CHAPTER ONE

1- INTRODUCTION

1-1 Basic Engine Types:

Heat engines can be classified as:

I - External Combustion Engine: the products of combustion of air and fuel transfer heat to a second fluid which then becomes the motive or working fluid for producing power. For example, steam engine, steam turbine, heat air engine and closed-cycle gas turbine.

II – Internal Combustion Engine: the products of combustion are directly the motive fluid. For example, gasoline engine, diesel engine, gas engine, jet engine, rocket engine.

1-2 Engines Classifications: Internal Combustion Engines (gasoline and diesel engines) can be classified in a number of different ways:

1-2-1 Cylinder Arrangements:

a) In-Line: cylinders are positioned in a straight line, one behind the other along the length of the crankshaft, as shown in figure (1-1) below. They consist of (2 to 11) cylinders or possibly more.

Figure (1-1): In-Line Engine.
b) **V-Engines**: two banks of cylinders at an angle with each other along a single crankshaft, as shown in figure (1-2). The angle between the banks of cylinders can be anywhere from \(15^\circ - 120^\circ\) with \(60-90^\circ\) being common.

![Figure (1-2): V-Engine.](image1)


c) **W-Engine**: same as a V-Engine with three banks of cylinders on the same crankshaft, as shown in figure (1-3). Not common, but some have been developed for racing automobiles, both modern and historic. Usually 12 cylinders with about 60\(^\circ\) angle between each bank.

![Figure (1-3): W–Engine.](image2)

d) **Radial Engine**: engine with pistons positioned in a circular plane around the central crankshaft, as shown in figure (1-4). The connecting rods of the pistons are connected to a master rod which in turn is connected to the crankshaft. A bank of cylinders on a radial engine always has an odd number of cylinders ranging from (3 to 13) or more. For large aircraft, two or more banks of cylinders are mounted together, one behind the other on a single crankshaft, making one powerful smooth engine. Very large ship engines exist with up to (54) cylinders, 6 banks of 9 cylinders each.
e) **Opposed Cylinder Engine:** two banks of cylinders opposite each other on a single crankshaft (also called V-Engine with a 180° angle), as shown in figure (1-5). These are common on small aircraft and automobiles with an even number of cylinders from (2 to 8) or more. These engines are often called (flat engines).

f) **Opposed Piston Engine:** two pistons in each cylinder with the combustion chamber in the center between the pistons. A single combustion process causes two power strokes at the same time with each piston being pushed away from center and delivering power to a separate crankshaft at each end of the cylinder, as shown in figure (1-6) below. Engine output is either on two rotating crankshafts or on one crankshaft incorporating complex mechanical linkage.
1-2-2 Basic Design (Piston Motion):

a) **Reciprocating Engine**: have one or more cylinders in which pistons reciprocate back and forth the combustion chamber is located in the closed end of each cylinder.

b) **Rotary Engine**: is made of a block (stator) built around a large non-concentric rotor and crankshaft. The combustion chambers are built in the non-rotating block.

1-2-3 Fuel Used:

a) Gasoline.

b) Diesel oil.

c) Gas, Natural Gas, Methane.

d) LPG (Low Pressure Gas).

e) Alcohol-Ethyl, Methyl.

f) Dual Fuel: there are a number of engines that use a combination of two or more fuels. For example, for CI (Diesel Engine) engines use a combination of methane and diesel fuel. Also, combined gasoline-alcohol fuels.

g) Gasohol: common fuel consisting 90% gasoline and 10% alcohol.

1-2-4 Engine Cycle:

a) **Four Stroke Cycle**: a four stroke cycle experiences four piston movements over two engine revolutions for each cycle.

b) **Two Stroke Cycle**: a two stroke cycle has two piston movements over one revolution for each cycle.
1-2-5 Types of Ignition:

a) Spark Ignition (SI): an SI engine starts the combustion process in each cycle by using of a spark plug. The spark plug gives a high-voltage electrical discharge (voltage of 5-15 KV) between two electrodes which ignites the air-fuel mixture in the combustion chamber surrounding the plug.

b) Compression Ignition (CI): the combustion process in CI engine starts when air-fuel mixture self-ignites due to high temperature in combustion chamber caused by high compression.

1-2-6 Valve Location:

a) Head Engine (I): An engine where both intake and exhaust valves are placed directly over the piston.

b) Head Engine (L): valves in block (flat head). Some historic engines with valves in block head, the intake valves on side of the cylinder and the exhaust valve on the other side. These were called T-head engines.
c) **Head Engine (F):** one valve in head (usually intake) and one in block, and this is much less common.

1-2-7 **Types of Cooling:**

a) Air Cooled, b) Liquid Cooled (Water Cooled).

1-2-8 **Applications:**

a) automobile, b) locomotive, c) stationary, d) marine, e) aircraft, f) small portable, chain saw, model airplane.

1-2-9 **Methods of Fuel Input for SI Engines:**

a) Carbureted.

b) Multipoint port fuel injection one or more injectors at each cylinder intake.

c) Throttle Body Fuel Injection. Injectors upstream in intake manifold.
1-2-10 Air Intake Process:

a) Naturally Aspirated: no intake air pressure boots system.

b) Turbocharged: intake air pressure increases with the turbo-compressor driven by the engine exhaust gases.

c) Supercharged: intake air pressure increased with the compressor driven off of the engine crankshaft.

d) Crankcase Compressed: two stroke cycle engine which uses the crankcase as the intake air compressor.

Figure (1-7): Supercharged & Turbocharged Air-Flow system.
1-2-11 Mean Piston Speed:

a) Low piston speed < 6 \text{ [m/s]}

b) Medium piston speed, (6 to 9) \text{ [m/s]}

c) High piston speed > 9 \text{ [m/s]}

1-3 Engine Components:

The following is a list of major components found in most reciprocating intake combustion engines, as shown in figure (1-8).

A- Block body of engine containing the cylinders made of cast-iron or aluminum. The block of water–cooled engine includes a water jacket cast around the cylinders. On air cooler engines, the exterior surface of the block has cooling fins.

B- Camshaft: rotating shaft used to push open valves at the proper time in the engine cycle either directly or through mechanical or hydraulic linkage (push rod, rocker arms). Camshafts are generally made of forged steel or cast-iron and are driven off the crankshaft by means of belts or chain. In four stroke engines, the camshaft rotates at half engine speed.

C- Combustion Chamber: the end of the cylinder between the head and the piston face where combustion occurs. The size of the combustion chamber continuously changes from a minimum volume when the piston is at TDC, a maximum value of volume is when the piston at BDC.

D- Connecting Rod: rod connecting the piston with the rotating crankshaft. Usually made of steel or alloy forging in most engines, but may be aluminum in some small engines.

E- Crankcase: part of the engine block surrounding the rotating crankshaft. In many engines, the oil pan makes up part of the crankcase housing.

F- Crankshaft: rotating shaft through which engine work output is supplied to external system. The crankshaft is connected to the engine block with the main bearings. It is rotated by reciprocating pistons through connecting rods connected to the crankshaft, offset from the axis of rotation. This offset is sometimes called crank throw or crank radius. Most crankshafts are made of forged steel, while some are made of cast-iron.

G- Cylinders: the circular cylinders in the engine block in which the pistons reciprocating back and forth. The walls of the cylinders have highly polished hard surfaces. Cylinders may be machined directly in the engine block, a hard metal (drown steel) may be pressed into the softer metal block.

H- Exhaust Manifold: piping system which carries exhaust gases away from the engine cylinders, usually made of cast-iron.
I- Head: the piece which closes the end of the cylinders usually containing part of the clearance volume of the combustion chamber.

J- Intake Manifold: piping system which delivers incoming air to the cylinders usually made of cast-iron metal, plastic or composite material. The individual pipe to a single cylinder is called runner.

L- Piston: is cylindrical-shaped mass that reciprocates back and forth in the cylinder transmitting the pressure forces in the combustion chamber to the rotating crankshaft. The top of the piston is called the (crown) and the sides are called the (skirt). Pistons are made of cast-iron, steel or aluminum.

M- Piston Rings: metal rings that fit into circumferential grooves around the piston and form a sliding surface against the cylinder walls. Near the top of the piston are usually two or more compression rings made of highly polished hard chrome steel. The purpose of these is to form a seal between the piston and cylinder walls and to restrict the high pressure gases in the combustion chamber from leaking past the piston into the crankcase. Below the compression rings on the piston is at least one oil ring which assists in lubricating the cylinder walls and scrapes away excess oil to reduce oil consumption.

N- Push Rods: mechanical linkages between the camshaft and valve.

O- Spark Plug: electrical device used to initiate combustion in a spark ignition engine (SIE) by creating a high-voltage discharge across an electrode gap. Spark plugs are usually made of metal surrounded with ceramic insulation.

P- Valves: used to allow flow into and out of the cylinder at the proper time in the cycle. Most engines use valves, which are spring loaded closed and pushed open by camshaft action. Valves are usually made of forged steel. Surfaces against which valves close are called valve seats and are made of hardened steel or ceramic. Rotary valves sometimes used, but are much less common.

Q- Water Jacket: system of liquid flow passages surrounding the cylinder, usually constructed as part of engine block and head. Engine coolant flows through the water jacket and keeps the cylinder walls from overheating. The coolant is usually a water-ethylene glycol mixture.
Figure (1-8): Engine Components.
1-4 Operating Characteristics - Engine Parameters:

Figure (1-9): Engine Parameters.

For an engine with bore (B), crank offset-radius (a), stroke length (S) and turning at engine speed of (N):

\[ S = 2a \]  
\[ \bar{U}_p = 2SN \ (m/\text{min}) \]  
\[ \text{or:} \]
\[ \bar{U}_p = \frac{S \cdot N}{30} \ (m/s) \]

Where:
\[ \bar{U}_p = \text{Average piston speed} \]
\[ S = \text{Stroke, displacement, length (m)} \]
\[ a = \text{Crank offset or crank radius (m)} \]
\[ N = \text{Engine speed (rpm)} \]
Average piston speed for all engines will normally be in the range of \([5 \text{ to } 15](m/s)\). There are two reasons why engines operate in this range:

a) For safe limit which can be tolerated by material strength of the engine components.

b) The gas flow into and out-of the cylinders, piston speed determines the instantaneous flow rate of air-fuel into the cylinder during intake and exhaust flow out-of the cylinder during the exhaust stroke. Higher piston speeds would require larger valves to allow for higher flow rates. In most engines, valves are at a maximum size with no room for enlargement.

The distance \((\ell)\) between crank axis and wrist pin axis is given by:

\[
\ell = a \cos \theta + \sqrt{r^2 - a^2 \sin^2 \theta} \quad \ldots \ (1-4)
\]

Where: \(\theta\) = Crank angle, which is measured from cylinder center line and is zero when the piston at TDC.

\[
V_d = V_T - V_C \quad \ldots \ (1-5)
\]

\(V_d\) = Displacement volume, swept volume \((m^3)\)

\(V_T\) = Total volume \((V_T = V_i)\) \((m^3)\)

\(V_C\) = Clearance volume \((V_C = V_2)\) \((m^3)\)

Displacement volume per one cylinder can be expressed as:

\[
V_d = \frac{\pi \cdot B^2}{4} \cdot S \quad (m^3) \quad \ldots \ (1-6)
\]

Also, displacement volume for the engine can be determined:

\[
V_D = \frac{\pi \cdot B^2}{4} \cdot S \cdot i \quad (m^3) \quad \ldots \ (1-7)
\]

Where \(i\) is No. of cylinders.
The compression ratio:

\[ r_c = \frac{V_d + V_c}{V_c} = \frac{V_r}{V_c} = \frac{V_1}{V_2} \quad \text{... (1-8)} \]

The cylinder volume \((V)\) at any crank angle is:

\[ V = V_c + \left(\frac{\pi B^2}{4}\right)(r + a - \ell) \quad \text{(m}^3\text{)} \quad \text{... (1-9)} \]

Piston area:

\[ A_p = \frac{\pi B^2}{4} \quad \text{(m}^2\text{)} \quad \text{... (1-10)} \]

Combustion chamber surface area:

\[ A = A_{ch} + A_p + \pi B(r + a - \ell) \quad \text{(m}^2\text{)} \quad \text{... (1-11)} \]

Where: \((A_{ch})\)-cylinder head surface area.

**1-4-1 Work**: the work is the result of force acting through a distance. Force that acts due to gas pressure on the moving piston generate the work in internal combustion engine cycle.

\[ W = \int F.dx = \int P.A_p.dx \quad \text{... (1-12)} \]

Where:

\[ P = \text{Pressure in combustion chamber} \quad \left[ \frac{N}{m^2} \right] \]

\[ x = \text{Distance that piston moves} \quad (m) \]
But:

\[ A_p . dx = dV \]  
\[ \therefore W = \int P . dV \]  

.. (1-13)  

\[ W_b = W_i - W_f \] (J)  

or: \[ W_b = W_i - \frac{W_f}{W_i} \] (J/kg)  

.. (1-15)

Where:

\( w_b \) = Brake work (actual work available at the crankshaft).

\( w_i \) = Indicated work generates inside combustion chamber.

\( w_f \) = Work lost due to friction and parasitic loads (parasitic load include the oil pump, supercharger, air-conditions, alternator, etc).

Mechanical Efficiency (\( \eta_m \)):

\[ \eta_m = \frac{w_b}{w_i} \]  

.. (1-16)

Mechanical efficiency will be on the order of (75% to 95%).
1-4-2 Mean Effective Pressure: From the figure (1-10) below, it can be seen that pressure in the cylinder of an engine is continuously changing during the cycle. An average or mean effective pressure \((mep)\) is defined by:

\[
\Delta v = v_{BDC} - v_{TDC} = V_T - V_C
\]  
\[\text{\ldots (1-19)}\]

Where:

\(w = \text{Work of one cycle.}\)

\(w = \text{Specific work of one cycle.}\)

\(V_d = \text{Displacement volume.}\)

Mean effective pressure is good parameter to compare of design or output because it is independent of engine size and/or speed.

\[
bmep(P_c) = \frac{W_b}{\Delta v}
\]  
\[\text{\ldots (1-20)}\]

\[
imep(P_i) = \frac{W_i}{\Delta v}
\]  
\[\text{\ldots (1-21)}\]

Where: \(bmep = \text{Brake Mean Effective Pressure}\)

\(imep = \text{Indicated Mean Effective Pressure}\)
Pump mean effective pressure (which can have negative value):

\[ p_{\text{mep}} = \frac{w_{\text{pump}}}{\Delta v} \quad \ldots (1-22) \]

Friction mean effective pressure:

\[ f_{\text{mep}} = \frac{w_f}{\Delta v} \quad \ldots (1-23) \]

\[ b_{\text{mep}} = i_{\text{mep}} - f_{\text{mep}} \quad \ldots (1-24) \]

\[ \eta_m = \frac{b_{\text{mep}}(P_e)}{i_{\text{mep}}(P_i)} \quad \ldots (1-25) \]

**1-4-3 Torque (T):** it is defined as a force acting at a moment distance.

\[ w_b = 2.\pi.T = \frac{(b_{\text{mep}}*V_d)}{n} \]

or:

\[ T = \frac{b_{\text{mep}}*V_d}{2.\pi.n} \quad (N.m) \quad \ldots (1-26) \]

Where:

- \( n \) = Number of revolutions per cycle, [\( n=1 \) for two stroke cycle & \( n=2 \) for four stroke cycle].
- \( w_b \) = Brake work of one revolution.
1-4-4 Power \((P)_{or}(\dot{W})\): it can be defined as the rate of work of the engine.

\[
\dot{W} = \frac{w_{cycle}}{\tau_{cycle}} 
\]  

... (1-27)

\[
w_{cycle} = PV_d 
\]  

... (1-28)

\[
\tau_{cycle} = \frac{Z}{2NS} 
\]  

... (1-29)

Where: \(Z\) = No of strokes.

From (1-27), (1-28) and (1-29) becomes:

\[
\therefore \dot{W} = \frac{2PV_dNS}{Z} 
\]  

... (1-30)

Where: \(\dot{W}\) (watt), \(N_s\) (rps), \(P\left(\frac{N}{m^2}\right)\) and \(V_d\) (m³).

\[
\dot{W} = \frac{PV_dNS}{Z/2} = \frac{wNS}{n} 
\]  

Also, \(w = \frac{Z}{2}\) 

... (1-31)

Also, \(\dot{W} = 2\pi NS.T = \frac{2\pi NT}{60}\) 

... (1-32)

\[
\dot{W} = \frac{mepA_pP_S}{2n} \quad (or: \overline{UP}) 
\]  

... (1-33)

Specific Power: \(SP = \frac{\dot{W}_b}{A_p}\) 

... (1-34)

Output per displacement: \(OPD = \frac{\dot{W}_b}{V_d}\) 

... (1-35)

Specific volume: \(SV = \frac{V_d}{\dot{W}_b}\) 

... (1-36)

Specific weight: \(SW = \frac{\text{engine.weight}}{\dot{W}_b}\) 

... (1-37)

Where: \(\dot{W}_b\) = Brake power.

\(A_p\) = Pistons face area of all pistons.
\[ V_d = \text{Displacement volume.} \]

**Note:** in case multi-cylinders, \((V_d)\) must be used, where: \((V_D = V_d \times i)\) and \((i)\) is the number of cylinders.

These parameters are important for engines used in transportation vehicles such as boats, automobiles and especially airplanes, where keeping weight to a minimum is necessary. For large stationary engines, weight is not important.

From equation (1-30):

\[
W = \frac{P \times i \times V_d \times N}{30Z} \quad (kW)
\]

Where:

\(\dot{W} = \text{Engine power} \quad (kW)\)

\(P = \text{Pressure} \quad (MPa)\)

\(V_d = \text{Cylinder displacement} \quad (\text{liter})\)

\(i = \text{Number of cylinders}\)

\(Z = \text{Number of strokes}\)

Power may be effective (brake) or may be indicated.

\[
(\dot{W}_b)_{or}(\dot{W}_e) = \frac{bmep \times V_D \times N}{30Z} \quad (kW)
\]

\[
\dot{W}_i = \frac{imep \times V_D \times N}{30Z} \quad (kW)
\]
Mean indicated power is higher than the mean effective power due to many losses such as:

1- Mechanical friction (piston, bearings)
2- Aerodynamic losses (flow losses)
3- Input of auxiliary devices (parasitic loads) such as; water pump, oil pump, fan generator, valve timing mechanism …etc.

\[ \dot{W}_e = \dot{W}_i - \dot{W}_{lost} \] \hspace{1cm} \text{... (1-41)}

Where: \( \dot{W}_{lost} \) = Lost power \((kW)\)

\[ \therefore \eta_m = \frac{\dot{W}_e}{\dot{W}_i} \] \hspace{1cm} \text{... (1-42)}

**1-4-5 Air-Fuel & Fuel-Air Ratio:**

\[ AF = \frac{m_a}{m_f} = \frac{\dot{m}_a}{\dot{m}_f} \] \hspace{1cm} \text{... (1-43)}

Where: \( AF \)= Air-fuel ratio.

\[ \dot{m}_a = \text{Air mass flow rate } \left( \frac{kg}{s} \right). \]

\[ \dot{m}_f = \text{Fuel mass flow rate } \left( \frac{kg}{s} \right). \]

\[ FA = \frac{m_f}{m_a} = \frac{\dot{m}_f}{\dot{m}_a} \] \hspace{1cm} \text{... (1-44)}

Where: \( FA \)= Fuel-air ratio.

\[ AF = \frac{1}{FA} \] \hspace{1cm} \text{... (1-45)}
Equivalence ratio ($\phi$) is defined as the actual ratio of fuel-air ratio to ideal (stoichiometric) fuel-air ratio.

$$\phi = \frac{(FA)_{\text{actual}}}{(FA)_{\text{stoich}}} = \frac{(AF)_{\text{stoich}}}{(AF)_{\text{actual}}} \quad \ldots \ (1-46)$$

1-4-6 Specific Fuel Consumption:

$$sfc = \frac{\dot{m}_f}{W} \quad \ldots \ (1-47)$$

Brake specific fuel consumption:

$$bsfc = \frac{\dot{m}_f}{\dot{W}_b} \quad \ldots \ (1-48)$$

Where: $m_{fe} = \text{Effective specific fuel consumption (} = bsfc\)$

Indicated specific fuel consumption:

$$isfc = \frac{\dot{m}_f}{\dot{W}_i} \quad \ldots \ (1-49)$$

Where: $m_{fi} = \text{Indicated specific fuel consumption (} = isfc\)$
1-4-7 Engine Efficiencies:

\[ Q_A = \dot{m}_f \cdot H_L \cdot \eta_c \]  \( \ldots \) (1-50)

\[ \eta_{th} = \frac{\dot{W}}{Q_A} = \frac{\dot{W}}{\dot{m}_f \cdot H_L \cdot \eta_c} \]  \( \ldots \) (1-51)

Where:

\((H_L)_{or}(Q_{HV})\) = Heating value of fuel

\(\eta_c\) = Combustion efficiency

\(\eta_{th}\) = Thermal efficiency

\[ \eta_f = \frac{\dot{W}}{\dot{m}_f \cdot H_L} \]  \( \ldots \) (1-52)

Where: \(\eta_f\) = Fuel conversion efficiency

\[ \eta_{th} = \frac{\dot{W}}{Q_A} = \frac{\dot{m}_f \cdot H_L \cdot \eta_f}{\dot{m}_f \cdot H_L \cdot \eta_c} = \frac{\eta_f}{\eta_c} \]

\[ \eta_{th} = \frac{\eta_f}{\eta_c} \]  \( \ldots \) (1-53)

\((\eta_{th})\) Could be indicated or effective as follows:

\[ \eta_{th}\_i = \eta_i = \frac{\dot{W}_i}{\dot{m}_f \cdot H_L \cdot \eta_c} \]  \( \ldots \) (1-54)

\[ \eta_{th}\_b = \eta_b = \eta_e = \frac{\dot{W}_b}{\dot{m}_f \cdot H_L \cdot \eta_c} \]  \( \ldots \) (1-55)

\[ \eta_i = \frac{1}{(isfc \cdot H_L \cdot \eta_c)} \]  \( \ldots \) (1-56)

\[ \eta_b = \frac{1}{(bsfc \cdot H_L \cdot \eta_c)} \]  \( \ldots \) (1-57)
Where:

\( \eta_i \) = Indicated thermal efficiency

\( \eta_b \) = Brake thermal efficiency = Total thermal efficiency = Effective thermal efficiency

\[
\eta_m = \frac{\eta_{th}}{\eta_{th}}_i \quad \ldots (1-58)
\]

\[
\therefore \eta_m = \frac{\dot{W}_b}{\dot{W}_i} = \frac{isfc}{bsfc} = \eta_b \quad \ldots (1-59)
\]

1-4-8 Volumetric Efficiency:

\[
\eta_v = \frac{m_a}{\rho_a * V_d} \quad \ldots (1-60)
\]

\[
\eta_v = \frac{n * \dot{m}_a}{\rho_a * V_d * N} \quad \ldots (1-61)
\]

Where: \( m_a \) = Mass of air into the engine or cylinder for one cycle

\( \rho_a \) = Air density evaluated at atmospheric conditions outside the engine

\[
\rho_a = \frac{P_o}{R * T_o}
\]

\( P_o = 101(KPa), T_o = 298(K) \) & \( R = 0.287(kJ/kg.K) \)
1-4-9 Emissions:

The four main engine exhaust emissions which must be controlled are oxides of nitrogen \((NO_x)\), carbon monoxide \((CO)\), hydrocarbons \((HC)\) and solid particulates parts. Two common methods of measuring the amounts of these pollutants are specific emission \((SE)\) and the emission index \((EI)\).

\[
(SE)_{NO_i} = \frac{\dot{m}_{NO}}{W_i} \quad \ldots \text{(1-62)}
\]

\[
(SE)_{CO} = \frac{\dot{m}_{CO}}{W_i} \quad \ldots \text{(1-63)}
\]

\[
(SE)_{HC} = \frac{\dot{m}_{HC}}{W_i} \quad \ldots \text{(1-64)}
\]

\[
(SE)_{part} = \frac{\dot{m}_{part}}{W_i} \quad \ldots \text{(1-65)}
\]

\[
(EI)_{NO_i} = \frac{\dot{m}_{NO}}{\dot{m}_f} \left[ \frac{gm}{sec} \right] \quad \ldots \text{(1-66)}
\]

\[
(EI)_{CO} = \frac{\dot{m}_{CO}}{\dot{m}_f} \left[ \frac{gm}{sec} \right] \quad \ldots \text{(1-67)}
\]

\[
(EI)_{HC} = \frac{\dot{m}_{HC}}{\dot{m}_f} \left[ \frac{gm}{sec} \right] \quad \ldots \text{(1-68)}
\]

\[
(EI)_{part} = \frac{\dot{m}_{part}}{\dot{m}_f} \left[ \frac{gm}{sec} \right] \quad \ldots \text{(1-69)}
\]
Example (1): John's automobile has a three liter SI V6 engine that operates on a four-stroke cycle at 3600 rpm. The compression ratio is 9.5, the length of connecting rods is 16.6 mm, and the engine is square (B=S). At this speed, combustion end at 20° a TDC. Calculate:

1- Cylinder bore and stroke length.
2- Average piston speed.
3- Clearance volume of one cylinder.
4- Distance the piston has traveled from TDC at the end of combustion.

