomes:

$$C_2H_5OH + 1.11\left(3O_2 + 3\times\frac{79}{21}N_2\right) \rightarrow 2CO_2 + 3H_2O + 0.11\times 3O_2 + 1.11\times 3\times\frac{79}{21}N_2$$

i.e. The total number of moles of products = 2+3+0.33+12.54 =17.87.

Hence wet analysis is:

Total dry moles = 2+0.33+12.54=14.87

Hence dry analysis is:

$$\frac{2}{17.87}\times 100 = 11.20\%\ CO_2; \qquad \frac{3}{17.87}\times 100 = 16.80\%\ H_2O$$

$$\frac{0.33}{17.87}\times 100 = 1.85\%\ O_2; \qquad \frac{12.54}{17.87}\times 100 = 70.20\%\ N_2$$

For $\phi = 1.2$:

$$\text{Actual A/F ratio} = \frac{8.96}{1.2} = 7.47:1$$

This means that $\frac{1}{1.2} = 0.834$ of the stoichiometric air is supplied. The combustion cannot be complete & is usual to assume that all the hydrogen is burned to H_2O, since H_2 atoms have a greater affinity for oxygen than C atoms. The carbon in the fuel will burn to CO and CO_2:

$$C_2H_5OH+0.834(3O_2+3\times 3.762N_2)\rightarrow aCO_2+bCO+3H_2O+0.834\times 3\times 3.762N_2$$

C balance: $2 = a + b$

O balance: $1+2\times 0.834\times 3=2a+b+3$

Subtracting the equations gives: a=1.004

and then: b =2 – 1.004 =0.996

i.e. The products are: 1.004 moles CO_2+ 0.996 moles CO+3 moles H_2+ 9.41 moles N_2

The total moles of products = 1.004+0.996 +3+9.41 =14.41

Hence wet analysis is:

$$\frac{1.004}{14.41}\times 100 = 6.97\%\ CO_2; \qquad \frac{0.996}{14.41}\times 100 = 6.91\%\ CO$$

$$\frac{3}{14.41}\times 100 = 20.8\%\ H_2O; \qquad \frac{9.41}{14.41}\times 100 = 65.3\%\ N_2$$

The total dry moles = 1.004+0.996+9.41=11.41

Hence dry analysis is:

$$\frac{1.004}{11.41}\times 100 = 8.80\%\ CO_2; \qquad \frac{0.996}{11.41}\times 100 = 8.73\%\ O_2$$

$$\frac{9.41}{14.41} = 82.47\%\ N_2$$

Dissociation:

It is found that during adiabatic combustion the maximum temperature reached is lower than that expected on the bases of elementary calculation. One important reason for this is that the exothermic combustion process can be reversed to some extent if the temperature is high enough. The reversed process is an endothermic one, i.e. energy is absorbed. In a real process the reaction proceeds in both directions simultaneously and ***chemical equilibrium*** is reached when the rate of break-up of product molecules is equal to their rate of formation. This is represented, for the combustion of CO and H_2, by:

$$2CO+O_2 \Leftrightarrow 2CO_2 \text{ and } 2H_2+O_2 \Leftrightarrow 2H_2O$$

Both of these equations can take place simultaneously in the same combustion process. The proportions of the constituents adjust themselves to satisfy the equilibrium conditions and their actual values depend on the particular pressure and temperature.

Dissociation may be neglected at temperature below 1500K. As the gases cool down in the expansion stroke, dissociation is reduced and the recombination of the dissociated molecules gives up heat.

Let the gases A & B react to form the products C & D in the proportions dictated by the stoichiometric equation:

$$\upsilon_a A + \upsilon_b B \Leftrightarrow \upsilon_c C + \upsilon_d D$$

The proportion of gases A, B, C and D in the balanced mixture is determined by the dissociation ***equilibrium constant***, K which is defined as:

$$K = \frac{P_C^{\upsilon_c} P_D^{\upsilon_d}}{P_A^{\upsilon_a} P_B^{\upsilon_b}}$$

K is a function of temperature and its values are tabulated against temperature for each reaction equation.

<u>Example (3 – 2):</u>

The products from the combustion of a stoichiometric mixture of CO and O_2 are at a pressure of 1atm and a certain temperature. The products analysis shows that 35 per cent of each kmol if CO_2 is dissociated. Determine the equilibrium constant for this temperature, and hence find the percentage dissociation when the products are at the same temperature but compressed to 10 atmospheres.

Solution:

The combustion equation is:

$$CO + \frac{1}{2}O_2 \rightarrow (1-a)CO_2 + aCO + \frac{a}{2}O_2$$

Where:

a is the fraction of CO_2 dissociated

At 1 atm. the products consists of $0.65CO_2 + 0.35\ CO + 0.175\ O_2 = n_t$

i.e. total number of moles of products = 1.175

$$P_{Co_2} = \frac{n_{co_2}}{n_t} P = \frac{0.65}{1.175} = 0.553 \quad \text{atm}$$

$$P_{co} = \frac{n_{co}}{n_t} P = \frac{0.35}{1.175} = 0.298 \quad \text{atm}$$

$$P_{o_2} = \frac{n_{o_2}}{n_t} P = \frac{0.175}{1.175} = 0.149 \quad \text{atm}$$

Based on the equilibrium equation $CO + \frac{1}{2} O_2 \Leftrightarrow CO_2$

$$K = \frac{P_{co_2}}{P_{co} (P_{O2})^{1/2}} = \frac{0.553}{0.298\,(0.149)^{\frac{1}{2}}} = 4.81 (atm)^{-\frac{1}{2}}$$

At pressure P, the partial pressures will be:

$$P_{co_2} = \frac{1-a}{1+\frac{a}{2}} P, \qquad P_{co} = \frac{a}{1+\frac{a}{2}} P, \qquad P_{o_2} = \frac{\frac{a}{2}}{1+\frac{a}{2}} P$$

And equilibrium constant will be:

$$K = \frac{\left(\frac{1-a}{1+\frac{a}{2}}\right)}{\left(\frac{a}{1+\frac{a}{2}}\right)\left(\frac{\frac{a}{2}}{1+\frac{a}{2}}\right)^{\frac{1}{2}}} \times \frac{P}{P \times P^{\frac{1}{2}}}$$

Since the temperature is unchanged, K will still equal $4.81 (\text{atm})^{\frac{1}{2}}$. At 10 atm:

$$4.81 = \frac{(1-a)(2+a)^{\frac{1}{2}}}{a^{\frac{3}{2}}} \times \frac{1}{10^{\frac{1}{2}}}$$

Squaring and simplifying, we have:

$$230.4\, a^3 + 3a = 2$$

And the solution, by trial and error; a=0.185

Dissociation = 18.5 percent of the CO_2

This indicates that although K is independent of pressure of the reacting mixture, the actual fraction of the dissociated product usually varies with pressure.

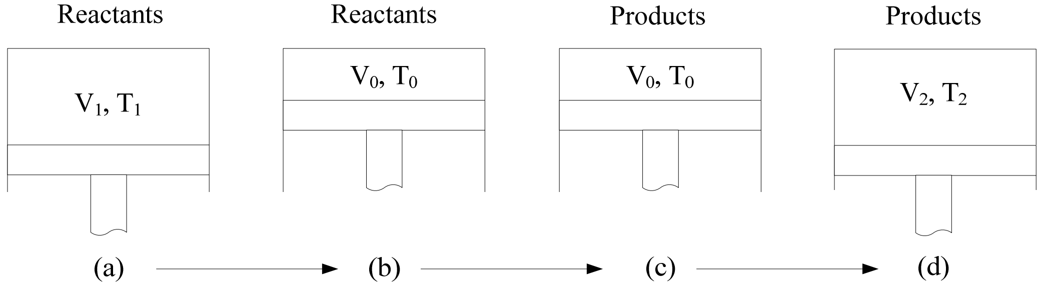
Enthalpy of Formation (Δh_f^o):

The enthalpy of formation of a chemical compound is; *the increase in enthalpy when a compound is formed from its constituent elements in their natural form and in a standard state.*

The natural state of the element is its most stable state. The normal form of oxygen (O_2) and hydrogen (H_2) are gaseous, so (Δh_f^o) for these can be put equal to zero. The standard reference state(STP) is 25°C and 1 atm. pressure. Property values at the standard reference state are indicated by a superscript (^{o}) such as h^o, u^o, etc. It was found adequate to assume that the enthalpy of all elements is zero at the reference state. The enthalpy at any temperature and pressure is given by: $h_{PT} = \Delta h_f^o + h(T)$, h(T) is given in tables.

First Law Applied to Combustion Process:

The first law of thermodynamics applies to any system, and the non-flow and steady- flow energy equation deduced from this law must be applicable to systems undergoing combustion processes. Since the internal energy is a function of state, a change in internal energy is independent of the path of the process. Consider a non-flow combustion process, starting with a mixture of fuel and air at state (V_1, T_1) and ending with products at state (V_2, T_2).



The change in internal energy between states 1 and 2 is:

$$U_2-U_1=U_{P2}-U_{R1}=(U_{P2}-U_{P0})+(U_{P0}-U_{R0})+(U_{R0}-U_{R1})$$

Where suffices R and P refer to reactants and products respectively, the first and third terms on the right – hand side represent changes of internal energy in processes not involving a chemical reaction, which can be evaluated by the methods applying to mixtures. The middle term has to be obtained from additional experimental information because it involves a chemical reaction.

Because b-c occurs at constant volume, no work is done and the non-flow energy equation becomes:

$$Q_{R_oP_o} = \left(U_{P_o} - U_{R_o}\right)$$

$\left(U_{P_o} - U_{R_o}\right)$Can there for be readily obtained by measuring the heat transferred in a constant – volume calorimetric process.

This term is called the "*internal energy of combustion at T_o*", or *the constant volume heat of combustion*, and it is symbolized by ΔU_0, thus:

$$(U_{P_2} - U_{R_1}) = (U_{P_2} - U_{P_0}) + \Delta U_o + (U_{R_0} - U_{R_1})$$

$$U_{P_2} - U_{P_0} = \sum_P m_i c_{vi}(T_2 - T_o) = \sum_P m_i (u_{i2} - u_{io})$$

And

$$U_{R_0} - U_{R_1} = \sum_R m_i c_{vi}(T_o - T_1) = \sum_R m_i (u_{io} - u_{i1})$$

m_i is the mass, and c_{vi} the mean specific heat at constant volume of the constituent i.

Similar analysis can be made for a steady flow or constant pressure combustion process:

$$H_{P_2} - H_{R_1} = (H_{P_2} - H_{P_0}) + \Delta H_o + (H_{R_0} - H_{R_1})$$

Where ΔH_0 = *enthalpy of combustion* at T_o, or the *constant pressure heat of combustion at T_o* and:

$$\Delta H_o = H_{P_o} - H_{R_o} \text{ ,and is always negative.}$$

$$H_{P_2} - H_{P_o} = \sum_P n_i (h_{i2} - h_{io}) \text{ ,to a mole basis}$$

$$= \sum_P m_i (h_{i2} - h_{io}) \text{ ,to a mass basis}$$

$$= \sum_P m_i c_{p_i}(T_2 - T_o) = (T_2 - T_o)\sum_P m_i c_{pi}$$

and

$$H_{R_o} - H_{R_1} = \sum_R n_i (h_{io} - h_{i1}) \text{ to a mole basis}$$

$$H_{R_o} - H_{R_1} = \sum_R m_i (h_{io} - h_{i1}) \text{ to a mass basis}$$

$$= \sum_R m_i c_{p_i}(T_o - T_1) = (T_o - T_1)\sum_R m_i c_{pi}$$

From the definition of the enthalpy of a perfect gas:

$$H = U+PV = U + nR_oT$$

$$H_{Po}=U_{Po}+n_PR_oT_o \text{ and } H_{Ro}= U_{Ro}+ n_RR_oT_o$$

$$\therefore \Delta H_o = \Delta U_o + (n_p - n_R)R_oT_o$$

If there is no change in number of moles (n) during the reaction, or if the reference temperature is absolute zero, thenΔH_0andΔU_0 will be equal.

Example (3 – 3):

Calculate ΔU_o in kJ/kg for the combustion of Benzene (C_6H_6) vapour at 25^oC given that ΔH_0= - 3169100 kJ/mole and the H_2O is in the vapourphase.

Solution:

Combustion equation is: $C_6H_6+7\ 1/2\ O_2 \rightarrow 6CO_2+3H_2O$ (vap.)

$$n_R = 1+7.5= 8.5,\ n_P = 6+3 = 9$$

$$\Delta U_o = \Delta H_o - (n_p - n_R)R_oT_o$$

$$= -3169100-(1/2) \times 8.314 \times 298= - 3170339 \text{ kJ/mol}$$

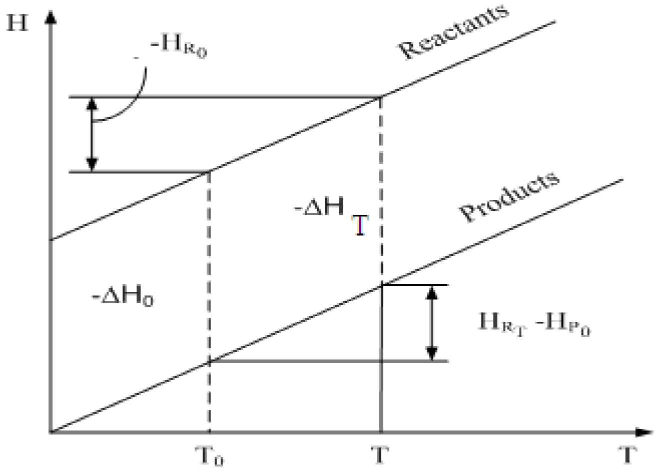
On mass basis, $\Delta U_o = \dfrac{-3170339}{(6\times 12 + 6\times 1)} = -40645$ kJ/kg

Change in reference temperature:

Internal energy and enthalpy of combustion depend on the temperature at which the reaction occurs. This is due to the change in enthalpy and internal energy of the reactants and products with temperature.

It can be seen from the property diagram that the enthalpy of combustion at temperature T,ΔH_T can be obtained from ΔH_0 at T_o by the relationship:

$$-\Delta H_T = -\Delta H_o + (H_{R_T} - H_{R_o}) - (H_{P_T} - H_{P_o})$$



Example (3 – 4):

ΔH_0For CO at 60°C is given as -285200 kJ/mole. CalculateΔH_T at 2500°C given the enthalpies of the gases concerned in kJ/mole, are as follows:

Gas	60°C	2500°C
CO	9705	94080
O	9696	99790
CO_2	10760	149100

Solution:

$$CO + \frac{1}{2}O_2 \rightarrow CO_2$$

$$-\Delta H_T = -\Delta H_o + (H_{R_T} - H_{R_0}) - (H_{P_T} - H_{P_o})$$

$$H_{R_0} = 1 \times 9705 + \frac{1}{2} \times 9696 = 14553 \quad kJ$$

$$H_{R_T} = 1 \times 94080 + \frac{1}{2} \times 99790 = 143975 \quad kJ$$

$$H_{P_0} = 10760 \quad kJ \quad \& \quad H_{P_T} = 1 \times 149100 = 149100 \quad kJ$$

$$\therefore \Delta H_T = -276282 \quad kJ/mole$$

Adiabatic flame temperature:

It is the temperature attained in combustion with no heat transfer, no work or changes in kinetic or potential energy; this is the maximum temperature that can be achieved.

For a given fuel and given pressure and temperature of the reactants, the maximum adiabatic flam temperature that can be achieved is with a stoichiometric mixture when complete combustion occurs.

Example (3 – 5):

Liquid heptane (C_7H_{16}) has an enthalpy of combustion of -44560 kJ/kg at 25°C, when the water in the products is in the vapour phase. This fuel is to be burnt adiabatically in a steady stream of air in stoichiometric proportion. If the initial temperature of the fuel and air is 15°C, estimate the temperature of the products. The specific heat of liquid heptane is 2.30kJ/kg K.

Solution:

The combustion equation for any hydrocarbon is:

$$C_nH_m+(n+\frac{m}{4})O_2+3.762(n+\frac{m}{4})N_2\rightarrow nCO_2+\frac{m}{2}H_2O+3.762(n+\frac{m}{4})N_2$$

i.e.

C_7H_{16}+11 O_2+41.382 $N_2 \rightarrow$ 7 CO_2+8 H_2O+41.382 N_2

The steady – flow energy equation:

$$H_R+\Delta Q=H_P+W$$

Since the process of combustion is adiabatic with no work, then:

$$H_R=H_P, \Delta Q=0 \ \& \ W=0$$

$$\text{i.e.}\,(H_{Pt}-H_{R15})=0$$

t represent products temperature. Expanding this equation we have:

$$(H_{Pt}-H_{P25})+\Delta H_{25}+(H_{R25}-H_{R15})=0$$

$$\left(H_{R25}-H_{R15}\right)=\sum_R n_i\left(h_{i25}-h_{i15}\right)=n_{C7H16}+n_{O_2}\left(h_{25}-h_{15}\right)_{o_2}+n_{N_2}\left(h_{25}-h_{15}\right)_{N_2}$$

$$=\left[nC_p(25-15)\right]_{C_7H_{16}}+11(731.5-438.2)+41.382\times(728.4-437)$$

$$=100\times2.3\times10+3229+120581=17587kJ$$

$$\left(H_{pt}-H_{p25}\right)=\sum_p n_1\left(h_{it}-h_{i25}\right)$$

Assume T=1800 K, as a first estimate

$(H_{pt}-H_{p25})$ = 7 (80530 – 913.8) + 8 (63270 – 840.5) + 41.382 × 49750 – 728.4

= 7 × 79616 + 8 × 6249.5 + 41.382 × 49022

= 3085361.3 kJ

17636.5 – 100 × 44560 + 3085361.3 = -1354040

Assume T=2000 K

$(H_{pt}-H_{p25})$ =7(92610 – 913.8) + 8(73200 – 840.5) + 41.382 × (56920 – 728.4)

= 7 × 91696.2 + 8 × 72360 + 41.382 × 56192

= 3546073 kJ

It is clear the equation will not equal to zero, take t=2500

$\therefore$ $(H_{pt}-H_{p25})$ = 7(123335 – 913.8) + 8(98950 – 840.5) + 41.382 × (75280 – 728.4)

= 7 × 122421.4 + 8 × 98110.2 + 41.382 × 74551.6

$= 7 \times 12242.1 + 8 \times 98110.2 + 41.382 \times 74551.6$

$= 4726921.2 kJ$

T		**ΔH**
1800	−	1354040
2000	−	894019
2500	+	282413.5

By linear interpolation, we have T=2380 K

Calorific value of fuels:

The quantities ΔH_0 andΔU_0 are approximated to, in fuel specification, by quantities called calorific values which are obtained by the combustion of the fuels in suitable apparatus. This may be of the constant volume type (e.g. *Bomb Calorimeter*) or constant pressure, steady flow type (e.g. *Boy's Calorimeter*).

1. **Higher (or Gross) Calorific value(H.C.V.) at constant volume**: It is the energy transferred at constant volume as heat to surroundings per unit quantity of fuel when burned at constant volume with the H_2O product of combustion in the liquid phase,($Q_{gr,v}$). This approximate to $-\Delta U_0$at the reference temperature of 25^oC with H_2O in the liquid phase.

   i.e. $Q_{gr,\,v} = -\Delta U_O$

   If H_2O products are in the vapour phase the energy released per unit quantity of fuel is called the **lower (or net) calorific value** (L.C.V).

2. **Higher calorific value at constant pressure**: It is the energy transferred at constant pressure as heat to the surrounding (cooling water) per unit quantity of fuel when burned at constant pressure with the H_2O products of combustion in the liquid phase, $Q_{gr,p} = - \Delta H_0$ at the reference temperature of 25^oC with the H_2O in the liquid phase.

Lower calorific value (L.C.V) at constant pressure; is the energy released with H_2O products in the vapour phase.

$Q_{gr,v} = Q_{net,v} + m_c v_{fg}$

$Q_{gr,p} = Q_{net,p} + m_c h_{fg}$

And

u_{fg} at 25°C for H_2O=2304.4 kJ/kg

h_{fg} at 25°Cfor H_2O =2441.8 kJ/kg

m_c: mass of condensate per unite quantity of fuel.

Example (3 – 6):

The calorific value $Q_{gr,v}$ of kerosene, whose ultimate analysis is 86 per cent carbon and 14 per cent hydrogen, was found by experiment to be 46890 kJ/kg. Calculate the other three calorific values.

Solution:

$$C_{7.167}H_{14}+10.667\ O_2 \rightarrow 7.167\ CO_2+7\ H_2O$$

On mass basis:

$$100\times 1\ C_{7.167}H_{14} + 32\times 10.667\ O_2 \longrightarrow 44\times 7.167\ CO_2 + 18\times 7\ H_2O$$

For one kg of fuel:

$$1\text{ kg fuel} + 3.41\text{ kg } O_2 \rightarrow 3.15\text{ kg } CO_2 + 1.26\text{ kg } H_2O$$

$Q_{net,v} = Q_{gr,v} - m_c u_{fg} = 46890 - 1.26 \times 2304 = 43987$ kJ/kg

The relation between $Q_{gr,v}$ and $Q_{gr,p}$ is identical with that between $-\Delta H_0$ and $-\Delta U_0$, i.e.

$Q_{gr,p} = Q_{gr,v} + R_oT_o(n_P-n_R)$

$$= 46890 + 8.314\times 298\left(\frac{3.15}{44} - \frac{3.41}{32}\right)$$, fuel & H_2O are liquids

= 46890+87 = 46977 kJ/kg

And

$Q_{net,p} = Q_{gr,p} - m_c h_{fg} = 46977-1.26 \times 2442 = 43900$ kJ/kg

CHAPTER (4)

Fuel Air Cycle

The theoretical cycle based on the actual properties of the cylinder contents is called the fuel – air cycle. The fuel – air cycle take into consideration the following:

1- The actual composition of the cylinder contents.
2- The variation in the specific heat of the gases in the cylinder.
3- The dissociation effect.
4- The variation in the number of moles present in the cylinder as the pressure and temperature change.
5- No chemical changes in either fuel or air prior to combustion.
6- Combustion takes place instantaneously at top dead center.
7- All processes are adiabatic.
8- The fuel is mixed well with air.

Variation of specific heats:

All gases, except mono-atomic gases, show an increase in specific heat as temperature increase. The specific heat may be written in the form:

$$C = a+bT+cT^2$$

where T is the absolute temperature and a, b and c are constants for any specific gas.

C_p and C_v increase with temperature but $\gamma = \frac{c_p}{c_v}$ decrease as the temperature increase. There are special tables and charts which gives the specific heat of different gases at different temperatures. Specific heats of a mixture of gases can be calculated if the constituents of the mixture are known, using the gas mixture relations.

