TECHNICAL UNIVERSITY OF BUDAPEST
Faculty of Mechanical Engineering

Internal Combustion Engines

(Heat Engines II.)

Lecture note for the undergraduate course
7th Semester

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CLASSIFICATION OF INTERNAL COMBUSTION ENGINES

Principle of operation

- four stroke engine or
- two stroke engine
- other

Charging system

- Naturally aspirated
- Mechanically charged
- Turbo charged

Fuel type

- gas fuels
    - -natural gas
    - -gasification (pirolysis)
    - -biogas, waste gas
    - -other
- liquid fuels
    - crude oil fractions (distillation fuels)
        - Crude-oil (Diesel fuel)
        - Benzine
        - Kerosene (JET-A)
        - Heavy (ends) oils, etc
    - Renewable fuels
        - Rape-, sunflower-seed oil, RME
        - Alcohols, Bioethanole, etc
    - Other

Air-fuel mixing methods

- Internal (CIE, GDI (SIE))
- External (SIE)

Control methods

- qualitative (SIE)
- quantitative (CIE, GDI (SIE))

Combustion chamber design

- single open combustion chamber
- divided combustion chamber
    - swirl chamber systems
    - prechamber systems

Start of Combustion

- External energy (Spark)
- Compression
- Hot Spot

Basic principals of mechanical construction

Arrangement of cylinders

- in line arrangement
- V arrangement
- opposed cylinder engine
- radial type engine

Fluid inlet-outlet control

- side valve (SV) arrangement
- overhead valve (OHV) arrangement
- overhead camshaft (OHC) arrangement

cooling system

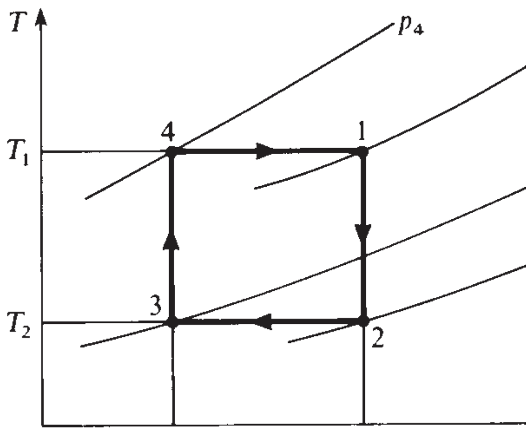
- air cooling
- water cooling

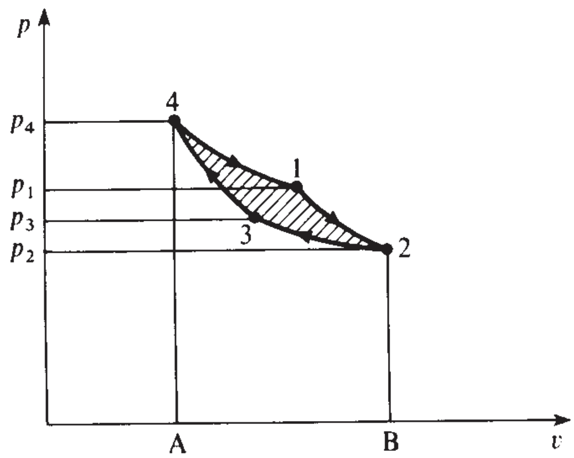
Ideal cycles.

To produce mechanical power from heat power, a cycle process is needed.

Carnot cycle would be ideal, but there is no machine which is working according to the Carnot cycle.

At the Carnot cycle:





Where:

Process 1 to 2 - isentropic expansion

Process 2 to 3 is isothermal heat rejection

Process 3 to 4 is isentropic compression

Process 4 to 1 is isothermal heat supply

Efficiency of the cycle:

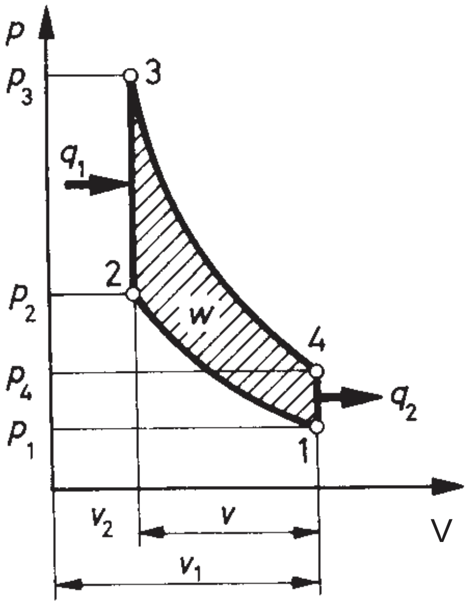
$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{(T_1 - T_2)\cdot(s_B - s_A)}{T_1 \cdot (s_B - s_A)} = 1 - \frac{T_2}{T_1}$$