Solution:

1- For one cylinder, using equation (1-6) with (S=B).

\[ V_d = \frac{\pi B^2}{4} \cdot S \]

\[ V_d = \frac{V_{total}}{6} = \frac{3L}{6} = 0.5L = 0.0005\,m^3 \]

\[ \Rightarrow B = 0.086\,(m) = 8.6\,(cm) = S \]

2- Using equation (1-2) to find average piston speed:

\[ \bar{U}_p = 2\cdot S\cdot N = 2\left( \frac{\text{strokes}}{\text{rev}} \right) \cdot 0.086 \left( \frac{\text{m}}{\text{stroke}} \right) \cdot \frac{3600\,(\text{rev})}{60\,(\text{ses})} = 10.32\,(m/\text{sec}) \]

3- Using equation (1-8) to find the clearance volume of one cylinder:

\[ r_c = 9.5 = \frac{V_d + V_c}{V_c} = \frac{0.0005 + V_c}{V_c} \]

\[ \Rightarrow V_c = 0.000059\,(m^3) = 59\,(cm^3) \]

4- Using equation (1-4) to find piston position:

\[ \ell = \sqrt{r^2 - a^2 \cdot \sin^2 \theta} \]

\[ \ell = 0.043 \cdot \cos 20° + \sqrt{(0.166)^2 - (0.043)^2 \cdot \sin 20°} \]

\[ \ell = 0.206\,(m) \]
Example (2): The engine of example (1) is connected to a dynamometer which gives a brake output torque reading of 205 (N.m) at 3600 rpm. At this speed, air enters the cylinders at 85 (kPa) and 60 (°C), and a mechanical efficiency of the engine is 85%. Calculate:

1. Brake power.
2. Indicated power.
3. Brake mean effective pressure.
4. Indicated mean effective pressure.
5. Friction mean effective pressure.
6. Power lost to friction.
7. Brake work per unit mass of gas in the cylinder.
8. Brake specific power.
9. Brake output per displacement.
10. Engine specific volume.

Solution:

1. Using equation (1-32) to find brake power:

\[ W = \frac{2\pi NT}{60} = 2\pi \left( \frac{\text{radians}}{\text{rev}} \right) \frac{3600}{60} \left( \frac{\text{rev}}{\text{sec}} \right) * 205(\text{N.m}) = 77300(\text{W}) = 77.3(\text{kW}) \]

2. Using equation (1-42) to find indicated power:

\[ \eta_m = \frac{W_i}{W_e} \Rightarrow W_i = \frac{W_e}{\eta_m} = \frac{77.3}{0.85} = 90.9(\text{kW}) \]

3. Using equation (1-26) to find the brake mean effective pressure:

\[ w_b = 2\pi T = \frac{\text{bme}(V_d)}{n} \]

or:

\[ T = \frac{\text{bme} N m}{2\pi n} \]

Where \( n = 2 \), because it is a four stroke cycle engine.

\[ T = \frac{\text{bme} N m}{2\pi n} \]

\[ \Rightarrow \text{bme} = 4\pi \left( \frac{\text{radians}}{\text{cycle}} \right) T(\text{N.m}) V_d \left( \frac{m^3}{\text{cycle}} \right) = 4\pi * 205 * 0.003 = 859000 \left( \frac{N}{m^2} \right) \]

\[ \therefore \text{bme} = 859(\text{KPa}) \]
4. Equation (1-25) gives indicated mean effective pressure:

\[ \eta_m = \frac{b_{mep}}{i_{mep}} \Rightarrow i_{mep} = b_{mep} \frac{\eta_m}{0.85} = 1010 (KPa) \]

5. Using equation (1-24) to calculate friction mean effective pressure:

\[ b_{mep} = i_{mep} - f_{mep} \Rightarrow f_{mep} = i_{mep} - b_{mep} = 1010 - 859 = 151 (KPa) \]

6. Using equation (1-33) to find friction power lost:

\[
\dot{W}_f = \frac{f_{mep} \cdot A_p \cdot \overline{P}_s (or: \overline{U}_p)}{2n}
\]

where: \( A_p = \frac{\pi}{4} B^2 \)

\[ \dot{W}_f = \frac{\pi}{8n} B^2 \cdot f_{mep} \cdot \overline{U}_p = \frac{\pi}{16} (0.086)^2 \cdot 151 \cdot 10.32 = 13.6 (kW) \]

Or it can be found from equation (1-41):

\[ \dot{W}_e = \dot{W}_i - \dot{W}_{lost} \Rightarrow \dot{W}_{lost} = \dot{W}_i - \dot{W}_e = 90.9 - 77.3 = 13.6 (kW) \]

7. Equation (1-18):

\[ W_b = m_{ep} \cdot V_d = 859 \cdot 0.0005 = 0.43 (kJ) \]

It can be assumed the gas entering the cylinder at BDC is air:

\[
\begin{align*}
m_u &= \frac{P \cdot V_{\text{BDC}}}{(R \cdot T)} = \frac{[P \cdot (V_d + V_e)]}{(R \cdot T)} \\
m_u &= \frac{85 \cdot (0.0005 + 0.000059)}{0.287 \cdot (60 + 273)} = 0.0005 (kg)
\end{align*}
\]

Brake specific work per unit mass:

\[ w_b = \frac{W_b}{m_u} = \frac{0.43}{0.0005} = 860 (kJ/kg) \]

8. Equation (1-34) gives brake specific power:

\[ BSP = \frac{W_b}{A_p} = \frac{(77.3)}{\left[\left(\frac{\pi}{4}\right)(0.086)^2 \cdot 6 \cdot \text{cylinders}\right]} = 2220 (kW/m^2) \]

9. Equation (1-35) gives brake output per displacement:
\[ BOPD = \frac{\dot{W}_b}{V_d} = \frac{(77.3\, kW)}{(3\, L)} = 25.8 \left( \frac{kW}{L} \right) \]

10. Equation (1-36) gives engine specific volume:

\[ BSV = \frac{V_d}{\dot{W}_b} = \frac{1}{BOPD} = \frac{1}{25.8} = 0.0388 \left( \frac{L}{kW} \right) \]

**Example (3):** the engine in *example (2)* is running with an air-fuel ratio \( AF = 15 \), a fuel heating value of 44000 (kJ/kg) and a combustion efficiency of 0.97. Calculate:

1. Rate of fuel flow into engine.
2. Brake thermal efficiency.
3. Indicated thermal efficiency.
4. Volumetric efficiency.
5. Brake specific fuel consumption.

**Solution:**

1. From example (2), the mass of air in one cylinder for one cycle is \( m_a = 0.0005\, kg \), then:

\[ m_f = \frac{m_a}{AF} = \frac{0.0005}{15} = 0.000033\, kg \] of fuel per cylinder per cycle

Therefore, the rate of fuel into the engine is:

\[ \dot{m}_f = 0.000033 \left( \frac{kg}{cylinder\,cycle} \right) \times 6(cylinder) \times \frac{3600\, (rev)}{60\, (sec)} \times \frac{1\,cycle}{2\,rev} \]

\[ \therefore \dot{m}_f = 0.006\, (kg/sec) \]

2. Using equation (1-55) to find brake thermal efficiency:

\[ \eta_{th.b} = \eta_b = \frac{\dot{W}_c}{\dot{m}_f \times H_L \times \eta_c} = \frac{77.3\, kW}{0.006\, (kg/sec) \times 44000\, (kJ/kg) \times 0.97} \]

\[ \therefore \eta_b = 0.302 = 30.2\% \]

Using equation (1-58):

\[ \eta_m = \frac{\eta_{th.b}}{\eta_{th.i}} \Rightarrow \eta_{th.i} = \frac{\eta_{th.b}}{\eta_m} = \frac{0.302}{0.85} = 0.355 \]

\[ \therefore \eta_{th.i} = 35.5\% \]
3. Using equation (1-60):

\[ \eta_v = \frac{m_a}{\rho_a \cdot V_d} = \frac{0.0005(kg)}{1.181\left(\frac{kg}{m^3}\right) \cdot 0.0005(m^3)} = 0.847 = 84.7\% \]

4. By using equation (1-48):

\[ bsfc = \frac{m_f}{W_b} = \frac{0.006\left(\frac{kg}{sec}\right)}{77.3(kW)} = 7.76 \times 10^{-5}\left(\frac{kg}{kW\cdot sec}\right) \]
1) A four-cylinder, two stroke cycle diesel engine with 10.9 (cm) bore and 12.6 (cm) stroke produces 88 (kW) of brake power at 2000 RPM. Compression ratio is 18:1. Determine:

a) Engine displacement \([cm^3 & L]\) … answer: (4703 & 4.703)
b) Brake mean effective pressure \([KPa]\) … answer: (561)
c) Torque \([N.m]\) … answer: (420)
d) Clearance volume of one cylinder \([cm^3]\) … answer: (69.2)

2) A four-cylinder, 2.4 liter engine operates on a four stroke cycle at 3200 RPM. The compression ratio is 9.4:1, the connecting rod length \((r = 18 \text{ cm})\), and the bore and stroke are related as \((S = 1.06* B)\). Calculate:

a) Clearance volume of one cylinder \([cm^3 & L]\)
b) Bore and stroke in \([cm & in]\)
c) Average piston speed in \([m/s]\)

3) A five-cylinder, 3.5 liter SI engine operates on a four-stroke cycle at 2500 RPM. At this condition, the mechanical efficiency of the engine is 62% and 1000 J of indicated work are produced each cycle in each cylinder. If \(B=S\), calculate:

a) Indicated mean effective pressure \([KPa]\) … answer: (1429)
b) Brake mean effective pressure \([KPa]\) … answer: (886)
c) Friction mean effective pressure \([KPa]\) … answer: (543)
d) Brake power in \([kW & hp]\) … answer: (64.4)
e) Torque \([N.m]\) … answer: (247)
f) Specific power \([kW/cm^2]\) … answer: (0.178)
g) Output per displacement \([kW/cm^3]\) … answer: (0.0185)
h) Specific volume \([cm^3/kW]\) … answer: (54.1)
i) Power lost in \([kW & hp]\) … answer: (39.6)

4) A small single-cylinder, two stroke cycle SI engine operates at 800 RPM with a volumetric efficiency of 0.85. The engine is square \((\text{bore}=\text{stroke})\) and has a displacement of 6.28 \((cm^3)\). The fuel-air ratio \(FA=0.067\). Determine:

a) Average piston speed \((m/s)\)
b) Flow rate of air into engine \((kg/s)\)
c) Flow rate of fuel into engine \((kg/s)\)
d) Fuel input for one cycle \((kg/cycle)\)
CHAPTER TWO
ENGINE CYCLES

2-1 Thermodynamics Principles:

2-1-1 For Isentropic Process:

\[ PV = mRT \leftrightarrow P_v = RT \leftrightarrow P = \rho RT \]  \hspace{1cm} \text{... (2-1)}

\[ dh = C_p dT \& du = C_v dT \]  \hspace{1cm} \text{... (2-2)}

\[ \frac{T_2}{T_1} = \left( \frac{v_1}{v_2} \right)^{\frac{1}{\gamma-1}} = \left( \frac{P_2}{P_1} \right)^{\frac{1}{\gamma}} \]  \hspace{1cm} \text{... (2-3)}

\[ w_{1-2} = \frac{P_1 v_1 - P_2 v_2}{\gamma - 1} = \frac{R(T_2 - T_1)}{1 - \gamma} \]  \text{ (in closed systems)}  \hspace{1cm} \text{... (2-4)}

2-1-2 For Polytropic Process:

\[ \frac{T_2}{T_1} = \left( \frac{v_1}{v_2} \right)^{\frac{k-1}{k}} = \left( \frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \]  \hspace{1cm} \text{... (2-5)}

\[ w_{1-2} = \frac{P_1 v_1 - P_2 v_2}{k - 1} = \frac{R(T_2 - T_1)}{1 - k} \]  \hspace{1cm} \text{... (2-6)}

\[ \gamma = \frac{C_p}{C_v} \geq k \]  \text{ (Specific Heat Ratio) (Index)}

\[ C = \sqrt{\gamma RT} \]  \hspace{1cm} \text{... (2-7)}

Where:  
\( P = \) Gas pressure in cylinder \([\text{KPa}]\)  
\( V = \) Gas volume \([\text{m}^3]\)  
\( v = \) Specific volume of gas \(= \frac{1}{\rho} \text{[m}^3/\text{kg}]\)  
\( R = \) Gas constant \([\text{kJ/kg.K}]\)  
\( T = \) Temperature \([\text{K}]\)  
\( m = \) Mass of gas in cylinder \([\text{kg}]\)  
\( \rho = \) Density of gas \([\text{kg/m}^3]\)  
\( h = \) Specific enthalpy \([\text{kJ/kg}]\)  
\( u = \) Specific internal energy \([\text{kJ/kg}]\)  
\( C_p \& C_v = \) Specific heat at constant pressure and constant volume  
\( C = \) Speed of sound
\( w = \text{Specific work [kJ/kg]} \)

The Ideal Cycles

2-2 Otto Cycle:

In 1876 Otto build a four-stroke internal combustion engine that compressed the air and gas before ignition.

\[
\begin{align*}
\text{Process (6-1):} & \quad \text{Constant pressure intake of air at } (P_o). \text{ Intake valve open and exhaust valve is closed.} \\
& \quad P_1 = P_6 = P_o \\
& \quad w_{6-1} = P_o \cdot (v_1 - v_6) \\
\end{align*}
\]

\[
\begin{align*}
\text{Process (1-2): Isentropic compression stroke. All valves are closed.} & \quad Q_{1-2} = 0 \\
& \quad w_{1-2} = \frac{P_1 \cdot v_1 - P_2 \cdot v_2}{\gamma - 1} = \frac{R \cdot (T_1 - T_2)}{\gamma - 1} \\
& \quad w_{1-2} = (u_1 - u_2) = C_v \cdot (T_1 - T_2) \\
\end{align*}
\]

\[
\begin{align*}
\text{Process (2-3): Constant-volume heat input (combustion). All valves are closed.} & \quad v_3 = v_2 = v_{TDC} \\
& \quad w_{2-3} = 0 \\
& \quad Q_{2-3} = Q_A = m_f \cdot H_L \cdot \eta_c = m_m \cdot C_v \cdot (T_3 - T_2) = (m_a + m_f) \cdot C_v \cdot (T_3 - T_2) \\
& \quad H_L \cdot \eta_c = (AF + 1) \cdot C_v \cdot (T_3 - T_2) \\
\text{Where: } m_m & = \text{maximum mass of mixture.}
\end{align*}
\]
\[ q_{2-3} = q_A = C_v(T_3 - T_2) = (u_3 - u_2) \]
\[ T_3 = T_{\text{max}} \text{ and } P_3 = P_{\text{max}} \]

Process (3-4): Isentropic power or expansion stroke. All valves are closed.

\[ q_{3-4} = 0 \]
\[ T_4 = T_3 \left( \frac{v_3}{v_4} \right)^{\gamma - 1} = P_3 \left( \frac{v_3}{v_4} \right)^{\gamma - 1} = T_3 \left( \frac{1}{r_c} \right)^{\gamma - 1} \]
\[ P_4 = P_3 \left( \frac{v_3}{v_4} \right)^{\gamma} = P_3 \left( \frac{v_3}{v_4} \right)^{\gamma} = P_3 \left( \frac{1}{r_c} \right)^{\gamma} \]
\[ w_{3-4} = P_3 \left( v_4 - v_3 \right) = \frac{R(T_4 - T_3)}{1 - \gamma} \]
\[ w_{3-4} = (u_3 - u_4) = C_v(T_3 - T_4) \]

Process (4-5): Constant-volume heat rejection (exhaust blow down). Exhaust valve open and intake valve is closed.

\[ v_5 = v_4 = v_1 = v_{\text{BDC}} \]
\[ w_{4-5} = 0 \]
\[ Q_{4-5} = Q_R = m_c C_v(T_5 - T_4) = m_c C_v(T_1 - T_4) \]
\[ q_{4-5} = q_R = C_v(T_5 - T_4) = C_v(T_1 - T_4) = (u_5 - u_4) \]

Process (5-6): Constant-pressure exhaust stroke at \( P_o \). Exhaust valve open and intake valve is closed.

\[ P_5 = P_6 = P_o \]
\[ w_{5-6} = P_o (v_6 - v_5) = P_o (v_6 - v_1) \]
\[ \eta_{\text{th}} = \frac{w_a}{q_A} = 1 - \frac{q_{\text{out}}(q_{\text{reject}})}{q_A} = 1 - \left[ \frac{C_v(T_4 - T_1)}{C_v(T_3 - T_2)} \right] = 1 - \left[ \frac{(T_4 - T_1)}{(T_3 - T_2)} \right] \]
\[ \frac{T_2}{T_1} = \left( \frac{v_1}{v_2} \right)^{\gamma - 1} = \left( \frac{v_4}{v_1} \right)^{\gamma - 1} = \frac{T_3}{T_4} \]
\[ \therefore \frac{T_4}{T_1} = \frac{T_3}{T_2} \]
\[ \eta_{\text{th}} = 1 - \frac{T_1}{T_2} \]
Using the above equations:

\[
\eta_{th,auto} = 1 - \left( \frac{1}{r_c} \right)^{\frac{\gamma-1}{\gamma}}
\]

\[\text{... (2-8)}\]

Figure (2-2): Real air-fuel cycle.

Work done during intake stroke:

\[w_{6-1} = P_{inlet}(V_1 - V_0) = P_{inlet}V_d\]

Work done during exhaust stroke:

\[w_{5-6} = P_{exh}(V_6 - V_5) = -P_{exh}V_d\]

Net indicated pumping work for the cycle:

\[(w_{pump})_{net} = (P_{inlet} - P_{exh})V_d\]
**Example (1):** A four-cylinder, 2.5 liter, SI automobile engine operates at WOT on four-stroke air standard Otto cycle at 3000 RPM. The engine has a compression ratio of 8.6:1, a mechanical efficiency of 86% and stroke-to-stroke ratio $S/B = 1.025$. Fuel is isoctane with $AF = 15$, a heating value of 44300 (kJ/kg), and combustion efficiency $\eta_c = 100\%$. At the start of compression stroke, condition in the cylinder combustion chamber are 100 kPa and 60 $^\circ$C. It can be assumed that there is a 4% exhaust residual left over the previous cycle. Do a complete thermodynamics analysis of this engine.