If the variation of specific is taken into account during the compression stroke, the final pressure and temperature would be lower if constant value of specific heats is used ($\bar{2}$) as shown in Figure (4-1).When taking variable specific heat, end of combustion will be ($\bar{3}$) instead of 3. Expansion process would be $\bar{3}\bar{4}$ when assumed isentropic, but expansion taking variable specific heats into accounts is above $\bar{3}\bar{4}$ and represented by $\bar{3}\bar{\bar{4}}$. Thus it is seen that the effect of variation of specific heats is to deliver less work.

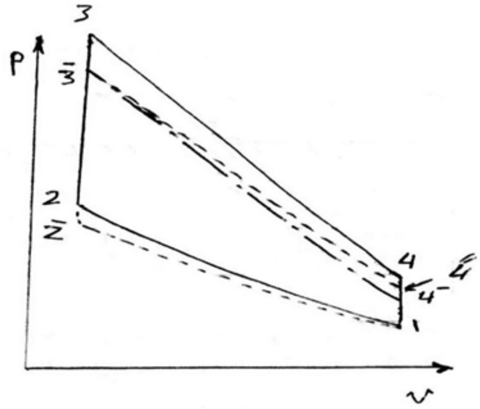


Figure (4-1): *Loss due to variation of specific heat*

Dissociation effect:

The effect of dissociation is a suppression of a part of the heat release during combustion and the liberation of it as expansion proceeds, a condition similar to the effect produced by the variation of specific heats. However, the effect of dissociation is much smaller than that of change of specific heats.

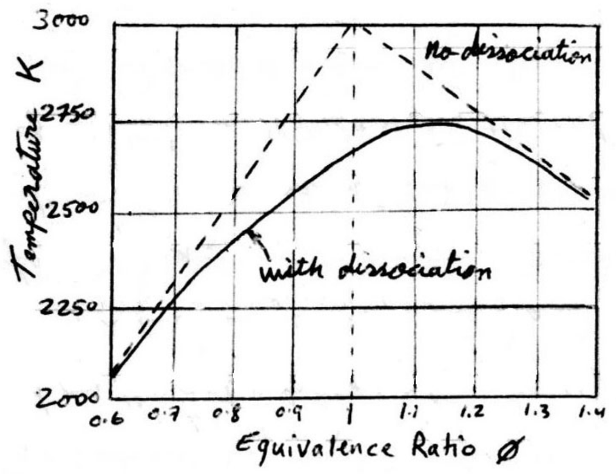


Figure (4-2): *Effect of dissociation temperature at different ϕ*

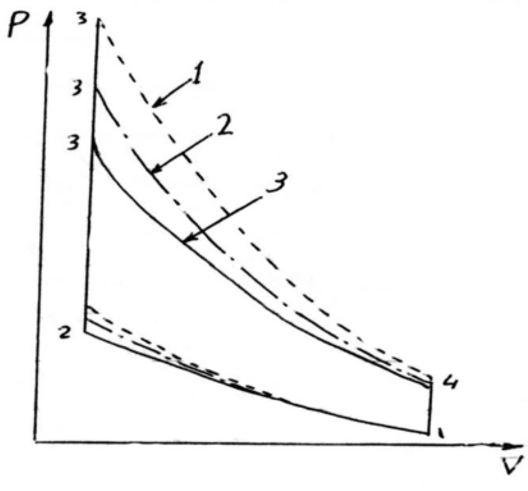


Figure (4-3): *Comparison of air-Standard cycle with fuel – air cycle*

The effect of dissociation on combustion temperature is as shown in figure (4-2), the dotted line represents the maximum combustion temperature attained with no dissociation, and the full line is with dissociation. Dissociation reduces the maximum temperature by about 300°C.

Comparison of P-V Diagram of Air-standard and Fuel – Air cycle for SI engine:

Diagram(4 – 3) shows three constant volume cycles:

1- Air – standard cycle.
2- Air – standard with variable specific heat.
3- Fuel – Air cycle with variable specific heat and dissociation.

Example (4-1):

What is the percentage change in the efficiency of Otto cycle having a compression ratio of 7, if the specific heat at constant volume increases by 1%?

Solution:

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

$$C_P - C_v = R$$

$$\gamma - 1 = \frac{R}{C_v}$$

$$\eta = 1 - \frac{1}{r^{\frac{R}{C_v}}} \qquad \Rightarrow 1-\eta = r^{-\frac{R}{C_v}}$$

$$\ln(1-\eta) = -\frac{R}{C_v}\ \ln r \qquad , r = \text{constant}$$

Differentiating:

$$\frac{1}{1-\eta} d\eta = \frac{-R}{C_v^2}\ \ln r \times d\,C_v$$

$$d\eta = -1(1-\eta)(\gamma-1)\ln r\,\frac{dC_v}{C_v}$$

$$\frac{d\eta}{\eta} = -\frac{(1-\eta)}{\eta}(\gamma-1)\ln r \times \frac{dC_v}{C_v}$$

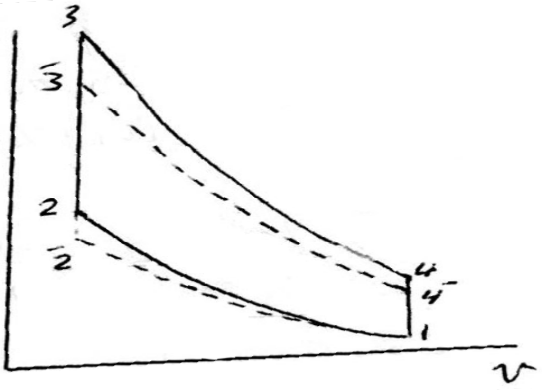
$$\eta = 1 - \frac{1}{7^{0.4}} = 0.54$$

$$\therefore \frac{d\eta}{\eta} = -\frac{(1-0.54)}{0.54} \times (1.4-1) \times \ln 7 = -0.663 \quad \text{(decrease)}$$

Example (4-2):

A petrol engine of compression ratio 6 uses a fuel of calorific value43950kJ/kg. The air – fuel ratio is 15:1. The temperature and pressure of the charge at the charge at the end of the suction stroke are 60°C and 1 bar. Determine the maximum pressure in the cylinder if the index of compression is 1.32 and the specific heat at constant volume is expressed by the expression; $C_v = 0.71 + 19 \times 10^{-5}T$ kJ/kg K, where T is the temperature in K.
Compare this value with that when constant specific heat C_v= 0.72 is used.

Solution:



$$p_1 v_1^n = p_2 v_2^n$$

$$p_2 = p_1\left(\frac{v_1}{v_2}\right)^n = 1 \times 6^{1.32} = 10.645\ bar$$

$$T_2 = T_1\left(\frac{p_2 v_2}{p_1 v_1}\right) = 333 \times \frac{10.645}{6}$$

$$= 590.8\text{K}$$

Mean specific heat $C_{v_{mean}} = 0.71 + 19 \times 10^{-5}\left[\frac{T_2 + T_3}{2}\right]$

Assume 1 kg of air in the cylinder, heat added per kg air= $\frac{43950}{15}$

$Q = C_v \times \text{mass of charge} \times (T_3 - T_2)$

$$2930 = [0.71 + 19 \times 10^{-5}\,(T_3 + 590.8)/2] \times \frac{16}{15} \times (T_3 - 590.8)$$

Solving we get $T_3 = 3090$ K

$$P_3 = P_2 \frac{T_3}{T_2} = 10.645 \times \frac{3090}{590} = 55.75\text{bar}$$

For constant specific heat, $2930 = 0.72 \times \frac{16}{15}(T_3 - 590)$

Solving we get $T_3 = 4405$ K

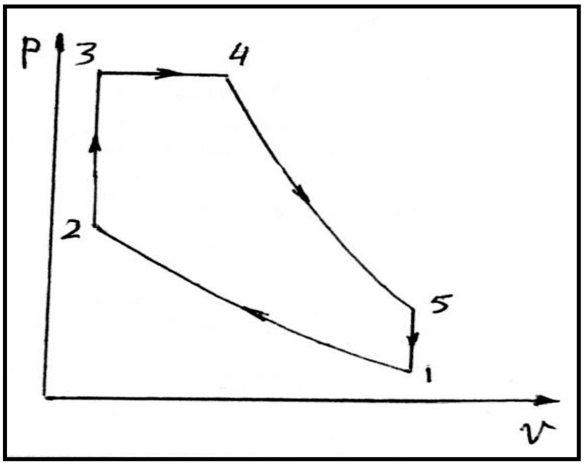
$$P_3 = 10.645 \times \frac{4405}{590} = 79.5 \text{ bar}$$

Example (3 - 3):

In an oil engine, working on dual combustion cycle, the temperature and pressure at the beginning of compression are 90^{o}C and 1 bar. The compression ratio is 13:1. The heat supplied per kg of air is 1674 kJ, half of which is supplied at constant volume and half at constant pressure. Calculate (i) the maximum pressure in the cycle (ii) the percentage of stroke at which cut-off occurs.

Take γ for compression 1.4, R= 0.293 kJ/kg K and C_v for products of combustion $(0.71 + 12 \times 10^{-5} T)$

Solution:



$$(i)\ P_2 = P_1\left(\frac{v_1}{v_2}\right)^{\gamma} = 1 \times (13)^{1.4} = 36.3 \text{ bar}$$

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{\gamma - 1} = 363(13)^{0.4} = 1013K$$

$$Q_{12} = m\int_{T_2}^{T_3} C_v dT = m\int_{T_2}^{T_3} (0.71 + 12 \times 10^{-5} T) dT$$

Neglect mass of fuel;

$$837 = 1\left[0.71T + 12 \times 10^{-5}\frac{T^2}{2}\right]_{T_2}^{T_3}$$

$$= 0.71(T_3 - 1013) + \frac{12 \times 10^{-5}}{2}(T_3^2 - 1013^2)$$

$$= 0.00006\,T_3^2 + 0.71\,T_3 - 1617.8$$

Solving we get $T_3 = 1955.6$ K

$$\therefore \text{Maximum pressure} = 36.3 \times \frac{1955.6}{1013} = 70.1 \text{ bar}$$

$(ii)\ C_p = C_v + R = 1.003 + 12\times10^{-5}T$

$$Q_{34} = m\int_{T_3}^{T_4} C_p dT = m\int_{T_3}^{T_4}(1.003 + 12\times10^{-5}T)dT$$

$$837 = 1\left[1.003\,T + 12\times10^{-5}\frac{T^2}{2}\right]_{1864}^{T_4}$$

$$837 = 1.003(T_4 - 1864) + 0.00006(T_4^2 - 1955.6^2)$$

$$= 1.003\,T_4 + 0.00006\,T_4^2 - 3028$$

$$0.00006\,T_4^2 + 1.003\,T_4 - 3865 = 0$$

Solving we get, T_4=2611 K

$$\frac{v_4}{v_3} = \frac{T_4}{T_3} = \frac{2611}{1955.6} = 1.335$$

$$\text{Cut} - \text{off} = v_4 - v_3 \qquad , \frac{v_1}{v_2} = 13$$

% of stroke at which cut off occurs = $\dfrac{v_4 - v_3}{v_1 - v_2}$

$$= \frac{1.335\,v_3 - v_3}{13\,v_2 - v_2} = \frac{(1.335-1)v_2}{12\ v_2} = 2.793\%, \qquad (V_3 = V_2)$$

Example (3 – 4):

A mixture of Heptane (C_7H_{16}) and air which is 10% rich is initially at a pressure of 1 atm. and a temperature of 100°C, is compressed through a volumetric ratio of 6:1. It is then ignited and adiabatic combustion proceeds at constant volume. The maximum temperature reached is 2627°C and at this temperature the equilibrium constants are;

$$\frac{P_{H2O}.P_{CO}}{P_{CO2}P_{H2}} = 6.72 \qquad and \qquad \frac{P^2{}_{CO}.P_{O2}}{P^2{}_{CO2}} = 0.054\ atm.$$

If the constituent of the product gases are; CO_2, CO, H_2O, H_2, O_2 and N_2, show that approximately 30.2% of the carbon has burned incompletely. Pressures are measured in atmospheres.

Solution:

The stoichiometric combustion equation for a hydrocarbon fuel is:

$$C_nH_m + (n + m/4)[O_2 + 3.762\,N_2] \rightarrow n\,CO_2 + (m/2)\,H_2O + (n + m/4)\times 3.762\,N_2$$

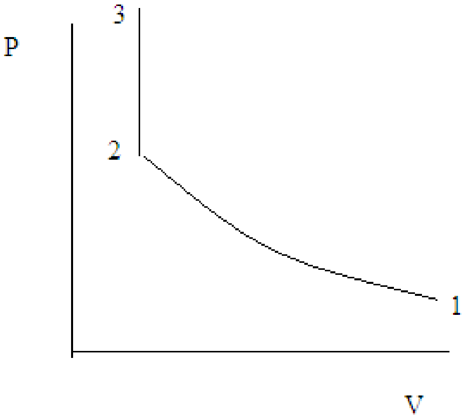
$$\therefore \qquad C_7H_{16} + 11(O_2 + 3.762\,N_2) \rightarrow 7\,CO_2 + 8\,H_2O + 41.36\,N_2$$

Actual combustion equation includes CO and H2 in the products also;

So; actual combustion equation may be written in the form:

$$C_7H_{16} + \frac{10}{11}(11O_2 + 41.36\,N_2) \rightarrow a\,CO_2 + b\,CO + c\,H_2O + d\,H_2 + e\,O_2 + 37.6\,N_2$$

Atom balance:



O_2 : a + b/2 + c/2 + e = 10

C : a + b = 7

H_2 : c + d = 8

n_1 = 1 + 10 + 37.6 = 48.6

n_3 = a + b + c + d + e + 37.6

$P_1V_1 = n_1 R_oT_1$ and $P_3V_3 = n_3R_oT_3$

By division: $\frac{P_3}{n_3} = \frac{P_1}{n_1}\frac{V_1}{V_3}\frac{T_3}{T_1} = \frac{1}{48.6}\frac{6}{1}\frac{2900}{373} = 0.958$

$P_{CO_2} = \frac{a}{n_3}P_3$; $P_{CO} = (b/n_3)\,P_3$; $P_{H2O} = (c/n_3)\,P_3$; $P_{H2} = (d/n_3)\,P_3$; and $P_{O2} = (e/n_3)\,P_3$

$$\frac{P_{H2O}.P_{CO}}{P_{CO2}P_{H2}} = 6.72 \qquad \frac{P^2{}_{CO}.P_{O2}}{P^2{}_{CO2}} = 0.054\,atm.$$

∴ $\frac{b.c}{a.d} = 6.72$ ……..(1) & $\frac{b^2}{a^2}\frac{e}{n_3}P_3 = 0.054$ ………(2)

The proportion of carbon incompletely burned is : $\frac{b}{7} = 0.302$

∴ b = 2.114 , then: a = 7 – 2.114 = 4.886

Substituting in (1), gives:

$$\frac{c}{d} = 6.72 \times \frac{a}{b} = 6.72 \times \frac{4.886}{2.114} = 15.5$$

C + d = 8, i.e. 15.5d + d = 8, then: d = 8/16.5 = 0.485

And c = 15.5 x 0.485 = 7.515

e = 10 – a – b/2 – c/2 = 10 – 4.886 – 2.114/2 – 7.515/2

= 0.297

Substitute in (2) gives:

$$\frac{b^2}{a^2}\frac{e}{n_3}P_3 = \left(\frac{2.114}{4.886}\right)^2 \times 0.297 \times 0.958 = 0.0531$$

Which gives sufficient agreement to the 0.054 quoted, showing that approximately 30.2% of carbon was burned to CO.

CHAPTER (5)

The Actual Cycle

The actual cycle experienced by internal combustion engines is an open cycle with changing composition, actual cycle efficiency is much lower than the air standard efficiency due to various losses occurring in the actual engine. These losses are as follows:

1- Losses due to variation of specific heats with temperature: already discussed.

2- Losses due to dissociation: already discussed.

3- time losses:

In theoretical cycles the burning is assumed to be instantaneous. Whereas, in actual cycle, burning is completed in a definite interval of time. The effect of this time is that the maximum pressure will not be produced when the volume is minimum; but sometime after T.D.C., causes a reduction in the work produced.

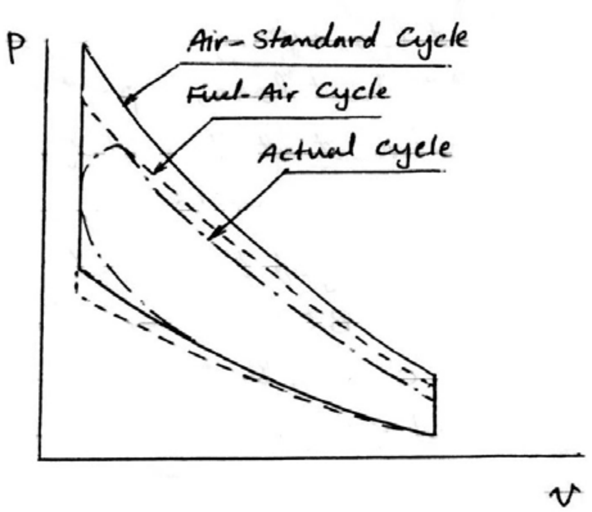


Figure (5-1): *The effect of time losses on P-V diagram*

In order that the maximum pressure is not reached too late in the expansion stroke, the time at which burning starts is varied by varying the spark timing (spark advance).

4- incomplete combustion loss:

Fuel vapour, air, and residual gas are present in the cylinder, this makes it impossible to obtain perfect homogeneous mixture.. Therefore some fuel does not burn to CO_2 or partially burns to CO, and O_2 will appear in the exhaust. Energy release in actual engine is about 90 to 93% of fuel energy input.

5- direct heat loss:

During combustion process and subsequent expansion stroke, the heat flows from cylinder gases through cylinder walls and cylinder head into the water jacket or cooling fins. Some heat enters the piston head and flows through piston rings into the walls of the cylinder or carried away by the engine oil.

The heat loss during combustion and expansion does not represent a complete heat loss; a part of the heat loss would be rejected in the exhaust at the end of the expansion stroke.

6- exhaust blow down loss:

The opening of the exhaust value before B.D.C. reducing cylinder pressure, causing the roundness of the end of the P-V diagram, this means a reduction in the work done per cycle.

7- pumping losses:

Pumping loss is due to expelling the exhaust gases and the induction of the fresh charge. In naturally aspirated engine this would be a negative work.

8- Friction losses:

These losses are due to the friction between the piston and cylinder walls, the various bearings, and the friction in the auxiliary equipment, such as pumps, fans, etc…

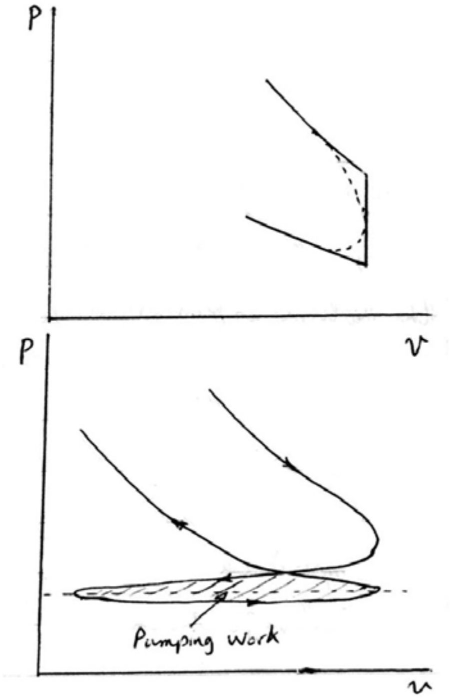


Figure (5-2, 3): *Effect of exhaust valve opening time and pumping*

Effect of throttle opening:

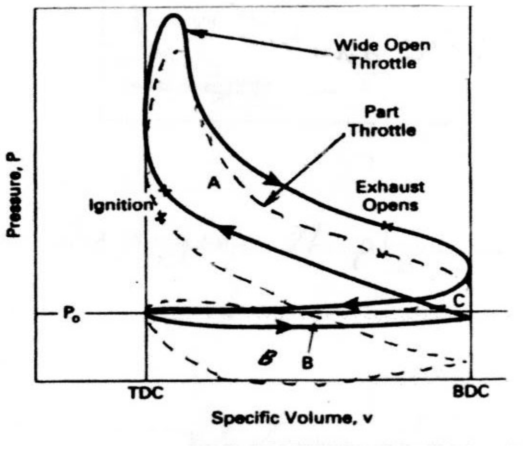


Figure (5-4): *Effect of throttle opening on the indicator diagram*

When a four- stroke SI engine is run at partially closed throttle, (throttle is a butterfly valve in the intake system), fuel supplied to the engine is reduced, and this would lead to less power output at part throttle opening as shown in figure (5-4). The upper loop represent positive work output (A) while the lower loop consisting of the exhaust and intake strokes is negative work (B). The more closed the throttle position, the lower will be the pressure during the intake stroke and the greater the negative pumping work.

Effect of spark timing:

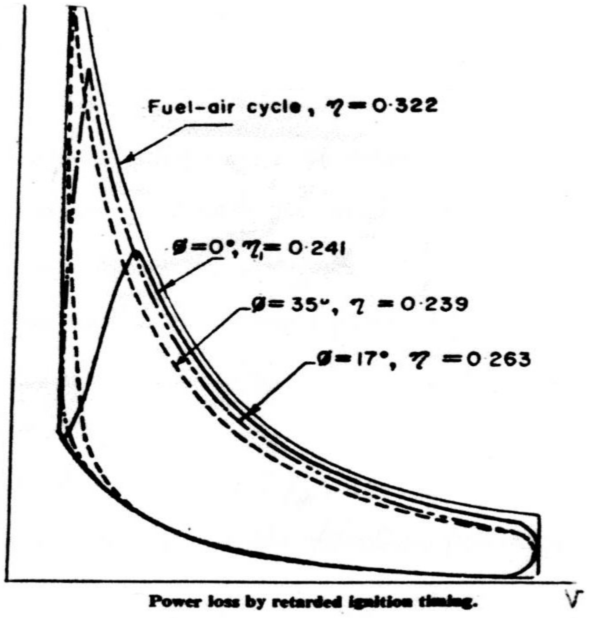


Figure (5-5): *Power loss by retarded ignition timing*

The effect of spark timing is shown in figure (5-5), $\phi = 0$ means spark timing at T.D.C, in this case the peak pressure is low and occurs nearly 40° after T.D.C., as spark timing is advanced to achieve combustion at TDC, additional work is required to compress the burning gases. Figure (5-5) shows the power loss by retarded ignition timing optimum loss in power. In actual practice a deliberate retard in spark from optimum may be done to avoid knocking and reduce exhaust emissions of HC and CO.