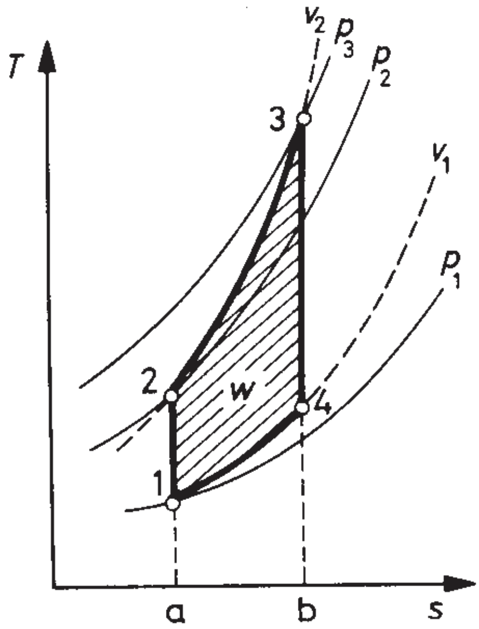
There are other types of cycles and machines which are connected with each other. These are called Reciprocating Internal Combustion Engines.

Otto cycle.

This cycle is named after Nicholaus August Otto, who invented his first engine in 1876.

The ideal air standard Otto cycle:





Where:

Process 1 to 2 is isentropic compression

Process 2 to 3 is reversible constant volume heating

Process 3 to 4 is isentropic expansion

Process 4 to 1 is reversible constant volume cooling

Efficiency of the cycle:

$$\eta = \frac{-\sum \boldsymbol{W}}{\boldsymbol{Q}_1} = \frac{\sum \boldsymbol{Q}}{\boldsymbol{Q}_1} = \frac{c_v(\boldsymbol{T}_3 - \boldsymbol{T}_2) - c_v(\boldsymbol{T}_4 - \boldsymbol{T}_1)}{c_v(\boldsymbol{T}_3 - \boldsymbol{T}_2)} = 1 - \frac{\boldsymbol{T}_4 - \boldsymbol{T}_1}{\boldsymbol{T}_3 - \boldsymbol{T}_2} = 1 - \frac{1}{\boldsymbol{\varepsilon}^{\kappa - 1}}$$

where: $\boldsymbol{\varepsilon} = \dfrac{\boldsymbol{V}_1}{\boldsymbol{V}_2} = \dfrac{\text{swept volume(V)} + \text{clearance volume}(\text{V}_2)}{\text{clearance volume}(\text{V}_2)}$ - the compression ratio

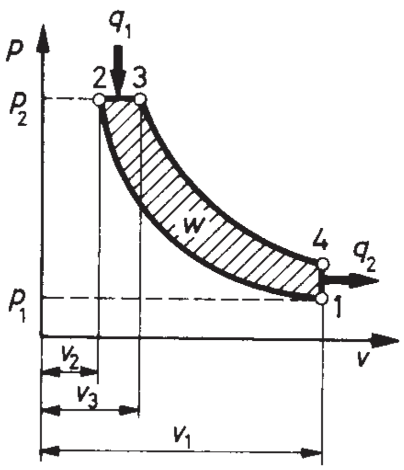
and since processes 1-2 and 3-4 are isentropic:

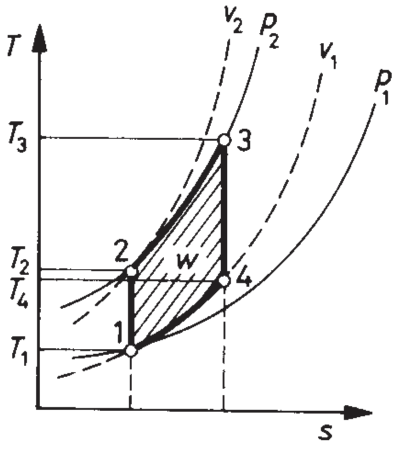
$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\kappa-1} = \left(\frac{v_4}{v_3}\right)^{\kappa-1} = \frac{T_3}{T_4} = \varepsilon^{\kappa-1}$$

The other type of cycle and machine called Diesel.

The original engine was invented by Rudolf Diesel in 1892.