**Solution:**
For one cylinder:

*Displacement volume:*

$$V_d = \frac{2.5 \text{ (liter)}}{4} = 0.625(L) = 0.000625(m^3)$$

*Clearance volume: by using the equation below:

$$r_c = \frac{V_1}{V_2} = \frac{V_T}{V_c} = \frac{(V_d + V_c)}{V_c} = 8.6 = \frac{(0.000625 + V_c)}{V_c}$$

$$\therefore V_c = 0.0000822(m^3) = 0.0822(L) = 82.2(cm^3)$$

*Bore and stroke:*

$$V_d = \frac{\pi B^2}{4} \cdot S = \frac{\pi B^2}{4} \cdot (1.025 \cdot B) = 0.000625(m^3)$$

*Then:*

$$B = 0.0919(m) = 9.19(cm)$$

$$S = 1.025 \cdot B = 0.0942(m) = 9.42(cm)$$

*State (1):*

$$T_1 = 60 ^\circ C = 333K \& P_1 = 100KPa$$

$$V_1 = V_T = V_d + V_c = 0.000625 + 0.0000822 = 0.000707(m^3)$$

*Mass of gas in cylinder* can be calculated at state (1). The mass within the cylinder will be then the same for the entire cycle.

$$m_m = \frac{P_1 V_1}{K T_1} = \frac{100(KPa) \cdot 0.000707(m^3)}{0.287 \left( \frac{kJ}{kg \cdot K} \right) \cdot 333(K)} = 0.00074(kg)$$
**State (2):** The compression stroke (1-2) is isentropic.

\[ P_2 = P_1 (r_c)^\gamma = 100(8.6)^{1.35} = 1826 \text{(kPa)} \]

\[ T_2 = T_1 (r_c)^{\gamma-1} = 333(8.6)^{1.35-1} = 707(\text{K}) = 434(\text{°C}) \]

\[ V_2 = \frac{(m.R.T_2)}{P_2} = \frac{(0.00074 \times 0.287 \times 707)}{1826} = 0.0000822(m^3) = V_c \]

This is the clearance volume of one cylinder, which agrees with the above.

Another way of getting this value is by using the following equation:

\[ V_2 = \frac{V_i}{r_c} = 0.000707/8.6 = 0.0000822(m^3) = V_c \]

The mass of gas mixture \((m_m)\) in the cylinder is made up of air \((m_a)\), fuel \((m_f)\), and exhaust residual \((m_{exh})\):

\[ m_a = \left(\frac{15}{16}\right) \times (0.96) \times (0.00074) = 0.000666(\text{kg}) \]

\[ m_f = \left(\frac{1}{16}\right) \times (0.96) \times (0.00074) = 0.000044(\text{kg}) \]

\[ m_{exh} = (0.04) \times (0.00074) = 0.000030(\text{kg}) \]

Total: \( m_m = m_a + m_f + m_{exh} = 0.000666 + 0.000044 + 0.000030 = 0.074(\text{kg}) \)

**State (3):**

For the heat added during one cycle:

\[ Q_m = Q_{2-3} = Q_A = m_f.H_l.\eta_c = m_m.C_c(T_3 - T_2) \]

\[ 0.000044 \times 44300 = 0.00074 \times 0.821 \times (T_3 - 707) \]

Solving this for \(T_3\):

\[ T_3 = 3915(\text{K}) = 3642(\text{°C}) = T_{\text{max}} \]

\[ V_3 = V_2 = 0.0000822(m^3) \]

For constant volume: \( P_3 = P_2 \left(\frac{T_3}{T_2}\right) = 1826 \times \left(\frac{3915}{707}\right) = 10111(\text{kPa}) = P_{\text{max}} \)

**State (4):**

For the power stroke (3-4) isentropic, and then we can find temperature and pressure:

\[ T_4 = T_3 \left(\frac{1}{r_c}\right)^{\gamma-1} = 3915 \times \left(\frac{1}{8.6}\right)^{1.35-1} = 1844 K = 1571\text{°C} \]

\[ P_4 = P_3 \left(\frac{1}{r_c}\right)^{\gamma} = 10111 \times \left(\frac{1}{8.6}\right)^{1.35} = 554(\text{kPa}) \]

\[ V_4 = \frac{(m.R.T_3)}{P_4} = \frac{(0.00074 \times 0.287 \times 1844)}{554} = 0.000707(m^3) = V_1 \]
This agrees with the value of \((V_i)\) found earlier.

Work produced in isentropic power stroke for one cylinder during one cycle:

\[
W_{3-4} = \frac{mR(T_4 - T_1)}{1 - \gamma} = \frac{0.00074 * 0.287 * (1844 - 3915)}{1 - 1.35} = 1.257\, (kJ)
\]

Work absorbed during isentropic compression stroke for one cylinder during one cycle:

\[
W_{1-2} = \frac{mR(T_2 - T_1)}{1 - \gamma} = \frac{0.00074 * 0.287 * (707 - 333)}{1 - 1.35} = -0.227\, (kJ)
\]

Net indicated work for one cylinder during one cycle is:

\[
W_{net} = W_{1-2} + W_{3-4} = (+1.257) + (-0.227) = +1.03\, (kJ)
\]

To find heat added for one cylinder during one cycle:

\[
Q_{in} = Q_A = m_f * Q_{in} * \eta_c = 0.000044 * 44300 * 1.00 = 1.949\, (kJ)
\]

Indicated thermal efficiency:

\[
\eta_{sh, ob} = 1 - \left( \frac{T_1}{T_2} \right)^{\gamma - 1} = 1 - \left( \frac{333}{707} \right) = 1 - \left( \frac{1}{8.6} \right) \approx 0.529 = 52.9\%
\]

Or it could be found:

\[
\eta_{sh, ob} = \frac{W_{net}}{Q_{in}} = \frac{1.03}{1.949} = 0.529 = 52.9\%
\]

Indicated mean effective pressure:

\[
imep = \frac{W_{net}}{V_1 - V_2} = \frac{W_i}{V_1 - V_2} = \frac{1.03}{0.000707 - 0.0000822} = 1649\, (kPa)
\]

Indicated power at 3000 RPM is obtained by using the equation below:
\[
\dot{W}_i = \frac{W.N}{n} = \left[ \frac{1.03 \left( \frac{kJ}{\text{cylinder - cycle}} \right) * 3000 \left( \frac{\text{rev}}{60 \text{ sec}} \right)}{2 \left( \frac{\text{rev}}{\text{cycle}} \right)} \right] * 4(\text{cylinder}) = 103(kW) = 138(hp)
\]

Mean piston speed:
\[
\overline{U}_p = 2.S.N = 2\left( \frac{\text{strokes}}{\text{rev}} \right) * 0.0942\left( \frac{m}{\text{stroke}} \right) * \frac{3000}{60}\left( \frac{\text{rev}}{\text{sec}} \right) = 9.42(m/\text{sec})
\]

Brake work for one cylinder during one cycle:
\[
W_b = \eta_m * W_i = 0.86 * 1.03 = 0.886(kJ)
\]

Brake power at 3000 RPM:
\[
\dot{W}_b = \eta_m * \dot{W}_i = 0.86 * 103 = 88.6(kJ)
\]

Torque is calculated from the equation:
\[
T = \frac{\dot{W}_b}{2.\pi.N} = \frac{88.6 \left( \frac{kJ}{\text{sec}} \right)}{2 * \pi \left( \frac{\text{radians}}{\text{rev}} \right) * \frac{3000}{60} \left( \frac{\text{rev}}{\text{sec}} \right)} = 0.282(kN.m) = 282(N.m)
\]

Friction power lost:
\[
\dot{W}_{\text{lost}} = \dot{W}_i - \dot{W}_b = 103 - 88.6 = 14.4(kW) = 19.3(hp)
\]

Brakes mean effective pressure:
\[
bmep = \eta_m \times imep = 0.86 * 1649 = 1418(kPa)
\]

Brake specific power:
\[
BSP = \frac{\dot{W}_b}{A_p} = \frac{88.6(kW)}{\frac{\pi}{4} \left( 9.19cm \right)^2} \times 4(\text{cylinder}) = 0.334\left( \frac{kW}{cm^2} \right)
\]
Output per displacement:

$$OPD = \frac{\dot{W}_b}{V_d} = \frac{88.6(kW)}{2.5(L)} = 35.4\left(kW/L\right)$$

Brake specific fluid consumption:

$$bsfc = \frac{\dot{m}_f}{W_b} = \frac{0.000044\left(\frac{kg}{cylinder - cycle}\right) \cdot 50\left(\frac{rev}{sec}\right) \cdot 0.5\left(\frac{cycle}{rev}\right) \cdot 4(cylinder)}{88.6(kW)}$$

$$bsfc = 0.00005\left(\frac{kg}{kW.sec}\right) = 180\left(\frac{gm}{kW.hr}\right)$$

Volumetric efficiency:

$$\eta_v = \frac{m_a}{\rho_a \cdot V_d} = \frac{0.000666(kg)}{1.181\left(\frac{kg}{m^3}\right) \cdot 0.000625(m^3)} = 0.902 = 90.2\%$$
2-3 Diesel Cycle:

In 1890/1892 Rudolf Diesel and Akroyd Stuart planned and produced a new type of engine which was burning coal dust as fuel.

Figure (2-3): Diesel Cycle.

Air-standard:

Process (6-1): Constant pressure intake of air at \( P_o \). Intake valve open and exhaust valve is closed.

\[
P_1 = P_6 = P_o
\]

\[
w_{6-1} = P_o \cdot (v_1 - v_6)
\]

Process (1-2): Isentropic compression stroke. All valves are closed.

\[
V_2 = V_{TDC} \quad \text{&} \quad q_{1-2} = 0
\]

\[
w_{1-2} = \frac{P_2 \cdot v_2 - P_2 \cdot v_2}{1-\gamma} = \frac{R \cdot (T_2 - T_1)}{1-\gamma}
\]

\[
w_{1-2} = (u_1 - u_2) = C \cdot (T_1 - T_2)
\]
Process (2-3): Constant-pressure heat input (combustion). All valves are closed.

\[ P_3 = P_2 \]

\[ Q_{2-3} = Q_A = m_f \cdot H_L \cdot n_{EC} = m_m \cdot c_p \cdot (T_3 - T_2) = (m_a + m_f) \cdot c_p \cdot (T_3 - T_2) \]

\[ H_L \cdot n_{EC} = (A F + 1) \cdot c_p \cdot (T_3 - T_2) \]

\[ q_{2-3} = q_A = c_p \cdot (T_3 - T_2) = (h_3 - h_2) \]

\[ w_{2-3} = q_{2-3} - (u_3 - u_2) = P_2 \cdot (v_3 - v_2) \]

\[ T_3 = T_{\text{max}} \]

Process (3-4): Isentropic power or expansion stroke. All valves are closed.

\[ q_{3-4} = 0 \]

\[ w_{3-4} = \frac{P_3 \cdot v_4 - P_3 \cdot v_3}{1 - \gamma} = \frac{R \cdot (T_4 - T_3)}{1 - \gamma} \]

\[ w_{3-4} = (u_3 - u_4) = c_v \cdot (T_3 - T_4) \]

Process (4-5): Constant-volume heat rejection (exhaust blow down). Exhaust valve open and intake valve is closed.

\[ v_5 = v_4 = v_1 = v_{BDC} \]

\[ w_{4-5} = 0 \]

\[ Q_{4-5} = Q_R = m_m \cdot c_v \cdot (T_5 - T_4) = m_m \cdot c_v \cdot (T_1 - T_4) \]

\[ q_{4-5} = q_R = c_v \cdot (T_5 - T_4) = c_v \cdot (T_1 - T_4) = (u_5 - u_4) \]

Process (5-6): Constant-pressure exhaust stroke at \( (P_o) \). Exhaust valve open and intake valve is closed.

\[ P_5 = P_6 = P_o \]

\[ w_{5-6} = P_o \cdot (v_6 - v_5) = P_o \cdot (v_6 - v_1) \]
\[ \eta_{th}^{\text{Diesel}} = \frac{w_u}{q_A} = 1 - \frac{q_{out}(q_{reject})}{q_A} = 1 - \left[ C_v \left( \frac{T_4 - T_1}{T_3 - T_2} \right) \right] = 1 - \left[ \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)} \right] \]

\[ \eta_{th}^{\text{Diesel}} = 1 - \frac{1}{\gamma} \left( \frac{T_1}{T_2} - 1 \right) \left( \frac{T_3}{T_2} - 1 \right) \]

\[ T_3/T_2 = \left( \frac{v_3}{v_1} \right)^{\gamma-1} \quad \text{and} \quad \left( \frac{v_3}{v_4} \right)^{\gamma-1} = \frac{T_4}{T_3} \]

\[ \frac{v_3}{v_2} = \frac{T_3}{T_2} \quad \text{and} \quad v_4 = v_1 \]

With rearrangement, this can be shown to equal:

\[ \eta_{th}^{\text{Diesel}} = 1 - \frac{1}{\gamma} \left( \frac{v_3}{v_2} \right)^{\gamma-1} - 1 \left[ \frac{\left( v_3/v_2 \right)^{\gamma-1}}{\left( v_3/v_1 \right)^{\gamma-1}} \right] \]

\[ \therefore \eta_{th}^{\text{Diesel}} = 1 - \frac{1}{r_c} \gamma - 1 \left[ \frac{\beta^\gamma - 1}{\gamma(\beta - 1)} \right] \]

Where: \( \beta = \text{cutoff ratio} = \frac{v_3}{v_2} = \frac{T_3}{T_2} \)

Cutoff ratio is defined as the change in volume that occurs during combustion.

**Note:** The Diesel cycle efficiency dependent on \((r_c \& \beta)\). The cutoff ratio lies in the range \((1 < \beta < r_c)\) and is thus always greater than unity. Consequently the expression in square brackets is always greater than unity and the Diesel cycle efficiency is less than Otto cycle efficiency for the same \((r_c)\). In the actual engines; the \((r_c)\) of Diesel engine is usually greater than for petrol engine, so the former is usually more efficient.
2-4 Dual Cycle: it is also called Limited-Pressure or Mixed Cycle.

Figure (2-4): Dual Cycle.

\[ \eta_{th} = 1 - \frac{Q_A}{Q_r} \]
\[ Q_A = C_v(T_3 - T_2) + C_p(T_4 - T_3) \]
\[ Q_r = C_v(T_4 - T_3) \]

\[ \therefore \eta_{th_{Dual}} = 1 - \frac{T_3 - T_1}{(T_3 - T_2) + \gamma(T_4 - T_3)} \]

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

\[ T_3 = T_2 \left( \frac{P_3}{P_2} \right) = T_1 \alpha (r_C)^{\gamma - 1} \]

Where: \( \alpha = \left( \frac{P_3}{P_2} \right) \) (pressure ratio or degree of pressure increasing during combustion.

\[ \frac{v_4}{v_1} = \frac{\beta}{r_c}, \text{ and hence } T_4 = T_1 \alpha \beta^\gamma \]

\[ \therefore \eta_{th_{Dual}} = 1 - \left( \frac{1}{r_c} \right)^{\gamma - 1} \left[ \frac{\alpha \beta^\gamma - 1}{(\alpha - 1) + \gamma \alpha (\beta - 1)} \right] \]

1) If \( \beta = 1 \), the above equation becomes the thermal efficiency of Otto cycle.
2) If \( \alpha = 1 \), the above equation becomes the Diesel cycle efficiency.

\[ \eta_{th_{Otto}} = f(r_c, \& \gamma) \]

Now:
\[ \eta_{th_{Diesel}} = f(r_c, \gamma, \beta \& \alpha) \]
\[ \eta_{th_{Dual}} = f(r_c, \gamma, \beta \& \alpha) \]
Example (2): A small truck has a four-cylinder, 4 liter CI engine that operates on the air-standard Dual cycle (Fig. 2-4) using light diesel fuel at an air-fuel ratio of 18. The compression ratio of the engine is 16:1 and the cylinder bore diameter is 10 cm. At the start of the compression stroke, conditions in the cylinders are 60°C and 100 KPa with a 2% exhaust residual. It can be assumed that half of the heat input from combustion is added at constant volume and half at constant pressure. If heating value of the diesel fuel is 42500 (kJ/kg), Calculate:

1. Temperature and pressure at each state of the cycle
2. Indicated thermal efficiency
3. Exhaust temperature
4. Engine volumetric efficiency

Solution:

1) For one cylinder:

\[ V_d = \frac{4L}{4} = 1(L) = 0.001(m^3) = 1000(cm^3) \]

\[ r_c = \frac{V_{BDC}}{V_{TDC}} = \frac{V_d + V_c}{V_c} \]

\[ 16 = \frac{1000 + V_c}{V_c} \Rightarrow V_c = 66.7(cm^3) = 0.0667(L) = 0.000667(m^3) \]

\[ V_d = \frac{\pi B^2}{4}.S \]

\[ 0.001(m^3) = \frac{\pi \times [0.1(m)]^2}{4} \times S \Rightarrow S = 0.127(m) = 12.7(cm) \]

State (1): \( T_1 = 60°C = 333(K) \) & \( P_1 = 100(kPa) \)

\[ V_1 = V_{BDC} = V_d + V_c = 0.001 + 0.000667 = 0.0010667(m^3) \]

Mass of gas in one cylinder at start of compression:

\[ m_n = \frac{P_1 \times V_1}{R \times T_1} = \frac{100(kPa) \times 0.0010667(m^3)}{0.287 \frac{kJ}{kg.K} \times 333(K)} = 0.00112(kg) \]

Mass of fuel injected per cylinder per cycle:

\[ m_f = 0.00112(kg) \times \left( \frac{1}{19} \right) \times 0.98 = 0.0000578(kg) \]
State (2):

\[ T_2 = T_1 \left( r_c \right)^{r-1} = 333(K) \times (16)^{0.35} = 879(K) = 606\,^\circ C \]

\[ P_2 = P_1 \left( r_c \right)^r = 100(kPa) \times (16)^{0.35} = 4222(kPa) \]

\[ V_2 = \frac{m \times R \times T_2}{P_2} = \frac{0.00112(kg) \times 0.287 \left( \frac{kJ}{kg.K} \right) \times 879(K)}{4222(kPa)} \]

\[ V_2 = 0.000067(m^3) = V_d \]

Or \((V_2)\) can be found by using the following equation:

\[ V_2 = \frac{V_1}{r_c} = \frac{0.0010667}{16} = 0.0000667(m^3) \]

State (3):

\[ Q_A = m_f \times Q_{hv} = 0.0000578(kg) \times 42500 \left( \frac{kJ}{kg} \right) = 2.46(kJ) \]

If half of \((Q_A)\) occurs at constant volume, then using the following equation gives us:

\[ Q_{2-3} = m_w \times C_v \times (T_3 - T_2) \]

\[ \frac{2.46}{2} \left( kJ \right) = 0.00112(kg) \times 0.821 \left( \frac{kJ}{kg.K} \right) \times (T_3 - 879K) \]

\[ \therefore T_3 = 2217(K) = 1944\,^\circ C \]

\[ V_3 = V_2 = 0.0000667(m^3) \]

\[ P_3 = \frac{m \times R \times T_3}{V_3} = \frac{0.00112(kg) \times 0.287 \left( \frac{kJ}{kg.K} \right) \times 2217(K)}{0.0000667(m^3)} = 10650(kPa) = P_{\text{max}} \]

or:

\[ P_3 = P_2 \times \left( \frac{T_3}{T_2} \right) = 4222(kPa) \times \left( \frac{2217}{879} \right) = 10650(kPa) \]

State (4):

\[ P_4 = P_3 = 10650(kPa) = P_{\text{max}} \]

\[ Q_{3-4} = m_w \times C_p \times (T_4 - T_3) \]

\[ 1.23(kJ) = 0.00112(kg) \times 1.108 \left( \frac{kJ}{kg.K} \right) \times (T_4 - 2217K) \]

\[ \therefore T_4 = 3208(K) = 2935\,^\circ C = T_{\text{max}} \]

\[ V_4 = \frac{m \times R \times T_4}{P_4} = \frac{0.00112 \times 0.287 \times 3208}{10650} = 0.000097(m^3) \]
State (\(4\)): 

\[ V_4 = V_1 = 0.0010667 \text{ (m}^3) \]

\[ T_4 = T_4 \left( \frac{V_4}{V_1} \right)^{\gamma-1} = 3208 \left( \frac{0.000097}{0.0010667} \right)^{0.35} = 1386 \text{ (K) = 1113° C} \]

\[ P_4 = P_4 \left( \frac{V_4}{V_1} \right)^{\gamma} = 10650 \left( \frac{0.000097}{0.0010667} \right)^{0.35} = 418 \text{ (kPa)} \]

Work out for process (3-4) for one cylinder for one cycle:

\[ W_{3-4} = P_{\text{max}} (V_4 - V_3) = 10650 \times (0.000097 - 0.0000667) = 0.323 \text{ (kJ)} \]

Work out for process (4–\(\bar{4}\)):

\[ W_{4-\bar{4}} = \frac{m \times R \times (T_4 \bar{4} - T_4)}{1 - \gamma} = \frac{0.00112 \times 0.287 \times (1386 - 3208)}{1 - 1.35} = 1.673 \text{ (kJ)} \]

Work in for process (1-2):

\[ W_{1-2} = \frac{m \times R \times (T_2 - T_1)}{1 - \gamma} = \frac{0.00112 \times 0.287 \times (897 - 3208)}{1 - 1.35} = -0.501 \text{ (kJ)} \]

\[ W_{\text{nec}} = W_{3-4} + W_{4-\bar{4}} + W_{1-2} = (+0.323) + (+1.673) + (-0.501) \]

\[ \therefore W_{\text{nec}} = +1.495 \text{ (kJ)} \]

2) Indicated thermal efficiency:

\[ \eta_{\text{th}} \text{ (Dual) = } \frac{|W_{\text{nec}}|}{Q_A} = \frac{1.495}{2.46} = 0.607 = 60.7\% \]

Pressure ratio:

\[ \alpha = \frac{P_3}{P_2} = \frac{10650}{4222} = 2.52 \]

Cutoff ratio:

\[ \beta = \frac{V_4}{V_3} = \frac{0.000097}{0.0000667} = 1.45 \]

Also, thermal efficiency can be found from the following equation

\[ \eta_{\text{th}} \text{ (Dual) = } 1 - \frac{1}{r_c} \left[ \frac{\alpha \beta^\gamma - 1}{\gamma \alpha (\beta - 1) + (\alpha - 1)} \right] = 1 - \left( \frac{1}{16} \right)^{0.35} \left[ \frac{(2.52 \times 1.45^{1.35}) - 1}{1.35 \times 2.52 \times (1.45 - 1) + 2.52 - 1} \right] = 0.607 \]

3) Assuming exhaust pressure is the same as intake pressure, then exhaust temperature is found from:

\[ T_{\text{exh}} = T_4 \left( \frac{P_{\text{exh}}}{P_2} \right)^{\gamma-1} = 1386 \times \left( \frac{100}{418} \right)^{0.35} = 957 \text{ (K) = 684° C} \]
4) Mass of air entering one cylinder during intake:
\[ m_a = 0.00112 \times 0.98 = 0.0011 \text{ (kg)} \]

Then, volumetric efficiency is found by using the equation below:
\[ \eta_v = \frac{m_a}{\rho_a \times V_d} = \frac{0.0011 \text{ (kg)}}{1.181 \left( \frac{\text{kg}}{m^3} \right) \times 0.001 \text{ (m^3)}} = 0.931 = 93.1\% \]

2-5 Comparison of Otto, Diesel and Dual Cycles

2-5-1 For the same inlet conditions and the same compression ratio:

Figure (2-5): Comparison of Otto, Diesel and Dual Cycles at same inlet conditions & compression ratio.