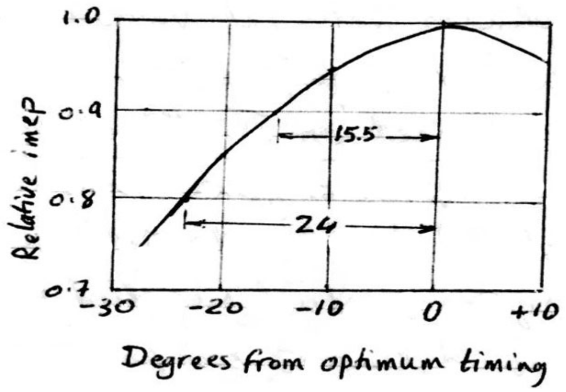


Figure (5-6): *Variation of imep with spark timing*

CHAPTER (6)

ENGINE PERFORMANCE

The basic performance parameters of internal combustion engine (I.C.E) may be summarized as follows:

1. Indicated power (i.p.):

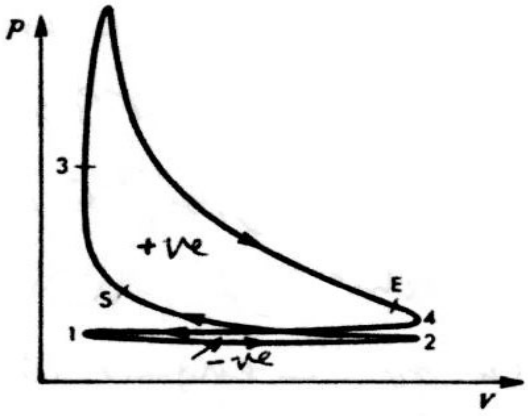


Figure (6-1): *indicator diagram of SI engine*

It is the actual rate of work done by the working fluid on the piston. As its name implies, the i.p. can be determined from an "indicator diagram" as show in figure (6-1), by subtracting the pumping loop area (- ve) from the positive area of the main diagram. i.p. power could be estimated by performing a Morse test on the engine. The physical equation for the i.p. is:

$$\text{i.p.} = P_m LAN$$

where N is the number of machine cycles per unit times, which is 1/2 the rotational speed for a four- stroke engine, and the rotational speed for a two- stroke engine.

2. Brake power (b.p.):

This is the measured output of the engine. It is usually obtained by a power absorption device such as a brake or dynamometer which can be loaded in such a way that the torque exerted by the engine can be measured. The break power is given by:

$\text{b.p.} = 2\pi NT$

Where T is the torque

3. Friction power (f.p.) and Mechanical efficiency (η_m):

The difference between the i.p. and the b.p is the friction power (f.p.) and is that power required to overcome the frictional resistance of the engine parts,

$$\text{f.p.} = \text{i.p.} - \text{b.p.}$$

The mechanical efficiency of the engine is defined as: $\eta_m = \dfrac{\text{b.p.}}{\text{i.p.}}$

η_m is usually between 80% and 90%

4. Indicated mean effective pressure (imep):

It is a hypothetical pressure which if acting on the engine piston during the working stroke would results in the indicated work of the engine. This means it is the height of a rectangle having the same length and area as the cycle plotted on a p- v diagram.

$$\text{imep } (P_i) = \frac{\text{Net area of the indicator diagram}}{\text{Swept volume}} \times \text{Indicator scale}$$

Consider one engine cylinder:

$$\text{Work done per cycle} = P_i\, AL$$

where: A = area of piston; L = length of stroke

$$\text{Work done per min.} = \text{work done per cycle} \times \text{active cycles per min.}$$

$$\text{i.p.} = P_i\, AL \times \text{active cycles/ min}$$

To obtain the total power of the engine this should be multiplied by the number of cylinder n, i.e.:

$$\text{Total i.p.} = P_i\, AL\, Nn/2 \qquad \text{for four- stroke engine}$$

and

$$= P_i ALNn \qquad \text{for Two- stroke engine}$$

5. Brake mean effective pressure (bmep) and brake thermal efficiency:

The bmep (P_b) may be thought of as that mean effective pressure acting on the pistons which would give the measured b.p., i.e.

$$\text{b.p.} = P_b\, AL \times \text{active cycles/ min}$$

The overall efficiency of the engine is given by the brake thermal efficiency, η_{BT} i.e.

$$\eta_{BT} = \frac{\text{Brake power}}{\text{Energy supplied}}$$

$$\eta_{BT} = \frac{b.p.}{\dot{m}_f \times Q_{net}}$$

where $\dot{m}_f$ is the mass of fuel consumed per unit time, and Q_{net} is the lower calorific value of the fuel.

6. Specific fuel consumption (s.f.c.):

It is the mass of fuel consumed per unit power output per hour, and is a criterion of economic power production.

$$sfc = \frac{\dot{m}_f}{b.p.}\ kg/kWh$$

Low values of s.f.c are obviously desired. Typical best values of bsfc for SI engines are about 270g/kW.h, and for C.I. engines are about 200g/kW.h.

7. Indicated thermal efficiency (η_{IT}):

It is defined in a similar way to η_{BT}

$$\eta_{IT} = \frac{i.p.}{\dot{m}_f \times Q_{net}}$$

Dividing η_{BT} by η_{IT} gives

$$\frac{\eta_{BT}}{\eta_{IT}} = \frac{\text{b.p.}}{\text{i.p.}} = \eta_m$$

$$\therefore \eta_{BT} = \eta_m \times \eta_{IT}$$

8. Volumetric efficiency (η_v):

Volumetric efficiency is only used with four- stroke cycle engines. It is defined as the ratio of the volume if air induced, measured at the free air conditions, to the swept volume of the cylinder:

$$\eta_v = \frac{\dot{v}}{v_s}$$

The air volume $(\dot{V})$may be refereed to N.T.P. to give a standard comparison.

The volumetric efficiency of an engine is affected by many variables such as compression ratio, valve timing, induction and port design, mixture strength, latent heat of evaporation of the fuel, heating of the induced charge, cylinder pressure, and the atmospheric conditions.

Example 6-1:

The peak pressure of a SI engine rotating at 1500 rpm occurs 0.003S after the spark, what will be the spark timing when peak pressure is at TDC. If the inlet valve opens at 10 degrees bTDC and closes at 45 degrees aBDC, how long the inlet valve opening period is in seconds.

Solution:

Number of revolutions per second $= \frac{1500}{60} = 25$ rev.

Number of revolutions between spark timing and TDC = 25 × 0.003 = 0.075 rev.

Crank shaft angle during this period = 0.075 × 360 = 27

i.e. spark must occurs 27 degree bTDC

inlet valve opening = 10 + 180 + 45 = 235 degrees

inlet valve opening time in seconds $= \frac{235}{360 \times 25} = 0.0265$.

Example 6-2:

In a four stroke single cylinder gas engine the indicated mean effective pressure is 0.46 MN/m^2, the brake power 9 kW, speed 250 rpm, mechanical efficiency, η_m = 0.8, and bore to stroke ratio = 0.66. Calculate cylinder diameter and mean piston speed.

Solution:

$$\eta_m = \frac{bp}{i.p} \quad , \ \text{i.p} = \frac{9}{0.8} = 11.25 \ \text{ kw}$$

$$\text{i.p} = \frac{\text{P}_\text{i} LANn}{2}$$

$$LA = \frac{2 \times i.p}{P_i Nn} = \frac{2 \times 11.25}{0.46 \times 1000 \times 250 \times 1} = 0.01174 \quad \text{m}^3$$

$$\frac{d}{L} = 0.66 \quad , \ \text{L} = \frac{\text{d}}{0.66}$$

$$\therefore \frac{d}{0.66} \times \frac{\pi}{4} d^2 = 0.01174$$

$$\text{d}^3 = 0.009866$$

$$\text{d} \ = 0.2145 \ \text{ m}$$

Mean piston speed = $\frac{2LN}{60} = \frac{2 \times 0.2145 \times 250}{0.66 \times 60} = 2.71 \quad \text{m/s}$

Example 6-3:

A four stroke petrol engine delivers 35.75kW with a mechanical efficiency of 80%, the fuel consumption of the engine is 0.4 kg per brake power hour, and the A/F ratio is 14:1.The heating value of the fuel is 41870 kJ/kg. Find: (a) i.p, (b) f.p., (c) η_{BT} , (d) η_{IT} , (e) fuel consumption per hour, (f) air consumption per hour.

Solution:

a) $\eta_m = \frac{b.p}{i.p} \quad , \text{i.p} = \frac{35.75}{0.8} = 44.7 \ \text{ kW}$

b) f.p= i.p – b.p=44.7– 35.75=8.95kW

c) $\eta_{BT} = \frac{b.p}{Q_{added}} = \frac{35.75 \times 3600}{0.4 \times 35.75 \times 41870} = 0.215$

d) $\eta_{BT} = \eta_{IT} \times \eta_m$

$$\eta_{IT} = \frac{0.215}{0.8} = 0.2687$$

e) fuel consumption per hour = 0.4 ×35.75=14.32kg

f) air consumption per hour = 14.32 × 14=200.5kg

Example 6-4:

The air flow to a four cylinder four – stroke engine is 2.15 m^3/min. During a test on the engine the following data were recorded:

Bore 10.5cm; stroke 12.5cm; engine speed 1200 rpm, torque 150 N.m, fuel consumption 5.5 kg/h, calorific value of fuel, 43124 kJ/kg, ambient temperature and pressure are 20^{o}C and 1.03 bars. Calculate:

1- The brake thermal efficiency.
2- The brakes mean effective pressure.
3- The volumetric efficiency.

Solution:

$$1\text{-}\ b.p.=\frac{2\pi NT}{60}=\frac{2\pi\times1200\times150}{60\times1000}=18.85\ \ \text{kW}$$

$$\eta_{BT}=\frac{b.p}{Q_{added}}=\frac{18.85\times3600}{5.5\times43124}=0.286$$

$$2\text{-}\ b.p=\frac{P_iLAN_n}{2}$$

$$\therefore P_i=\frac{2\times18.85\times4\times60}{0.125\times\pi\times(0.105)^2\times1200\times4}=435.4\quad\text{kPa}$$

$$3\text{-}\ \eta_v=\frac{\dot{V}}{V_s}$$

$$V_s=LA\frac{Nn}{2}=0.125\times\frac{\pi}{4}(0.105)^2\times\frac{1200\times4}{2}=2.6\quad m^3/\min$$

$$\therefore\eta_v=\frac{2.15}{2.6}=0.83$$

Testing of Internal Combustion Engines:

There are a wide variety of engine tests, starting from simple fuel and air- flow measurements to taking of complicated injector needle lift diagram, swirl patterns and photographs of the combustion process, etc..

Here only certain basic tests and measurement will be considered.

1-Measurement of speed:

A wide variety of speed measuring devices are available they range from a mechanical tachometer to digital and triggered electrical tachometers.

The best method of measurement is to count the number of revolution in a given time; this could be done either mechanically or electrically.

2-Fuel consumption measurement:

The fuel consumption of an engine is measured by determining the volume flow in a given time interval, or to measure the time required for the consumption of a given volume (or mass) of fuel.

3-Air consumption measurement:

The measurement of the air flow in the engine intake is not an easy task, because of the cyclic nature of the engine which causes a pulsating air flow:

a) Air box method:

In this method the intake air is drawn from a large surge tank, and measurement of air flow into the surge tank is performed using a calibrated orifice or a flow nozzle (see fig.6-2).

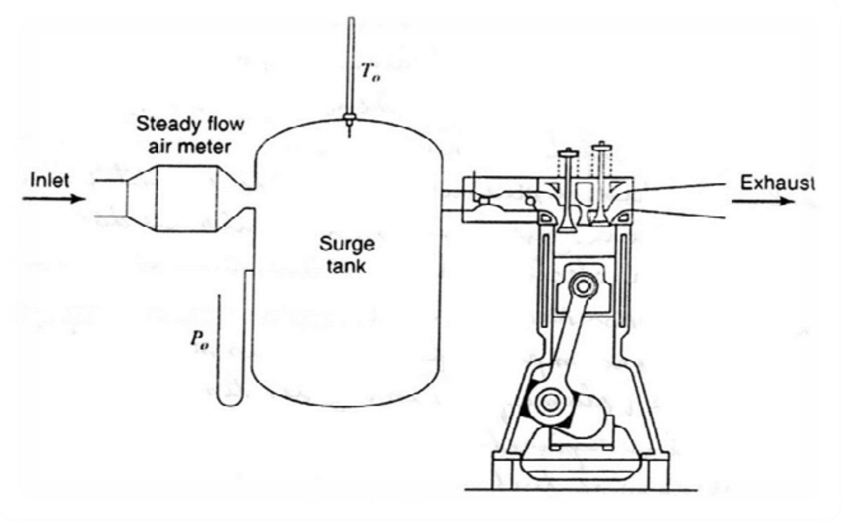


Figure (6-2): *Testing equipment for measured of air consumption*

b) Positive – Displacement meters:

Positive – displacement meters are very accurate, their working principle is shown in figure (6-3), as the impellers rotate, a fixed volume of air is alternately trapped between each impeller and the casing. This occurs four times for each complete revolution of both impellers.

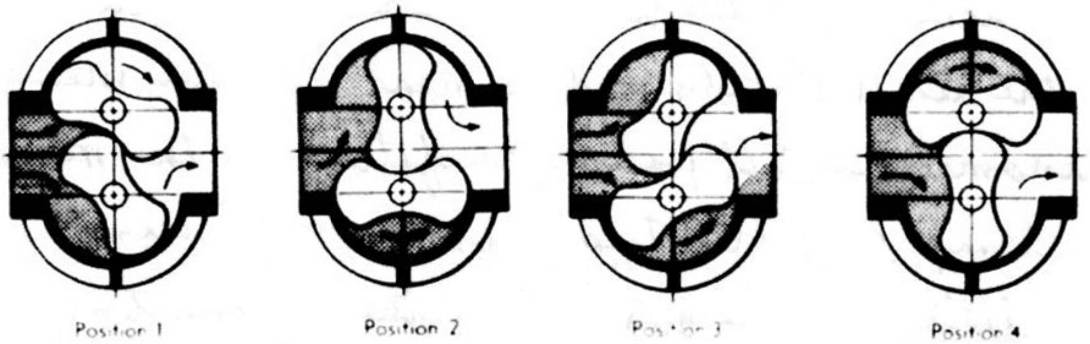


Figure (6-3):*Rotary positive displacement meter*

c) Viscous – flow air meter:

The meter is show diagrammatically in figure 6-4. It uses an element where viscous resistance is the principle source of pressure loss and kinetic effects are small. With the air box the flow is proportional to the square root of the pressure difference, while the viscous resistance is directly proportional to the air velocity and is measured by means of an inclined manometer. Felt pads are fitted in the manometer connections to damp out fluctuations, an additional damping vessel is fitted between the meter and the engine to increase the accuracy by reducing the effect of pulsations.

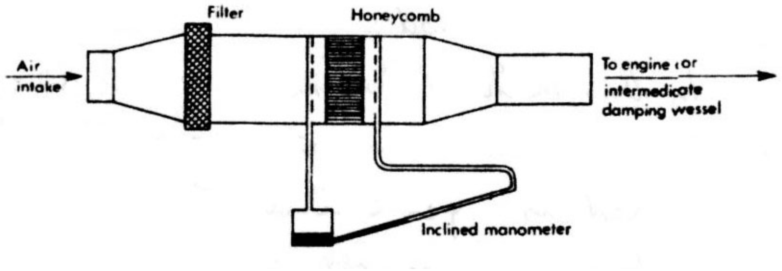
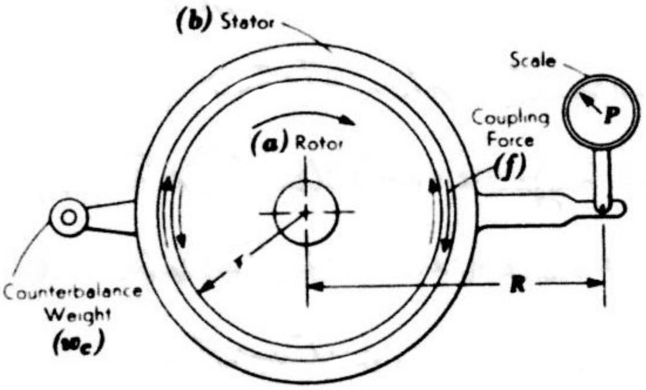


Figure (6-4): *Viscous- flow air meter*

4-Measurement of engine torque and power:

Any apparatus that permits the measurement of torque and power of the engine is called a "***dynamometer***". There are many types of dynamometers; all operate on the principle illustrated in fig. (6-5). Here the rotor (a), driven by the engine to be tested, is couple (electrically, magnetically, hydraulically or by friction) to the stator (b). In one revolution of the shaft, the peripherally of the rotor moves through a distance($2\pi r$) against the coupling force f (drag force).

Fig.6-5 *The dynamometer principle*



Thus the work per revolution is:

$$\text{Work} = 2\pi r f$$

The external moment, which is the product of the reading p of the scale (could be a beam balance or weights) and the arm R, must just balance the turning moment, which is r×f;

$r \times f = R \times P$

$\text{Work} = 2\pi RP$

Work per minute $= 2\pi RPN$ (N is engine speed in rpm)

Power is defined as the time rate of doing work, i.e.

$\text{Power} = 2\pi RPN$

where R in meters and P in Newton, then

$$power = \frac{2\pi RPN}{1000 \times 60} \quad kW$$

a) Fluid Dynamometers:

They absorb engine energy in water or oil pumped through orifices or dissipated with viscous losses in a rotor– stator combination. Large energy can be absorbed in this manner. Fluid brakes fall into two classes; the “friction" and the “agitator" type. In the friction type the coupling force arises from the viscous shearing of fluid between the rotor and stator, while in the agitator type the coupling force arises from the change in momentum of fluid as it is transported from rotor vanes to the stator vanes and back again.

Figure (6-6) illustrates Heenan – Froude hydraulic dynamometer. Here, the vanes of the rotor direct the water outward toward the stator vanes which redirect it back into the rotor. This highly turbulent process repeats itself again and again. The change of momentum experienced by the water as it changes direction is manifested as a reaction force on the stator housing.

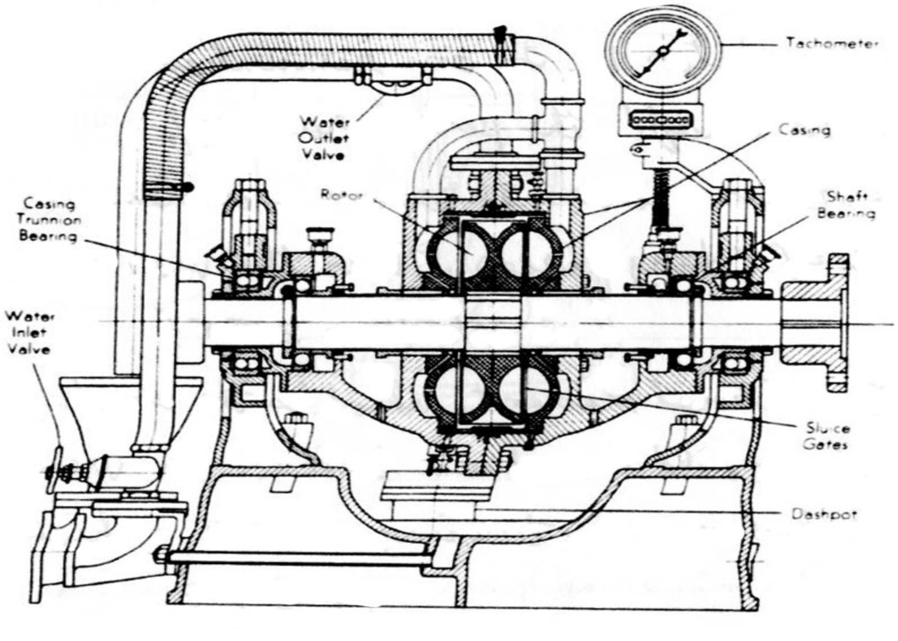


Figure (6-6): *Cross-section through casing of Froud dynamometer*

b) The eddy – current Dynamometer:

It consist of disk (d) which, driven by the engine under test, turns in a magnetic field, see figure (6-7). The strength of the field is controlled by varying the current through a series of coils (g) located on both sides of the disk (d).The revolving disk; act as a conductor cutting the magnetic field. Currents are induced in the disk and, since no external circuit exists, the induced current heats the disk. The temperature rise of the stator (a) is controlled by the flow of water in channels (h).

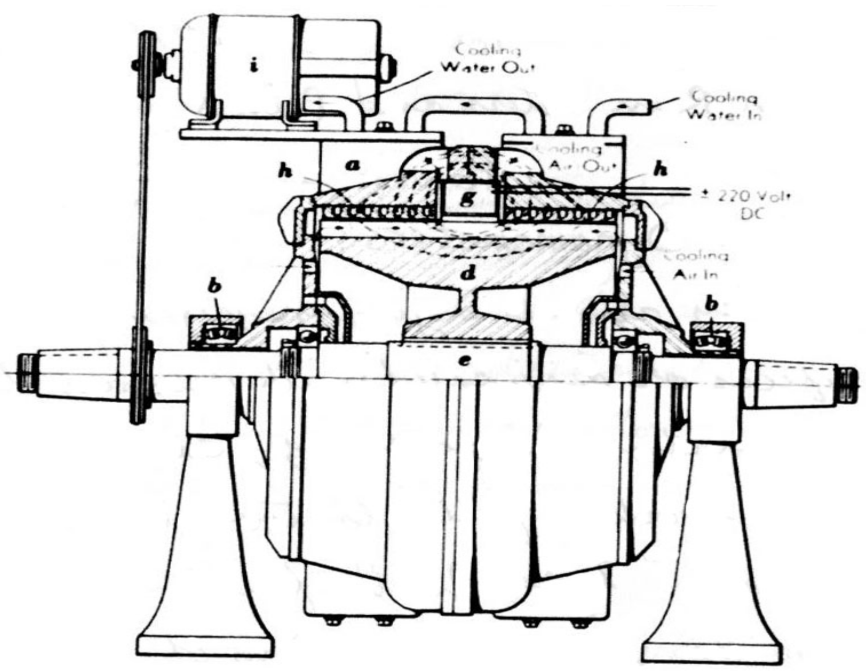
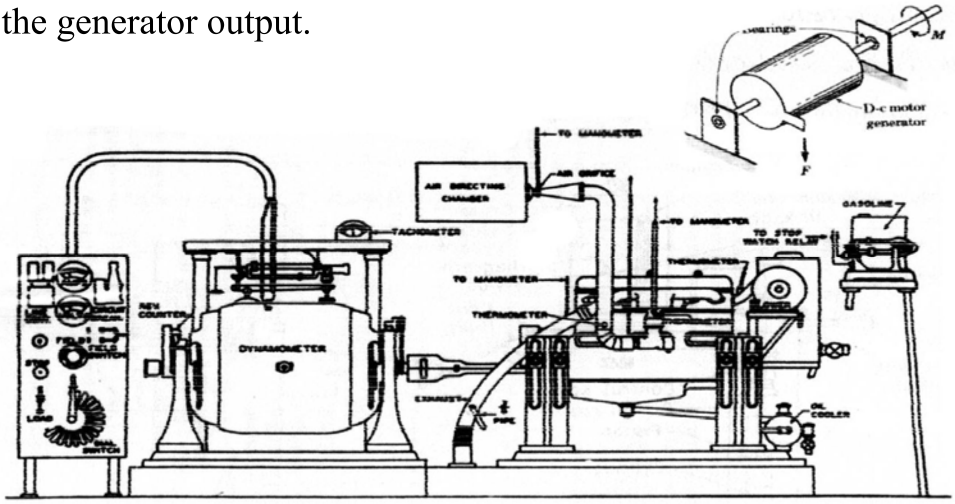


Figure (6-7): *Cross- section of eddy- current dynamometer*

c) The electric dynamometer:

The electric dynamometer; as shown in fig.(6-8) can operate either as a motor to start and drive the engine at various speeds or as a generator to absorb the power output of the engine. The load is easily varied by changing the amount of resistances in the circuit connected to the generator output.