The Ideal air standard Diesel cycle:





Where:

Process 1 to 2 is isentropic compression

Process 2 to 3 is reversible constant pressure heating

Process 3 to 4 is isentropic expansion

Process 4 to 1 is reversible constant volume cooling

Efficiency of the cycle:

$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^{\kappa} - 1}{\kappa \cdot (\rho - 1)}$$

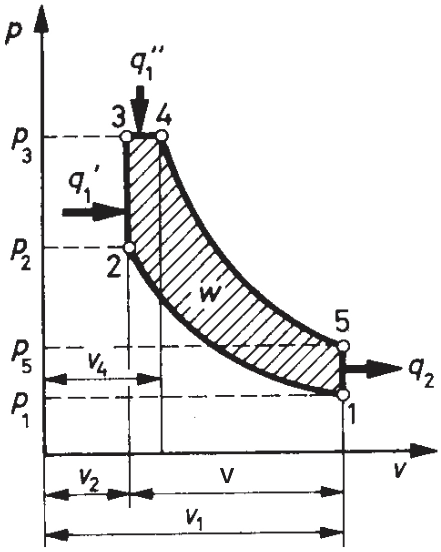
where: $\boldsymbol{\varepsilon} = \frac{\boldsymbol{V}_1}{\boldsymbol{V}_2} = \frac{\text{swept volume(V) + clearance volume}(\text{V}_2)}{\text{clearance volume}(\text{V}_2)}$ - the compression ratio

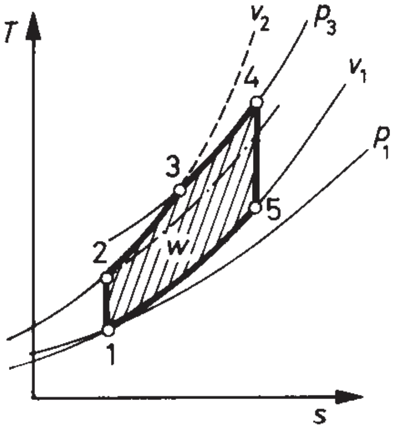
swept volume: $\boldsymbol{V} = \boldsymbol{V}_1 - \boldsymbol{V}_2$

$\rho = \frac{\boldsymbol{v}_3}{\boldsymbol{v}_2}$ - cut off ratio

The Dual-combustion cycle:

In this process heating is divided in two parts, a constant volume, and a constant pressure part:





Where:

Process 1 to 2 is isentropic compression

Process 2 to 3 is reversible constant volume heating

Process 3 to 4 is reversible constant pressure heating

Process 4 to 5 is isentropic expansion

Process 5 to 1 is reversible constant volume cooling

Efficiency of the cycle:

$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_v(T_3 - T_2) + c_v(T_4 - T_3) - c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_v(T_4 - T_3)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^{\kappa} \cdot \lambda - 1}{(\lambda - 1) + \kappa \cdot \lambda \cdot (\rho - 1)}$$

where: $\varepsilon = \frac{V_1}{V_2} = \frac{\text{swept volume(V)} + \text{clearance volume}(V_2)}{\text{clearance volume}(V_2)}$ - the compression ratio

$\lambda = \frac{p_3}{p_2}$ - pressure ratio

$\rho = \frac{v_4}{v_3}$ - cut off ratio

Using this type of cycle, choosing appropriate ratios, you can study closer to the real processes both Otto and Diesel engines.

<u>Classification of Internal Combustion (IC) engines</u>

There are several methods of classification of IC engines:

One of the most important is the principle of operation.

It can be four stroke engine or
two stroke engine
(one stroke is half revolution of the crankshaft)

The four stroke engine operation:

(one cycle is two crankshaft revolution)

Strokes:

I. Induction: piston is moving down, inlet valve is open, and air, or air and fuel mixture flow into the cylinder

II. Compression: piston is moving up, both valve are closed fluid inside the cylinder is being comprimised

At TDC: Ignition and after it combustion.