Heat input for Otto is the area under \((2-3)\)
Heat input for Diesel is the area under \((2-3')\)
Heat input for Dual is the area under \((2-2'-3')\).
Heat rejection for Otto = Heat rejection for Diesel = Heat rejection for Dual Cycles = Area under \((1-4)\).

\[ Q_A(\text{Otto}) > Q_A(\text{Diesel}) \text{ also: } Q_A(\text{Otto}) > Q_A(\text{Dual}) \]
\[ \therefore W_{\text{Otto}} > W_{\text{Dual}} > W_{\text{Diesel}} \]
\[ &\eta_{\text{th}}(\text{Otto}) > \eta_{\text{th}}(\text{Dual}) > \eta_{\text{th}}(\text{Diesel}) \]
2-5-2 *For the same inlet conditions and same maximum temperature*:

Heat input \( (Q_A) \) for Otto is the area under \((2-3)\)
Heat input \( (Q_A) \) for Diesel is the area under \((2'-3)\)
Heat input \( (Q_A) \) for Dual is the area under \((2''-3''-3)\).
Heat rejection \( (Q_R) \) for Otto = Heat rejection \( (Q_R) \) for Diesel = Heat rejection \( (Q_R) \) for Dual Cycles = Area under \((1-4)\).

\[
\begin{align*}
Q_A^{\text{Diesel}} &> Q_A^{\text{Dual}} > Q_A^{\text{Otto}} \\
\therefore W_{\text{Diesel}} &> W_{\text{Dual}} > W_{\text{Otto}} > \\
&\& \eta_{th}^{\text{Diesel}} > \eta_{th}^{\text{Dual}} > \eta_{th}^{\text{Otto}}
\end{align*}
\]
2-5-3 For the same heat input and same maximum pressure:

Heat input \( (Q_A) \) for Otto is the area under \((2-3)\)
Heat input \( (Q_A) \) for Diesel is the area under \((2'-3')\)

But: \( Q_A )_{\text{Otto}} = Q_A )_{\text{Diesel}} \)

Heat rejection \( (Q_R) \) for Otto = area under \((1-4)\)
Heat rejection \( (Q_R) \) for Diesel = area under \((1-4')\)

\( Q_A )_{\text{Diesel}} < Q_A )_{\text{Otto}} \)

\( W_{\text{Diesel}} > W_{\text{Otto}} \)

\( \eta_{th} )_{\text{Diesel}} > \eta_{th} )_{\text{Otto}} \)
2-5-4 For the same maximum pressure and same maximum temperature:

![Diagram showing comparison of Otto and Diesel cycles.](image)

Figure (2-8): Comparison of Otto, Diesel Cycles at same maximum temperature & maximum pressure.

**Otto**: 1 → 2 → 3 → 4 → 1

**Diesel**: 1 → 2' → 3 → 4 → 1

\[ Q_A^{\text{Otto}} \underbrace{\text{under}(2 - 3)} \]

\[ Q_A^{\text{Diesel}} \underbrace{\text{under}(2' - 3)} \]

\[ Q_R^{\text{Otto}} \underbrace{\text{under}(4 - 1)} \]

\[ Q_R^{\text{Diesel}} \underbrace{\text{under}(4 - 1)} \]

\[ Q_A^{\text{Otto}} < Q_A^{\text{Diesel}} \]

\[ Q_R^{\text{Otto}} = Q_R^{\text{Diesel}} \]

\[ \therefore W_{\text{Diesel}} > W_{\text{Otto}} \text{ & } \eta_{th}^{\text{Diesel}} > \eta_{th}^{\text{Otto}} \]
2-5-5 For the same heat input and same compression ratio:

Figure (2-9): Comparison of Otto, Diesel and Dual Cycles at same heat input & same compression ratio.

\(Otto: 1 \rightarrow 2 \rightarrow 3 \rightarrow 4 \rightarrow 1\)

\(Diesel: 1 \rightarrow 2 \rightarrow 3' \rightarrow 4' \rightarrow 1\)

\(Dual: 1 \rightarrow 2 \rightarrow 2' \rightarrow 3'' \rightarrow 4'' \rightarrow 1\)

\(Q_A)_{Otto} = Q_A)_{Diesel} = Q_A)_{Dual}\)

\(Q_R)_{Otto} under(1-4)\)

\(Q_R)_{Diesel} under(1-4')\)

\(Q_R)_{Dual} under(1-4'')\)

\(Q_A)_{Otto} < Q_R)_{Dual} < Q_A)_{Diesel}\)

\(\therefore W_{Otto} > W_{Dual} > W_{Diesel}\)

\&

\(\eta_{th})_{Otto} > \eta_{th})_{Dual} > \eta_{th})_{Diesel}\)
2-6 Atkinson Cycle (also called: Over-expanded Cycle):

In Otto and Diesel cycles, when the exhaust valve is opened near the end of expansion stroke, pressure in the cylinder is still on the order of three to five atmospheres. A potential for doing additional work during the power stroke is there lost when the exhaust valve is opened and pressure is reduced to atmospheric. If the exhaust valve is not opened until the gas in the cylinder is allowed to expand to atmospheric pressure, a greater amount of work would be obtained in the expansion stroke with an increase in engine thermal efficiency. Atkinson cycle is illustrated in figure (2-10) below.

This cycle starting in 1885, a number of crank and valve mechanisms were tried to achieve this cycle, which has a longer expansion stroke than compression stroke. No large number of these engines has ever been marked, indicating the failure of this development.

\[
\text{Compression ratio: } r_c = \frac{v_1}{v_2}
\]

\[
\text{Expansion ratio: } r_e = \frac{v_4}{v_2}
\]

\[
\eta_{th} \text{ (Atkinson)} = 1 - \gamma \left( \frac{r_c - r_e}{r_e^\gamma - r_c^\gamma} \right)
\]
2-7 Miller Cycle:

This cycle is named after R. M. Miller (1890-1967), it is a modern modification of the Atkinson cycle and has an expansion ratio greater than the compression ratio. Miller cycle is shown in figure (2-11).

Air intake in a Miller cycle is un-throttled. The amount of air ingested into each cylinder is then controlled by closing the intake valve at the proper time, long before BDC (point-7). As the piston then continues towards BDC during the lasted part of the intake stroke, cylinder pressure is reduced along process (7-1). When the piston reaches BDC and starts back toward TDC, cylinder pressure is again increased during process (1-7). The work produced in the first part of the intake process (6-7) is canceled by part of the exhaust stroke (7-6), process (7-1) is canceled by process (1-7), and the net indicated work is the area within loop (7-2-3-4-5-7). There is essentially no pump work.

The shorter compression stroke which absorbs work combined with the longer expansion stroke which produces work, results in a greater net indicated work per cycle.
Compression ratio: \( r_c = \frac{v_7}{v_2} \)

Expansion ratio: \( r_e = \frac{v_4}{v_2} = \frac{v_4}{v_3} \)

\[
\eta_{th, Miller} = \frac{W}{Q_A}
\]

Automobiles with Miller cycle engines were first marked in the later half of the 1990s. A typical value of the compression ratio is about \( (r_c = 8:1) \) with an expansion ratio about \( (r_e = 10:1) \).
2-7 Joule (Brayton) Cycle:

Compression ratio: \( r_c = \frac{v_1}{v_2} \)

Pressure ratio: \( r_p = \frac{P_2}{P_1} \)

\[
\eta_{th, Brayton} = 1 - \frac{Q_R}{Q_A} = 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}
\]

\[
T_4 = T_3 \left( \frac{r_c}{(r_c)^{\gamma - 1}} \right) \quad \text{and} \quad T_1 = T_2 \left( \frac{r_c}{(r_c)^{\gamma - 1}} \right)
\]

\[
\eta_{th} = 1 - \frac{T_3 \left( \frac{T_3}{(r_c)^{\gamma - 1}} - \frac{T_2}{(r_c)^{\gamma - 1}} \right)}{T_3 - T_2} = 1 - \frac{1}{r_c^{\gamma - 1}}
\]

also;

\[
\eta_{th} = 1 - \frac{1}{r_p^{\gamma - 1}}
\]
The Actual Cycle

The actual cycle experienced by the internal combustion engine is not in the true sense thermodynamic cycle. An ideal air-standard thermodynamic cycle occurs on a closed system of constant composition. This is not what actually happens in an IC engine and for this reason air-standard gives a best only approximations to actual conditions and output. For actual cycle the efficiency is lower than the ideal efficiency and this duo to the following losses:

1) **Losses due to variation of Specific Heats with Temperature:**

All gases except monatomic gases show an increase in specific heat at high temperatures.

\[
C_p = a + k_1 T_1 + k_2 T_2 + \ldots \quad \text{for temper range from} \ (300-1500)K
\]

\[
C_v = b + k_1 T_1 + k_2 T_2 + \ldots
\]

But \( (\gamma) \) decrease with increasing temperature, because the difference between \( (C_p \& C_v) \) constant.

\[
\gamma = \frac{C_p}{C_v}
\]

\[\therefore R = C_p - C_v \Rightarrow C_p = R + C_v\]

\[\therefore \gamma = \frac{R + C_v}{C_v} = \frac{R}{C_v} + 1\]

2) **Dissociation Losses:**

Dissociation increases with increasing temperature, also considerable amount of heat is absorbed. This heat will be liberated when the element recombine as the temperature falls.

\[C + O_2 \leftrightarrow CO_2\]

\[2CO_2 + Heat \leftrightarrow 2CO + O_2\]

This effect appears at temperature:

\[(1000-1500)K\]

---

\[
P
\]

\[P
\]

\[P
\]
3) **Time Losses:**
Burning in theoretical cycle is assumed to be completed instantaneously, while burning in actual cycle is completed in a definite interval of time during which there will be a change in volume (the burning combustion is completed during $40^\circ$ of crank turn, i.e. $10^\circ$ before BDC and $30^\circ$ after TDC) and thus the maximum pressure dose not occurs sometime after TDC.

4) **Losses due incomplete combustion:**
Some fuel does not burn or burn partially and thus CO and \( \text{CO}_2 \) or even fuel will appear in the exhaust gases. This is due hydrogenous mixture, i.e. excess fuel in another part. Therefore, energy release in actual engine is about (90-93) % of fuel energy input.

5) **Direct Heat Losses:**
During the combustion process and subsequent expansion (power) stroke, heat will be loosed according to:

- **a.** From hot gases to cylinder wall to water or cooling fins.
- **b.** Heat into piston or cylinder wall carried away by engine oil.

*There for \( C_p \) and \( C_v \) are not constant.
6) *Exhaust Blow-down Losses*:

The cylinder pressure at the end of expansion stroke is about (3-5) bars know:

a. If the exhaust valve is opened at BDC, the piston has to do work against high cylinder pressure.

b. If the exhaust valve is opened too early (point 2), part of the expansion stroke is lost.

c. The best compromise is to open exhaust valve (40°–60°) before BDC, thus reducing the cylinder pressure to half way to atmospheric before the exhaust stroke begins:
   - Exhaust valve open at point (4) too late.
   - Exhaust valve open at point (1) too early.
   - The best is to open at point (2).

![Diagram of engine cycle](image-url)

7) *Pumping Losses*:

These losses are due to pumping gas from atmospheric (low inlet pressure) to higher exhaust pressure. Pumping losses increase as a part throttle because throttling reduces the suction pressure also increases with speed.
8) Rubbing Friction Losses:

These losses are due to:

- a. Friction between piston and cylinder walls.
- b. Friction in various bearings.
- c. Friction in auxiliary equipment as pumps (oil, water, fuel … etc), fans generators … etc.

These losses increased with increasing engine speed.
CHAPTER THREE
Induction and Exhaust Processes

3-1 General:

In reciprocating internal combustion engines, the induction and exhaust processes are non-steady flow processes (pulsation flow). For cycle analysis, the flow could be assumed steady. This assumption is reasonable especially for multi-cylinder engines with some form of silencing in induction and exhaust passages.

The inlet and exhaust manifold designs are often determined by consideration of cost, easy of manufacturing, space, easy of assembly, low pressure drop and optimum flow.

3-2 Intake System:

The intake system consists of an air filter, a carburetor and throttle valve or fuel injectors and throttle or throttle with individual fuel injectors in each intake port, and intake manifold. The intake manifold is designed to deliver air or air and fuel to the engine through pipes to each cylinder called runners. The inside diameter of the runners must be large enough so that a high flow resistance and the resulting low volumetric efficiency do not occur. On the other hand, the diameter must be small enough to assure high air velocity and turbulence which enhances its capability of carrying fuel droplets and increases evaporation and air-fuel mixing. The manifold should have:

1) Similar passage ways to all cylinders (same length, diameter and symmetry).
2) Turbulence inducers (high velocities are one means).
3) Hot spots to reduce large liquid droplets.
4) Smooth inner wall to reduce the thickness of liquid film and to minimize flow resistance.
3-3 Valve Types:

3-3-1 Mushroom-Shaped Poppet Valve:

- **Advantages:** cheep, good seating, easy lubrication, good heat transfer to cylinder head and good flow properties (i.e.; it give larger values of valve flow area to piston area than most other types and it has excellent flow coefficient if properly designed.

- **Flow Calculation:** the vale lift ($\ell$) depending on the engine size, usually about [5 to 10] (mm) for automobile engines.

Generally:

$$\ell_{\text{max}} \leq \frac{d_v}{4}$$

Where: $\ell_{\text{max}}$ = Value lift when valve is fully open.

$$\frac{d_v}{4} = \text{Diameter of valve}$$

Figure (3-1): Mushroom's valve calculation.

$$C_D = \frac{A_{\text{act}}}{A_{\text{pass}}} \quad \ldots (3-1)$$

Where:

$C_D$ = Discharge coefficient.

$A_{\text{act}}$ = Actual flow area (effective area).

$A_{\text{pass}}$ = Passage area of flow (curtain area) = $\pi d_v \ell_{\text{max}}$
\[ A_i = 1.3B^2 \left[ \frac{(P_i)_{\text{max}}}{C_i} \right] = \frac{\pi d_v^2}{4} \]  

... (3-2)

Where:

\( A_i \) = Minimum intake valve area for one cylinder.

\( C_i \) = Speed of sound at inlet conditions = \( \sqrt{\gamma RT_i} \), where: \( T_i \) = inlet air temperature.

**Note:** The angle of valve surface at the interface with the valve seat is generally designed to give minimum flow restriction. As air flow around corners, the stream line separate from the surface and the actual area of flow is less than the flow passage area (\( A_{\text{act}} \) is less than the \( A_{\text{pass}} \)). Figure (3-2) below illustrate this case.

\[ A_{\text{ex}} = 1.3B^2 \left[ \frac{(P_{\text{ex}})_{\text{max}}}{C_{\text{ex}}} \right] \]  

... (3-

3)

Where:

\( A_{\text{ex}} \) = Exhaust valve area

\( C_{\text{ex}} \) = Sonic speed at exhaust temperature

\[ \phi = \frac{A_{\text{ex}}}{A_i} = \frac{C_i}{C_{\text{ex}}} = \frac{\sqrt{\gamma RT_i}}{\sqrt{\gamma RT_{\text{ex}}}} = \sqrt{\frac{T_i}{T_{\text{ex}}}} \]
Where:
\[ \phi = \text{Valve area ratio} \]
\[ T_i = \text{Flow temperature at inlet} \]
\[ T_e = \text{Flow temperature at exhaust} \]

In actual engines, \((\phi)\) usually has a value of about \((0.8-0.9)\). To find the valves diameter:

\[
\frac{A}{x} = \frac{\pi d_v^2}{4}
\]

Where:
\[ x = \text{Number of intake valves or number of exhaust valves per cylinder.} \]

Mach index (Mach number):

\[
M = \frac{\text{Velocity of flow}}{\text{Inlet sonic velocity}}
\]

Assume the flow is incompressible, then:

**Volume flow rate through the piston** = **Volume flow rate through the valve**

\[
\bar{P}_s \times A_s = V \times A_i
\]

or:

\[
V = \frac{\bar{P}_s \times A_s}{A_i}
\]

Where:

\[ A_i = \text{Piston area} = \frac{\pi B^2}{4} \]

\[ V = \text{Velocity of the flow passing the valve.} \]

Substitute equation (3-5) into (3-4):

\[
M = \frac{\bar{P}_s \times A_s}{A_i} \times \frac{1}{C_i}
\]

... (3-)
Valve Operation System:

1) Over head valve (ohv): The valves are operating from the camshaft via cam followers. Figure (3-2) illustrates the over-head valve construction.

![Figure (3-2): over-head valve.](image)

2) Over head Camshaft (ohc): The camshaft can be mounted directly over the valve stems or it can be offset when the camshaft is offset the valves are operated by rockers.

![Figure (3-3): Over-head camshaft.](image)
3-3-2 Sleeve Valve:

They are consisted of a single sleeve or pair of sleeves between piston and the cylinder with inlet and exhaust ports. Figure (3-5) below shows the sleeve valve in details.
Advantages of Sleeve Valves:

1) Elimination the hot spot associated with a poppet valve (this is very important when only low octane fuel are available).
2) Compact engine when compared with engine using poppet valve.
3) Piston lubrication is improved since there was always relative motion between the piston strokes.

Disadvantages of Sleeve Valves:

1) Cost
2) Difficulty of manufacturing
3) Lubrication and friction problems between cylinder and sleeve
4) Heat transfer problem from piston through the sleeve and oil film to the cylinder.

Note: Motion of sleeve valve is half engine speed and under went vertical and rotary oscillation.

3-3-2 Rotary, Disc and Slide Valves: are still sometimes used, but are subject to heat transfer, lubrication and clearance problems.

Figure (3-6): Rotary Valve.
The Advantages of Four Valves per Cylinder:

1) Large valves throat area for gas flow.
2) Good heat transfer due to larger seat area.
3) Smaller valve forces occur since a lighter valve with lighter spring can be used. This will also reduce the hammering effect on the valve seat.

3-4 Exhaust System:

After leaving the cylinders by passing out of the exhaust valves, exhaust gases pass through the exhaust manifold, a piping system that directs the flow into one or more exhaust pipes.

The pressure pulses in the exhaust system are much greater than those in the inlet system, since in a naturally aspirated engine the pressures in the inlet have to be less than about (1 bar).

In designing the exhaust system for a multi-cylinder engine, advantage should be taken of the pulsed nature of the flow. The system should avoid sending pulses from the separate cylinders into the same pipe at the same time, since this will lead to increase the flow losses. However, it is sensible to have two or three (2 or 3) cylinders that are out of phase ultimately feeding into the same pipe. When there is a junction, [see figure (3-7) below], a compression wave will also reflect an expansion wave back. If the expansion wave returns to the exhaust valve at the end of the exhaust valve opening, then it helps to scavenge the combustion products if the inlet valve is also open then it will help to draw in the next charge. Obviously the cancellation of compression and expansion wave must be avoided.
Example: Arrange the exhaust system for a 4-cylinder, 4-strokes engine having firing order [1-3-4-2].
**Explanation:**
Consider the engine operating with the exhaust valve just opening on (cylinder-1). A compression wave will travel to the first junction; since the exhaust valve on (cylinder-4) is closed, an expansion wave will be reflected back to the open exhaust valve. The same process occurs (180°) later in the junction connecting cylinders (2 and 3). At the second junction, the flow is significantly steadier and ready for silencing.

**3-5 Carburetor Arrangement for Multi-Cylinder:**
Inlet manifolds are usually designed for easy of production and assembly, even on turbocharged engines when a single carburetor per cylinder is used. The flow pulsation will cause a rich mixture at full throttle as the carburetor will feed fuel for flow in either direction. In engines with a carburetor supplying more than one cylinder, the flow at the carburetor will be steadier because the interaction between compression and expansion waves.

**3-5-1 For Single Carburetor:**

![Diagram of carburetor arrangement for multi-cylinder engines with poor volumetric efficiency and poor mixture distribution.]

* Poor Volumetric Efficiency.
* Uniform Distribution.

![Diagram of carburetor arrangement for multi-cylinder engines with good volumetric efficiency and poor mixture distribution.]

* Good Volumetric Efficiency.
* Poor Mixture Distribution.

Figure (3-8): Single Carburetor.
3-5-2 For Twin Carburetor:

* Uniform Inlet Passages.
* Simple & Equally Effective.
* More widely used.

3-6 Valve Timing Diagram:

3-6-1 P-V Diagram:

\[ P_\alpha = \text{Atmospheric Pressure} \]
\[ \text{I.V.O} = \text{Inlet Valve Open} \]
\[ \text{I.V.C} = \text{Inlet Valve Closed} \]
\[ \text{E.V.O} = \text{Exhaust Valve Open} \]
\[ \text{E.V.C} = \text{Exhaust Valve Closed} \]

Figure (3-9): Twin Carburetor.

Figure (3-10): P-V Diagram.
3-6-2 Open Diagram for 4-Strokes Engine:

Figure (3-11): Open Diagram for 4-Strokes Engine.

\[ \theta = \text{angle of valve over-lap}. \]

3-6-3 Polar Diagram for 4-Strokes Engine:

Figure (3-12): Polar Diagram for 4-Strokes Engine.

- Power Stroke
- Intake Stroke
- Compression Stroke
- Exhaust Stroke

**Legend:**
- I.V.O = Inlet Valve Open
- I.V.C = Inlet Valve Closed
- E.V.O = Exhaust Valve Open
- E.V.C = Exhaust Valve Closed
3-7 Silencing:

The most effect approach to silencing is the reduction of the peaks, especially those in the most sensitive frequency rang of the air.