Figure (6-8*): set- up*
of engine and dynamometer



6-Measurment of engine indicated power:

There are two methods of finding the indicated power of an engine:
i-By taking the indicator diagram with the help of an indicator.
ii-By measuring b.p and f.p separately and adding the two.

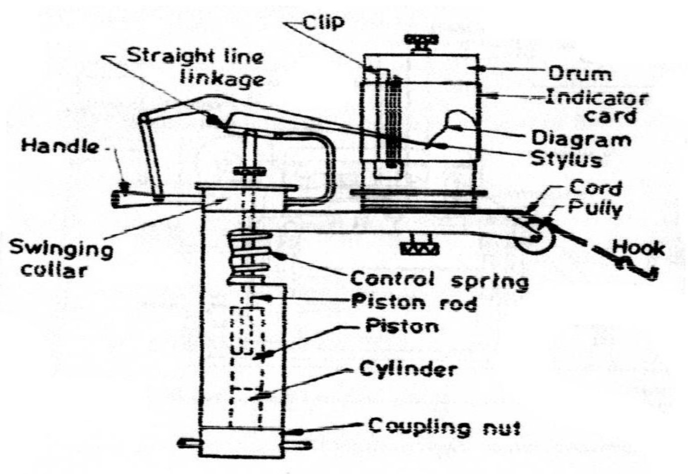
i- Indicator Diagram:

The device which measures the variation of the pressure in the cylinder over the cycle is called an ***indicator*** and the plot (diagram) of such information obtained is called ***indicator diagram***. There are two types of indicator diagrams which can be taken from various indicators, these are:

1- Pressure – volume (p–v) plot.
2- Pressure – crank angle (p–θ) plot.

There are number of indicators in use. However, only some representative types would show here:

a) Piston indicator (fig. 6-9)
b) Balance– Diaphragm (Farnborough balanced engine) indicator (fig. 6-10)



Figure(6-9) *Piston indicator*

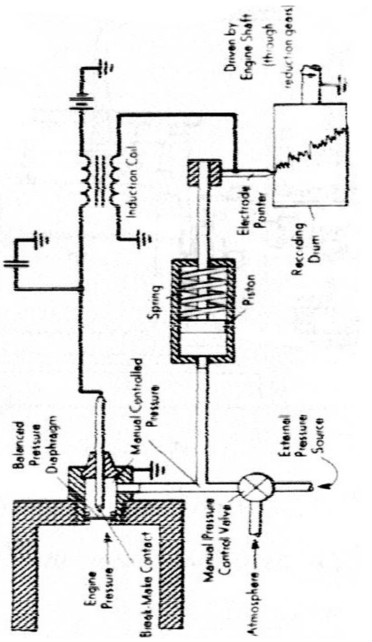


Figure (6-10)*: Schematic diagram of balanced- diaphragm type indicator*

b) Transducers and electronic indicators:

In general, a transducer is any device which converts a non-electrical quantity into an electrical signal. Examples of quantities which can be converted to electrical signals are; displacement, velocity, acceleration, and force. The electrical properties of many materials change when the material is subjected to a mechanical deformation. This is the characteristics upon which all pressure transducers depend. Resistive (strain– gauge), capacitive, or piezoelectric elements are the most common types of pressure pickups for engine work.

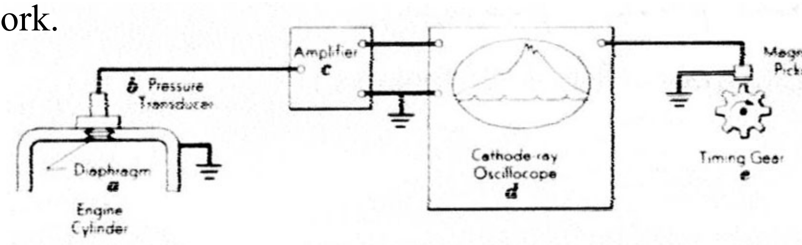
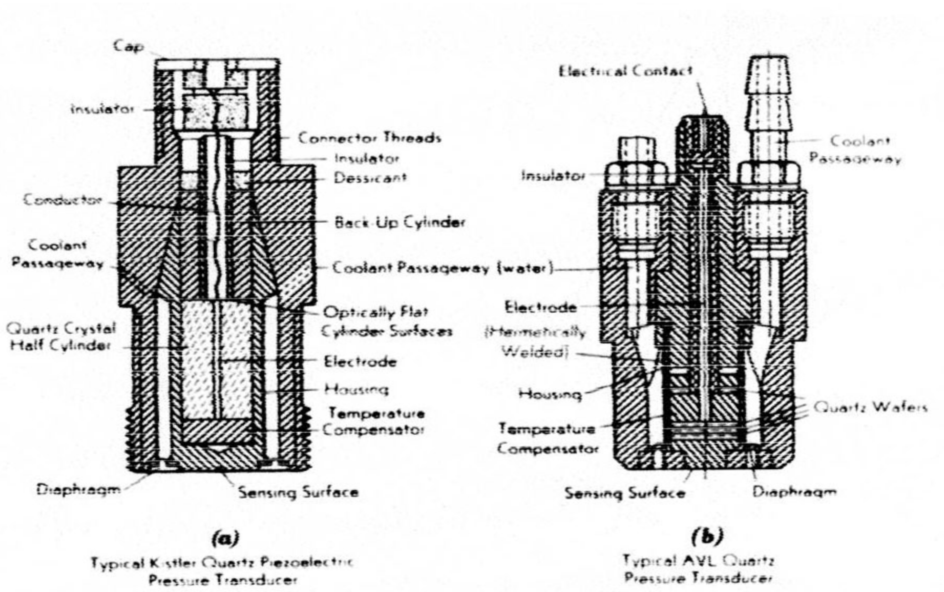


FIG. **5-4.** Elements of an electrical instantaneous pressure indicator

Figure (6-11) *Elements of an electrical instantaneous pressure* transducer

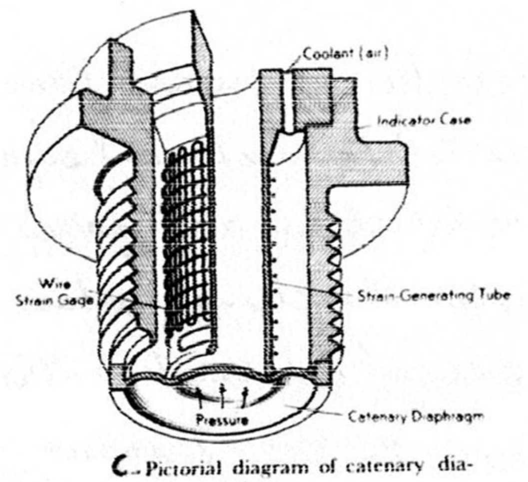
Fig. (6-11) shows a continuous – pressure system with a pressure pickup (b) (various types of pressure pickups are feasible). Fig. 6-12 shows different examples of pressure transducers, fig. (6-12 a and b) explain piezoelectric transducers, fig. (6-12b), indicate a strain – gauge transducer, while fig.6-12 d shows an electromagnetic pressure pickup.



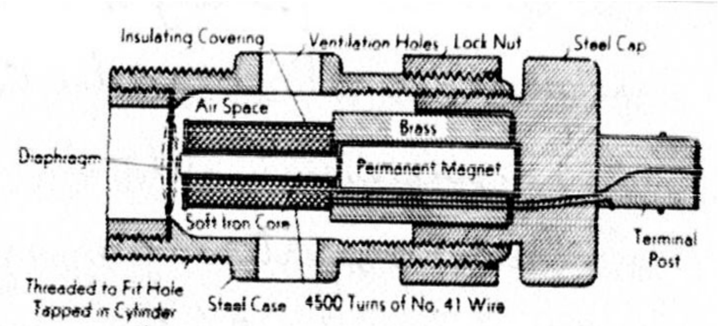
(a)
Typical Kistler Quartz Piezoelectric Pressure Transducer

(b)
Typical AVL Quartz Pressure Transducer

Figure (6 – 12): *a & b*



C- Pictorial diagram of catenary diaphragm pressure pickup.



d- Electromagnetic pressure pickup. (Courtesy of Electro Products Laboratory, Chicago.)

Figure (6-12)*: c & d*

7-Pressure Diagrams for I.C engines:

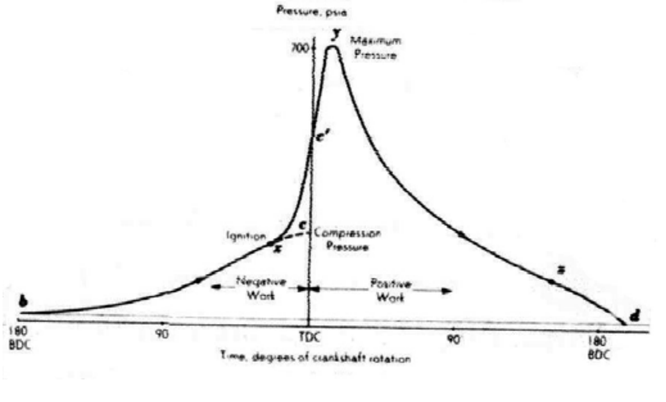


Figure (6-13): *Typical p−t diagram for SI engine at wide- open throttle*

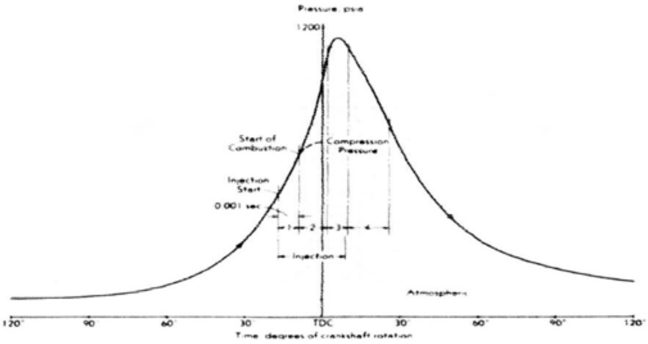


***Figure (6-14)** p-t diagram for mechanical- injection CI engine at full load*

7-Measurement of friction power (f.p):

The friction power is nearly constant at a given engine speed. Friction has a dominating effect on the performance of the engine. Frictional losses are dissipated to the cooling system as they appear in the form of heat.

Measurement of friction power is important for having better understanding on how the engine output can be increased. Methods of measuring the friction power are as follows:

i-Measurement of the i.p. and b.p. by the methods described previously for the engine at identical working conditions.

ii-Motoring test:

In this test; the engine is first run to measure the b.p at a given speed, then the fuel supply (or the spark) is cut-off and the dynamometer is converted to run as motor to drive the engine (motoring) at the same speed and keeping other parameters the same. The power supplied to the motor is measured which is a measure of the friction power (f.p).

The main objection to this method is that the engine is not firing, which leads to make running conditions are not similar. The pressure and temperature of cylinder contents, cylinder and piston surfaces are not the same.

iii-Morse test:

This test is only applicable to multi-cylinder engines. The engine is run at the required speed and the torque is measured. One cylinder is cut out, the speed falls because of the loss of power with one cylinder cut out, but is restored by reducing the load. The torque is measured again when the speed has reached its original value. If the values of i.p. of cylinders are denoted by I_1, I_2, I_3, and I_4(considering a four – cylinder engine), and the power losses in each cylinder are denoted by L_1, L_2, L_3 and L_4, then the value of b.p, B, at the test speed with all cylinders firing is given by:

$$B=(I_1-L_1)+(I_2-L_2)+(I_3-L_3)+(I_4-L_4)$$

If number 1 cylinder is cut out, then the contribution I_1 is lost; and if the losses due to that cylinder remain the same as when it is firing, then the b.p B_1 now obtained at the same speed is:

$$B_1= (0 -L_1)+(I_2-L_2)+(I_3-L_3)+(I_4-L_4)$$

Subtracting the second equation from the first given

$$B - B_1 = I_1$$

By cutting out each cylinder in turn the values I_2, I_3 and I_4 can be obtained, then:

$$I = I_1 + I_2 + I_3 + I_4$$

iv- Willan's line:

In this method gross fuel consumption versus b.p at a constant speed is plotted. The graph drawn is called the "Willan's line" and extrapolated back to cut the b.p axis at the point A. OA represent the power loss of the engine at this speed. The fuel consumption at zero b.p is given by OB; this would be equivalent to the power loss OA. This test is applicable to C.I. engines only.

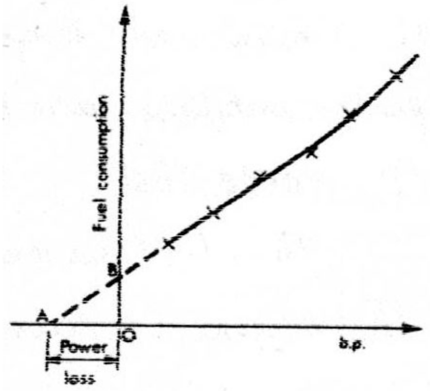


Figure (6-15): *Willan's line*

8 – Heat balance of Engine:

The main components of the heat balance are:

1- Heat equivalent to the b.p of the engine.
2- Heat rejected to the cooling medium.
3- Heat carried away from the engine with the exhaust gases.
4- Unaccounted losses.

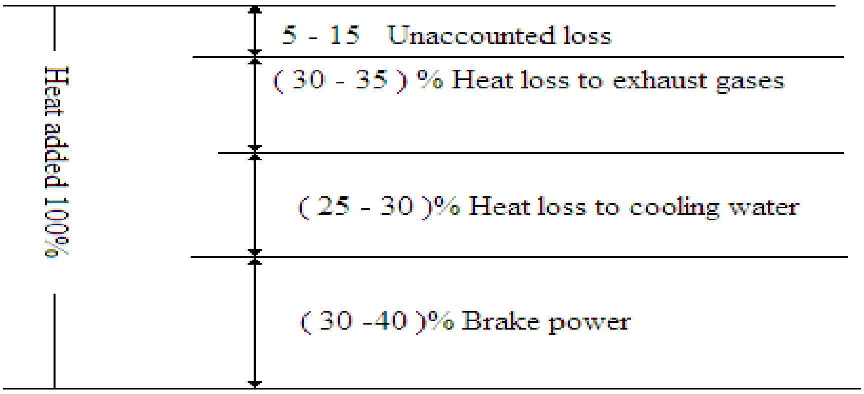


Figure (6-16) Heat balance diagram (or chart)

The following table gives the approximate percentage values of various losses in SI and CI engines:

Engine	% b.p	% heat to cooling water	% heat to exhaust gases	% unaccounted loss
S.I.	21-28	12-27	30-55	0-15
C.I.	29-42	15-35	25-45	10-20

Performance Characteristics:

The modern I.C engines have; higher ratios of power /weight than earlier types, increased values of (bmep) and thermal efficiency, and lower (s.f.c). At present time in the automotive field; the petrol engine is highly developed and flexible, but there is an increasing competition from the diesel engine. Brake thermal efficiencies of 25 to 35% are usual with S.I. engines and may reach 50% in diesel engines.

For comparing the performance of engines, a number of standards are available:
1-Specific fuel consumption (kg/kW.h).
2-Brake means effective pressure, bmep (kPa).
3-Specific weight (Weight of engine per kW, kg/kW)
4-Output per unit displacement kW per m^3)

Most of the performance factors are directly related to atmospheric conditions, so comparison between engines should be performed at similar atmospheric conditions.

The tests on I.C. engines can be divided into two types:
1-Variable – speed test.
2-Constant – speed test.

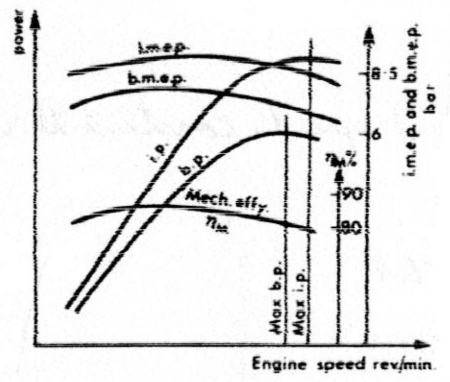
1-Variable – sped test:

Variable – speed tests can be divided into full – load tests, where maximum power and minimum s.f.c at each different speed are the objectives, and part – load tests to determine variation in the s.f.c.

a) Full – load test with SI engine:

The throttle is fully opened and the lowest desired speed is maintained by brake load adjustment. The spark is adjusted to give maximum power at this speed. The test is started by the watch governing the fuel consumption, the test ended at the time the fuel- consumption test has been completed. During this interval of time, the average speed, brake load, temperatures, fuel weight … etc., are recorded, then load is adjusted for the next run at different speed.

After the completion of the test, the required results are calculated, and performance curves are drawn and a typical example is shown in fig. 5.15. The variation of volumetric efficiency with speed is indicated in fig. 5-16, and that of mechanical efficiency with speed in fig. 6-15.



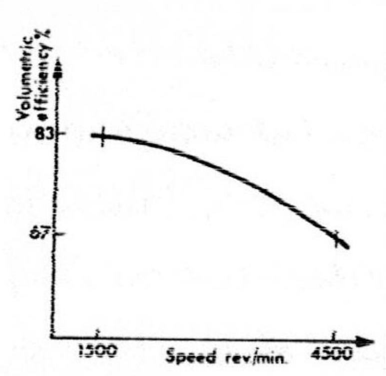


Figure (6 – 15) *Figure (6 – 16)*

b) Part – load test:

To run a part – load test at variable speed, say $\frac{1}{2}$ load, power reading of half the maximum power at each speed are obtained by varying the throttle and brake setting.

2-Constant – speed test:

Constant – speed test is run with variable throttle from no load to full load in suitable steps of load to give smooth curves. Starting at zero load, the throttle is opened to give the desired speed. Then a load is put on the engine and the throttle is opened wider to maintain the same constant speed as before, and the second run is ready to start. The last run of the test is made at wide-open throttle. In a CI-engine test the last run would show smoke in the exhaust gas.

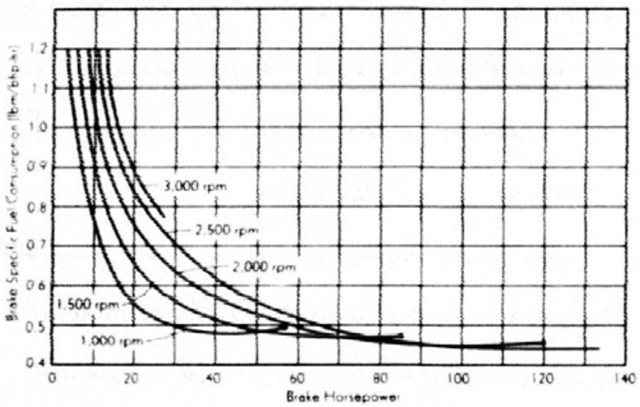


Figure (6 – 17) Constant speed, Variable throttle, test of automotive S.I. engine

3-Consumption loop test:

This test is carried out at constant speed, constant throttle opening, and constant ignition setting. The specific fuel consumption is plotted to a base of "bmep" and a "hook curve" is obtained. For a single cylinder at full throttle the curve is defined as in fig. 6-18.

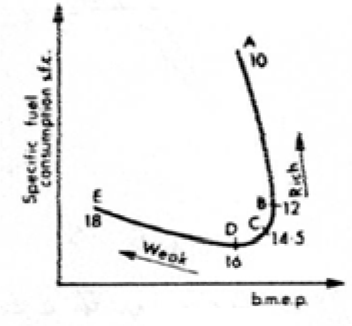


Figure (6 – 18)

The A/F ratio is a minimum at A(i.e. richest mixture). As the A/F ratio is increased the "bmep" increases until a maximum is reached at B (usually for an A/F ratio between 10/1 and 13/1). Further increase in A/F ratio produce a decrease in "bmep" with increasing economy until the position of maximum economy is reached at D. beyond D, for increasing A/F ratios, both "bmep" and consumption values are adversely affected. Near the point A the engine could be running unsteadily and there may be combustion of the mixture in the exhaust system. At E, with the weakest mixture, running will be unsteady and the combustion may be slow. Point C is the point of chemically correct A/F ratio.

For multi-cylinder engines the consumption loops are less distinct, but are generally similar to that for the single – cylinder engine. This is also true for tests made at part throttle opening. A series of reading obtained at different throttle positions at constant speed is shown in fig 6-19.

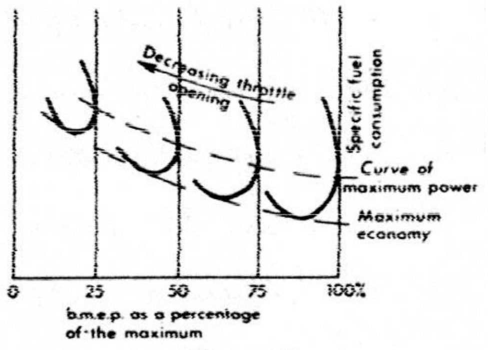


Fig.5 .19

Example 6-5:

A four – cylinder petrol engine has a bore of 57mm and a stroke of 90mm. its rated speed is 2800 rpm and it is tested at this speed against a brake which has a torque arm of 0.356m. The net brake load is 155N and the fuel consumption is 6.74 l/h. The specific gravity of the petrol used is 0.735 and it has a lower calorific value of;44200 kJ/kg. A Morse test is carried out and the cylinders are cut out in the order 1,2,3,4, with corresponding brake load of 111,106.5, 104.2 and 111 N, respectively. Calculate for this speed, the engine torque, the bmep, the brake thermal efficiency, the specific fuel consumption, the mechanical efficiency and the imep.