III. Expansion: piston is moving down both valve are closed

IV. Exhaust: piston is moving up and exhaust valve is open flue gas flow out

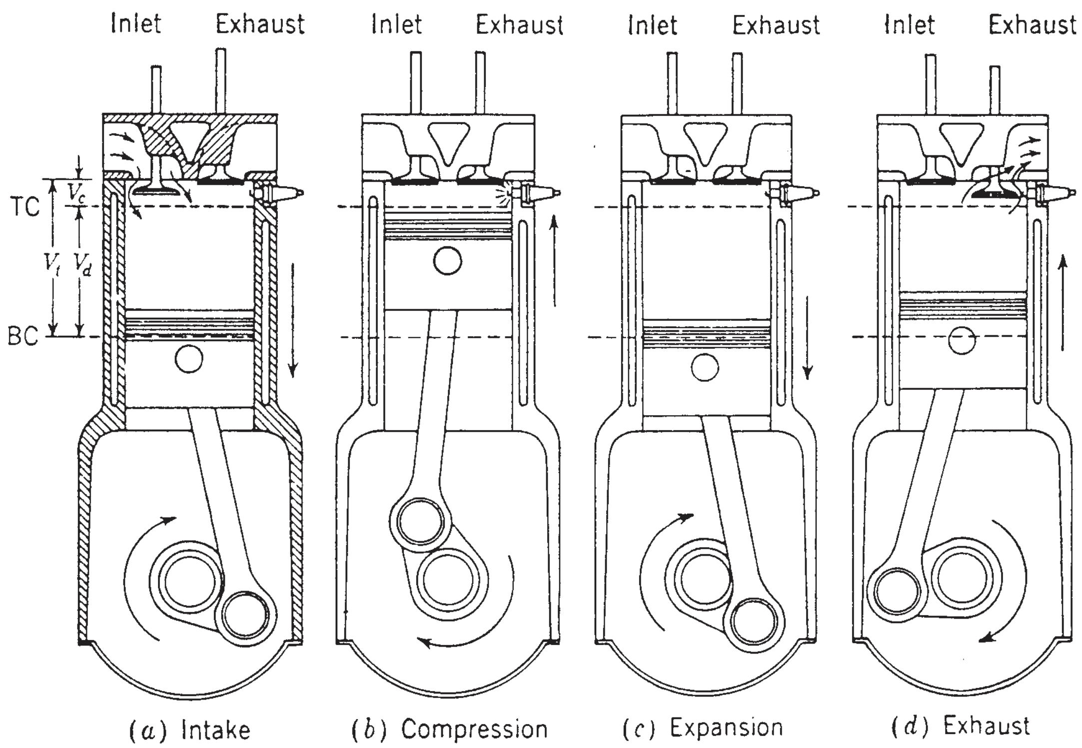


Fig. 1.

The two stroke cycle:

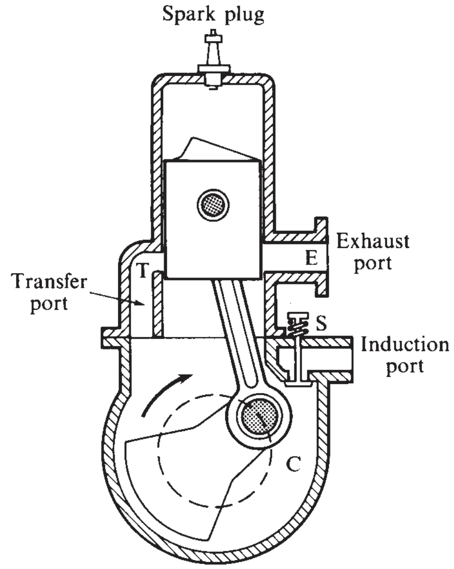


Fig. 2.

Strokes:

I. Induction and compression: piston is moving up compressing the fluid above it, and into the crankhouse new fluid is inducting

At TDC: Ignition and after it combustion.

II. Expansion and precompression:

piston is moving down, and precomprimising the fluid in the crankhouse

At BDC: Precomprimised fresh fuel is streaming into the cylinder and flue gas flow out