The inlet noise is attenuated by the air filter and its housing, the air filter is also acts as a flame trap if the engine back-fire.

Exhaust silencers work either by absorption or by modifying the pressure wave in such away as to lead to cancellation and reduction in sound. Absorption silencers work by dissipating the sound energy in porous medium.

3-7-1 Absorptive Silencer:

![Absorptive Silencer Diagram]

3-7-2 Modifying Pressure Waves:

a) Expansion Box
b) Side Resonator
c) Constriction
d) Interface Filter

![Modifying Pressure Waves Diagram]

(a) Expansion Box
(b) Side Resonator
(c) Constriction
(d) Interface Filter
CHAPTER FOUR

Fuels

Over 99% of the world’s internal combustion engines use liquid fuel derived from petroleum. In some countries where natural petroleum is scarce, fuel having very similar composition and characteristics are being produced. There are four significant sources of crude oil: (1) petroleum; (2) coal liquefaction; (3) shale oil and (4) tar sands. Most of the crude oil used to date has been petroleum derived since what is found in the ground requires processing before delivering to a refinery. Coal, on the other hand must be treated to increase its hydrogen content and removes undesirable elements such as nitrogen, sulfur, arsenic, mercury, cadmium or phosphorous. Shale oil is difficult to get out ground since it is soaked up in rocks. Tar sands contain hydrocarbons mixed with sand and are more difficult to remove from the ground than petroleum.

Crude oil contains a large number of different hydrocarbons. For example, 2500 different compounds have been found in one sample of petroleum-derived crude oil subjected to an extraordinarily thorough analysis. The compounds range from gases to viscous liquids and waxes.

Crude petroleum is a mixture of hydrocarbons compounds, small quantities of sulfur, small quantities of oxygen and nitrogen, small quantities of metallic compound such as iron, radium, nickel, …etc. The exact composition of crude oil differs widely according to its source.

Refining of crude oil usually starts with distillation at atmospheric pressure during which process of the distillate separated into various fractions according to volatility. The resulting distillates are called straight run products. Distillates may then be subjected to heat treatments and chemical treatments at various pressures and temperatures, such treatments are define as cracking, when they chiefly tend to reduce the average molecular size and as polymerization when the reverse predominates. The products resulting from the retirement are
classified by their usage and according to their specific gravity and their volatility. The important products of refining process are:

1) **Natural gas**: these gases are usually associated with liquid petroleum either standing above the liquid in the earth or dissolved in it. Some of these gases are methane, propane, butane … etc.

2) **Gasoline**: this type is used in S.I.E; the chemical composition of its constituents varies widely depending on the base crude and the methods used in retiring.

3) **Kerosene**: it is used as a fuel in for C.I.E engines and aircraft gas turbines also as a jet fuel. Also, used in lamps, heaters … etc.

4) **Distillate**: is slightly heavier than kerosene.

5) **Diesel oils**: are petroleum fractions that lie between kerosene and the lubricating oils. These oils cover a wide range of distillation having a wide range of specific gravity. They are used in various types of compression ignition engines.

6) **Fuel oils**: they are used in continuous burners.

7) **Lubricating oils**: they are made up in part from heavy distillates of petroleum and in part from residual oils, that is, oils remaining after distillation.

8) **Tar, Asphalt and Petroleum coke** are solid or semisolids products, which remain undistilled.

### 4.1 Chemical Structure of Petrol

Fuel are often mixtures of hydrocarbons, the carbon and hydrogen atoms may be linked in different ways and these bonds or linked influence the chemical and physical properties of the different hydrocarbon groups. The following classifications cover the more important groups in petroleum fuels:

1) **Straight Chain, or Normal Paraffins**: the general chemical formula for this series is $[C_n H_{2n+2}]$, where (n) is the number of carbon atoms. The carbon atoms
are connected together as a chain with hydrogen atoms filling the empty valences. For example; normal heptane \((C_7H_{16})\):

\[
\begin{align*}
\text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} \\
\text{H} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{H} \\
\text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H}
\end{align*}
\]

2) **Branched Chain or Isoparaffins**: the general formula is \([C_nH_{2n+2}]\). In these compounds, the chain of carbon atoms is branched, for example isoctane \((C_8H_{18})\)

\[
\begin{align*}
\text{H} & \quad \text{H} \\
\text{H} & \quad \text{HCH} & \quad \text{H} & \quad \text{HCH} & \quad \text{H} \\
\mid & \quad \mid & \quad \mid & \quad \mid & \quad \mid \\
\text{H} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{H} \\
\mid & \quad \mid & \quad \mid & \quad \mid & \quad \mid \\
\text{H} & \quad \text{HCH} & \quad \text{H} & \quad \text{H} & \quad \text{H} \\
\text{H}
\end{align*}
\]

3) **Olefins**: are compounds with one or more double bonded carbon atoms in a straight chain. The general formulas are: (1) for the mono-olefins (one double bond) is \([C_nH_{2n}]\) for example; Hexane \((C_6H_{12})\) and (2) for the diolefins (two double bonds), for example; Butadiene \((C_4H_6)\) is \([C_nH_{2n-2}]\).

**Hexane**

\[
\begin{align*}
\text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} \\
\mid & \quad \mid & \quad \mid & \quad \mid & \quad \mid & \quad \mid \\
\text{H} & \quad \text{C} & \quad \text{C} & \quad \text{C} & \quad \text{C} = \text{C} & \quad \text{H} \\
\mid & \quad \mid & \quad \mid & \quad \mid \\
\text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H}
\end{align*}
\]

**Butadiene**

\[
\begin{align*}
\text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H} \\
\mid & \quad \mid & \quad \mid & \quad \mid & \quad \mid & \quad \mid \\
\text{H} & \quad \text{C} = \text{C} & \quad \text{C} = \text{C} & \quad \text{H} \\
\mid & \quad \mid & \quad \mid \\
\text{H} & \quad \text{H} & \quad \text{H} & \quad \text{H}
\end{align*}
\]
4) **Naphthenes**: are characterized by a ring structure, examples:

![Cyclopropane](image1)

![Cyclobutane](image2)

5) **Aromatics**: are hydrocarbons with carbon-carbon double bonds internal to a ring structure. The most common aromatic is benzene ($C_6H_6$):

![Benzene](image3)

Some common aromatics have groups substituted for hydrogen atoms:

![Substituted Aromatics](image4)
4.2 Fuel for S. I. E:

The following characteristics are important for fuel used in S.I.E.

1. **Volutility:**

   It defines as the tendency of fuel to evaporate under a given set of conditions (i.e. tendency of fuel to go from a liquid to a gaseous state). Volatility is an important characteristic of gasoline that affects the engine performance and fuel economy. It is expressed in term of the volume percentage that is distilled at or below fixed temperature.

   Volatility can be divided into three regions called front end (0%-20% evaporated), the mid range (20%-80% evaporated) and tail end (80%-100% evaporated). The above affect could be distributed as follows:

   **A) Front End Volatility:**

   **a) Cold Starting:**

   The problem of cold starting is to have sufficient fuel vapor, thus a more volatile fuel is desirable. The approximate limits of inflammability of air-gasoline mixture are; **8:1**(for rich mixture), **20:1** (for lean mixture), and **12:1** (the best for starting from cold).

   **b) Hot Starting:**

   Good cold starting fuel causing hot starting problems, because the amount of fuel evaporating and so going into the inlet manifold under hot shutdown condition is high and the mixture formed is too rich to ignite. This could be avoided by proper placement and design of the fuel system.
c) **Vapor Lock:**

The vapor lock is situation where too lean mixture is supplied to the engine which cause uneven running of an engine, stalling while irregular acceleration, difficult starting when hot. The vapor lock can be reduced by:

i) Keeping the fuel system element away from heat

ii) Improving the vapor handling capacity of fuel system

iii) Limiting the fuel from propane and butane

**Mid Range Volatility:**

a) **Engine Warm up:**

When an engine is first starting, it dose not respond as rapidly to change in operating conditions as it does after having been run for sometime. The interval between starting-up and the time at which flexible operation is possible, referred to as the warm-up period. The mid range portion should
be volatile enough to give satisfactory A/F ratios under various operation conditions.

b) **Engine acceleration, Smoothness and Fuel economy:**

A part from the size, design and its mechanical condition the acceleration of an engine depends upon its ability to deliver suddenly to the intake an extra supply of fuel-air mixture in a sufficiently vaporized form to burn quickly (good acceleration occurs when an A/F vapor ratio is about 12:1). For power and smoothness of operation correct fuel-air ratio should be delivered to all cylinders. As low as mid-range and tail end volatility as practical will help in good mixture distribution and hence good fuel economy.

c) **Short and Long trip economy:**

Short trip economy is related to warm-up characteristics. Keeping the warm-up period to a minimum by having a fuel, which is relatively volatile in mid-range permits more efficient operation and great economy. In long trip, a gasoline that has more weight per liter will give more (kms/liter) in a warmed-up engine, since the higher boiling hydrocarbons are generally heavier than the more volatile ones, the mid-range and tail-end parts of the fuel should be as involatile as practical for good long trip economy.

d) **Carburetor Icing:**

When the gasoline is vaporized in the carburetor there will be drop in temperature of carburetor body and if the humidity is high (greater than 75%) and the air temperature lower than 10° C, water condenses out of air and freezes on carburetor. The presence of ice-up-sets carburetion resulting in poorer economy, because ice formation restricts the air path and thus engine stalls due to richness of mixture and even can stop completely due to air starvation. Two types of gasoline additives are used to overcome this problem. One is a freezing point depressant type, such as dipropylene glycol (DPG) to reduce the temperature at which ice forms.
The other is a surfactant type, which coats the carburetor surface to prevent the ice sticking to it.

D) Tail End Volatility:

a) Crankcase Dilution: If part of fuel has too high evaporation temperature, this part will not be completely vaporized and will be carried as fuel droplets into the combustion chamber. This liquid fuel gets past the piston rings into the crankcase where it dilutes the oil and decrease viscosity. It also washes away the lubricating oil film on cylinder walls. Crankcase dilution is more at low engine operating temperature, also engines using heavy fuels, such as kerosene and distillate.

b) Varnish and Sludge Deposite: Certain types of high boiling hydrocarbons contribute to varnish and sludge deposition inside an engine. These deposits can cause piston ring plugging, sticking, and valve sticking resulting in poor operation and poor fuel economy.

c) Spark Plug Fouling: Some high boiling hydrocarbons form deposits leading to spark plug fouling. Lower the tail end volatility less are the chances of spark plug fouling.

d) Evaporation Loss: Evaporation loss from storage tanks and carburetor depends on vapor pressure, which is a function of fraction components, and initial temperature, vapor pressure of the gasoline is also responsible for evaporation losses due to venting from the tanks. This loss decrease the fuel economy also decreases its anti-knock (the fuel anti-knock) quality as the lighter fraction has higher anti-knock properties.
2. **Knock Characteristics:**

Octane rating is a measure of its anti-knock performance of fuel. A scale of (0-100) is devised by assigning a value of (0 to n-heptanes) and a value of 100 to isooctane [a fuel resistant to knock]. For example, a 95-octane fuel has the performance equivalent to that of a mixture of 95% isooctane and 5% normal heptane by volume. The octane requirement of an engine depends:

a) Engine pressure ratio  
b) Operation conditions  
c) Geometrical and mechanical considerations

![Graph showing the relationship between octane number and pressure ratio.](image)

The octane number of a fuel is determined by using special standard single cylinder engine "CFR" (Cooperating Fuel Research) in which the fuel being tested is compared with a reference mixture of two chemically pure hydrocarbons; isooctane \((C_8H_{18})\) and normal heptane \((C_7H_{16})\). The test must be done under standard condition and setting compression ratio to give standard knock intensity as measured by the knock indicator. While holding compression ratio and other test conditions constant, various mixtures of the two reference fuels (called Primary Reference Fuels "PRF") are tried until a mixture, which gives the same knock intensity as the fuel under, is discovered. The percentage of isooctane in the matching mixture is called the octane number of the fuel. The octane number of gasoline engine fuel is improved by adding additives. These additives or antiknock must have the following properties:
a) Low cost per unit increase in octane rating  
b) No deposits left in the engine or exhaust system  
c) Relatively low boiling temperature to ensure good distribution in multi cylinder engines  
d) Complete solubility  
e) Nontoxic, and nontoxic exhaust emissions  
f) Stable

The primary commercial antiknock is: (1) Lead additives taking the form of lead alkyls, either tetramethyl lead $[\text{(CH}_3\text{)}_4\text{Pb}]$, or tetraethyl lead $[\text{(C}_2\text{H}_5\text{)}_4\text{Pb}]$. Most countries now have restrictions on use of lead in fuels for environmental reasons (the danger of lead pollution). (2) Iron carbonyl has been tried in Europe, but the product of combustion (iron oxide), which tends to short the spark plug and to cause extreme wear of the cylinder and rings. Also, ethyl alcohol is used as antiknock additives. For example;

<table>
<thead>
<tr>
<th>Name</th>
<th>Grams to match gallon gasoline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tetraethyl lead</td>
<td>1</td>
</tr>
<tr>
<td>Methyl triethyl lead</td>
<td>1.05</td>
</tr>
<tr>
<td>Dimethyl diethyl lead</td>
<td>1.1</td>
</tr>
<tr>
<td>Tetramethyl lead</td>
<td>1.3</td>
</tr>
<tr>
<td>Ethyl alcohol</td>
<td>158</td>
</tr>
</tbody>
</table>

Sensitivity: is the difference between research octane number (RON) and motor octane number (MON).

$$\text{Sensitivity} = \text{RON} - \text{MON}$$

The higher the sensitivity the poorer its performance under service condition.
3. **Gum Deposits:**

All petroleum motor fuels oxidize slowly in presence of air. The oxidation of unsaturated hydrocarbons (also unstable sulfur and nitrogen compounds) results in formation of resinous materials called gum. High gum content fuels may clog carburetor jet, promote sticking of automatic chokes, sticking of the intake valve, piston rings and promote formation of manifold deposit reducing volumetric efficiency ($\eta_v$), thus the gum content as well as the tendency to form gum is limited in gasoline specifications.

4. **Sulfur:**

Hydrocarbon fuel (motor fuels) may contain free sulfur, hydrogen sulfide and other sulfur compounds, which are objectionable for several reasons:

   a) They are corrosive elements and corrode fuel lines, carburetor and injection pump

   b) The sulfur will unite with oxygen to form sulfur dioxide that in the presence of water at low temperature may form sulfurous acid or the sulfur dioxide to unit with other substance to form products that could cause engine wear at low or even high temperatures.

   c) Sulfur has a low ignition temperature, the presence of sulfur can reduce the self-ignition temperature, and thus promoting knock in the S.I.E. and tending to decrease knock in the C.I.E.

   d) It is found that the response of S.I. fuel to tetraethyl lead reduces by the presence of sulfur.
4.3 Non Petroleum Fuels:

1) Methyl alcohol, or methanol \((CH_{3}OH)\) and ethyl alcohol or ethanol \((C_{2}H_{5}OH)\), methanol is used in racing engines on account of the cooling effect produced by its very large (negative) heat of vaporization. Ethanol blended with gasoline has been used in locations where it is punctual as a byproduct of sugar refining.

2) Benzol: it is a mixture of about 70% benzene \((C_{6}H_{6})\), 20% toluene \((C_{7}H_{8})\) and 10% xylene \((C_{8}H_{10})\) and some sulfur components. It has a very high knock resistively, but its freezing point is 5.5°C so it cannot be used in cold climate. Also, the heating value is lower than gasoline. It is used as a blending agent with gasoline.

3) Gaseous: the main gaseous fuel are used in internal combustion engines are natural gas, liquid petroleum gas (LPG), producer gas, blast-furnace gas, coke oven gas.

4.4 Fuel for Diesel Engines:

The requirement for good C.I. fuel cannot be as were these for gasoline. This situation arises because of the added complexity of the C.I. engine from its heterogeneous combustion process, which is strongly affected by injection characteristics. However, the following general observation can be made:

1) Knock characteristics:

Cetane rating (octane number) is the measure of knock characteristics. The best fuel in general will have a cetane rating sufficiently high to avoid objectionable knock.

\[
CN = \frac{104 - ON}{2.75}
\]

Cetane number: the cetane rating of a Diesel fuel is a measure of its ability to auto ignites quickly (ignitability) when it injected into the compressed and heated air in the engine. Through ignition is affected by:

i) Engine design parameters such as compression ratio, injection rate, injection time, inlet air temperature,…etc

ii) Hydrocarbon composition of the fuel

iii) Volatility of fuel
2) **Starting Characteristics:**

The fuel should start the engine easily. The requirement demands high volatility to form readily a combustible mixture and a high cetane rating in order that the self-ignition temperature will be low.

3) **Smoking and Odor:**

The fuel should not promote either smoke or odor from the exhaust pipe. In general, good volatility is demanded as the first prerequisite to ensure good mixing and therefore complete combustion.

4) **Corrosion and Wear:**

The fuel should not cause corrosion before combustion, or corrosion and wear after combustion. These requirements appear to be directly related to the sulfur ash and residue contains of the fuel.

**Combustion of Fuels:**

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

\[
\text{Fuel} + \text{Air} \rightarrow \text{Products of Combustion} + \text{Heat}
\]

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.
**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 kmol C + 1 kmol O\(_2\) → 1 kmol CO\(_2\)

12 kg C + 32 kg O\(_2\) → 44 kg CO\(_2\)

*ii- Combustion of Hydrogen*

\[ H_2 + 1/2 O_2 \rightarrow H_2O \]

1 kmol H\(_2\) + 1/2 kmol O\(_2\) → 1 kmol H\(_2\)O

2 kg H\(_2\) + 16 kg O\(_2\) → 18 kg H\(_2\)O


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,

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

A frequently used quantity in the analysis of combustion process is the air fuel ratio $A/F$. It is defined as the ratio of the mass of air to the mass of fuel for a combustion process.

$$A/F = \frac{m_a}{m_f} = \text{mass of air} \div \text{mass of fuel}$$

The mass $m$ of a substance is related to the number of moles $n$ through the relation: $m = n \times M$, 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.

$$\text{Equivalence ratio} = \phi = \frac{(A/F)_{\text{stoich}}}{(A/F)_{\text{actual}}} = \frac{(F/A)_{\text{actual}}}{(F/A)_{\text{stoich}}}$$
Where:
\( \phi = 1 \): stoichiometric
\( \phi < 1 \): lean (week) mixture - excess of air.
\( \phi > 1 \): rich mixture - deficiency of air.

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

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

Example (1): Isooctane is burned with 120% theoretical air in small three cylinder turbocharged automobile engine. Calculate:

1) air fuel ratio
2) fuel air ratio
3) equivalence ratio

Solution:
Stoichiometric reaction:
\[
C_8H_{18} + 12.5O_2 + 12.5(3.76)N_2 \rightarrow 8CO_2 + 9H_2O + 12.5(3.76)N_2
\]

With 20% excess air:
\[
C_8H_{18} + 15O_2 + 15(3.76)N_2 \rightarrow 8CO_2 + 9H_2O + 15(3.76)N_2 + 2.5O_2
\]

1) \( A / F = \frac{m_a}{m_f} = \frac{N_aM_a}{N_fM_f} = \frac{(15)(4.76)(29)}{(1)(114)} = 18.16 \)

2) \( F / A = \frac{m_f}{m_a} = \frac{N_fM_f}{N_aM_a} = \frac{1}{A / F} = \frac{1}{18.16} = 0.055 \)

3) Fuel-air ratio of stoichiometric combustion:
\[
(F / A)_{stoich} = \frac{m_f}{m_a} = \frac{N_fM_f}{N_aM_a} = \frac{(1)((12*8)+(2*9))}{(12.5)(4.76)(29)} = 0.066
\]
Equivalence ratio is obtained from:

\[
\phi = \frac{(A/F)_{\text{stoich}}}{(A/F)_{\text{actual}}} = \frac{(F/A)_{\text{actual}}}{(F/A)_{\text{stoich}}} = \frac{0.055}{0.066} = 0.833
\]

**Example (2):** The four-cylinder engine of a light truck owned by a utility company has been converted to run on propane fuel \((C_3H_8)\). A dry analysis of the engine exhaust gives the following volumetric percentages:

- \(CO_2 = 4.90\%\)
- \(CO = 9.79\%\)
- \(O_2 = 2.45\%\)

Calculate the equivalence ratio at which the engine is operating.

**Solution:**

The three components identified sum up to \((4.90+9.79+2.45=17.14\%)\) of the total, which means that the remaining gas (nitrogen) accounts for \((82.86\%)\) of the total.

\[
xC_3H_8 + yO_2 + y(3.76)N_2 \rightarrow 4.90CO_2 + 9.79CO + 2.45O_2 + 82.86N_2 + zH_2O
\]

Conservation of nitrogen (N2) during reaction gives:

\[
y(3.76) = 82.86 \Rightarrow y = 22.037
\]

Conservation of carbon (C):

\[
3x = 4.90 + 9.79 \Rightarrow x = 4.897
\]

Conservation of hydrogen (H2):

\[
8x = 2z \\
8 \times 4.897 = 2z \\
\therefore z = 19.588
\]

The reaction is:

\[
4.9C_3H_8 + 22.037O_2 + 22.037(3.76)N_2 \rightarrow 4.9CO_2 + 9.79CO + 2.45O_2 + 82.86N_2 + 19.588H_2O
\]

Dividing by 4.9:

\[
C_3H_8 + 4.5O_2 + 4.5(3.76)N_2 \rightarrow CO_2 + 2CO + 0.5O_2 + 16.92N_2 + 4H_2O
\]
Actual air-fuel ratio:

\[
(A/F)_{\text{actual}} = \frac{m_a}{m_f} = \frac{(4.5)(4.76)(29)}{(1)(3*12)+(2*4)} = 14.12
\]

Stoichiometric combustion:

\[
C_3H_8 + 5O_2 + 5(3.76)N_2 \rightarrow 3CO_2 + 4H_2O + 5(3.76)N_2
\]

Stoichiometric air-fuel ratio:

\[
(A/F)_{\text{stoich}} = \frac{m_a}{m_f} = \frac{(5)(4.76)(29)}{(1)(3*12)+(2*4)} = 15.69
\]

Equivalence ratio can be calculated as following:

\[
\phi = \frac{(A/F)_{\text{stoich}}}{(A/F)_{\text{actual}}} = \frac{15.69}{14.12} = 1.11
\]

Example (3): Find the stoichiometric A/F ratio for the combustion of ethyl alcohol (C2H5OH) in a petrol engine. Calculate the A/F ratios for 0.9 & 1.2 equivalence ratios (\( \phi \)). Determine the wet and dry analyses by volume of the exhaust gas for each equivalence ratio.