Solution:

Torque T=RP=0.356 × 155 = 55.2 Nm

$$b.p = 2\pi\, NT = \frac{2\pi \times 2800 \times 55.2}{60 \times 10^3} = 16.2\ kw$$

$$bmep = \frac{b.p \times 2}{ALNn} = \frac{16.2 \times 2 \times 4 \times 60 \times 10^3}{\pi \times 0.057^2 \times 0.09 \times 2800 \times 4 \times 10^5} = 7.55 \text{ bar}$$

$$\eta_{BT} = \frac{b.p}{\dot{m}_f \times C.V} = \frac{16.2}{0.001377 \times 44200} = 0.266 \text{ or } 26.6\%$$

$$\text{Where } m_f = \frac{6.74}{3600} \times 1 \times 0.735 = 0.001377 \quad \text{kg/s}$$

$$\text{sfc} = \frac{\dot{m}_f}{\text{b.p}} = \frac{0.001377 \times 3600}{16.2} = 0.306 \quad \text{kg/kW.h}$$

The indicated load for the engine is calculated by the Morse test method as:

$I=I_1+I_2+I_3+I_4$

and: $I_1= B-B_1=155 - 111=44$ N

$I_2= B-B_2=155-106.5=48.5$ N

$I_3= B-B_3=155-104.2=50.8$ N

$I_4= B-B_4=155 - 111=44$ N

$I=44+48.5+50.8+44=187.3$ N

$$\eta_M=\frac{b.p}{i.p}=\frac{155}{187.3}=0.828 \text{ or } 82.8\%$$

$$i.p=\frac{16.2}{0.828}=19.57 \text{ kw}$$

$$\text{bmep}=\eta_\text{M}\times imep$$

$$i.e \quad \text{imep}=\frac{7.55}{0.828}=9.12 \text{ bar}$$

Solved problems – chapter (6):

Ex.1-31 six – cylinders SI engine operates on a four – stroke cycle and run at 3600 rpm. The compression ratio is 9.5 the length of connecting rode is 16.6cm, and the bore equal the stroke. Combustion ends at 20° after TDC calculate: (1) Cylinder bore and stroke, (2) average piston speed, (3) clearance volume of one cylinder, (4) the distance piston has traveled from TDC at the end of combustion, (5) volume of the combustion chamber at the end of combustion.

Solution

1- Volume of one cylinder, $V_s=\frac{3000}{6}=500 \text{ cc }=0.0005\text{m}^3=\frac{\pi}{4}B^2S$

$0.000637=B^3 \Rightarrow B=0.086m=8.6cm=S$

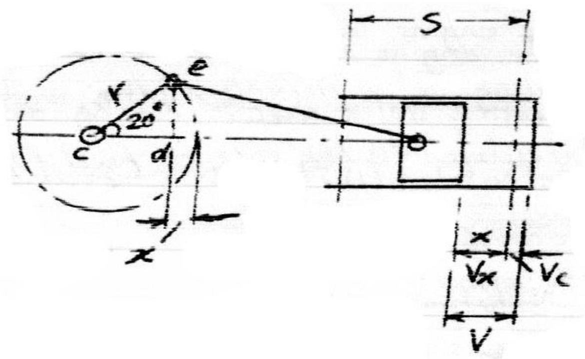
2-$V_m=\frac{2SN}{60}=\frac{2\times0.086\times3600}{60}=10.32 \text{ m/s}$

3- $r=\frac{V_s+V_c}{V_c}=9.5=\frac{0.0005+V_c}{V_c}$

i.e. $V_c=0.000059\text{m}^3=59 \text{ cm}^2$

4- Volume at any C.A. = V_c+V_x

$V=V_c+x\times\frac{\pi}{4}B^2$ (B=bore)



$5-\text{x} = \text{r}(1\text{-}\cos\theta)$, $\quad \text{r} = \frac{\text{S}}{2} = 4.3$ cm

x = 4.3 (1-cos 20) = 0.26 cm, $\quad$ V=59+ $\frac{\pi}{4}(9)^2\times$ 0.26 = 75.54 cm^3

Ex.2- The engine in example 1 is connected to a dynamometer which gives a brake output torque of 205 Nm at 3600 rpm. At this speed air enters the cylinder at 85 kPa and 60°C, and the mechanical efficiency of the engine is 85%. Calculate: (1) b.p, (2) i.p, (3) bmep, (4) imep, (5)fmep, (6) f.p, (7) engine specific volume.

Solution

$$1\text{-}\ b.p = 2\pi\,NT = 2\pi \times \frac{3600}{60} \times 205 = 77.3 \ \text{ kW}$$

$$2\text{-}\ i.p = \frac{b.p}{\eta_M} = \frac{77.3}{0.85} = 90.9 \ \text{ kW}$$

$$3\text{-}\ bmep = \frac{b.p \times z}{LANn} = \frac{77.3 \times 60 \times 2}{0.0005 \times 3600 \times 6} = 859 \ \text{ kPa}$$

$$4\text{-}\ imep = \frac{859}{0.85} = 1010.5 \ \text{ kPa}$$

$$5\text{-}\ fmep(P_f) = imep - bmep = 1010.5 - 859 = 151.57 \ \text{ kPa}$$

$$6\text{-}\ f.p = P_f LAN \times \frac{n}{z} = 151.57 \times 0.0005 \times 3600 \times \frac{6}{2} \times \frac{1}{60} = 13.64 \ \text{ kW}$$

$$7\text{-Engine specific volume} = \frac{\text{swept volume}}{\text{brake power}} = \frac{3\text{l}}{77.3} = 0.0388 \ \text{ L/kW}$$

The inverse of the specific volume is = 25.8 kW/L

Ex.3-The engine in example 2 is running with A/F ratio =15,afuel of heating value;44000kJ/kg and a combustion efficiency of 97% calculate: (1) the rate of fuel flow. (2) η_{BT} , (3) η_{IT} ,(4) η_V , and brake specific consumption.

Solution:

1- The clearance volume of the engine = 0.000059 m^3(example 1)

$$m_\alpha = \frac{PV_{BDC}}{RT} = \frac{P(V_c + V_s)}{RT} = \frac{85(kPa)(0.0005 + 0.000059)m^3}{0.287(kJ/kgK)(333K)} = 0.0005 \ \text{ kg}$$

$$m_f = \frac{m_a}{\frac{A}{F}} = \frac{0.0005}{15} = 0.000033\,\text{kg}_\text{f} \text{ Per cylinder per cycle}$$

$$\dot{m}_f = (0.000033 \frac{kg}{cycle.cylinder})(6\,\text{cylinder})(\frac{3600}{60} rev/s)(\frac{1}{2} effective\,\text{cycle/cylinder})$$

$$= 0.006 \ \text{ kg/s}$$

$$2\text{-}\ \eta_{BT} = \frac{\text{b.p}}{\dot{\text{m}}_\text{f} \times \text{C.V} \times \eta_\text{c}} = \frac{77.3}{0.006 \times 44000 \times 0.97} = 0.302 \ \text{ or } \ 30.2\%$$

$$3\text{-}\ \eta_{IT} = \frac{\eta_{BT}}{\eta_m} = \frac{0.302}{0.85} = 0.355 \ \text{ or } \ 35.5\%$$

$$4\text{-}\ \eta_V = \frac{m_a}{\rho_a V_s} = \frac{0.0005\,\text{kg}}{(P/RT)(0.0005\,\text{m}^3)} = \frac{1}{1.181} = 0.847 \qquad \text{or } \ 84.7\%$$

5- $bsfc = \dfrac{\dot{m}_f}{b.p} = \dfrac{0.006\,\frac{\text{kg}}{\text{s}}}{77.3\,\text{kw}} = 7.76\,\text{kg/kW.s} = 279\,\text{kg/kW.h}$

$$\rho_{\text{air}} = \frac{\text{P}}{\text{RT}} = \frac{1.013 \times 10^{2}}{0.287\ (15 \times 273)}$$

$$\eta_{\text{V}} = 0.76 = 76\,\%$$

Ex.4- A six-cylinder 4-stroke cycle petrol engine is to be designed to develop 300 kW of (b.p) at 2500 rpm the bore / stroke ratio is to be 1:1.25. Assuming η_m=83% and an indicated mean effective pressure of 9.5 bar, determine the required bore and stroke. If the compression ratio of the engine is to be 6.5 to 1, determine consumption of petrol in kg/h and in kg/bp.hr. Take the ratio of the indicated thermal efficiency of the engine to that of the constant volume air standard cycle as 0.55 and the calorific value of the petrol as; 44770kJ/kg.

Solution

b.p=300 kW

$$\eta_m = \frac{b.p}{i.p} \quad ; \qquad \text{i.p} = \frac{300}{0.83} = 361\ kW$$

$$P_i = 9.5\ \text{bar} \quad \&\ \ \text{N} = 2500\ \text{rpm}$$

$$\text{i.p} = \frac{\text{P}_\text{i} \times L \times A \times N \times n}{60} \times \frac{1}{2} \quad (4 - strok\ \ engine\ )$$

$$1000 \times 361 = \frac{9.5 \times 10^{5} \times (LA) \times 2500 \times 6}{60 \times 2}$$

$$(LA) = 0.00304 \quad \text{m}^{3}$$

Let Diameter = D, $\therefore$ L=1.25D &A=$\dfrac{\pi}{4}$ D^2

$$1.25D \times \frac{\pi}{4} D^{2} = 0.00304$$

$$D^{3} = 0.003096 \quad \Rightarrow D = 0.146\ m$$

$$D = 14.6\ \text{cm} \quad \&\ \text{L} = 1.25\text{D} = 18.25\ \text{cm}$$

r = 6.5 & γ = 1.4

$$\eta_{a.s} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{6.5^{1.4-1}} = 52.6\%$$

$$\eta_r = \frac{\eta_{th}}{\eta_{A.S}} \times 100$$

$$55 = \frac{\eta_{th}}{52.6} \times 100$$

$$\eta_{th} = \frac{55 \times 52.6}{100} = 28.9\%$$

$$\eta_{th} = \frac{i.p \times 60}{\text{heat in fuel supplied /min}} \times 100$$

$\therefore$ Heat in fuel supplied /min = $\dfrac{361 \times 60}{0.289} = 74948\ \text{kJ}$

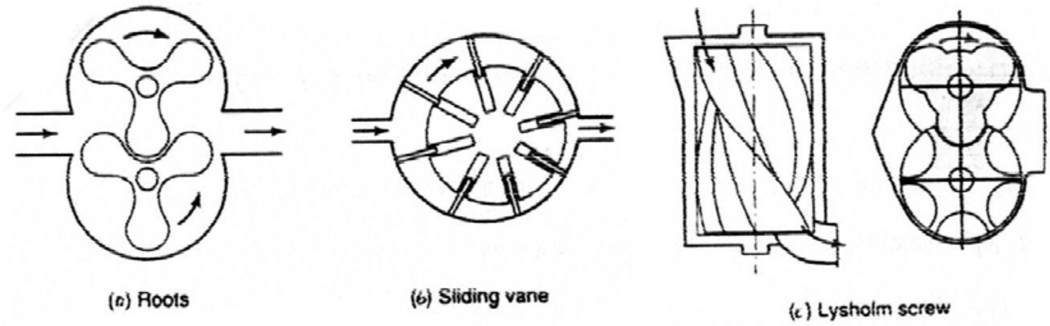
$\therefore$ Consumption of petrol in kg/h $= \dfrac{74948}{44770} = 100.4$ & $\dfrac{\text{kg}}{\text{kW.h}} = \dfrac{100.4}{300} = 0.33$

CHAPTER (7)

SUPERCHARGING

The power and efficiency of an internal combusting engine can be increased with the use of an air compression device such as a supercharger or turbocharger. Increasing the pressure and density of the inlet air will allow additional fuel to be induced into the cylinder, increasing the power produced by the engine. Spark ignition engines are knock limited, restricting the allowable compressor pressure increase, in many cases the compression ratio of a SI engine is reduced. Superchargers and turbochargers are used extensively on a wide range of diesel engines, since they are not knock limited.

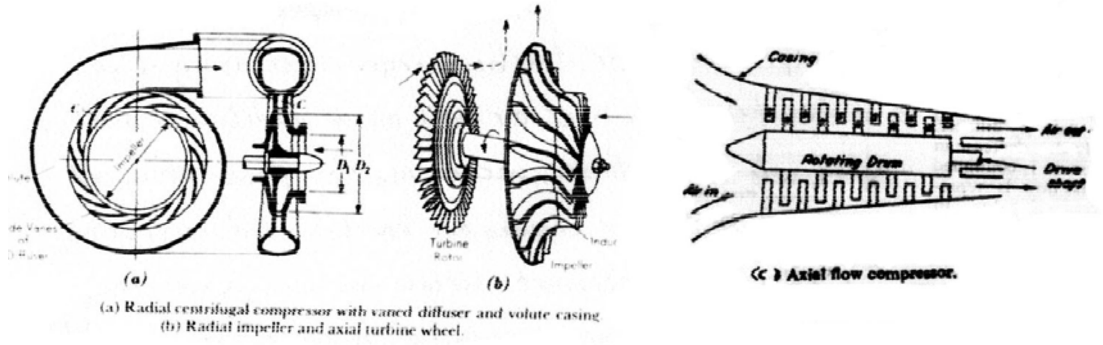
The types of compressors used on internal combustion engines are primarily of two types: positive displacement and dynamic. With a positive displacement compressor, a volume of gas is trapped, and compressed by movement of a compressor boundary element. Three types of positive displacement compressors are the roots, vane, and screw compressor, as shown in figure 7-1.



(a) Roots (b) Sliding vane (c) Lysholm screw

Figure (7 – 1) Types of positive displacement compressors

A dynamic compressor has a rotating element that adds tangential velocity to the flow which is converted to pressure in a diffuser. Two types of dynamic compressors and turbines are; radial (centrifugal) and axial as shown in figure 7-2.



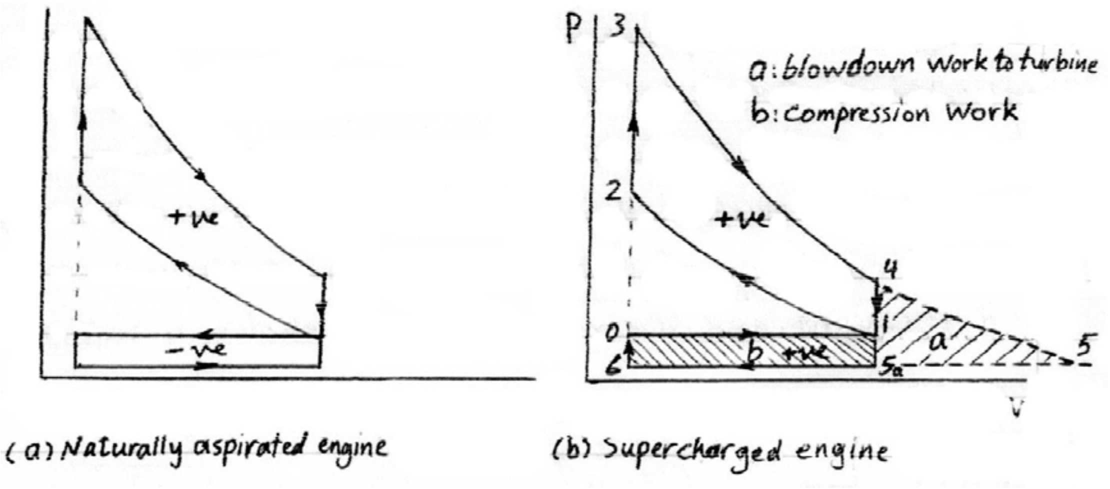
(a) (b)

(a) Radial centrifugal compressor with vaned diffuser and volute casing
(b) Radial impeller and axial turbine wheel.

(c) Axial flow compressor.

Figure (7 – 2)

7-1 Thermodynamic Cycle with Supercharging:



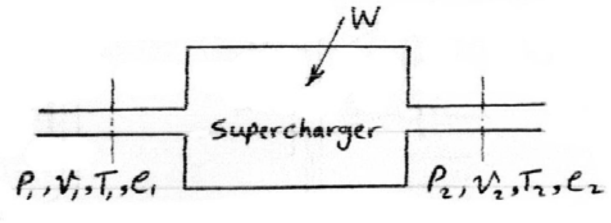
(a) Naturally aspirated engine (b) Supercharged engine

Figure (7 – 3)

The pumping loop of a supercharged engine is positive instead of negative. Hence to get the net indicated power (i.p), the power represented by pumping loop is to be added:

i.p=area 12341 + area 015a60

The power required for driving the supercharger can be calculated by considering the steady flow process as given in the figure. The air enters the supercharger at a pressure p_1 and has an internal energy e_1. The work supplied to the supercharger is W. The air leaves the supercharger at a pressure P_2 and has an internal energy e_2, then:



$$e_1+P_1V_1+W=e_2+P_2V_2+Q$$

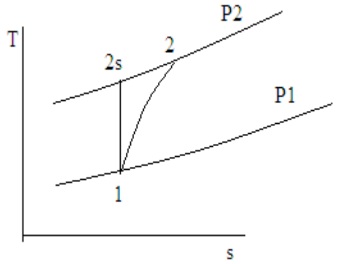
If we assume that the heat loss Q from the supercharger is zero, we get:

$$W=(e_2+P_2V_2)-(e_1+P_1V_1)$$

$$=h_2-h_1=C_p\,(T_2-T_1)$$

T_2 is the temperature at the end of compression in the supercharger, is given by:

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



η_s; is the adiabatic efficiency of the supercharger.

The power required to drive the supercharger is then equal to;

$$Wc=maC_pT_1\left\{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right\}\times\frac{1}{\eta_s}$$

This power can be supplied by a separate drive for the supercharger or by connecting the supercharger directly to the engine shaft or to gas turbine driven by the engine exhaust gases. In all cases the gain in the power output of the engine would be many times the power required to drive the compressor.

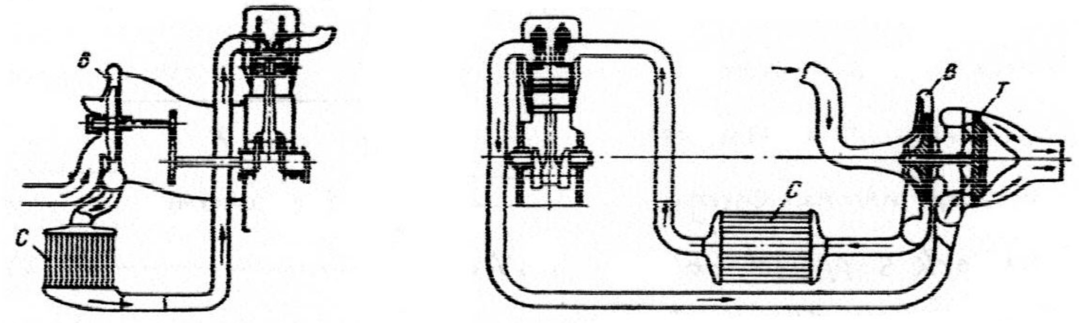


Fig. 6-4a Diagram of a mechanically supercharged engine:
B—centrifugal blower, C—supercharging air cooler

Fig 6-4b Diagram of a turbocharged engine:
B—centrifugal blower; C—supercharging air cooler; T—gas turbine

Figure (7 – 4 a & b)

7-2 TURBOCHARGERS:

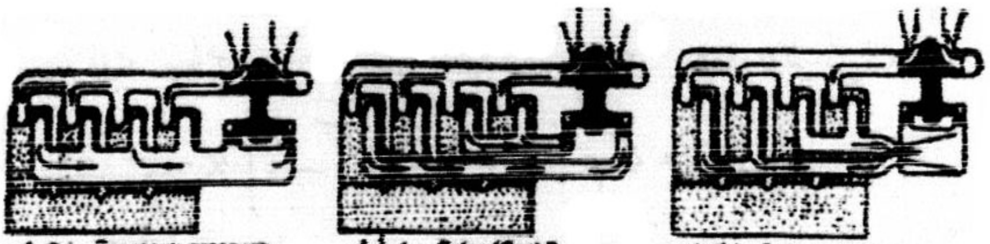
Turbochargers are made in all sizes to fit the smallest as well as the largest engines. Typical example is shown in fig.7-4.

In order to supply sufficient energy to the turbocharger the exhaust valve is opened much before the BDC as compared to naturally aspirated engine. This allows the exhaust gasses to escape at a higher pressure and temperature giving the turbocharger enough energy to drive the compressor.

Methods of turbo charging:

Figure 7-5 shows various methods used for turbo charging of 4-stroke and 2-stroke diesel engines. The main types of turbo charging methods are:

a) **Constant pressure:**



(a) Constant pressure (b) Pulse (Buchi) (c) Pulse converter

Figure (7 – 5) Methods of turbo charging

The exhaust from various cylinders, discharge into a common manifold at pressures higher than atmospheric pressure and maintained at constant pressure during the whole cycle so that a pure reaction turbine can be used. This objective dictates a large exhaust manifold to absorb pressure fluctuations and therefore the kinetic energy in the exhaust blow down is dissipated.

b) Pulse (or Buchi):

In this system the objectives is to use the kinetic energy in the blow down process to drive the turbine, ideally, without increase in 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. The turbine has separate inlets and nozzle segments for each exhaust pipe.

c) Pulse converter:

Pulse converter allows converting the K.E. in the blow down process into a pressure rise at the turbine by means of one or more diffusers. Ideally, the advantages of both the pulse system and the constant – pressure system are gained.

<u>Arrangement of exhaust manifolds:</u>

1-Constant pressure:

The exhaust piping system in this arrangement is not complex. The exhaust pipe diameter must be 1.4 times the piston diameter.

2-Pulse blow down:

With this system the exhaust system and turbo charger location must be carefully designed to avoid interference with the scavenging process, especially for two-stroke engines.

Figure 7-6 shows the exhaust pressure diagram for a 4-stroke diesel engine with a firing order 1-3-4-2, having two exhaust manifolds, Cylinders 1 and 4 exhausting in one manifold, and cylinders 2 and 3 into another. The pressure diagrams of the two manifolds are super-imposed.

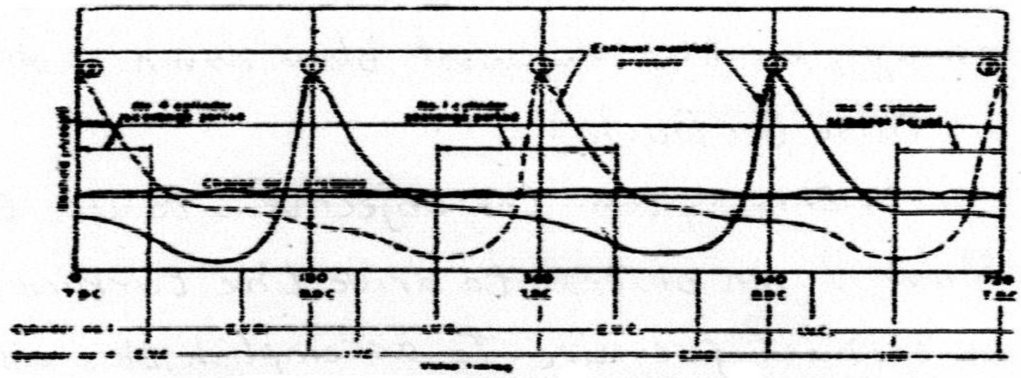


Figure (7 – 6) Exhaust manifold pressure diagram for a 4 – Stroke diesel engine with firing order 1 – 3 – 4 – 2.

The figure shows that during scavenge period the charge pressure is always more than the exhaust pressure. If the four cylinders are made to exhaust in a common exhaust, the exhaust pressure in cylinder 3 will be more than the charging air. That means a four cylinder engine requires two exhaust manifold, an eight cylinder engine require four exhaust manifolds and a five cylinder engine having equally spaced ranks require three manifold. Figure 7-7a shows the manifold arrangements used for four strokes in – line engines and figure 7-7b shows the manifold arrangements for 2-stroke engines.