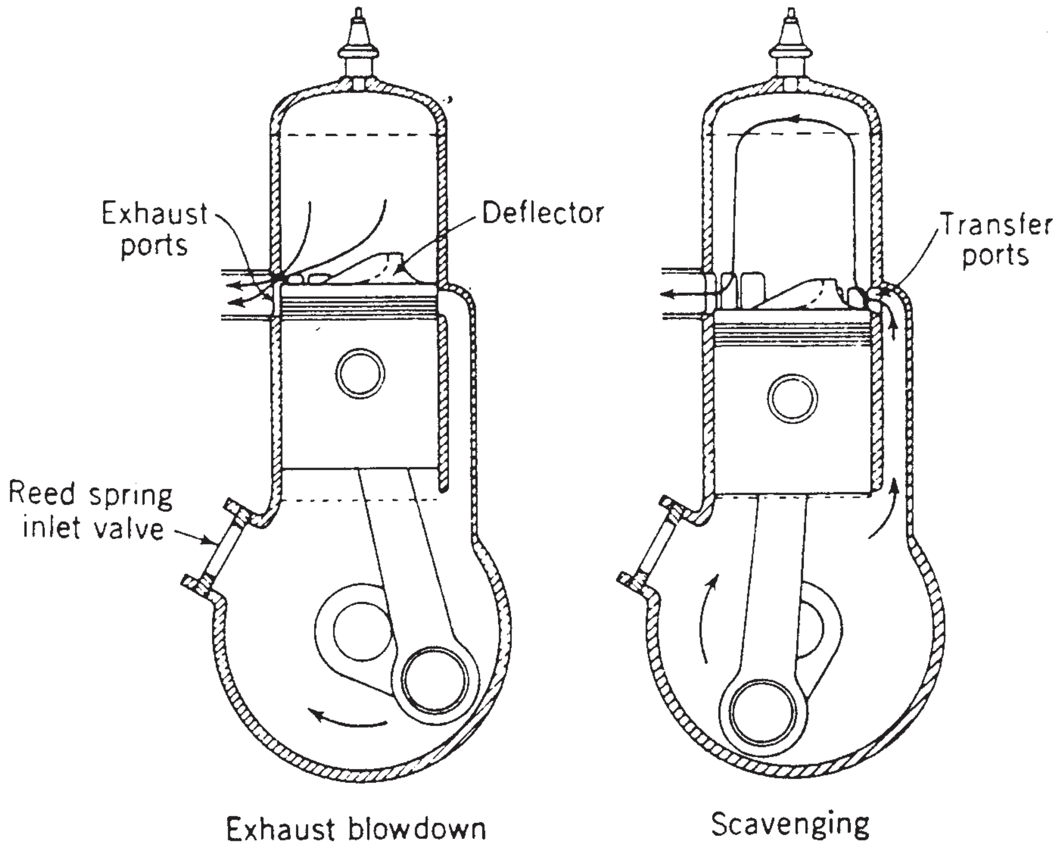


Fig. 3.

Several features are connected with one another.

"Otto" engine:

- Fuel and air mixture is produced outside the cylinderand the mixture of fuel and air flow into the cylindermixture can produced by- carburetor or- injection system (to the inlet pipe)
- mixture in the cyclinder is ignited by a spurk-plug. That`s why these type of engines called"Spark Ignition" (SI) engines.
- Fuel can be petrol or gas (or alcohol and petrol mixture). That`s why these types of engines generally called petrol engine.
- Petrol engine can be two stroke, or four stroke type as well. Two stroke type engines generally used at low power rate.

- Compression ratio (ε) can be maximum ε=10-11, because at higher compression ratio the fuel air mixture ignited spontaneously, because of the heat produced by compression.

"Diesel" engine:

- The piston in the cylinder is compressing pure air. The fuel is injected at the Top Dead Centre to the compressed air. The fuel-air mixture is produced inside the cylinder.
- The air became hot because of the compression, and has to be hot enoughto ignite the fuel. That`s why it is called "Compression Ignition" (CI) engine.
- Fuel can be diesel oil, or fuel oil.
- Compression ratio (ε) is at the range of ε=15-30 at the difference type of diesel engines, because high compression ratio need the fuel ignition. That`s the main reason which produce higher efficiency of Diesel engines compared with petrol engines.
- For Diesel engine operation generally are made four stroke operation engines. Rearly, mainly at very large power rate, two stroke operation type engines are built as well.

Following type of classification is the <u>charging system</u>.

It can be:

- Naturally aspirated

That means the vacuum produced by the piston mooving down forced the air or mixture to flow in to the cylinder.

- Mechanically charged

That means that a compressor driven by the engine charge the cylinder.

- Turbo charged

That means that the charging compressor is driven by an exhaust gas driven turbine.

Following type of classification is the fluid inlet-outlet control.

- At four stroke engines valve control is used. Valves and camshaft can be at various arrangements. Generally used arrangments:

  -side valve (SV) arrangement That means that valves and camshaft are nearbythe cylinder (nowadays not used)

  -overhead valve (OHV) arrangement That means that valves are built in the cylinder headand camshaft are nearby the cylinder

  -overhead camshaft (OHC) arrangement That means that valves and camshaft are built in the cylinder head (further improvement which used mainly at sport cars DOHC system, which means that two camshaft open and close valves)

- At two stroke engines port or piston control is used.

Following type of classification is the cooling system.

Engines have to be cooled, so as not to demage biult in materials.