Solution:

Combustion equation of ethyl alcohol is:

\[
C_2H_5OH + 3O_2 + 3(3.76)N_2 \rightarrow 2CO_2 + 3H_2O + 3(3.76)N_2
\]

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:

\[
(3*4.76*29) = 414.12
\]

\[
\therefore \text{Stoichiometric (A/F) ratio} = \frac{414.12}{46} = 9.002 \approx 9
\]

\[
\phi = \frac{(A/F)_{\text{stoich}}}{(A/F)_{\text{actual}}}
\]

\[
0.9 = \frac{9}{(A/F)_{\text{actual}}}
\]

\[
\therefore (A/F)_{\text{actual}} = \frac{9}{0.9} = 10
\]
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:

\[
C_2H_5OH + 1.11[3O_2 + 3(3.76)N_2] → 2CO_2 + (0.11)3O_2 + 3H_2O + 1.11(3)(3.76)N_2
\]

The total number of moles of products = 2+3+0.33+12.52 = 17.85.
Total dry moles = 2+0.33+12.52 = 14.85
Hence dry analysis is:

\[
CO_2 = \frac{2}{17.85} \times 100 = 11.204\%
\]
\[
O_2 = \frac{0.33}{17.85} \times 100 = 1.848\%
\]
\[
H_2O = \frac{3}{17.85} \times 100 = 16.806\%
\]
\[
N_2 = \frac{12.52}{17.85} \times 100 = 70.144\%
\]

⇒ For φ = 1.2:

\[
(A/F)_{\text{actual}} = \frac{9}{1.2} = 7.5
\]

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 H2O, since H₂ atoms have a greater affinity for oxygen than C atoms. The carbon in the fuel will burn to CO and CO₂:

\[
C_2H_5OH + 0.834[3O_2 + 3(3.76)N_2] → aCO_2 + bCO + 3H_2O + 0.834(3)(3.76)N_2
\]

C balance: 2 = a + b
O balance: 1+2 × 0.834 × 3 = 2a + b + 3
Subtracting the equations gives: a = 1.004
Then: b = 2 − 1.004 = 0.996

The rest of question is homework
H. W.

1) \( C_4H_8 \) is burned in an engine with a rich fuel-air ratio. Dry analysis of the exhaust gives the following volume percentages: \( \text{CO}_2 = 14.95\% \), \( C_4H_8 = 0.75\% \), CO = 0\%, \( \text{O}_2 = 0\% \), with the rest being \( \text{N}_2 \). Calculate:
   a) Air-fuel ratio
   b) Equivalence ratio

2) Methanol (\( CH_3OH \)) is burned in an engine with air at equivalence ratio of 0.75. Write the balanced chemical equation for this reaction and air-fuel ratio.

3) Isooctane (\( C_8H_{18} \)) is burned with air in an engine at an equivalence ratio of 0.833, find air-fuel ratio and write the balanced chemical reaction equation.
CHAPTER FIVE

Fuel Metering For S.I. Engines

Carburetor or the injection system meters the amount of fuel to be delivered into the air stream, which is, depends on load. Any two of the following could determine load:

a) Torque, b) Engine speed, c) Throttle position, d) Rate of airflow, e) Pressure in the inlet manifold.

5-A Carburetor

5.1 Carburetion:

Is the process of preparing fuel-air mixture, this process is done outside the engine through a device called a carburetor.

The basic or main element of most carburetors consists of an air passage of fixed geometry containing a venturi-shaped restriction. A fuel nozzle is located in the venturi throat and is supplied with fuel from a constant-level float chamber. A throttle down stream in the venture controls air.

As air enters the engine due to the pressure differential between the surrounding atmospheric air and the partial vacuum in the cylinders during intake strokes, it is accelerated to high velocity in the throat of the venturi.
By Bernoulli's principle, this causes the pressure in the throat \((P_d)\) to be reduced to a value less than the surrounding pressure \((P_a)\), which is about one atmosphere. The pressure above the fuel in the fuel-floating chamber is equal to atmospheric pressure as the floating chamber is vented to surroundings. Therefore, there is a pressure differential through the fuel supply capillary tube and this forces fuel flow into the venturi throat. As the fuel flows out of the end of the capillary tube, it breaks into very small droplets, which are carried away by the high velocity of air. These droplets then evaporated and mixed with the air in following intake manifold. As engine speed is increased, the higher flow rate of air will create an even lower pressure in the venturi throat. This creates a greater pressure differential through the fuel capillary tube, which increases the fuel flow rate to keep up with the greater air flow rate and engine demand.

The level in the fuel reservoir (floating chamber) is controlled by a float shut off (needle valve). Fuel comes from a fuel tank supplied by an electric fuel pump on most modern automobiles, by a mechanical driven fuel pump on older automobiles, or even by gravity on some small engines and historic automobiles.

The throttle controls the air flow and thus the engine speed. There is an idle speed adjustment which sets the closed throttle position such that some air can flow even at fully closed throttle. An idle jet is added, which gives better fuel flow control, at idle and almost closed throttle position.

**The process of carburetion is affected by the following factors:**

1) Time: when the engine speed increases the time available for mixture formation is small.

2) Quality of fuel: petrol consists of various hydrocarbons having different volatility.

3) Operation condition: (temperature and pressure of the ambient).

4) The design of the induction system and combustion chamber.
5.2 Carburetor Flow Equations:

5.2.1 Air Mass Flow Rate:

\[
\dot{m}_{\text{air}} = \dot{m}_a = C_a A_d \sqrt{2 \rho_a \Delta P_d}
\]

(5-1)

Where:
\( \dot{m}_a \) = Air mass flow rate \((kg/s)\)
\( C_a \) = Air-flow discharge coefficient
\( A_d \) = Venturi throat area \( (m^2) \)
\( \rho_a \) = Air density \((kg/m^3)\)
\( \Delta P_d \) = Static pressure difference between atmospheric and venturi throat area

\[
\Delta P_d = P_a - P_d \ (N/m^2)
\]

(5-2)

\( P_a \) = Atmospheric static pressure
\( P_d \) = Static pressure at venturi throat

\[
\Delta P_d = \Delta h \cdot \rho \cdot g \ (N/m^2)
\]

(5-3)

\( \rho \) = Manometer fluid density \((kg/m^3)\)
\( g = 9.8 \ (m/sec^2) \)
\( \Delta h = \) Pressure head difference across (1) and (2) in (meter)

### 5.2.2 Fuel Mass Flow Rate:

\[
\dot{m}_{fuel} = \dot{m}_f = C_f \cdot A_y \cdot \sqrt{2 \rho_f \cdot \left( \Delta P_d - g \cdot \rho_f \cdot x \right)}
\]

(\(5-4\))

Where:
- \( \dot{m}_f = \) Fuel mass flow rate (\( kg/s \))
- \( A_y = \) Fuel orifice area, jet area (fuel jet area) (\( m^2 \))
- \( C_f = \) Fuel discharge coefficient
- \( \rho_f = \) Fuel density (\( kg/m^3 \))
- \( x = \) Distance between fuel level and the high of capillary tube (\( m \))
5.2.3 Excess Air Coefficient ($\lambda$):

Is the ratio of the amount of air enters the engine during combustion (actual mass of air) to the theoretical amount needed during combustion (stoichiometric mass of air).

$$\lambda = \frac{\dot{m}_a_{\text{actual}}}{\dot{m}_a_{\text{stoich}}}$$

$$\lambda = \frac{A_d \cdot C_a \cdot \sqrt{2 \cdot \rho_a \cdot \Delta P_d}}{AF_{\text{stoich}} \cdot A_y \cdot C_f \cdot \sqrt{2 \cdot \rho_f \cdot \left(\Delta P_d - g \cdot \rho_f \cdot \lambda\right)}}$$

$$\lambda = \frac{1}{AF_{\text{stoich}}} \cdot \frac{A_d}{A_y} \cdot \frac{\sqrt{2 \cdot \rho_a}}{\sqrt{2 \cdot \rho_f}} \cdot \frac{C_a}{C_f} \cdot \sqrt{\frac{\Delta P_d}{\Delta P_d - g \cdot \rho_f \cdot \lambda}}$$

(5-5)

**Assume:** $(g \cdot \rho_f \cdot \lambda)$ is small compared to $(\Delta P_d)$, therefore it could be neglected.

$$\frac{1}{AF_{\text{stoich}}} \cdot \frac{A_d}{A_y}, \quad \text{and} \quad \frac{\sqrt{2 \cdot \rho_a}}{\sqrt{2 \cdot \rho_f}}$$

are constants

$$\lambda \approx \text{const.} \cdot \frac{C_a}{C_f}$$

(5-6)

At the same temperature:

- $\Delta P_d$ is small, $\lambda > 1$ (lean mixture)
- $\Delta P_d$ is large, $\lambda < 1$ (rich mixture)
5.2.4 The Effect of Altitude on AF ratio:

The temperature and pressure of air decreases with increasing altitude as given by the following expressions:

\[ T_{alti} = T_s - 0.0065h \] \hspace{1cm} \text{... (5-7)}

\[ T_{alti} = \text{Temperature of air at altitude (K)} \]
\[ T_s = \text{Temperature of air at certain level (K)} \]
\[ h = \text{Altitude (m)} \]

\[ h = 19200 \log_{10} \left( \frac{1.03}{P_{alti}} \right) \] \hspace{1cm} \text{... (5-8)}

\[ P_{alti} = \text{Pressure (kgf/cm}^2) \]

\[ \frac{(AF)_{altitude}}{(AF)_{at \text{ certain level}}} = \sqrt{\frac{\rho_{altitude}}{\rho_s}} \]

\( \rho_s = \text{Air density at certain level} \)

\[ \therefore \frac{(AF)_{alti}}{(AF)_s} = \sqrt{\frac{P_{alti}}{P_s}} = \sqrt{\frac{T_{alti}}{T_s} \cdot \frac{T_s}{P_s}} \] \hspace{1cm} \text{... (5-9)}

Where:

\( P_s = \text{Air pressure at certain level} \)
5.3 **Types of Carburetors:**

There are two basic types of carburetors:

5.3.1 Fixed Venturi (F.V.) or fixed jet type

5.3.2 Variable Venturi (V.V.) or variable jet type

The size of venturi and the size of the fuel jet are changed with speed. Both types may be:

a) **Down draft carburetor** (vertical venturi tube with air flowing from the bottom): is best in that gravity assists in keeping the fuel droplets flowing in same direction as the air flow. Along runner (passage between throttle and intake manifold) that allows more distance and time for evaporation and mixing is also good.

b) **Updraft carburetor:** for special reasons of space and/or other considerations, some engines are fitted with updraft carburetors. These need high flow velocities to carry the fuel droplets in suspension against the action of gravity.

c) **Side draft carburetors:** were developed with air flowing horizontally. These generally need higher flow velocities to keep the fuel droplets suspended in the air flow, and with higher velocities come greater pressure losses.

5.4 **Auxiliary Devices and System of Carburetors:**

a) **Choke system** (*starting system*); when the choke is partly closed, less air is admitted into the carburetor and a high vacuum is built up in the venturi. This causes intensive outflow of the fuel from idling jet.

b) **Idle running system**; this system insure operation of the engine without load, especially at low speed (throttle is almost completely closed).

c) **Main jet** (*main metering jet*); control the economy or cruise range. The main metering system or jet must be compensated to provide essentially constant lean or stoichiometric mixture over the (20% to 80%) air flow range.
d) *Economizer or power compensating system*; used to supplies an additional amount of fuel under full load (maximum power) as wide-open throttle is approached.

e) *Altitude compensation system*; is required to adjust the fuel flow to changes in air density.

f) *Acceleration pump*; this pump enriches the mixture during acceleration of the engine. When the throttle is sharply opened, the air response is almost instantaneous but the fuel flow lags, thus the pump help to overcome this lag.

### 5.5 Altitude Compensate System:

The effects of increasing in altitude on the carburetor could be consider as;

Air density changes with ambient pressure and temperature, with changes due to changes in pressure with altitude being most significant, while ambient temperature variation, winter to summer can produce change of comparable magnitude, the temperature of the air entering the carburetor for warmed-up engine operations is controlled to within much closer tolerance by drawing an appropriate fraction of the air from around the exhaust manifold. A number of methods can be used to compensate for changes in ambient pressure with altitude:

1) **Venturi Bypass Method**: to keep the air volumetric flow rate through the venturi equal to what it was at sea level atmospheric pressure (calibration edition), a bypass circuit around the venturi for the additional volume is provided.

2) **Auxiliary Jet Method**: an auxiliary fuel-metering orifice with a pressure controlled tapered metering rod connects the fuel bowl to the main wall in parallel with the main metering orifice.

3) **Fuel Block-Section Method**: as altitude increases, an aneroid bellows moves a tapered metering rod from an orifice near the venturi throat
admitting to the bowl an increasing amount of the vacuum single developed at the throat.

4) **Compensate Air-Bleed Method**: the orifices in the bleed circuits to each carburetor system are filled with tapered metering pins actuated by a single aneroid bellows.

**Example (1)**: for simple carburetor; venturi diameter is 20 (mm), $C_a = 0.85$, $\rho_a = 1.2$, $C_f = 0.66$, fuel orifice diameter (fuel jet diameter) = 12.5 (mm), $x = 5$ (mm). Determine the $AF$ ratio when the pressure drop = 0.07 (bar) and $\rho_f = 750 (kg/m^3)$.

**Solution**:

$$m_{air} = m_a = C_a \cdot A_d \cdot \sqrt{2 \cdot \rho_a \cdot \Delta P_d}$$

$$m_{fuel} = m_f = C_f \cdot A_y \cdot \sqrt{2 \cdot \rho_f \cdot (\Delta P_d - g \cdot \rho_f \cdot x)}$$

$$AF = \frac{A_d \cdot C_a \cdot \sqrt{2 \cdot \rho_a \cdot \Delta P_d}}{A_y \cdot C_f \cdot \sqrt{2 \cdot \rho_f \cdot (\Delta P_d - g \cdot \rho_f \cdot x)}}$$

$$AF = \left( \frac{20}{1.25} \right)^2 \cdot \left( \frac{0.85}{0.66} \right) \cdot \left( \frac{\sqrt{2 \cdot 1.2 \cdot 0.07}}{2 \cdot 750 \cdot (0.07 - 3.678 \cdot 10^{-4})} \right) = 13.22$$

**Notice**: $g \cdot \rho_f \cdot x = 750 \cdot 9.81 \cdot \frac{5}{1000} = 36.78 \approx 3.678 \cdot 10^{-4} (bar)$

In case neglecting the value of $(x)$ and assuming that the fuel at the edge, the solution could be (عند إهمال قيمة $(x)$ واعتبار الوقود عند الحافة):

$$AF = \frac{A_d \cdot C_a \cdot \sqrt{\rho_a}}{A_y \cdot C_f \cdot \sqrt{\rho_f}} = \frac{0.85}{0.66} \cdot \left( \frac{20}{1.25} \right)^2 \cdot \sqrt{\frac{1.2}{750}} = 13.2$$
Example (2): A petrol engine has a fuel consumption of 10 (liter/hr). The air-fuel ratio is 15. The venturi throat diameter is 20 (mm). Determine the diameter of jet if the top of the jet is 5 (mm) above the fuel level in the float chamber. The barometer reads 750mmHg, the temperature is 32°C, \( C_a = 0.85 \), \( \rho_f = 750(\text{kg/m}^3) \) and \( C_f = 0.7 \).

Solution: air density at 32°C and \( \left( \frac{750}{760} \right) \text{bar} \) is:

\[
\rho_a = \frac{750 * 1.013 * 10^5}{287 * (32 + 273)} = 1.143(\text{kg/m}^3)
\]

\[
AF * \dot{m}_f = \dot{m}_a = A_y * C_a * \sqrt{\frac{2}{\rho_a} \Delta P_d}
\]

\[
\frac{15}{1} \left( \frac{10 * 10^{-3}}{3600} \right) * 700 = \frac{\pi}{4} \left( \frac{20}{1000} \right)^2 * 0.85 * \sqrt{2 * 1.143 * \Delta P_d}
\]

\[\Delta P_d = 5218(\text{N/m}^2)\]

\[
\dot{m}_f = A_y * C_f * \sqrt{2 \rho_f * \left[ 5218 - \left( 700 * 9.81 * \frac{5}{1000} \right) \right]}
\]

\[
\frac{10 * 10^{-3}}{3600} * 700 = A_y * 0.7 * \sqrt{2 * 700(5218 - 34.335)}
\]

\[
A_y = 1.031(\text{mm}^2) = \frac{\pi}{4} * d^2
\]

\[\therefore d = 1.145(\text{mm})\]

Example (3): A petrol engine has the following parameters at sea level: \( AF = 14 \), \( T = 27°C \), \( P = 1.03 \text{bar} \). Calculate the AF ratio at the altitude of 5000 (m).

Solution:

\[T_{alt} = T_s - 0.0065h = 27 - 0.0065 * 5000 = -5.5°C\]

\[h = 19200 * \log_{10} \frac{1.03}{P_{alt}}\]

\[5000 = 19200 * \log_{10} \frac{1.03}{P_{alt}}\]

\[\log_{10} \frac{1.03}{P_{alt}} = 0.2604 \Rightarrow \frac{1.03}{P_{alt}} = 1.821\]

\[\therefore P_{alt} = 0.565(\text{bar})\]
\[
\frac{(AF)_{\text{alti}}}{(AF)_{s}} = \sqrt{\frac{\rho_{\text{alti}}}{\rho_{s}}}
\]

\[
(AF)_{\text{alti}} = 14 \times \sqrt{\frac{0.565 \times 300}{267.5 \times 1.03}} = 11
\]
5-B **Electronic Fuel Injection System (EFI)**

The Electronic Fuel Injection System can be divided into three basic sub-systems. These are the: **(1) Fuel Delivery System**, **(2) Air Induction System**, and **(3) Electronic Control System**.

1) **The Fuel Delivery System:**

   - The fuel delivery system consists of the fuel tank, fuel pump, fuel filter, fuel delivery pipe (fuel rail), fuel injector, fuel pressure regulator, and fuel return pipe.

   - Fuel is delivered from the tank to the injector by means of an electric fuel pump. The pump is typically located in/or near the fuel tank. Contaminants are filtered out by a high capacity in line fuel filter.

   - Fuel is maintained at a constant pressure by means of a fuel pressure regulator. Any fuel that is not delivered to the intake manifold by the injectors returned to the tank through a fuel return pipe.

The fuel delivery system is illustrated in the figure below.
2) **The Air Induction System:**

- The Air Induction System consists of the air cleaner, air flow meter, throttle valve, air intake chamber, intake manifold runner, and intake valve.
- When the throttle valve is opened, air flows through the air cleaner, through the air flow meter, past the throttle valve, and through a well-tuned intake manifold runner to intake valve.
- Air delivered to the engine is a function of drive demand. As the throttle valve is opened further, more air is allowed to enter the engine cylinders.

The air induction system is illustrated in the figure below.

3) **Electronic Control System:**

- The electronic control system consists of various engine sensors, Electronic Control Unit (ECU), fuel injector assemblies, and related wiring.
- The ECU determines how much fuel needs to be delivered by the injector by monitoring the engine sensor.
The ECU turns the injectors on for a precise amount of time, referred to as injection pulse width or injection duration, to deliver the proper air/fuel ratio to the engine.

The electronic control system is illustrated in the figure below.
Basic System Operation:

1) Air enters the engine through the air induction system where it is measured by the air flow meter. As the air flows into the cylinder, fuel is mixed into the air by the fuel injector.

2) Fuel injectors are arranged in the intake manifold behind each intake valve. The injectors are electrical solenoids, which are operated by the ECU.

3) The ECU pulses the injector by switching the injector ground circuit on and off.

4) When the injector is turned on, it opens, spraying atomized fuel at the back-side of the intake valve.

5) As fuel sprayed into the intake air stream, it mixes with the incoming air and vaporizes due to the low pressures in the intake manifold. The ECU signals the injector to deliver just enough fuel to achieve an ideal air/fuel ratio of 14.7:1, often referred to as stoichiometry.

6) The precise amount of fuel delivered to the engine is a function of ECU control.

7) The ECU determines the basic injection quantity based upon measured intake air volume and engine RPM.

8) Depending on engine operating conditions, injection quantity will vary. The ECU monitors variables such as coolant temperature, engine speed, throttle angle, and exhaust oxygen content and makes injection corrections which determine final injection quantity.

Advantages of Electrical Fuel Injection:

1) Uniform air/fuel mixture distribution

2) Highly accurate air/fuel ratio control throughout all engine operating conditions

3) Superior throttle response and power

4) Excellent fuel economy with improved control

5) Improved cold engine start-ability and operation

6) Simple mechanics, reduce adjustment sensitivity
6.1. General:

The basic difference between S.I.E and D.E. is the means by which the fuel and air are mixed and burned. In S.I.E, the fuel and air are mixed external to the cylinder volume; during intake a fuel-air mixture is inducted through the intake valve into the cylinder. In diesel engine, fuel and air are mixed internally; during intake only air is inducted into the cylinder.

Diesel engines are able to operate at higher compression ratios than gasoline engines, because the fuel is mixed with air at the time of combustion is to commence. The compression ratio is deliberately selected to be high enough so that the gases near the end of the compression stroke are hot enough so that the fuel auto-ignites very soon after injection starts. The remaining fuel to be injected can then burn no faster than it is injected. The period between the start of injection and auto-ignition is called ignition delay. Its duration depends upon the design of engine and the fuel type.