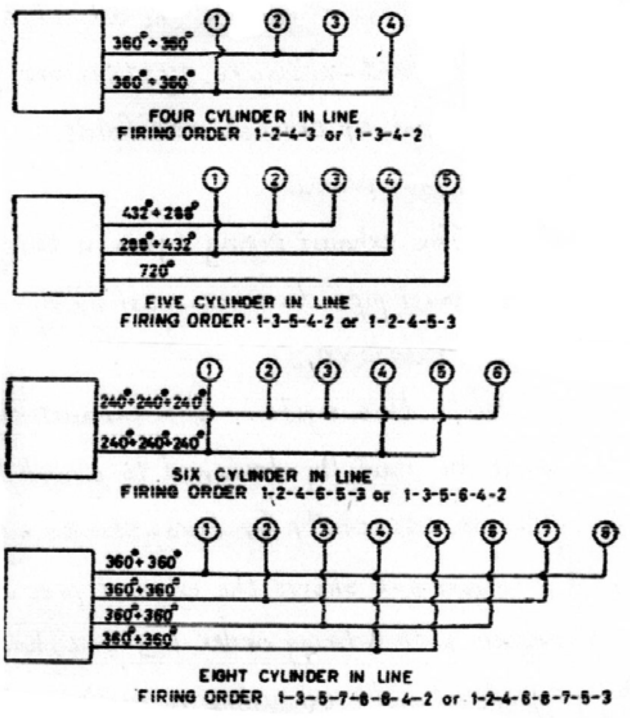


Figure (7 – 7a) Typical exhaust arrangement and cylinder Combination for 4 – stroke in line engines using pulse Turbo charging.

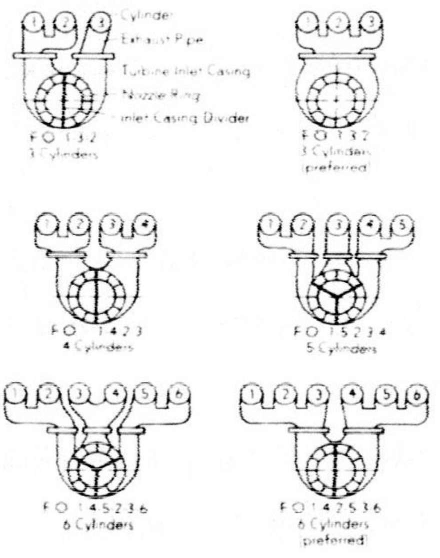


Figure (7 – 7b) Typical arrangement for pulse turbocharged, two – stroke engines.

Example 7-1

A 4-stroke diesel engine is to be designed to operate with the following characteristics at sea level, where the ambient conditions are 103 kPa and 10°C.

b.p=260 kW, s.f.c=0.244 kg/kW.h,

Speed = 1500 rpm, volumetric efficiency =0.78.

A/F ratio = 17:1,

Calculate the required engine capacity and the anticipated bmep.

The engine is fitted with a supercharger so that it may be operated at an altitude of 2700m where the atmospheric pressure is 73kPa.The power taken by the supercharger is 8% of the total power produced by the engine and the temperature of the air leaving the supercharger is 32°C. The A/F ratio and thermal efficiency remain the same, so as the η_v.Calculate the increase of pressure at the supercharger to maintain the same net power output. Take R=0.287 kJ/kg.K.

Solution:

Naturally aspirated:

Fuel consumption = $0.244 \times 260 = 63.44$ kg/h = 1.057 kg/min.

Air consumption = 1.057×17=17.97 kg/min.

$$\eta_v=\frac{\dot{V}}{V_s}=\frac{17.97\times0.287\times283}{1.03\times V_s\times750\times100}\qquad\text{(PV=mRT)}$$

$V_s = 0.0242\ m^3$

$$bmep=\frac{b.p.\times n}{LAN}=\frac{260\times2\times60}{0.0242\times1500}=859.5\ \text{kPa}$$

Supercharged:

Total power produced by the engine = $260+0.08\times260$=280.8 kW

This is by assuming ideal supercharging, i.e. all the power taken by the supercharger is added to the engine power.

Mass of fuel required per minute = $280.8\times\frac{0.244}{60}$=1.142kg/min

Mass of air/min. =1.142×17=19.41kg/min.

Volume of air induced at supercharger outlet conditions = $\frac{mRT}{P}$

$$\dot{V}=\frac{19.41\times0.287\times305}{p_2}=\frac{1699}{p_2}\ \text{m}^3/\text{min}$$

$$\therefore 0.78=\frac{1699}{p_2\times0.0242\times750}$$

$$p_2=120\ \text{kPa}$$

Increase of pressure required = 120 – 73= 47kPa

Pressure ratio of supercharger = $\frac{120}{73}$=1.644

Example 7-2:

A 4-stroke diesel engine of 3000 C.C. capacity develops 13.42 kW per $m^3$of free air induced per minute. When running at 3500 rev/min., it has a volumetric efficiency of 80 per cent referred to free air conditions of 1.03 bars and 27^oC. It is proposed to boost the power of the engine by supercharging by a blower of pressure ratio 1.7 and isentropic efficiency of 75%. Assuming that at the end of induction the cylinders contains a volume of charge equal to the swept volume, at the pressure and temperature of the delivery from the blower, estimate the increase in brake power to be expected from the engine. Take overall mechanical efficiency as 80 per cent.

Solution:

Swept volume, $V_s=3000\times\frac{3500}{2}=5.25\times10^6$ C.C or 5.25 m^3/min

Naturally aspirated induced air volume = 0.8×5.25=4.2 m^3/min.

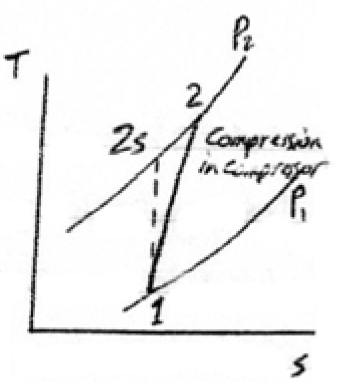
Blower delivery pressure = 1.7×1.03=1.751 bar

Temperature after isentropic compression = $T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 300(1.7)^{\frac{0.4}{1.4}} = 350\ K$

$$\eta_s = \frac{T_{2s} - T_1}{T_2 - T_1}$$

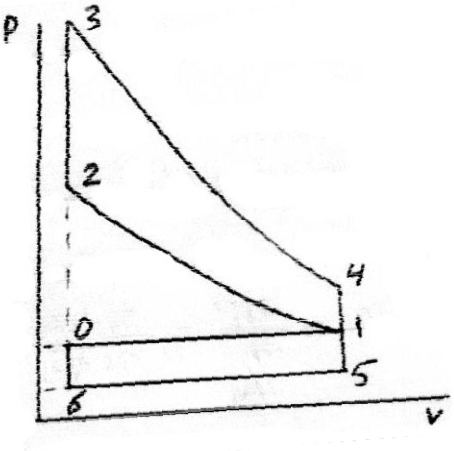
$$0.75 = \frac{350 - 300}{T_2 - 300}$$

$$T_2 = 300 + \frac{50}{0.75} = 366\ K$$



Blower delivery temperature = 366 – 273=93 K
The blower delivers; 5.25 m^3/min., at 1.751 bars and 366K, equivalent to a volume of:

$\frac{5.25\times1.751\times300}{1.03\times366} = 7.31\ \ m^3/\min.$ $\frac{p_1 v_1}{T_1}=\frac{p_2 v_2}{T_2}$, (at the ambient conditions of; 1.03 bars and 300 K).



By assuming ideal supercharging; from the P-V diagram:
Increase in (i.p) due to increased induction pressure

$$= \frac{\Delta P\times V_s}{60} = (1.751-1.03)\times5.25\times\frac{100}{60}$$
$$= 6.31\quad kW$$

Increase in induced volume = 7.31− 4.2=3.11 m^3/min
Increase in power from air induced = 13.42 × 3.11 =41.736 kW

Mass of air delivered by blower = $(1.751\times5.25\times10^2)/(0.287\times366)$
=8.75 kg/min.

Power required by the blower = $\dot{m}_a C_p T_1\left\{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} -1\right\}\times\frac{1}{\eta_s}$

$$= 8.75\times1.005\times366\ \{(1.7)^{0.286} -1\}\times\frac{1}{0.75}\times\frac{1}{60}$$
$$= 11.72\ \text{kW}$$

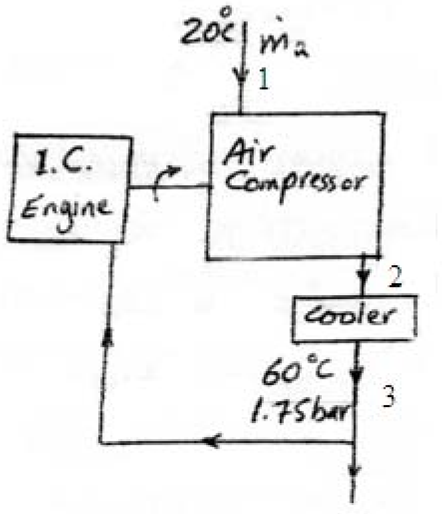
Total increase in b.p=41.736+6.31 × 0.8=46.78 kW
Net increase in b.p =46.78 − 11.72=35.06 kW

Example 7-3:

Four – stroke oil engine is used to drive an air compressor, the air enters the compressor at 20°C and is delivered to a cooler which removes heat at the rate of 1340 kJ/min. The air leaves the cooler at 60°C and 1.75 bars. Part of this air flow is used to supercharge the engine which has a volumetric efficiency of 0.7 based on induction conditions of 60°C and 1.75 bars. The engine which has six cylinders of 90mm bore and 100mm stroke runs at 2000rpm and delivers an output torque of 150 Nm. The mechanical efficiency of the engine is 0.75. Determine:

1- The engine indicated mean effective pressure.
2- The air consumption in kg/min.
3- The air flow into the compressor in kg/min.

Solution



b.p= $2\pi\,\mathrm{N\,T} = 2\pi\times2000\times150\times\frac{1}{60\times1000}$

$=31.42\ \mathrm{kW} = W_c$

i.p= $\frac{\mathrm{b.p}}{\eta_m} = \frac{31.42}{0.75} = 41.89\ \mathrm{kW}$

$$P_i = \frac{i.p\times z}{LAN\times n} = \frac{41.89\times2\times4\times60}{0.1\times\pi\times(0.09)^2\times6\times2000\times100}$$

$= 6.58\ \mathrm{bar}$

Engine swept volume = $6\times\frac{\pi}{4}(0.09)^2\times0.1\times\frac{2000}{2}$

$= 3.82\ \mathrm{m^3/min}.$

$\eta_v = \frac{\dot{V}}{V_s}$ $\qquad \dot{V} = 0.7\times3.82 = 2.674\,\mathrm{m^3/min}$

Aspirated air mass flow into the engine = $\frac{PV}{RT} = \frac{1.75\times2.674\times10^2}{0.287\times333} = 4.9\ \mathrm{kg/min}$

Now, work done on air in compressor = gain in enthalpy of air in compressor.

$W_c = \dot{m}_a\, C_{pa}\,(T_2 - T_1)$

$Q = \dot{m}a\, C_{pa}\,(T_2 - T_3)$

$$\frac{W_c}{Q} = \frac{T2 - T1}{T2 - T3} = \frac{T2 - 293}{-333 + T2} = \frac{31.42\times60}{1340} = 1.407$$

$\therefore T_2 = 431.17\ \mathrm{K}$

$\frac{1340}{60} = 1.005\,\dot{m}_a\times98.1 \quad\Rightarrow \dot{m}_a = 0.2265\ \mathrm{kg/s}$

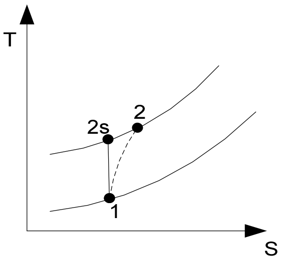
Example 7-4

Six-cylinder, 4.8 lit supercharged engine operating at 3500 rpmhas an overall volumetric efficiency of 158%. The supercharger has an isentropic efficiency of 92% and mechanical efficiency 87%.It is desired that air to be delivered to the cylinder at 65°C and 180 kPa, while ambient conditions are 23°C and 98 kPa.

Calculate: (a) Amount of after cooling needed.

(b) Engine power lost to run the supercharger

Solution:



$$\rho_{air} = \frac{P}{RT} = \frac{98}{0.287 \times 296} = 1.81\,\text{kg/m}^3$$

$$\eta_v = \frac{\dot{V}}{V_s}$$

$$\dot{V} = 1.58 \times \frac{4.8}{10^3} \times \frac{3500}{2 \times 60} = 0.2212 \ \ \text{m}^3/s$$

$$\dot{m}_a = \dot{V}\rho_{air} = 0.2212 \times 1.81 = 0.4 \ \ \text{kg/s}$$

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{2s} = 296 \times \left(\frac{180}{98}\right)^{0.286} = 352.15K$$

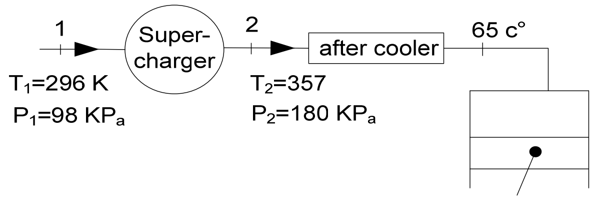
$$\eta_s = \frac{T_{2s} - T_1}{T_2 - T_1} \Rightarrow 0.92 = \frac{352.15 - 296}{T_2 - 296}$$

$$T_2 = 357\ \text{K} \quad \text{or}\ 84^{\text{o}}_{\text{C}}$$

a) The amount of after cooling needed to reduce air temperature from 84 to 65^{o}C is:

$Q = m_a C_{pa} (T_2 - T_m) = 0.4 \times 1.005(84-65)$

$= 7.64$ kW



b) To find the engine power lost to drive the supercharger:

$$P = \frac{m_a C_{pa}(T_{out} - T_{in})}{\eta_m} = \frac{0.4 \times 1.005(357 - 296)}{0.87} = 28.2kW$$

Example 7-5: A diesel engine is fitted with a turbocharger. The engine is tested at constant speed of 500 r.p.m at atmospheric conditions of 1 bar and 27°C, the power output is 5000 kW, bmep is 15 bars and fuel consumption is 1250 kg/h an air intake manifold pressure is 2 bars, exhaust manifold pressure is 1.6 bars, and turbine inlet gas temperature is 650°C and leave the turbine at a pressure of 1 bar. The volumetric eff.of the engine is 100%. The air compressor of the turbocharger has an isentropic eff. of 70%. Calculate the isentropic eff. of the turbine. If the friction mean effective pressure is 1.5 bar, calculate the i.sfc of the engine. Assume Cp = 1.005 kJ/kg.Kγ = 1.4 for air, & Cp$_g$ = 1.05 kJ/ kg.K , $\gamma = 1\frac{1}{3}$ for exhaust gasses.

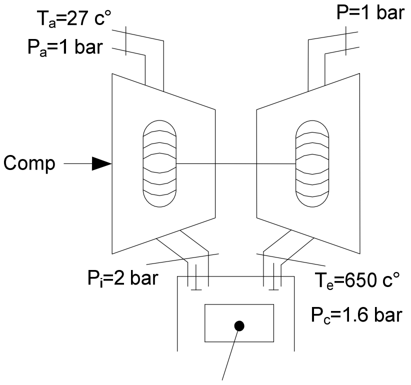
Solution:

$$bp = P_{bmep} \times \dot{V}_s$$

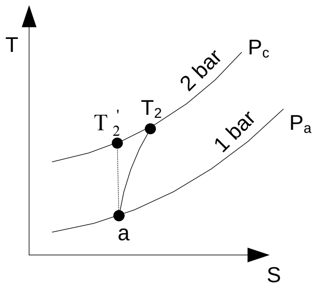
$$5000kW = 15 \times 100 \frac{kN}{m^2} \times \dot{V}_s \frac{m^3}{\sec.}$$

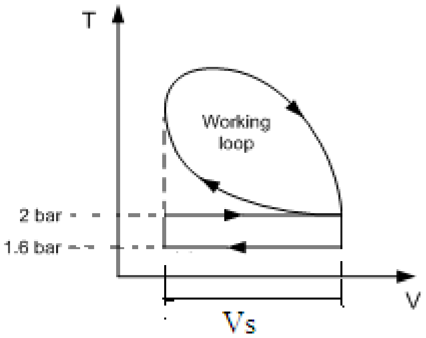
$$\therefore \dot{V}_s = \frac{5000}{1500} = 3.333 \frac{m^3}{\sec.} = \dot{V}_{air}$$

$$\eta_v = \frac{\dot{V}_{act}}{\dot{V}_s} = 100\%$$



At intake condition P = P_1 = 2bar and the temperature is:





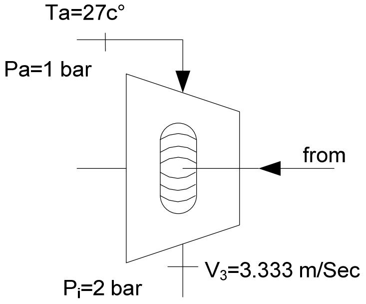
$$\frac{T_2'}{T_1} = \left(\frac{P_c}{P_a}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2' = (273+27)\left(\frac{2}{1}\right)^{\frac{0.4}{0.4}} \Rightarrow \qquad T_2' = 365.7k$$

$$0.7 = \frac{365.7-300}{T_2-300} \qquad \therefore T_2 = 393.86k$$

$$P_{comp} = 2 \times 100 = 200kpa, \quad T_c = 393.86k$$

$$P_sV_s = \dot{m}_sRT_s, \Rightarrow \dot{m}_s = \frac{2 \times 100 \times 3.333}{0.286 \times 393.86} = 5.898\ kg/s$$

Actual compressor (work) power is:



$\text{Work}_{comp.} = m_aC_{pa}(T_2 - T_a)$

$= 5.898 \times 1.005(393.86-300) = 556.35$ kW

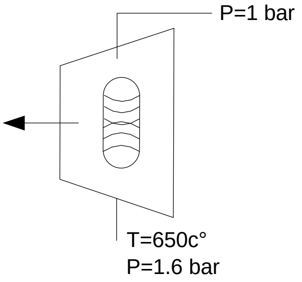
The power consumed by compressor actually developed by the turbocharger:

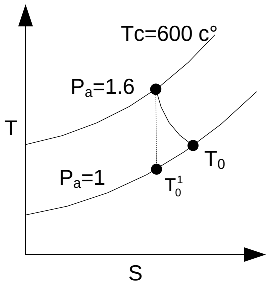
$\text{Work}_{comp.} = \text{Work}_{turbo}$

For turbine:

$$\frac{T_a'}{T_e} = \left(\frac{P_a}{P_e}\right)^{\frac{n-1}{n}} \qquad\qquad T_o^1 = \left(\frac{P_a}{P_e}\right)^{\frac{n-1}{n}}$$

$$T_0^1 = (650+273)\left(\frac{1}{1.6}\right)^{\frac{\frac{1}{3}}{1\frac{1}{3}}} = 820.7K$$





Isentropic turbine power is:

$$(W_t)_{isen} = \dot{m}_{ge} \times cp_{ge}(T_e - T_0')$$

$$= (\mathrm{m_a} + m_f)cp_{ge}(T_e - T_0'),\ ,\ \mathrm{m_a} = 5.898$$

$$\therefore \dot{m}g = \dot{m}_a + \dot{m}_f = 5.898 + \frac{1250}{3600} = 6.245\ \text{kg/sec.}$$

$(W_t)_{isent} = 6.245 \times 1.05(923 - 820.7) = 668.8\,\text{kW}$, isentropic eff. Of turbine is

$$= \frac{\text{actual power}}{\text{Isentropic power}}$$

$$\therefore \eta_{is} = \frac{556.35}{668.8} = 83\%$$

$$b.p = i.p + p.p. - f.p$$

p.p.: is pumping power

p.p. $= \Delta p \times \dot{V}_s = (200-160) \times 3.333 = 133\ \text{kW}$

$$\text{f.p} = \mathrm{P_{fmep}} \times \dot{\mathrm{V}}\mathrm{s} = 1.5 \times 100 \times 3.333 = 500\,\text{kW}$$

i.p = b.p – p.p. + f.p = 5000 – 133 + 500
= 5366.7 kW

i.sfc = $\frac{m_f}{Ip} = \frac{1250}{5366.7} = 0.233\,\text{kg/kW.hr}$

Example7-6:The compression ratio for a six cylinder, 4-stroke, supercharged C.I engine is 15:1. The bore is 130 mm and stroke 150 mm. The indicated thermal efficiency is 0.41 and the A/F is $\frac{18.5}{1}$. The lower heating value of the fuel is 44 MJ/kg. The intake and delivery pressure for the rotary blower are −5 kPa and 42 kPa gauge respectively. The blower mechanical eff. is 0.65. The average exhaust pressure is 14 kPa gauges and the average pressure during intake is 32 kPa gauges. The volumetric eff. for engine is 1.08. The mechanical friction power for engine is equivalent to 12.5% of the indicated power. The engine operates at 900 rpm and the atmospheric pressure and temperature are 100 kPa absolute and 25^{o}C respectively. Determine:

a)Brake power for the engine.

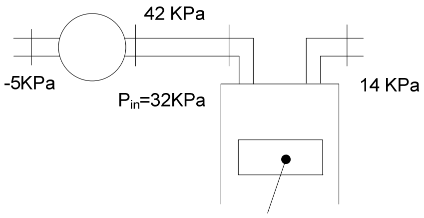
b) Brake specific fuel consumption.

c) Brake thermal efficiency.