Cooling system can be: - air cooled, or - water cooled

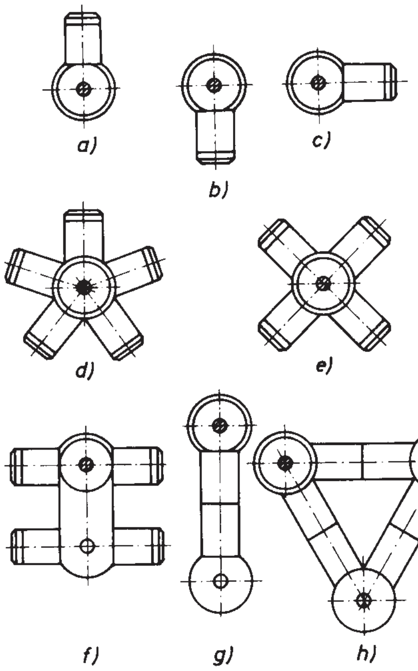
Engines which consists of only one cylinder generally at small power rate are built.

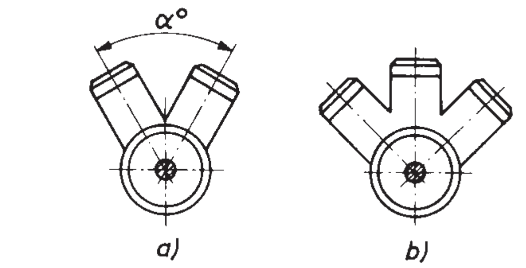
Engines generally have more than one cylinder, up to 12.

Arrangement of cylinders can be very various.

Generally used arrangements are:
- in line arrangement
- V arrangement
- opposed cylinder engine
- radial type engine

At the next drawings you can see some other arrangements as well.





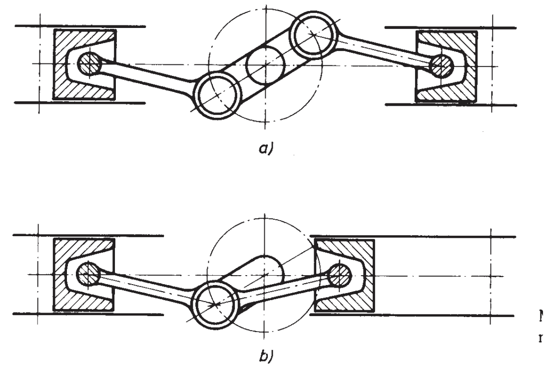


Fig. 4.