The usual compression ratios for diesel engines vary from (12:1 to 20:1) or can be to (30:1), the corresponding pressure and temperature at the end of compression stroke being (28 kg/cm$^2$ to 100 kg/cm$^2$) and (500$^\circ$C to 900$^\circ$C) respectively.

6.2. Mixture Formation:

In diesel engine, the mixture is formed at the end of compression stroke during a short and very brief interval of time corresponding to (20$^\circ$ to 60$^\circ$) of crank travel. Thus, the process takes place during combustion. This condition makes the mixture formation difficult, because of:

1) Short time injection and mixture formation

2) Greater viscosity of diesel fuel

The energy used to distribute fuel throughout the combustion chamber (this energy consists of kinetic energy of fuel jet and kinetic energy of air, "air
charges energy"), depends on the method of mixture employed and the shape of combustion chamber.

The mixture formation consists of a number of physical processes:

1) Splitting of spray into droplets
2) Heating and evaporation of fuel
3) Distribution throughout the combustion chamber (those take place inside the cylinder).

Most of these processes take place simultaneously. There are three principle methods of mixture formation employed in modern automotive diesel engines, these methods are:

A) Volumetric Method:

A jet of fuel is issued from the nozzle of the injector, broken into droplets (5-40 micron) in size and mixing with air and rapid combustion. The stream of fuel is broken up into droplets by:

1) The forces of aerodynamic resistance of air medium into which fuel is injected.
2) The internal disturbances appearing when the fuel moves in the nozzle duct.

B) Film Method:

The idea of this method is to allow a minimum amount of fuel to evaporate and mix with air during the ignition lag. This is done by directed the fuel to the wall of combustion chamber at an acute angle so that the droplets are not reflected, but spread over the surface in the form of a thin film (0.012 to 0.014 mm) thick. The path of the fuel spray from the orifice nozzle to the wall should be as short as possible to reduce the quantity of fuel vaporized as the stream moves in the combustion chamber.

The advantages of this method:

1) The engine operates satisfactory with various fuel (multi-fuel operation)
2) Lower maximum combustion pressure
3) Lower fuel consumption
**The disadvantages of this method:**

1) It needs a special heater for starting the engine from cold
2) High temperature is needed to evaporate the fuel, this ensures by piston crown, therefore, piston crown needs cooling with oil (for supercharging engine), and otherwise increasing the piston temperature beyond the safe limit is not advice.

**C) Combined Method**

![Diagram](image)

**6.3. Air Flow Inside Combustion Chamber:**

Air flow is important factor in mixture formation due to the following reasons:

1) Distribute the fuel among the chamber
2) Mixing the fuel with air
3) Assist combustion
4) Reduce after burning

⇒ **Types of air flow:**

1) **Direct:**
   
   a) Tangential flow (swirl)
   
   b) Radial flow (squish)
   
   c) Axial flow
   
   d) Combined flow (radial + tangential)
2) Non-Direct:
   a) Turbulent
   b) Swirl

**6.4. Jet Geometry:**

The geometry of the jet spray depends upon the injection angle or nozzle angle ($\beta$) and penetration length ($L$).

*Injection (nozzle) angle ($\beta$) and penetration length ($L$) are depend on:*

1) Pressure inside the cylinder
2) Density of the air
3) Viscosity of the fuel
4) Diameter of the fuel nozzle

*Penetration length depends on:*

a) *Injection pressure*; increasing the injection pressure increases the spray penetration in proportion to:
Where:
\[ L = (P_N)^{0.4} \]

\( P_N \) = Injection pressure

Above a certain pressure, the spray becomes finely atomized and has insufficient momentum to penetration as far.

b) **Aspect ratio of the nozzle hole (ratio of length to diameter of the nozzle)**: long aspect ratio holes produce a jet that diverges less and penetrates further.

c) **Density of the fuel**: increasing the density of the fuel increases penetration, also denser fuels are more viscous and this causes the jet to diverge and to atomize less.

d) **Engine speed**: the jet penetration increases with engine speed. The injection period occupies an approximately constant fraction of the cycle for a given load, so as engine speed increases, the jet velocity and thus penetration also increase.

### 6.5. Methods of Injection:

Diesel engines are divided into two basic categories according to their combustion chamber design:

1. **Direct injection (D.I) engines**, which have a single open combustion chamber (undivided combustion chamber) into which fuel is injected directly.
2. **Indirect injection (I.D.I) engines**, where the chamber is divided into two regions and the fuel is injected into the "prechamber" which is connected to main chamber (situated above the piston crown) via a nozzle or one or more orifices.

I.D.I engine designs are only used in the smallest engine size. Within each category there are several different chamber geometry, air-flow, and fuel-injection arrangements.
6.5.1 Direct Injection Systems (D.I):

In the largest size engines where mixing rate requirements are least stringent, quiescent direct injection systems are used. The momentum and energy of the injected fuel jets are sufficient to achieve adequate fuel distribution and rates of mixing with air additional organized air motion is not required. The combustion chamber shape is usually a shallow bowl in the crown of the piston, and a central multi-hole injector is used. This type could be divided into:

a) Semi quiescent and low swirl open chamber  
b) Medium swirl open chamber  
c) High swirl open chamber

Open combustion chambers have the following performance:

a) Good performance in term of fuel economy, power and emission when properly developed  
b) The shape is less critical than careful design of the air motion and fuel injection (the most air motion in D.I. is swirl)

Optimum swirl ratio for both optimum economy and power output is about (10.5).

The following is specification for performance:

i- Compression ratio (12:1 to 16:1)  
ii- (Stroke / Bore) ratio > 1  
iii- Maximum mean piston speed (12 m/s)

6.5.2 Indirect Injection Systems (I.D.I):

Indirect injection systems have a divided combustion chamber with some form of pre-chamber in which the fuel is injected, and a main chamber with the
piston and valves (see figure (6-3)). The purpose of a divided combustion chamber is to speed up the combustion process in order to increase the engine output by increasing engine speed. **There are three principle classes of divided combustion chamber:**

A) *Pre-combustion chamber*; this type rely (depends) on turbulence to increase combustion speed

B) *Swirl combustion chamber*; this type rely on an ordered air motion to raise combustion speed

C) *Air-Cell and Energy Cell*; fuel is injected into the main chamber and ignites. As combustion proceeds, fuel and air will be forced into secondary chamber or air cell, so producing turbulence. As the expansion, stroke continuous the air, fuel and combustion products will flow out of the air cell, so generally further turbulence.

**Advantages of divided combustion chamber:**

a) This type giving a high (bmep – brake mean effective pressure)

b) The fuel injection requirements is less sever and lower fuel injection pressure are satisfactory. A single orifice in the nozzle is sufficient, but the spray direction should be into the air for good starting and on the chamber wall for good running

c) The delay period (or ignition delay) is reduced, therefore great air utilization and faster combustion; also lower quality fuel can be used. These conditions permit small engines to run at higher speed with larger output.

**Disadvantages of divided combustion chamber:**

a) Smoke emission

b) During compression, the high gas velocities into the pre-chamber cause high heat transfer coefficient that reduce the air temperature. This means that compression ratios in the range (18:1 to 24:1) have to use to ensure reliable ignition when starting.
e) The high heat transfer coefficient in swirl chamber can cause problems with injectors, if temperature rise above $14\,^\circ C$ carbonization of fuel can occur.

d) Complicated combustion chamber design, also there is a loss of (5% to 15%) of fuel economy than other types.

**Types of CI Engines**

**Direct injection:**
(Quiescent chamber)

**Direct injection:**
(Swirl in chamber)

**Indirect injection:**
(Turbulent and Swirl prechamber)
6.6 Cold Starting of Diesel Engines (D.E.):

Starting compression ignition engines from cold is a problem due to the following means:

a) Poor quantity fuel  
b) Low temperature  
c) Poorly seated valves  
d) Leakage past the piston ring  
e) Low starting speed

The following are various methods for starting the engine from cold:

A) Various aids for starting can be fitted to fuel injection system:

a) *Excess fuel injection*; this method is beneficial for the following reasons:

1) Its bulk raises the compression ratio  
2) Un-burnt fuel helps to seal the piston rings and valves  
3) Increases the probability of combustion starting  

b) *Late injection (Retarded injection timing)*; means that fuel is injected when the temperature and pressure are higher. It is better to have an
auxiliary nozzle in the injector in order to direct a spray of the fuel in
the air than impinges the fuel spray on the combustion chamber surface.
c) *Extra nozzles in the injector*

B) Introducing with air a volatile liquid which self-ignites readily

C) Heaters in combustion chamber; air in combustion chamber can be
heated electrically by:

   (a) Heater plugs, especially in divided combustion chambers,
   heater plugs are either exposed loops of thick resistance wire or finer multi-
turned wire insulated by a refractory material and then sheathed.

   (b) High voltage surface discharge plug

D) Variable compression ratio pistons; in these pistons the distance
   between the top of the piston (the crown) and the gudgeon pin (little
   end) can be varied hydraulically. Ignition in D.E. relies on both high
   temperature and a high pressure.

6.7 Fuel Metering:

The injection system of C.I. engine should fulfill the following objectives
consistently:

1) Meter the quantity of fuel demanded by the speed and load of the engine
2) Inject the fuel at correct time in the cycle
3) Distribute the metered fuel equally among the cylinders
4) Inject the fuel at correct rate
5) Inject the fuel with the spray pattern and atomization demanded by the
   design of the combustion chamber
6) Begin and end the injection sharply without dribbling or after injection.

*To accomplish these objectives, a number of functional elements are required:*

A) Pumping elements: to move the fuel from fuel tank to cylinders. The
   pumping element consists of:
   
   (a) Low pressure pump; to lift the fuel from fuel tank and provides a
   constant pressure of about (0.75 bar) to injector pump.
(b) High pressure pump (injection pump); fuel injection pressure in the range (200 to 1700 bars) are provided by this pump. This pressure depends on the engine size and the type of combustion chamber.

(c) Fuel filters and low-high piping

B) Metering and controls elements to measure and adjust the fuel flow rate demanded by engine speed and load

C) Distributing elements to divide the metered fuel equally among the cylinders

D) Timing controls to adjust the start and the stop of injection

E) Mixing elements to atomize and distribute the fuel within the combustion chamber.

6.8 Types of Injection Systems:

A) Air Injection – absolute due to size, cost and power required

B) Solid or Mechanical Injection; every solid injection system should have:
   a) Pressurizing unit (high pressure pump)
   b) Atomizing unit (injector and its nozzle)

The different types of solid injection system vary only in the manner of operation and control of these two basic elements. The main types of solid injection are:

1) Individual Pump or Jerk System: this could be:
   a) Separate pump and separate injector for each cylinder.

Disadvantages:

(i) Noisy with jerking.

(ii) Required heavy valve gears
Unit injector: the high-pressure pump and injector are combined together into one unit for each cylinder.

2) Common Rail System:

A high-pressure fuel pump delivers fuel to an accumulator where the pressure is kept constant with help of pressure regulating valve. A common rail or a pipe starts from the accumulator and leads to the different distributing elements for each cylinder.
For each cylinder, there is a separate metering and timing element, which is connected to an automatic injector.

*The main disadvantages of the common rail system* are that in case of injection needle sticking in an open position, an excess amount of fuel may be injected into cylinder.

3) Distributor System:

In this system the pump, which pressurizes the fuel, also meters and times it. The fuel pump after metering the required amount of fuel, supplies it to a rotary distributor at correct time for supply to each cylinder. The number of injection strokes per cycle of the pump is equal to the number of cylinders.
To Cylinder No. 1
To Cylinder No. 2
To Cylinder No. 3
To Cylinder No. 4

Distributor

Engine

High Pressure Fuel Pump

L. P. Fuel Line

Injectors

Distributor

$N(distr.)$
6.9 Injectors:

There are two main types of fuel injectors:

1) **Blast Injector:** this type was used with air injection system. When the fuel valve is opened the blast air sweeps along with it the fuel and well atomized fuel spray is sent to the combustion chamber.

2) **Mechanically Operated Injectors:** these types are classified according to the design of their nozzles. The type of nozzle to be used is selected according to:
   (i) Shape of the combustion chamber
   (ii) Operation condition of the diesel engine
   (iii) The design of the fuel injection system

**Types of injectors according to their nozzles:**

A) **Open Type Injectors:** these types are designed without shut-off valve between the delivering duct and the orifices of the spray nozzle. They are cheap, but less efficient.

**Note:** *(Characteristics of Fuel Nozzle Applied for all Nozzles) (Open or Closed):*

Bernoulli equation between (1) and (2), assume steady state condition:
\[ P_N - P_C = \frac{\rho_f w_f^2}{2} - \frac{\rho_f w_N^2}{2} = \Delta P_N \]

or:
\[ \Delta P_N = \frac{\rho_f}{2}(w_f^2 - w_N^2) \]

\[ \dot{V}_f = A_z\mu_f w_f \]

\[ w_N \ll w_f \Rightarrow w_N \approx 0 \]

Substitute equations (2) and (3) into equation (1), becomes:
\[ \Delta P_N = \frac{\rho_f}{2} \left( \frac{\dot{V}_f}{\mu_f A_z} \right)^2 \]

But:
\[ \dot{V}_f = f(N) = \text{const} \times N \]

\[ \therefore \Delta P_N = \frac{\rho_f}{2} \left( \frac{C \times N^2}{\mu_f^2 A_z^2} \right) \]

or:
\[ \Delta P_N = \text{Const.} \times N^2 \]

Where:
- \( w_f \) = Fuel velocity at orifice \( (m/s) \)
- \( w_N \) = Fuel velocity inside nozzle \( (m/s) \)
- \( A_z \) = Area of nozzle \( (m^2) \)
- \( P_N \) = Injection pressure (pressure inside nozzle) \( (N/m^2) \)
- \( P_C \) = Pressure of gas inside cylinder \( (N/m^2) \)
- \( \mu_f \) = Discharge coefficient (fuel discharge coefficient)

**B) Closed Type Injectors:** these types are provided with a spring loaded needle valve in their nozzle.

**Advantages of this type:**

i) Avoidance of pressure drops

ii) Control of injection pressure

*The following are the main types of nozzles used with closed injectors:*

a) **Single Hole Nozzle:** single-hole nozzles are used in open (undivided) combustion chamber. The size of the hole is usually larger than (0.2 mm). The
hole may be drilled centrally or at angle to the centerline of the nozzle. The disadvantages of the single-hole nozzles are:

1) Very high injection pressure is needed
2) Single-hole has a tendency to dribble
3) The spray angle is very narrow (usually about $15^\circ$) and this does not facilitate good mixing unless higher air velocities are provided

b) Multi-Hole Nozzle: in order to mix the fuel properly even with slow air movement available with many types of open combustion chambers a multi-hole nozzle is used. The numbers of holes varies from (4 to 18) and the size varies from (0.15 mm to 0.35 mm).
c) **Pintle Nozzle**: the Pintle nozzle has the form of two truncated cone with their smaller base together or cylindrical shape. The spray obtained by the Pintle nozzle is hollow conical spray and the cone angle can be varied from (0° to 60°) by giving different tapers to the Pintle. This type of nozzle is used with pre-combustion chamber, or high swirl chamber.

![Pintle Nozzle Diagram](image)

*The advantages of Pintle nozzle are:*

1) Good atomization with reducing penetration
2) The injection pressure is lower than that used for single or multi-hole nozzles
3) This type is avoid weak injection

![Rate of Injection vs Crank Angle Graph](image)

d) **Pintaux Nozzle (Pintle with Auxiliary Hole)**: to improve cold starting performance without determinate effect on efficiency a Pintaux nozzle is used (if the fuel is injected in a direction upstream the direction of air, the delay period is reduced due to increasing heat transfer between air and fuel). This is a development of Pintle nozzle and has auxiliary hole drilled in the nozzle body. This hole allows a small amount of pilot fuel injection in the upstream at a time.
slightly in advance of the main down-stream injection. At low speeds, the needle valve does not lift fully and most of the fuel is injected through the side hole resulting in very good cold starting performance.
Example: A four stroke, 6 cylinders, diesel engine has the following parameters: Bore = (100mm), Stroke = (120mm), Piston Speed = (8m/s), Clearance Volume = (0.3liter), Power Output = (373kW), Brake Specific Fuel Consumption = (90g/kW – h), Pressure at the beginning of compression = (1bar), Compression Index = (1.34), Fuel Velocity inside Injector = (0.9m/s), Injection Pressure \( P_N = 150kg/cm^2 \), \( \rho_f = 740kg/m^3 \). \( \mu_f = 0.9 \). Calculate:

A) Nozzle orifice area \( (A_z = ?) \)

B) Fuel velocity at nozzle orifice \( (w_f = ?) \)

C) Engine speed \( (N = ?) \)

Solution:

\[
V_D = \frac{\pi}{4} B^2 s_i = \frac{\pi}{4} (0.1)^2 * 0.12 * 6 = 5.655 * 10^{-3} \text{(m}^3\text{)}
\]

\[
r_C = \frac{V_T}{V_C} = \frac{5.655 * 10^{-3} + 0.3 * 10^{-3}}{0.3 * 10^{-3}} = 19.84
\]

\[
P_C = P_2 = P_1 * (r_C)^k = 1 * (19.84)^{1.34} = 5.48 * 10^6 \text{ (N/m}^2\text{)}
\]

1) \[
P_s = \frac{S * N}{30} \Rightarrow N = \frac{30 * 8}{0.12} = 2000 \text{(rpm)}
\]

\[
sfc = \frac{\dot{m_f}}{P_b} \Rightarrow \dot{m_f} = 90 * 10^{-3} * 373 = 33.556 \text{(kg/hr)}
\]

\[
\dot{V}_f = \frac{33.556}{3600 * 740} = 1.25963 * 10^{-5} \text{ (m}^3\text{/s)}
\]

\[
\Delta P_N = P_N - P_C = 150 * 10^{-4} * 9.81 - 5.48 * 10^6 = 9217613 \text{ (N/m)}
\]

\[
A_z = 8.867 * 10^{-8} \text{ (m}^2\text{)} = 0.0886 \text{ (mm}^2\text{)}
\]

2) \[
A_z = \frac{8.867 * 10^{-8}}{6} = 0.014 \text{ (mm}^2\text{)} \Rightarrow \text{(For one cylinder)}
\]

3) \[
\Delta P_N = \frac{\rho_f}{2} (w_f^2 - w_N^2)
\]

9217613 = \frac{740}{2} * (w_f^2 - 9^2)

\[
w_f = 157 \text{ (m/s)}
\]
Combustion may be defined as a relatively rapid chemical combination of
hydrogen and carbon in fuel with oxygen in air resulting in liberation of energy
in the form of heat. Following conditions are necessary for combustion to take
place:
1. The presence of combustible mixture
2. Some means to initiate mixture
3. Stabilization and propagation of flame in Combustion Chamber

In S I Engines, carburetor supplies a combustible mixture of petrol and air
and spark plug initiates combustion.

**Ignition Limits**

Ignition of charge is only possible within certain limits of fuel-air ratio. Ignition limits correspond approximately to those mixture ratios, at lean and rich
ends of scale, where heat released by spark is no longer sufficient to initiate
combustion in neighboring unburnt mixture. For hydrocarbons fuel the
stoichiometric fuel air ratio is 1:15 and hence the fuel air ratio must be about
1:30 and 1:7.

![Ignition Limits Diagram](image-url)

**Theories of Combustion in SI Engine**

Combustion in SI engine may roughly is divided into two general types:
Normal and Abnormal (knock free or knocking). Theoretical diagram of
pressure crank angle diagram is shown in figure below. (a → b) is compression
process, (b → c) is combustion process and (c → d) is an expansion process. In
an ideal cycle it can be seen from the diagram, the entire pressure rise during
combustion takes place at constant volume i.e., at TDC. However, in actual cycle this does not happen.

Richard Theory of Combustion:
Sir Ricardo, known as father of engine research describes the combustion process can be imagined as if it is developing in two stages:
1. Growth and development of a self-propagating nucleus flame. (Ignition lag)
2. Spread of flame through the combustion chamber

Three Stage of Combustion:
According to Ricardo, There are three stages of combustion in SI Engine as shown
1. Ignition lag stage
2. Flame propagation stage
3. After burning stage

1. Ignition lag stage:
There is a certain time interval between instant of spark and instant where there is a noticeable rise in pressure due to combustion. This time lag is called IGNITION LAG.
Ignition lag is the time interval in the process of chemical reaction during which molecules get heated up to self ignition temperature, get ignited and produce a self propagating nucleus of flame. The ignition lag is generally expressed in terms of crank angle ($\theta_i$). The period of ignition lag is shown by path (a-b).
Ignition lag is very small and lies between 0.00015 to 0.0002 seconds. An ignition lag of 0.002 seconds corresponds to 35 deg crank rotation when the engine is running at 3000 RPM. Angle of advance increase with the speed. This is a chemical process depending upon the nature of fuel, temperature and pressure, proportions of exhaust gas and rate of oxidation or burning.

2. Flame propagation stage:
   Once the flame is formed at “b”, it should be self sustained and must be able to propagate through the mixture. This is possible when the rate of heat generation by burning is greater than heat lost by flame to surrounding. After the point “b”, the flame propagation is abnormally low at the beginning as heat lost is more than heat generated. Therefore pressure rise is also slow as mass of mixture burned is small. Therefore, it is necessary to provide angle of advance (30-35) degrees, if the peak pressure to be attained (5-10) degrees after TDC. The time required for crank to rotate through an angle ($\theta_2$) is known as combustion period during which propagation of flame takes place.

3. After burning:
   Combustion will not stop at point “c” but continue after attaining peak pressure and this combustion is known as after burning. This generally happens when the rich mixture is supplied to engine.
Factors Affecting the Flame Propagation:

Rate of flame propagation affects the combustion process in SI engines. Higher flame propagation velocities can achieve higher combustion efficiency and fuel economy. Unfortunately flame velocities for most of fuel range between \([10-30]\) (m/second).