Solution:

$$V_s = (V_s \times n) = \frac{\pi}{4} b^2 LNn$$

$$= \frac{\pi}{4}\left(\frac{130}{1000}\right)^2 \times \frac{150}{1000} \times \frac{900}{2 \times 60} \times 6 = 0.0869 \quad m^3/\sec.$$



$$\eta_v = \frac{\dot{V}_{air}}{\dot{V}_s} \Rightarrow \dot{V}_{air} = 1.08 \times 0.0896 = 0.096762$$

$$\rho_{air} = \frac{P}{RT} = \frac{100}{0.287(25+273)} = 1.169\,\text{kg/m}^3$$

$$\therefore \dot{m}_a = \rho_{air} V_{air} = 1.169 \times 0.096762 = 0.11314\,\text{kg/sec.}$$

$$\therefore \dot{m}_f = \frac{\dot{m}_a}{A/F} = \frac{0.11314}{18.5} = 6.116 * 10^{-3}\,\text{kg/sec.}$$

$$\eta_{Ith} = \frac{i.p}{m_f * L.C.V} \qquad \therefore \text{i.p} = 0.41 * 6.116 * 10^{-3} * 44000 = 110.324\,\text{kW}$$

$$f.p_m = \frac{12.5}{100} \times i.p = \frac{12.5}{100} * 110.324 = 13.79\,\text{kW}$$

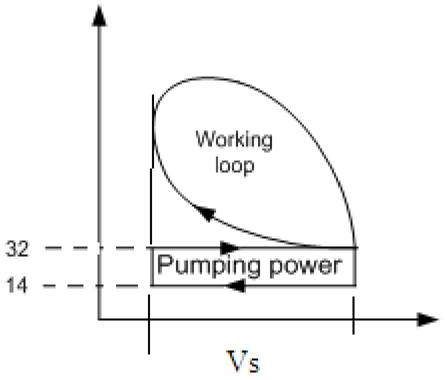
$$P_{blower} = \frac{\Delta P \ \dot{m}_a V_{air}}{\eta_m}$$

$$V_{air} = \frac{RT}{P} = \frac{0.287(25+273)}{100-5} = 0.9\,\text{m}^3/\text{kg}$$

P= {[42-(-5)]×0.11314×0.9}/ 0.65 = 7.365 kW

Pumping power (p.p.) = $\Delta p \times \dot{V}_s$

p.p. = (32-14)×0.0896=1.613 kW



a) b.p = i.p+p.p.−f.p$_m$−P$_{blower}$

b.p =110.324+1.613 −13.79 −7.365

= 90.782 kW

b) bsfc = $\frac{6.11 \times 10^{-3}}{90.7} \times 3600 = 0.24$ kg/kW.hr

c) $\eta_{bth} = \frac{b.p}{\dot{m}_f \times LCV} = \frac{90.782}{6.116 \times 10^{-3} \times 44000} = 0.3373 = 33.73\%$

CHAPTER (8)

TWO – STROKE ENGINE

Both S.I. and C.I. engines can be designed so that one complete cycle of events in the cylinder is completed in two strokes instead of four. The difference between two-strokes and four stroke engines is in the method of filling the cylinder with fresh charge and removing the burned gases from the cylinder. The piston stroke in two strokes engine is longer because part of the compression and expansion strokes are used for the process of exhaust and induction. Fig. 8-1 shows a section of a common type of two stroke engine.

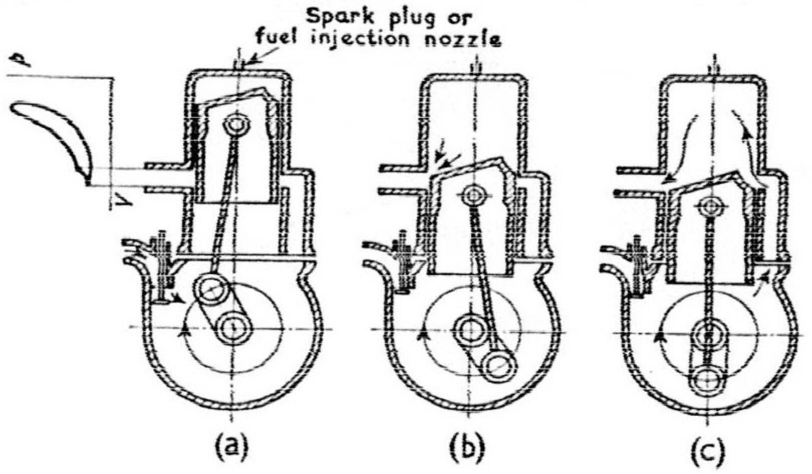


Figure (8 – 1) *The two – stroke engine*

In Figure(8-1a) the piston is shown near the end of compression stroke. The upward motion has decompressed the crank – case and air has been admitted through the self-acting valve to the crank-case. During the expansion stroke the air in the crank-case is compressed, and near the end of this stroke (b) the exhaust port is uncovered to allow the hot gases to blow down the exhaust duct. Further movement of the piston uncovers the inlet port (c), and compressed air from the crank-case flows into the cylinder. The exhaust and inlet ports are open simultaneously for a short period so that the incoming air can assist in cleaning the cylinder from combustion products.

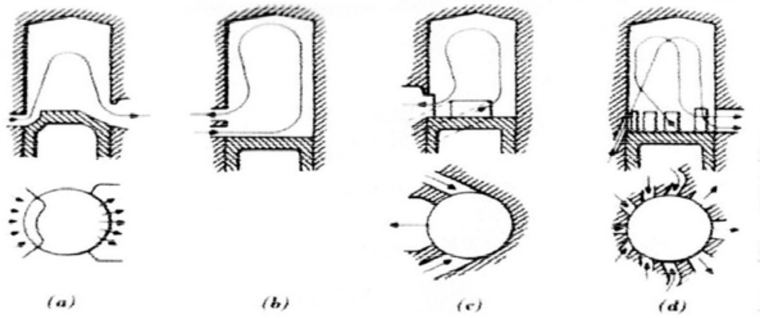


Figure (8 – 2)
(a) Cross scavenging;
(b) MAN loop scavenging;
(c) Schnuerle loop scavenging;
(d) Curtiss loop scavenging.

The piston can be shaped to deflect the fresh gas across the cylinder to assist the "scavenging" of the cylinder; this is called ***cross-flow scavenging*** (Fig 8-2a). The scavenging may be done by using two transfer ports (Fig 8-2c) which direct the incoming air up the cylinder; this is called ***inverted flow scavenging***. In ***loop scavenging*** the inlet and exhaust ports are on the same side (fig 8-2b). Ideal scavenging can be obtained by locating exhaust (or inlet) valve in the head, ***uniflow scavenging***.

For the same power output, more air is required in a two – stroke engine than in a four – stroke engine. This is because some of the air is lost in the overlap period of the scavenging process. Volumetric efficiency of the four- stroke engine is replaced by either delivery ratio or charging efficiency.

Delivery ratio $\lambda_{dr} = \frac{m_{mi}}{V_s \rho_a} \equiv \text{scavenging ratio } \lambda = \frac{\dot{m}_{mi}}{\dot{m}_{cy}}$

Charging efficiency, $\eta_{CH} = \frac{m_{mt}}{V_s \rho_a} \equiv \text{scavenging efficiency } \lambda_{sc} = \frac{m_{mi}}{m_{cy}}$

Trapped efficiency $\eta_{tr} = \frac{\dot{m}_{mt}}{m_{mi}}$

Where: m_{mi} =mass of air-fuel mixture supplied

m_{mt} = mass of air-fuel mixture trapped in cylinder after valves closes

m_{cy} = mass of air-fuel mixture which could be trapped in cylinder volume

V_s= swept volume

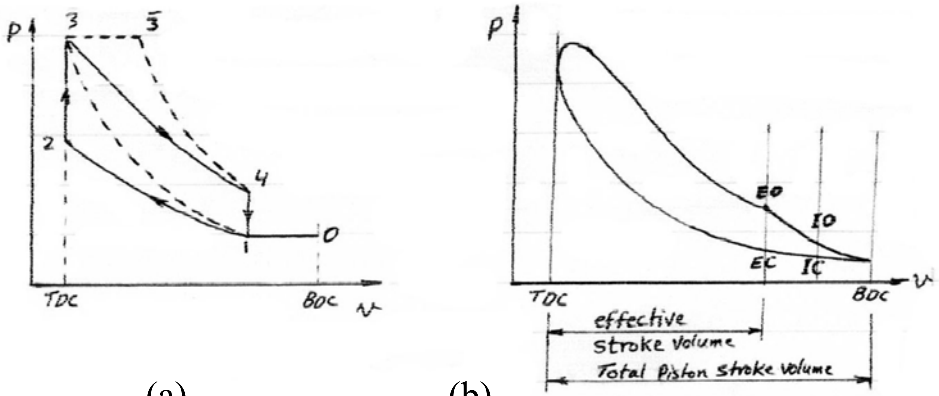
ρ_a=density of air at ambient conditions

Delivery ratio (λ_{dr}) is greater than charging efficiency (η_{ch}), because some of the air-fuel mixture supplied is lost though the exhaust port before it is closed. For engines that inject fuel after the values are closed, the mass of mixture should be replaced with mass of air.

The compression ratio of two- stroke engine is defined as :

$$C\bar{R} = \frac{\text{total volume above exhaust ports}}{\text{clearence volume}}$$

Theoretical and actual cycle for two-stroke engine:



(a) (b)

Figure (8 – 3) *(a) Theoretical p – v diagram; (b) Actual p – v diagram*

The m.e.p of the 2-stroke cycle engine may be measured on the basis of effective stroke or on the basis of total stroke. It is obvious the former is greater than the later.

Valve Timing Diagram for 2-Stroke Engine:

The valve timing diagram for two-stroke engine is shown in fig.8-4; (a) petrol engine,(b) diesel engine,(c) gas engine. These timing diagrams are self-explanatory.

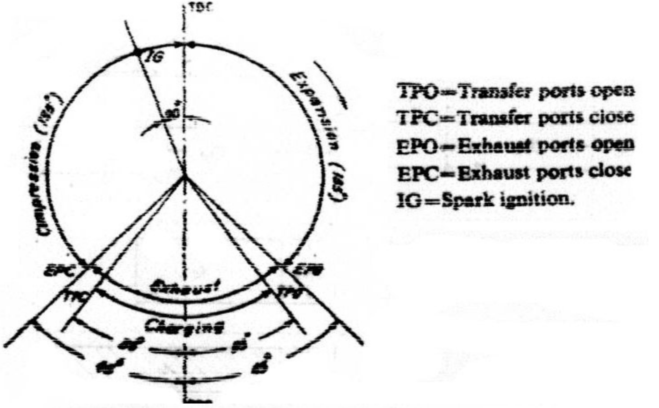


Figure (8 – 4a) *Valve timing Diagram for 2 – stroke petrol engine*

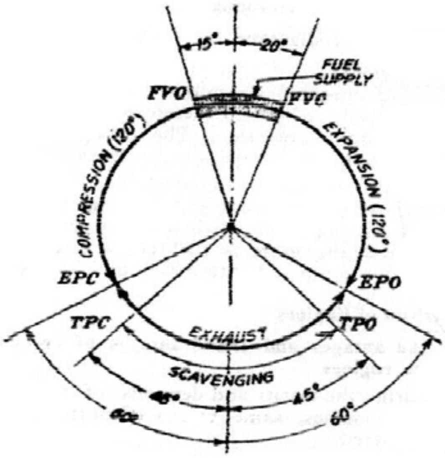


Figure (8 – 4b) *Valve timing for two – stroke diesel engine*

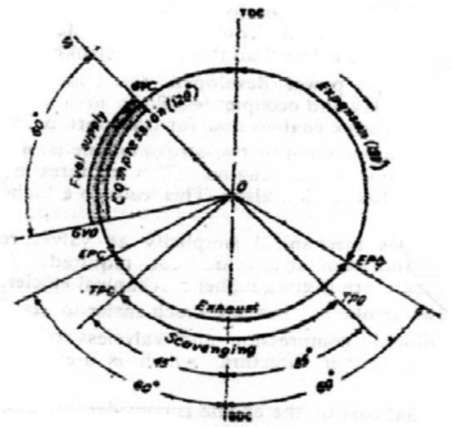


Figure (8 – 4c) *Valve timing diagram for 2 – stroke gas engine*

Comparison of Two-Stroke and Four Stroke Engines:

Advantages of two – stroke engine:

1. 2- stroke engine gives twice as many power strokes as a four stroke cycle engine at the same speed; therefore, the two-stroke engine should develop twice the power of four- stroke engine.
2. For the same power developed, the –stroke engine is much lighter.
3. Turning moment of 2-stroke engine is more uniform.
4. It provides mechanical simplicity.
5. Easier to start.
6. Initial cost is less.

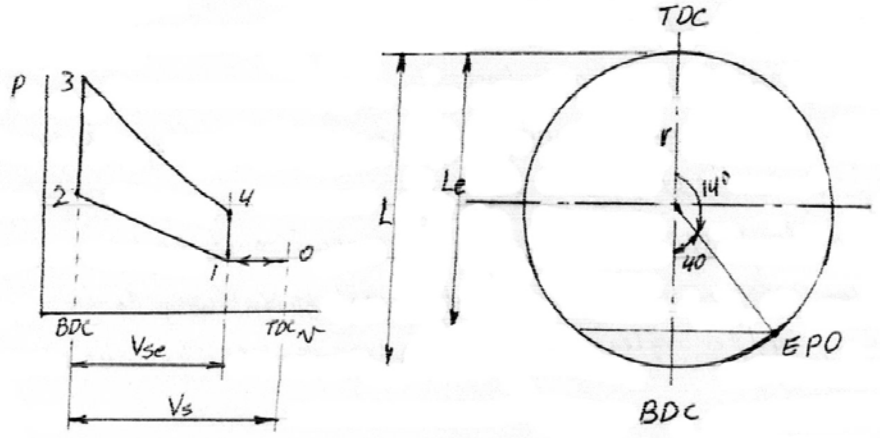
Disadvantages:

1- Lower thermodynamic efficiency, because the effective compression ratio is less.
2- Portion of the fresh charge is escape through the exhaust port,therefore,sfc is higher.
3- The capacity of the cooling system must be higher.
4- Consumption of lubricating oil is sufficiently large.
5- Sudden release of burnt gases makes the exhaust more noisy.
6- The fresh charge is highly polluted by residuals from previous cycle.

Example 8-1:

The stroke and diameter of the two stroke petrol engine are 14 cm and 10 cm respectively. The clearance volume is 157 cm^3/ If the exhaust ports open after 140^o after TDC, find the air standard efficiency of the cycle.

Solution:



Effective stroke = r +r cos θ

Where r is crank radius which is $\frac{1}{2}$ stroke and θ=40^o

Le=7+7 cos 40 = 12.43 cm

Effective stroke volume Ve = $\frac{\pi}{4}d^2 * Le = \frac{\pi}{4}(10)^2 * 12.43 = 975$ cm

Effective compression ratio = $\frac{V_{se}+V_c}{V_c} = \frac{975+157}{157} = 7.23$

$$\eta_a = 1-\frac{1}{r_e^{\gamma-1}} = 1-\frac{1}{7.23^{1.4-1}} = 1-\frac{1}{2.206} = 0.547$$

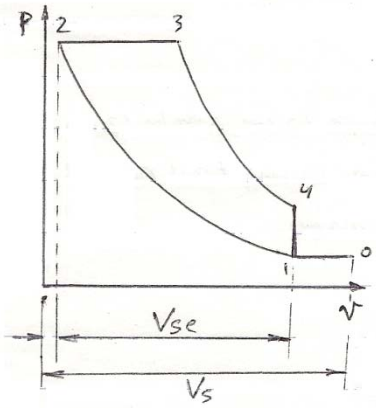
Example 8-2:

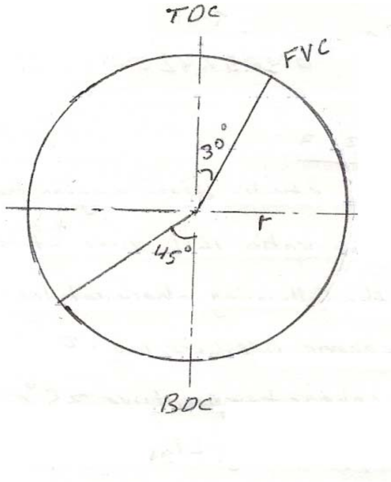
The diameter and stroke of two stroke diesel engine are 8 cm and 12 cm respectively. The clearance volume is 36.2 cm^3.The exhaust ports close after 45^o crank angle from BDC.

The fuel valve opens at TDC and closes 30^o crank angle after TDC. Find the air-standard efficiency of the cycle.

If the relative efficiency is 50% find the specific fuel consumption assuming the C.V. of the fuel used is; 42000 kJ/kg.

Solution:





$L_e = r\,(1+\cos\theta) = 6\,(1+\cos 45) = 10.242$ cm

$V_3 = (r - r\cos 30)\,A + V_c = 6\,(1-\cos 30)\frac{\pi}{4}\times 8^2 + 36.2$

$= 36.2 + 6.734 = 42.934\ \text{cm}^3$

$V_{se} = \frac{\pi}{4}d^2 \,.\, L_e = \frac{\pi}{4}\times 64 \times 10.242 = 515\text{cm}^3$

Effective compression ratio $= \dfrac{V_{se}+V_c}{V_c} = \dfrac{515+36.2}{36.2} = 15$

Cut- off ratio $= \dfrac{V_3}{V_2} = \dfrac{42.934}{36.2} = 1.186$

$$\eta_a = 1 - \frac{1}{r_e^{\gamma-1}}\left[\frac{1}{\gamma}\left(\frac{\left(\frac{V_3}{V_2}\right)^{\gamma}-1}{\left(\frac{V_3}{V_2}\right)^{\gamma}-1}\right)\right] = 1 - \frac{1}{1.4\times 15^{0.4}}\left(\frac{1.186^{1.4}-1}{1.186-1}\right)$$

$$= 1 - \frac{1}{4.136}\left(\frac{1.269-1}{0.186}\right) = 0.65$$

$$\eta_r = \frac{\eta_{IT}}{\eta_a}$$

$$\eta_{IT} = 0.5\times 0.65 = 0.325$$

$$\eta_{IT} = \frac{ip}{m_f \times CV}$$

$$m_f = \frac{ip}{0.325\times 42000}$$

$$sfc = \frac{3600}{0.325\times 42000} = 0.264 \quad \text{kg/kw.h}$$

Example 8-3:

A 2-stroke diesel engine having bore 10 cm, stroke 12 cm, compression ratio 16:1, runs at 1500 rpm. During trail run of the engine, the following observations were made:

Atmospheric pressure 103 kPa
Atmospheric temperature 35^{o}C
Air supplied by blower = 125 kg/hr

If the scavenging efficiency of the engine is 57.4%, calculate the scavenging ratio, trapped efficiency, and charging efficient.

Solution:

Scavenging efficiency = $\frac{m_{mt}}{m_{cy}} = \eta_{sc}$

$m_{cy}=V_{cy}\times P_a$

$$V_{cy} = \frac{r}{r-1}V_s = \frac{16}{16-1}V_s$$

$$\eta_{CH} = \frac{m_{mt}}{V_s P_a}$$

$$\rho_a = \frac{P}{RT} = \frac{103}{0.287\times 308} = 1.16 \quad \text{kg/m}^3$$

$$\dot{V}_S = \frac{\pi}{4}\times\left(\frac{10}{100}\right)^2\times\frac{12}{100}\times\frac{1500}{60\times 1} = 0.02356 \quad \text{m}^3/s \quad \text{or} \quad 1.4136\,\text{m}^3/\text{min}.$$

$$\dot{V}_{cy} = \frac{16}{15}\times 0.02356 = 0.025133 \ \ \text{m}^3/s$$

$$\dot{m}_{mt} = \eta_{sc}\times m_{cy} = 0.574\times 0.025133\times 1.16 = 0.01673 \quad \text{kg/s}$$

Scavenging ratio $\lambda = \frac{m_{mi}}{m_{cy}} = \frac{125}{3600\times 0.025133\times 1.16} = 1.19$

Trapped efficiency $\eta_{trap} = \frac{m_{mt}}{m_{mi}} = \frac{0.01673\times 3600}{125} = 0.4818$

$$\eta_{CH} = \frac{m_{mt}}{V_s P_a} = \frac{0.01673}{0.02356\times 1.16} = 0.612$$

Example 8-4:

A 2-stroke single cylinder diesel engine has a bore 125 mm, stroke 150 mm, compression ratio 15:1, runs on 1800 rpm, the atmospheric conditions are 300 k and 1 atm. The trapping efficiency is 60%, air / fuel ratio= 30:1, LCV = 43000 kJ/kg, imep = 4.36 bar and η_{IT} =35%. Evaluate:

a) Scavenging ratio	b) charging efficiency
c) Delivery ratio	d) scavenging efficiency

Solution:

$$\dot{V}_s = \frac{\pi}{4}\left(\frac{125}{1000}\right)^2 \times \frac{150}{1000} \times \frac{1800}{60\times 1} = 0.055\,\mathrm{m^3/s}$$

$$\dot{V}_{cy} = \frac{r}{r-1}V_s = \frac{15}{14}\times 0.055 = 0.0591\,\mathrm{m^3/s}$$

$$\rho_a = \frac{100}{0.287\times 300} = 1.161$$

$$i.p = p_i \dot{V}_s = 4.36\times 100\times 0.055 = 24\,\mathrm{kW}$$

$$m_f = \frac{ip}{\eta_{IT}\times LCV} = \frac{24}{0.35\times 43000} = 0.0016\,\mathrm{kg/s}$$

$$m_{mt} = m_f \times \frac{A}{F} = 0.0016\times 30 = 0.048\,\mathrm{kg/s}$$

$$\eta_{trap} = \frac{m_{mt}}{m_{mi}} \Rightarrow m_{mi} = \frac{0.048}{0.6} = 0.08\,\mathrm{kg/s}$$

a) $\lambda = \dfrac{m_{mi}}{m_{cy}} = \dfrac{0.08}{0.0591\times 1.16} = \dfrac{0.08}{0.0684} = 1.169$

b) $\eta_{CH} = \dfrac{m_{mt}}{V_s\rho_a} = \dfrac{0.048}{0.055\times 1.16} = 0.7523$ or 75.23%

c) $\lambda_{dr} = \dfrac{m_{mi}}{V_s\rho_a} = \dfrac{0.08}{0.0638} = 1.25$

d) $\eta_{sc} = \dfrac{m_{mt}}{m_{cy}} = \dfrac{0.048}{0.0684} = 0.7$ or 70%

CHAPTER (9)

COMBUSTION IN I.C. ENGINES

A- Combustion in SI Engines

Combustion in spark ignition engines normally begins at the spark pluck where the molecules in and around the spark discharge are activated to a level where reaction is self- sustaining. Combustion is SI engines may be one or more of the following types of combustion:

1- Normal combustion.
2- Abnormal combustion.
3- Uncontrolled combustion.

9-1 Normal combustion:

Once ignition has started the flame front expands across the chamber until it engulfs the whole mixture. Two stages may be distinguished during the normal combustion. The first stage, (AB) correspond to the time for the formation of the self propagation nucleus of the flame. This is mainly a chemical process and depends on the nature of mixture composition, temperature and turbulence.

The second stage (BC) corresponds to the propagation of the flame throughout the combustion chamber. The second stage begin at the point where first measurable rise of pressure can be seen on the indicator diagram. The rate of pressure rise is proportional to the rate of heat release because during this stage the combustion volume is nearly constant.

The spark occurs at the point A, there is a "**delay period**" between the occurrence of the spark and the noticeable pressure rise from that of motoring compression. This is a time delay which is independent of engine speed so that as the engine speed is increased the point A must occur earlier in the cycle to obtain the best position of the peak pressure.