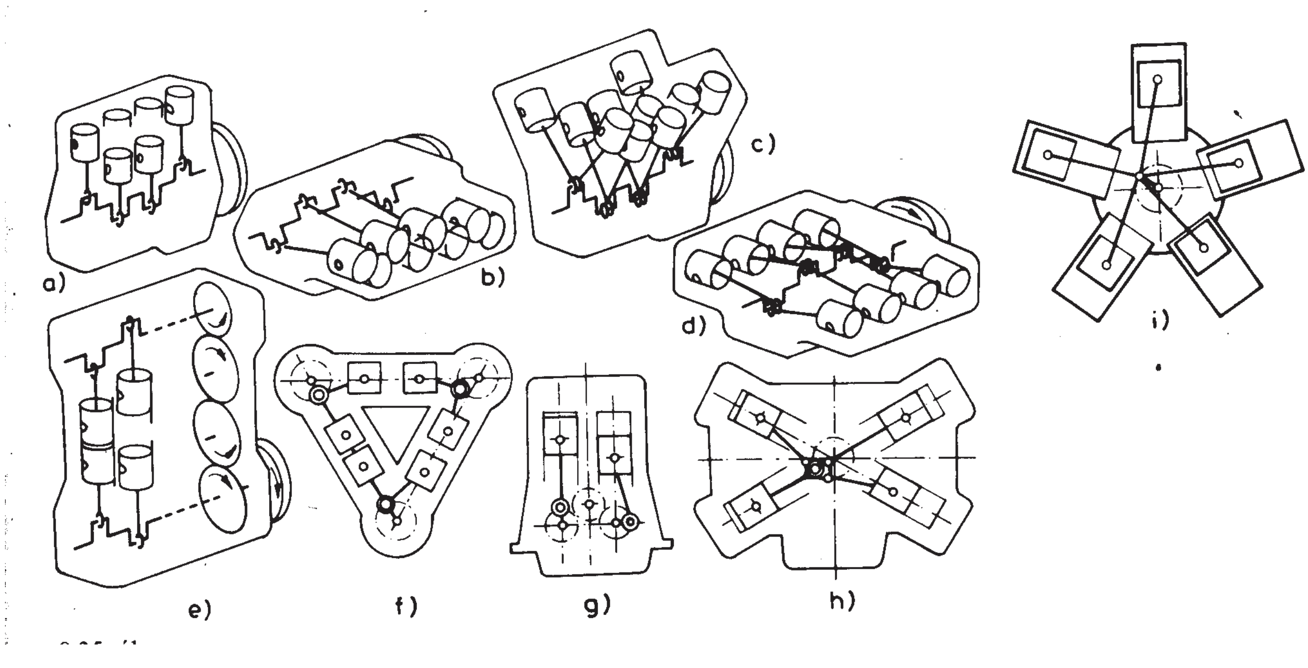


Fig. 5.

Basic principals of mechanical construction

The function of the major components of Internal Combustion Engines and their construction materials will now be reviewed.

The engine cylinders are contained in the engine block. The block has traditionally been made of gray cast iron because of its good wear resistance and low cost. Passages for the cooling water are cast into the block. Heavy-duty and truck engines often use removable cylinder sleeves pressed into the block that can be replaced when worn. These are called *wet liners* or *dry liners* depending on whether the sleeve is in direct contact with the cooling water. Aluminium is being used increasingly in smaller SI engine blocks to reduce engine weight. Iron cylinder liners may be inserted at the casting stage, or later on in the machining and assembly process. The crankcase is often integral with the cylinder block.

The crankshaft has traditionally been a steel forging; nodular cast iron crankshafts are also accepted normal practice in automotive engines. The crankshaft is supported in main bearings. The maximum number of main bearings is one more than the number of cylinders; there may be less. The crank has eccentric portions (crank throws). The connecting rod big-end bearings attach to the crank pin on each throw.

Both main and connecting rod bearings use steel backed precision inserts with bronze, babbit, or aluminium as the bearing materials. The crankcase is sealed at the bottom with a pressed steel or cast aluminium oil pan which acts as an oil reservoir for the lubricating system.

Pistons are made of aluminium in small engines or cast iron in larger slower-speed engines. The piston both seals the cylinder and transmits the combustion-generated gas pressure to the crank pin via the connecting rod. The connecting rod, usually a steel or alloy forging (though sometimes aluminium in small engines), is fastened to the piston by means of a steel piston pin through the rod upper end.

The piston pin is usually hollow to reduce its weight.

The oscillating motion of the connecting rod exerts an oscillating force on the cylinder walls via the piston skirt (the region below the piston rings). The piston skirt is usually shaped to provide appropriate thrust surfaces. The piston is fitted with rings which ride in grooves cut in the piston head to seal against gas leakage and control oil flow.

The upper rings are compression rings which are forced outward against the cylinder wall and downward onto the groove face. The lower rings scrape the surplus oil from the cylinder wall and return it to the crankcase. These crankcase must be ventilated to remove gases which blow by the piston rings, to prevent pressure build-up.

The cylinder head (or heads in V engines) seals off the cylinders and is made of cast iron or aluminium. It must be strong and rigid to distribute the gas forces acting on the head as uniformly as possible through the engine block. The cylinder head contains the spark plug (for an SI engine) or fuel injector (for a CI engine), and in overhead valve engines, parts of the valve mechanism.

The valve type normally used in four-stroke engines, is poppet valve. Valves are made from forged alloy steel. The cooling of the exhaust valve which operates at about 700°C may be enhanced by using a hollow stem partially filled with sodium which through evaporation and condensation carries heat from the hot valve head to the cooler stem. Most modern spark-ignition engines have overhead valve locations

(sometimes called valve-in-head or I-head configurations). This geometry leads to a compact combustion chamber with minimum heat losses and flame travel time, and improves the breathing capacity. Previous geometries such as the L head where valves are to one side of the cylinder are now only used in small engines.

The valve stem moves in a valve guide, which can be an integral part of the cylinder head (or engine block for L-head engines), or may be a separate unit pressed into the head (or block). The valve seats may be cut in the head or block metal (if cast iron) or hard steel inserts may be pressed into the head or block.

A valve spring, attached to the valve stem with a spring washer and split keeper, holds the valve closed. A valve rotator torns the valves a few degrees on opening to wipe the valve seat, avoid local hot spots, and prevent deposits building up in the valve guide.

A camshaft made of cast iron or forged steel with one cam per valve is used to open and close the valves. The cam surfaces are hardened to obtain adequate life.

In four-stroke cycle engines, camshafts turn at one-half the crankshaft speed.

Mechanical or hydraulic lifters or tappets slide in the block and ride on the cam.