The factors that affect the flame propagations are:

1. Air fuel ratio
2. Compression ratio
3. Load on engine
4. Turbulence and engine speed
5. Other factors

1. **A/F ratio**: The mixture strength influences the rate of combustion and amount of heat generated. The maximum flame speed for all hydrocarbon fuels occurs at nearly 10% rich mixture. Flame speed is reduced both for lean and as well as for very rich mixture. Lean mixture releases less heat resulting lower flame temperature and lower flame speed. Very rich mixture results incomplete combustion \((C\alpha)\) instead of \(C\alpha\), and also results in production of less heat and flame speed remains low. The effects of \((A/F)\) ratio on P-v diagram and P-\(\alpha\) diagram are shown below, (where \(\alpha\) is crank angle):
2. **Compression ratio**: The higher compression ratio increases the pressure and temperature of the mixture and also decreases the concentration of residual gases. All these factors reduce the ignition lag and help to speed up the second phase of combustion. The maximum pressure of the cycle as well as mean effective pressure of the cycle with increase in compression ratio. Figure below shows the effect of compression ratio on pressure (indirectly on the speed of combustion) with respect to crank angle for same (A/F) ratio and same angle of advance. Higher compression ratio increases the surface to volume ratio and thereby increases the part of the mixture which after-burns in the third phase.

![Compression ratio diagram](image_url)

3. **Load on Engine**: With increase in load, the cycle pressures increase and the flame speed also increases. In S.I. engine, the power developed by an engine is controlled by throttling. At lower load and higher throttle, the initial and final pressure of the mixture after compression decrease and mixture is also diluted by the more residual gases. This reduces the flame propagation and prolongs the ignition lag. This is the reason, the advance mechanism is also provided with change in load on the engine. This difficulty can be partly overcome by providing rich mixture at part loads but this definitely increases the chances of afterburning. The after burning
is prolonged with richer mixture. In fact, poor combustion at part loads and necessity of providing richer mixture are the main disadvantages of S.I engines which causes wastage of fuel and discharge of large amount of CO with exhaust gases.

4. **Turbulence**: Turbulence plays very important role in combustion of fuel as the flame speed is directly proportional to the turbulence of the mixture. This is because, the turbulence increases the mixing and heat transfer coefficient or heat transfer rate between the burned and unburned mixture. The turbulence of the mixture can be increased at the end of compression by suitable design of the combustion chamber (geometry of cylinder head and piston crown). Insufficient turbulence provides low flame velocity and incomplete combustion and reduces the power output. But excessive turbulence is also not desirable as it increases the combustion rapidly and leads to detonation. Excessive turbulence causes to cool the flame generated and flame propagation is reduced. Moderate turbulence is always desirable as it accelerates the chemical reaction, reduces ignition lag, increases flame propagation and even allows weak mixture to burn efficiently.

**A) Engine Speed:**

The turbulence of the mixture increases with an increase in engine speed. For this reason, the flame speed almost increases linearly with engine speed. If the engine speed is doubled, flame to traverse the combustion chamber is halved. Double the original speed and half the original time give the same number of crank degrees for flame propagation. The crank angle required for the flame propagation, which is main phase of combustion will remain almost constant at all speeds. This is an important characteristics of all petrol engines.
B) Engine Size:

Engines of similar design generally run at the same piston speed. This is achieved by using small engines having larger RPM and larger engines having smaller RPM. Due to same piston speed, the inlet velocity, degree of turbulence and flame speed are nearly same in similar engines regardless of the size. However, in small engines the flame travel is small and in large engines large. Therefore, if the engine size is doubled the time required for propagation of flame through combustion space is also doubled. But with lower RPM of large engines the time for flame propagation in terms of crank would be nearly same as in small engines. In other words, the number of crank degrees required for flame travel will be about the same irrespective of engine size provided the engines are similar.

5. Other Factors: Among the other factors, the factors that increase the flame speed are supercharging of the engine, spark timing and residual gases left in the engine at the end of exhaust stroke. The air humidity also affects the flame velocity but its exact effect is not known. Anyhow, its effect is not large compared with (A/F) ratio and turbulence.

A) PHENOMENON OF KNOCKING IN SI ENGINE: Knocking is due to auto ignition of end portion of unburned charge in combustion chamber. As the normal flame proceeds across the chamber, pressure and temperature of unburned charge increase due to compression by burned portion of charge. This unburned compressed charge may auto ignite under certain temperature condition and release the energy at a very rapid rate compared to normal combustion process in cylinder. This rapid release of energy during auto ignition causes a high-pressure differential in combustion chamber and a high-pressure wave is released from auto ignition region. The motion of high-pressure compression waves inside the cylinder causes vibration of engine parts and pinging noise and it is
known as knocking or detonation. This pressure frequency or vibration frequency in SI engine can be up to 5000 Cycles per second.

Denotation is undesirable as it affects the engine performance and life, as it abruptly increases sudden large amount of heat energy. It also put a limit on compression ratio at which engine can be operated which directly affects the engine efficiency and output.
**B) AUTO IGINITION**: A mixture of fuel and air can react spontaneously and produce heat by chemical reaction in the absence of flame to initiate the combustion or self-ignition. This type of self-ignition in the absence of flame is known as Auto-Ignition. The temperature at which the self-ignition takes place is known as self-igniting temperature. The pressure and temperature abruptly increase due to auto-ignition because of sudden release of chemical energy. This auto-ignition leads to abnormal combustion known as detonation which is undesirable because its bad effect on the engine performance and life as it abruptly increases sudden large amount of heat energy. In addition to this knocking puts a limit on the compression ratio at which an engine can be operated which directly affects the engine efficiency and output. Auto-ignition of the mixture does not occur instantaneously as soon as its temperature rises above the self-ignition temperature. Auto-ignition occurs only when the mixture stays at a temperature equal to or higher than the self-ignition temperature for a “finite time”. This time is known as delay period or reaction time for auto-ignition. This delay time as a function of compression ratio is shown in adjacent figure. As the compression ratio increases, the delay period decreases and this is because of increase in initial (before combustion) pressure and temperature of the charge. The self-ignition temperature is a characteristic of fuel air mixture and it varies from fuel to fuel and mixture strength to mixture - strength of the same fuel.
C) **PRE-IGNITION:** Pre-ignition is the ignition of the homogeneous mixture of charge as it comes in contact with hot surfaces, in the absence of spark. Auto ignition may overheat the spark plug and exhaust valve and it remains so hot that its temperature is sufficient to ignite the charge in next cycle during the compression stroke before spark occurs and this causes the pre-ignition of the charge. Pre-ignition is initiated by some overheated projecting part such as the sparking plug electrodes, exhaust valve head, metal corners in the combustion chamber, carbon deposits or protruding cylinder head gasket rim etc. Pre-ignition is also caused by persistent detonating pressure shockwaves scoring away the stagnant gases which normally protect the combustion chamber walls. The resulting increased heat flow through the walls, raises the surface temperature of any protruding poorly cooled part of the chamber, and this therefore provides a focal point for pre-ignition.
**Effects of Pre-ignition**

- It increases the tendency of denotation in the engine
- It increases heat transfer to cylinder walls because high temperature gas remains in contact with for a longer time
- Pre-ignition in a single cylinder will reduce the speed and power output
- Pre-ignition may cause seizure in the multi-cylinder engines, only if only cylinders have pre-ignition

**Effects of Detonation**

The harmful effects of detonation are as follows:

1. *Noise and Roughness*: Knocking produces a loud pulsating noise and pressure waves. These waves which vibrates back and forth across the cylinder. The presence of vibratory motion causes crankshaft vibrations and the engine runs rough.

2. *Mechanical Damage*:
   
   (a) High pressure waves generated during knocking can increase rate of wear of parts of combustion chamber. Sever erosion of piston crown (in a manner similar to that of marine propeller blades by capitation), cylinder head and pitting of inlet and outlet valves may result in complete wreckage of the engine.

   (b) Detonation is very dangerous in engines having high noise level. In small engines the knocking noise is easily detected and the corrective measures can be taken but in aero-engines it is difficult to detect knocking noise and hence corrective measures cannot be taken. Hence severe detonation may persist for a long time which may ultimately result in complete wreckage of the piston.


4. *Increase in heat transfer*: Knocking is accompanied by an increase in the rate of heat transfer to the combustion chamber walls.

*The increase in heat transfer is due to two reasons*:

- The minor reason is that the maximum temperature in a detonating engine is about 150°C higher than in a non-detonating engine, due to rapid completion of combustion
• The major reason for increased heat transfer is the scouring away of protective layer of inactive stagnant gas on the cylinder walls due to pressure waves. The inactive layer of gas normally reduces the heat transfer by protecting the combustion and piston crown from direct contact with flame.

5. Decrease in power output and efficiency. Due to increase in the rate of a detonating engine decreases.

6. Pre-ignition: The increase in the rate of heat transfer to the walls has yet another effect. It may cause local overheating, especially of the sparking plug, which may reach a temperature high enough to ignite the charge before the passage of spark, thus causing pre-ignition. An engine detonating for a long period would most probably lead to pre-ignition and this is the real danger of detonation.
EFFECT OF ENGINE OPERATING VARIABLES ON THE ENGINE KNOCKING DETONATION

The various engine variables affecting knocking can be classified as:

- Temperature factors
- Density factors
- Time factors
- Composition factors

(A) TEMPERATURE FACTORS

Increasing the temperature of the unburned mixture increase the possibility of knock in the SI engine. We shall now discuss the effect of following engine parameters on the temperature of the unburned mixture:

- RAISING THE COMPRESSION RATIO: Increasing the compression ratio increases both the temperature and pressure (density of the unburned mixture). Increase in temperature reduces the delay period of the end gas, which in turn increases the tendency to knock.

- SUPERCHARGING: It also increases both temperature and density, which increase the knocking tendency of engine

- COOLANT TEMPERATURE: Delay period decreases with increase of coolant temperature, decreased delay period increase the tendency to knock

- TEMPERATURE OF THE CYLINDER AND COMBUSTION CHAMBER WALLS: The temperature of the end gas depends on the design of combustion chamber. Sparking plug and exhaust valve are two hottest parts in the combustion chamber and uneven temperature leads to pre-ignition and hence the knocking.

(B) DENSITY FACTORS

Increasing the density of unburnt mixture will increase the possibility of knock in the engine. The engine parameters that affect the density are as follows:

- Increased compression ratio increase the density

- Increasing the load opens the throttle valve more and thus the density
Supercharging increases the density of the mixture.

Increasing the inlet pressure increases the overall pressure during the cycle. The high-pressure end gas decreases the delay period, which increases the tendency of knocking.

Advanced spark timing: quantity of fuel burnt per cycle before and after TDC, position depends on spark timing. The temperature of charge increases by increasing the spark advance and it increases with rate of burning and does not allow sufficient time to the end mixture to dissipate the heat and increase the knocking tendency.

(C) TIME FACTORS

Increasing the time of exposure of the unburned mixture to auto-ignition conditions increase the possibility of knock in SI engines.

Flame travel distance: If the distance of flame travel is more, then possibility of knocking is also more. This problem can be solved by combustion chamber design, spark plug location and engine size. Compact combustion chamber will have better anti-knock characteristics, since the flame travel and combustion time will be shorter. Further, if the combustion chamber is highly turbulent, the combustion rate is high and consequently combustion time is further reduced; this further reduces the tendency to knock.

Location of spark plug: A spark plug that is centrally located in the combustion chamber has minimum tendency to knock, as the flame travel is minimum. The flame travel can be reduced by using two or more spark plugs.

Location of exhaust valve: The exhaust valve should be located close to the spark plug so that it is not in the end gas region; otherwise, there will be a tendency to knock.

Engine size: Large engines have a greater knocking tendency because flame requires a longer time to travel across the combustion chamber. In SI engine therefore, generally limited to 100mm
Turbulence of mixture decreasing the turbulence of the mixture decreases the flame speed and hence increases the tendency to knock. Turbulence depends on the design of combustion chamber and one engine speed.

(D) COMPOSITION

The properties of fuel and A/F ratio are primary means to control knock:

(a) Molecular Structure: The knocking tendency is markedly affected by the type of the fuel used. Petroleum fuels usually consist of many hydrocarbons of different molecular structure. The structure of the fuel molecule has enormous effect on knocking tendency. Increasing the carbon-chain increases the knocking tendency and centralizing the carbon atoms decreases the knocking tendency. Unsaturated hydrocarbons have less knocking tendency than saturated hydrocarbons.

Paraffins

- Increasing the length of carbon chain increases the knocking tendency.
- Centralizing the carbon atoms decreases the knocking tendency.
- Adding methyl group (CH to the side of the carbon chain in the centre position) decreases the knocking tendency.

Olefins

Introduction of one double bond has little effect on anti-knock quality but two or three double bond results less knocking tendency except C and C

Napthenes and Aromatics

- Napthenes have greater knocking tendency than corresponding aromatics.
- With increasing double-bonds, the knocking tendency is reduced.
- Lengthening the side chains increases the knocking tendency whereas branching of the side chain decreases the knocking tendency.

(b) Fuel-air ratio: The most important effect of fuel-air ratio is on the reaction time or
ignition delay. When the mixture is nearly 10% richer than stoichiometric (fuel-air ratio = 0.08) ignition lag of the end gas is minimum and the velocity of flame propagation is maximum. By making the mixture leaner or richer (than F/A 0.08) the tendency to knock is decreased. A too rich mixture is especially effective in decreasing or eliminating the knock due to longer delay and lower temperature of compression.

(c) **Humidity of air**: Increasing atmospheric humidity decreases the tendency to knock by decreasing the reaction time of the fuel.

**Effect of engine variables on Knocking in SI engine**

1. Compression ratio: The pressure and temperature at the end of compression increases with increase in compression ratio. This in turn increases the maximum pressure during the combustion and creates a tendency to knock.
2. Supercharging: increase the temperature and density of mixture and thus the tendency to knock is increased.
3. Turbulence: decreasing the turbulence of mixture decreases the flame speed and hence increase the tendency to knock.
4. Octane rating of fuel: higher the octane number, less the tendency to knock. Parafins have maximum tendency to knock and aromatic series have minimum tendency to knock.

**Factors that limits the compression ratio in petrol engine**

In petrol engine we use the mixture of air and petrol and thermal efficiency of petrol engine increase with increase in compression ratio. However, the value of compression ration is limited by phenomenon of knocking. The pressure and temperature at the end of compression increases with increase in compression ratio. This in turn increases the maximum pressure during the combustion and creates a tendency to knock. Thus Higher compression ratio, higher is the tendency to knock, therefore the value of compression ratio in petrol engine is limited to 6 to 10. Compression ratio can be marginally improved by using fuel with Tetra-ethyl lead. TEL delays the auto ignition allows it to occur at higher temperature and thus reduces knocking. The
use of TEL is now in disfavor because of atmospheric pollution (lead is toxic and has serious environmental and health hazards).

**KNOCK RATING OF SI ENGINE FUELS (OCTANE NUMBER)**

The tendency to detonate depends on composition of fuel. Fuel differ widely in their ability to resist knock. The property of fuel that describes how fuel will or will nor self ignite is called the OCTANE NUMBER. It is defined as the percentage of Isooctane by volume in a mixture of Isooctane and n-heptane, which exactly matches the knocking tendency of a given fuel, in a standard fuel under given standard operating conditions. The rating of a particular SI fuel is done by comparing its antiknock performance with that of standard reference fuel that is usually combination of Isooctane and n-heptane. Isooctane (C8H18) which has a very high resistance to knock and therefore it is arbitrarily assigned a rating of (100 octane number). N-heptane (C7H16) which is very prone to knock and therefore given a zero value.

For example: Octane number 80 means that the fuel has same knocking tendency as mixture of 80% isoctane and 20% n-heptane (by volume basis).

A fuel having an octane number of 110 means fuel has the same tendency to resist as a mixture of 10 cc of Tetra ethyl lead (TEL) in one U.S gallon of Isooctane.
CHAPTER EIGHT
Hybrid Electrical Vehicles

Introduction

A hybrid electric vehicle (HEV) has two types of energy storage units, electricity and fuel. Electricity means that a battery (sometimes assisted by ultra-caps) is used to store the energy, and that an electromotor (from now on called motor) will be used as traction motor.

Fuel means that a tank is required, and that an Internal Combustion Engine (ICE, from now on called engine) is used to generate mechanical power, or that a fuel cell will be used to convert fuel to electrical energy. In the latter case, traction will be performed by the electromotor only. In the first case, the vehicle will have both an engine and a motor.

- **Motors** are the "work horses" of Hybrid Electric Vehicle drive systems. The electric traction motor drives the wheels of the vehicle.

  A *main advantage of an electromotor (Motor)* is the possibility to function as generator. In all HEV systems, mechanical braking energy is regenerated. The maximum operational braking torque is less than the maximum traction torque; there is always a mechanical braking system integrated in a car.

- **The battery pack** in a HEV has a much higher voltage than the SIL automotive 12 Volts battery, in order to reduce the currents and the I2R losses.

- **Accessories** such as power steering and air conditioning are powered by electric motors instead of being attached to the combustion engine. This allows efficiency gains as the accessories can run at a constant speed or can be switched off, regardless of how fast the combustion engine is running. Especially in long haul trucks, electrical power steering saves a lot of energy.
1. **Types by drive-train structure**

1.1. **Series hybrid**

In a series hybrid system, the combustion engine drives an electric generator (usually a three-phase alternator plus rectifier) instead of directly driving the wheels. The electric motor is the only means of providing power to the wheels. The generator both charges a battery and powers an electric motor that moves the vehicle. When large amounts of power are required, the motor draws electricity from both the batteries and the generator.

![Diagram of series hybrid system](image)

**Advantages of series hybrid vehicles:**

1. There is no mechanical link between the combustion engine and the wheels. The engine-generator group can be located everywhere.

2. There are no conventional mechanical transmission elements (gearbox, transmission shafts). Separate electric wheel motors can be implemented easily.

3. The combustion engine can operate in a narrow rpm range (its most efficient range), even as the car changes speed.

4. Series hybrids are relatively the most efficient during stop-and-go city driving.
1.2. Parallel hybrid

Parallel hybrid systems have both an internal combustion engine (ICE) and an electric motor in parallel connected to a mechanical transmission.

**Operation Modes:**

The parallel configuration supports diverse operating modes:

(a) Electric power only: Up to speeds of usually 40 km/h, the electric motor works with only the energy of the batteries, which are not recharged by the ICE. This is the usual way of operating around the city, as well as in reverse gear, since during reverse gear the speed is limited.

(b) ICE power only: At speeds superior to 40 km/h, only the heat engine operates. This is the normal operating way at the road.

(b) ICE + electric power: if more energy is needed (during acceleration or at high speed), the electric motor starts working in parallel to the heat engine, achieving greater power.

(c) ICE + battery charging: if less power is required, excess of energy is used to charge the batteries. Operating the engine at higher torque than necessary, it runs at a higher efficiency.

(d) Regenerative breaking: While braking or decelerating,
Advantages of parallel hybrid vehicles:

1. Total efficiency is higher during cruising and long-distance highway driving.

2. Large flexibility to switch between electric and ICE power

3. Compared to series hybrids, the electromotor can be designed less powerful than the ICE, as it is assisting traction. Only one electrical motor/generator is required.

1.3. Combined hybrid

Combined hybrid systems have features of both series and parallel hybrids. There is a double connection between the engine and the drive axle: mechanical and electrical. This split power path allows interconnecting mechanical and electrical power, at some cost in complexity.

Power-split devices are incorporated in the power-train. The power to the wheels can be either mechanical or electrical or both. This is also the case in parallel hybrids. But the main principle behind the combined system is the decoupling of the power supplied by the engine from the power demanded by the driver.
Advantages of combined hybrid vehicles:

1. Maximum flexibility to switch between electric and ICE power
2. Decoupling of the power supplied by the engine from the power demanded by the driver allows for a smaller, lighter, and more efficient ICE design.

2. Types by degree of hybridization

Parallel and combined hybrids can be categorized depending upon how balanced the different portions are at providing motive power. In some cases, the combustion engine is the dominant portion; the electric motor turns on only when a boost is needed. Others can run with just the electric system operating.

2.1. Strong hybrid ( = full hybrid): A full hybrid EV can run on just the engine, just the batteries, or a combination of both. A large, high-capacity battery pack is needed for battery-only operation.

2.2. Medium hybrid ( = motor assist hybrid): Motor assist hybrids use the engine for primary power, with a torque-boosting electric motor connected in parallel to a largely conventional power-train. EV mode is only possible for a very limited period of time, and this is not a standard mode. Compared to full hybrids, the amount of electrical
power needed is smaller, thus the size of the battery system can be reduced.

2.3. **Mild hybrid/micro hybrid (= start/stop systems with energy recuperation):** Mild hybrids are essentially conventional vehicles with oversized starter motors, allowing the engine to be turned off whenever the car is coasting, braking, or stopped, yet restart quickly and cleanly.

2.4. **Plug-in hybrid (= grid connected hybrid = vehicle to grid V2G):** A plug-in hybrid electric vehicle (PHEV) is a full hybrid, able to run in electric-only mode, with larger batteries and the ability to recharge from the electric power grid. Their main benefit is that they can be gasoline-independent for daily commuting, but also have the extended range of a hybrid for long trips.

3. **Types by nature of the power source**

3.1. **Electric-internal combustion engine hybrid:** There are many ways to create an electric-internal combustion hybrid. The variety of electric-ICE designs can be differentiated by how the electric and combustion portions of the power-train connect, at what times each portion is in operation, and what percent of the power is provided by each hybrid component.

3.2. **Fuel cell hybrid:** Fuel cell vehicles have a series hybrid configuration. They are often fitted with a battery or super capacitor to deliver peak acceleration power and to reduce the size and power constraints on the fuel cell.

3.3. **Human power and environmental power hybrids:** Many land and water vehicles use human power combined with a further power source. Common are parallel hybrids, e.g. a boat being rowed and also having a sail set, or motorized bicycles. In addition, some series hybrids exist. Such vehicles can be tribrid vehicles, combining at the
same time three power sources e.g. from on-board solar cells, from grid-charged batteries, and from pedals.

3.4. **Pneumatic hybrid:** Compressed air can also power a hybrid car with a gasoline compressor to provide the power.

3.5. **Hydraulic hybrid:** A hydraulic hybrid vehicle uses hydraulic and mechanical components instead of electrical ones. A variable displacement pump replaces the motor/generator, and a hydraulic accumulator (which stores energy as highly compressed nitrogen gas) replaces the batteries. The hydraulic accumulator, which is essentially a pressure tank, is potentially cheaper and more durable than batteries.