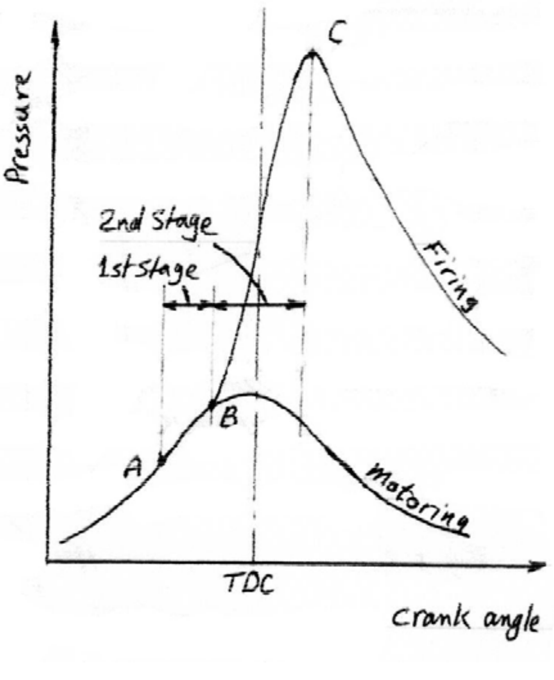


Figure (9 – 1)

Although the point C marks the completion of the flame travel, it does not follow that at this point the whole of the heat of the fuel has been liberated, some further chemical adjustments due to re-association, etc., and what is generally referred to as after – burning, will to a greater or less degree continue throughout the expansion stroke.

Effect of engine variables on first stage:

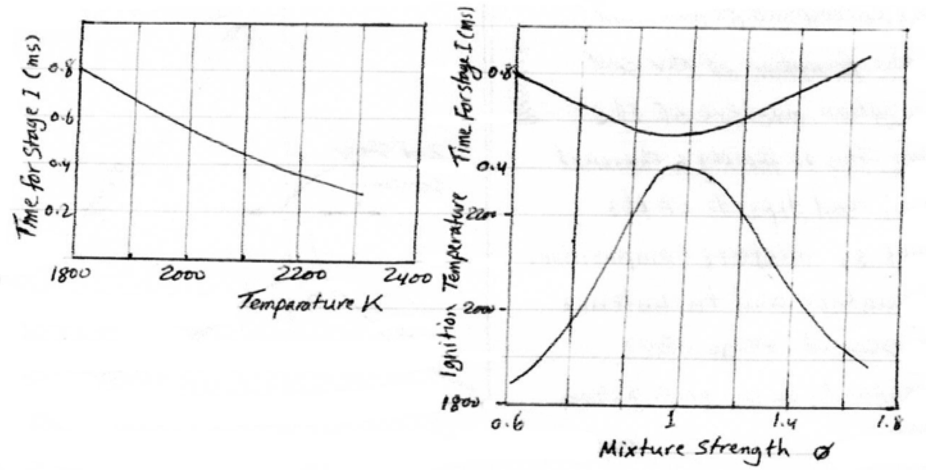
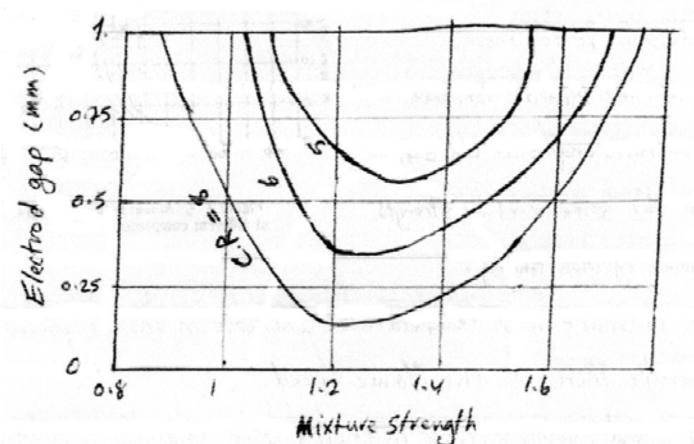


Figure (9 – 2)

Electrode gap: If the gap is too small, quenching of the flame nucleus may occur and the range of fuel / air ratio for the development of a flame nucleus is reduced.

Figure (9 – 3)

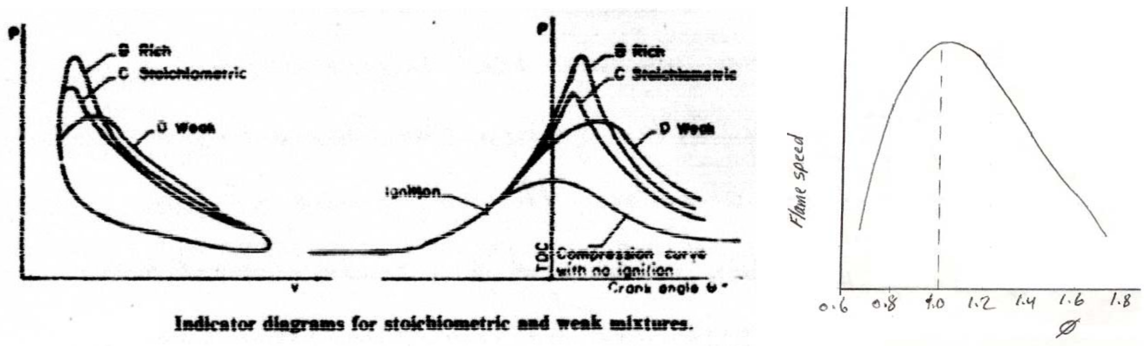


Turbulence: Turbulence is directly proportional to engine speed. Therefore increase in engine speed does not affect much ignition lag measured in millisecond.

Effect of engine variable on second stage:

There are several factors which affect the second stage (flame speed) such as:

1- fuel / air ratio:



Indicator diagrams for stoichiometric and weak mixtures.

Figure (9 – 4)

2- Compression Ratio:

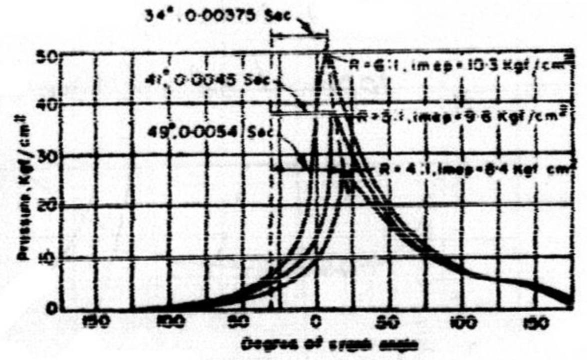


Figure (9 – 5) *Actual indicator diagram at different compression ratios.*

Fig. 9-5 shows the increased speed of combustion with increase of compression ratio. These diagrams are for Ricardo variable compression ratio engine at CR=4, 5, 6with the same mixture strength and the same ignition timing.

3-Intake pressure and temperature: increase in intake pressure and temperature increases the flame speed.

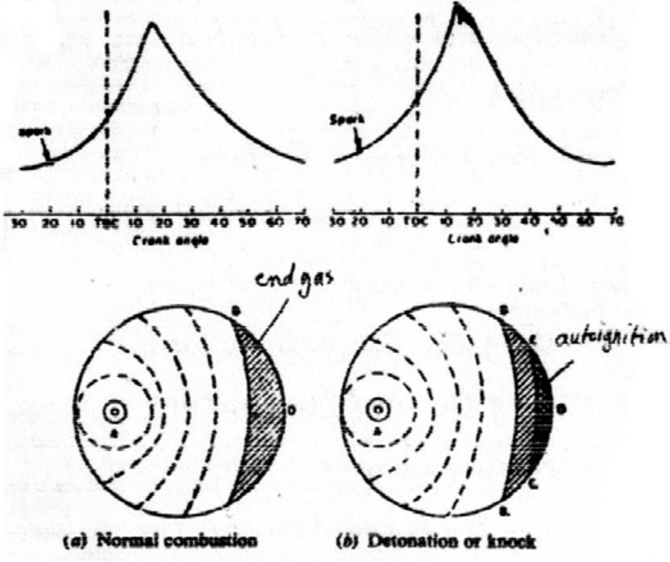
4-Engine load: with increase in engine load the cycle pressure increases hence the flame speed increase.

5-Turbulence: Turbulence plays a very vital role in combustion phenomenon. The flame speed is very low in non-turbulent miniatures. A turbulent motion of the mixture intensifies the process of heat transfer and mixing of the burned and unburned portion in the flame front. These two factors cause the velocity of turbulent flame to increase.

9-2 Abnormal Combustion:

Normal combustion rarely occurs in a real engine without some trace of auto ignition appearing. After ignition, the flame front travels across the combustion chamber. The gas a heat of the flame front called the"**end gas** ". The end gas receives heat due to compression by expanding gases and by radiation from the advancing flame front, therefore, its temperature and density increases. If the temperature exceeds the self – ignition temperature and the un-burnt gas remains at or above this temperature for a period of time equal to/or greater the delay period, spontaneous ignition (or auto ignition) will occurs at various locations. Shortly after words an audible sound called **knock** appears.

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



(a) Normal combustion (b) Detonation or knock

Figure (9 – 6): *Combustion in SI engine.*

Effect of Knock:

Knock has the following effects on engine operation:

1. Noise and Roughness.
2. Mechanical damage: increase in engine wear, cylinder head and valves may be pitted.
3. Carbon deposits.
4. Increase in heat transfer.
5. Decrease in power output and efficiency.
6. 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.

9-3 Knock rating of S.I. Fuels:

The Knock rating of a gasoline is found by comparing the Knock of the fuel under test with that of a blend of primary reference fuels (PRF). These fuels are n-heptane (C_7H_{16}), which have a very low auto ignition reaction time and treated as fuel with octane number (ON) of O and 2,2,4- trimethyl pentane (iso- octane), which has inactive auto ignition reaction and is treated as ON equal to 100. The fuel is rated by the percent of iso - octane in the n- heptane and iso- octane mixture. The scale of octane rating is extended above 100 by adding Tetra- ethyl- lead (TEL) to iso- octane, and:

$$ON\,(above\ 100) = 100 + \frac{6.221\,T}{1 + 0.1619\,T + \sqrt{1 + 0.3238\,T - 0.001704\,T^2}}$$

Where T = ml TEL/ lt

There are several methods of Knock rating to suit the various matching conditions for different engines and operating variables, and in each of these methods, a standard engine built to exacting specifications must be run under prescribed operating conditions.

The standard engine used for either the research or motor method is the CFR (Co - operative Fuel research) engine.

Motor and research methods: the engine must first be calibrated under specified conditions, such as those indicated below:

Test method	**Engine speed (rpm)**	**Coolant temp. (°C)**	**A/f ratio**	**Spark timing (bTDC)**
Research	600	100	Max	13
Motor	900	100	Max	14-26

The sensitivity of a fuel to knock is measured by the difference in the two knock ratings; the greater the difference the greater the fuel sensitivity. The Octane rating of fuel is lower in the motor method than in the research method.

Influence of fuel additive on knock:

It is the possible to raise the Octane number of fuel by refining methods alone, but these methods are in general expensive. These additives are used to raise ON of the fuel, control surface ignition, reduce spark plug fouling, resist gum formation, prevent rust, reduce carburetor icing, remove carburetor or injector deposits, minimize deposits in intake system, and prevent valve sticking. The most effective antiknock agents are lead alkyls. Tetraethyl lead (TEL), $(C_2H_5)_4$ Pb, was first introduced in 1923. Tetramethyl lead (TML), $(CH_3)_4$ Pb, was introduced in 1960. In 1959 manganese antiknock compound (methylcyclopentadienyl manganese tricarbonyl), MMT, was introduced.

The intereste in unleaded gasoline was started in 1970 because of the issue of legislation in many developing countries which control harmful pollutants. Lead has toxicological effect in the urban environment and the use of catalytic devices for emission control was introduced. The expanding use of unleaded fuels has increased interest in other methods of increasing the ON of gasoline, one of these methods is; the oxygenates (alcohols and ethers) as fuel additives to increase Octane rating.

9-4 Uncontrolled Combustion:

Under certain conditions the fuel- air mixture is ignited by hot spot in the cylinder. The hot spot might be the spark plug insulator or electrode, or combustion deposits etc…. When ignition occurs before the spark the phenomenon is called *pre-ignition*. When the phenomenon occurs after ignition is switched off it is called *running- on*. Combustion deposit ignition is called *rumbling*.

B- Combustion in CI engines:

In compression ignition engine usually air is drawn during intake stroke. A compression ratio between 12 and 20 is used, so that temperature of the air near the end of compression stroke is quite high. Just before TDC, fuel is sprayed into the combustion chamber. This spray of fuel with the aid air movement distributes the fuel through the air. Owing to the high temperature of the air, the fuel ignites and burns almost as soon as introduced.

Since the cylinder contents are not homogeneous (heterogeneous). They vary from fuel, F/A combustion products mixture to air alone in different regions. These regions may exist close to each other. Because of the dependence of combustion on the process of the fuel finding oxygen, C.I. engines require excess air to a greater extent than do S.I. engines if high efficiency and low smoke levels are to be achieved.

9-5 stage of combustion in C.I. engine:

From the commencement of injection, the combustion process may be divided into several stages.

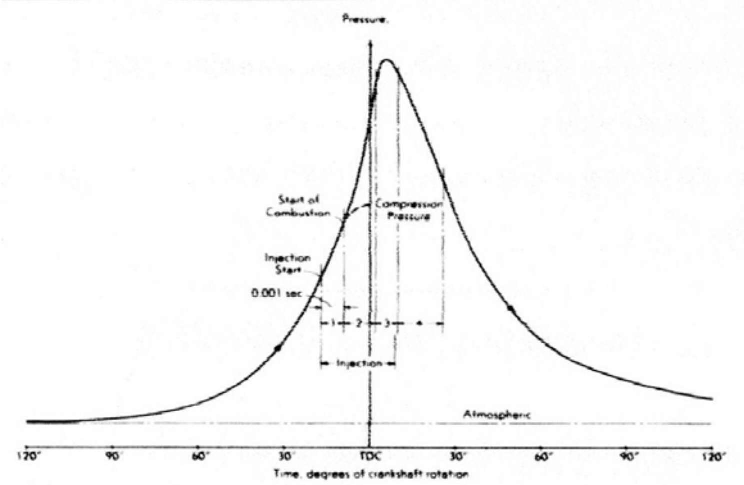


Figure (9 – 7) *p – t diagram for C.I. engine at full load*

1. Delay Period:

During this period there is no visible pressure rise. The fuel is injected through nozzle, but does not ignite. There is a definite period for mixing of liquid fuel with air, vaporizing and mixing of fuel vapour with air, preflame reactions of fuel and then ignition. The delay period can be divided into two overlapping parts:

a- Physical delay: during which the cold fuel droplets are heated, vaporized, mixed with air and raised in temperature.

b- Chemical delay: during which chemical reaction proceeds very slowly and then accelerate until local inflammation or ignition takes place.

2. Rapid (uncontrolled) combustion:

During the delay period an appreciable amount of fuel has been prepared, fuel burns very rapidly, the velocity of burning is controlled by chemical kinetics. There is also partial combustion of fuel which continues to be injected during this period.

3. Controlled combustion (diffusion flame):

The rate of combustion in this stage is determined mainly by how rapidly the fuel vapour are heated and mixed with air. During this stage, the moving piston increase the volume of the cylinder and thus the end of this stage is to the right of TDC, this end is characterized by the point of maximum cycle temperature. This period is affected by: rate of injection of fuel- Turbulence in the cylinder- injection pressure.

4. Tail of combustion (after burning):

This stage commences from the point of maximum cycle temperature, the rate of heat release in this phase of combustion gradually drops to zero.

9-6 knock in the C.I. engine:

In C.I. engine, the fuel is injected into hot air and combustion begins with auto – ignition, if the first peak in the pressure rate diagram is high enough a knocking sound appears. This is due to very long delay period and a large amount of fuel would be prepared during the delay period.

If diesel knock is experienced in an engine a cure may be by means that either reduce the delay period or reduce the rate of injection during the delay period.

Knocking in C.I. engine is an auto ignition phenomenon same as in SI engine, but diesel knock requires cures diametrically opposite to those required for the SI engines.

Knock rating of CI fuels:

The knock rating of a diesel fuel is found by comparing the fuel under test in a special engine with primary reference fuels. These fuels are n- cetane (hexeadecane, $C_{16}H_{34}$) which has a low self ignition temperature and given cetane number of 100 and heptamethylnonane with a cetane number of 15. The cetane number for a blend is calculated by:

$$CN = \%\ \text{cetane} + 0.15 * \%\ \text{heptamethylnonane}.$$

In some text books the reference fuels are: cetane (100 CN) and α- methylnaphthalene (CN = 0).

9-7 Firing order in multi- cylinder engines:

In multi- cylinder engines; the expansion strokes for the different pistons must be arranged to give suitable distribution of force, in this way the engine runs more quietly and smaller the flywheel would be. The crank angle between any two explosions, ensuring the best uniformity if crankshaft rotation should be as follows:

Four- stroke engines: $\phi = 720/ n$

Two- stroke engines: $\phi = 360/ n$

Where n is the number of cylinders.

Figure 9-8 shows diagrams of forces for individual cylinders and the summation force of an 8- cylinder four stroke engine.

The firing order is of more importance in multi- cylinder engines, because the exhaust valves remain open for some interval of crank motion, so two exhaust valves of two adjacent cylinders may open simultaneously. This overlapping will cause the exhaust of one of the two adjacent cylinders to 'below- over' into the other in which the exhaust stroke is nearly completing, thus interfering with the evacuation of the latter.

Blow- over can be minimized by using such a firing order that adjacent cylinders never fire in succession. The following table gives examples of the possible crank arrangements and firing order for four- stroke and two- stroke in line engines.

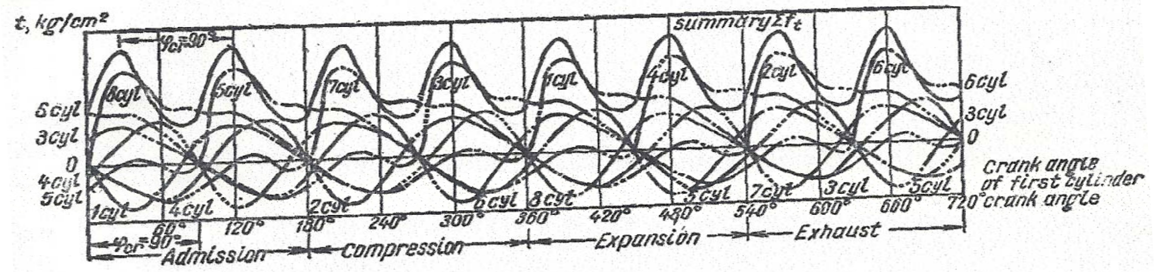
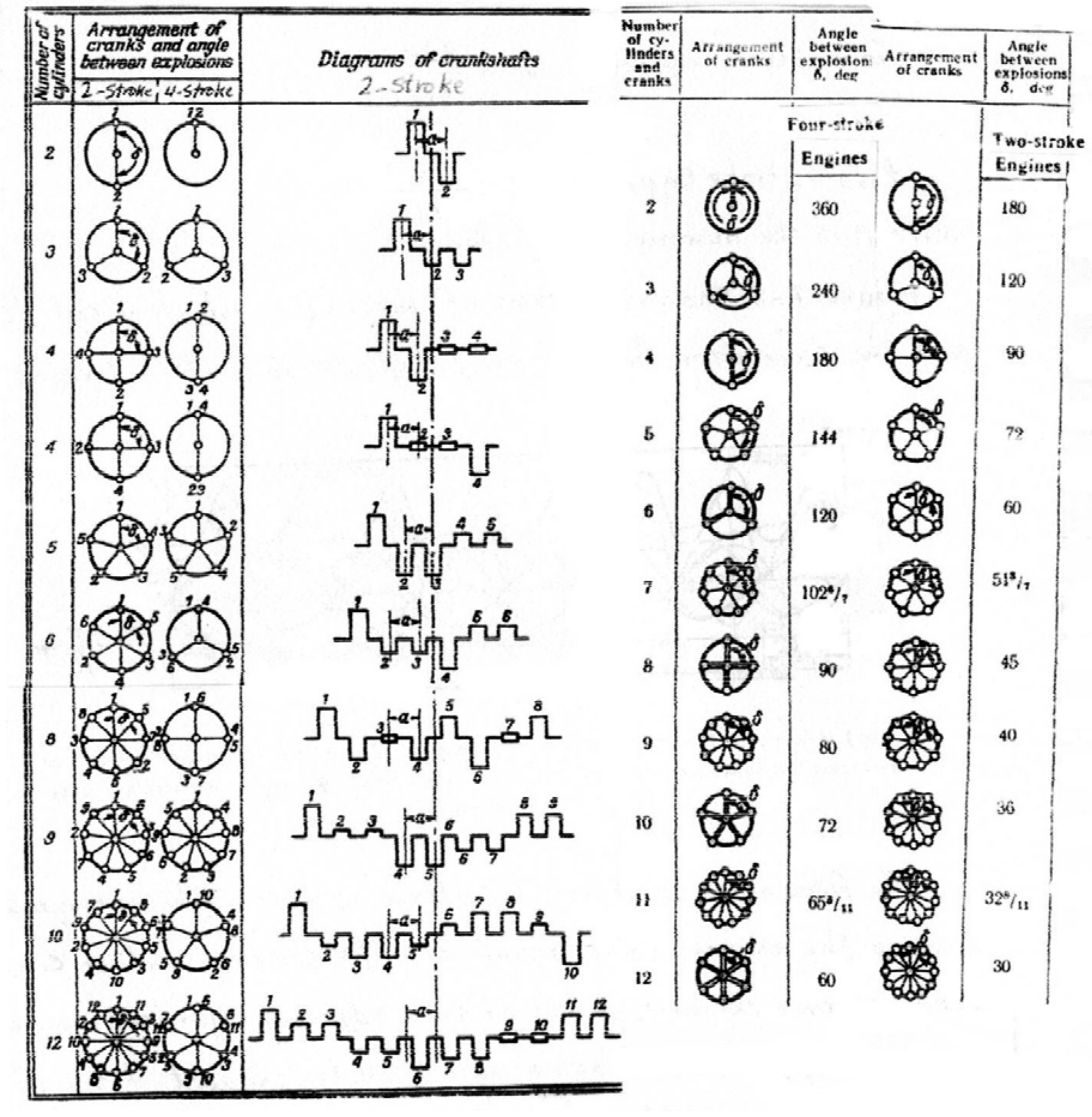


Fig. 8 Diagrams of tangential forces for individual cylinders and the total tangential force of an 8-cylinder four-stroke engine. Engine firing order 1-4-2-6-8-5-7-3; $\phi = \frac{720}{8} = 90°$— angular spacing of cranks



Number of cy-linders and cranks	Arrangement of cranks	Angle between explosion δ, deg	Arrangement of cranks	Angle between explosions δ, deg
		Four-stroke Engines		Two-stroke Engines
2		360		180
3		240		120
4		180		90
5		144		72
6		120		60
7		$102^6/_7$		$51^3/_7$
8		90		45
9		80		40
10		72		36
11		$65^5/_{11}$		$32^8/_{11}$
12		60		30

Figure (9 – 9)