Depending on valve and camshaft location, additional members are required to transmit the tappet motion to the valve stem; e.g., in in-head valve engines with the camshaft at the side (OHV systems), a push rod and rocker arm are used. A recent trend in automotive engines is to mount the camshaft over the head with the cams acting (OHC systems) either directly or through a pivoted follower on the valve.

Camshafts are gear, belt, or chain driven from the crankshaft.

An intake manifold (aluminium or cast iron) and an exhaust manifold (generally of cast iron) complete the engine assembly.

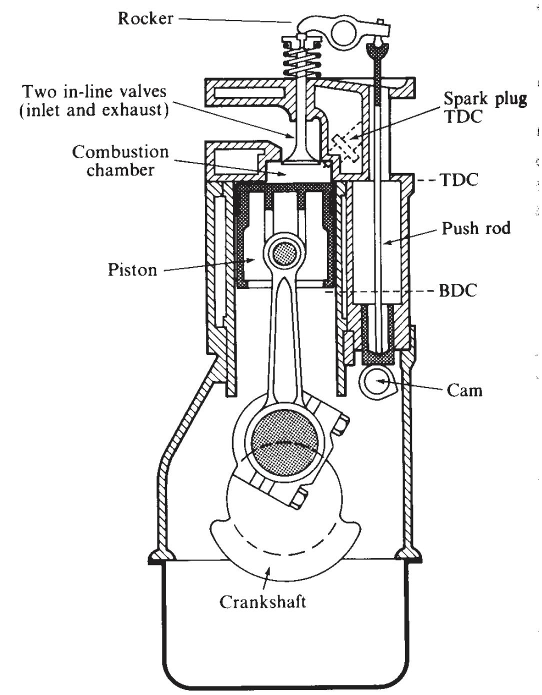


Fig. 6. Basic parts of IC engine

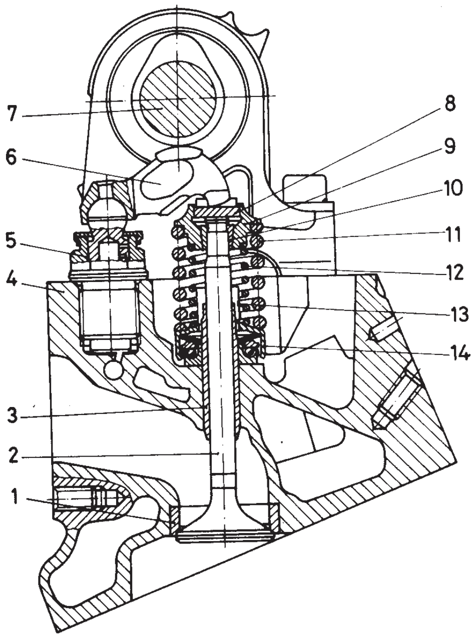


Fig. 7. OHC valve arrangement

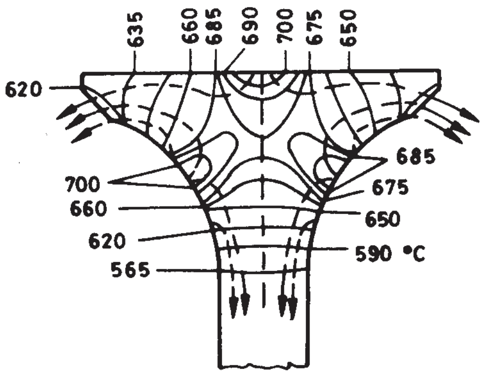


Fig. 8. Temperature distribution at an exhaust valve during operation.

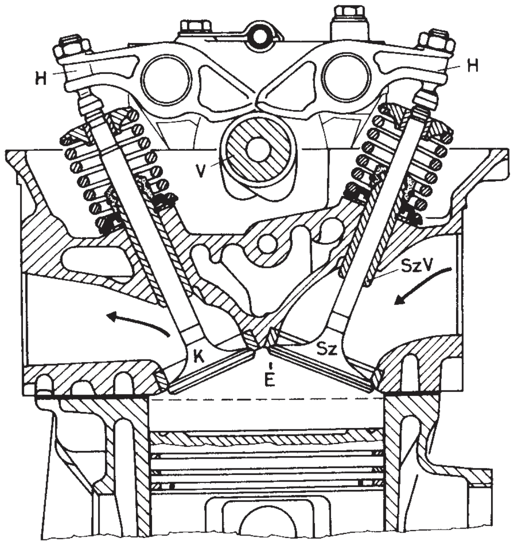
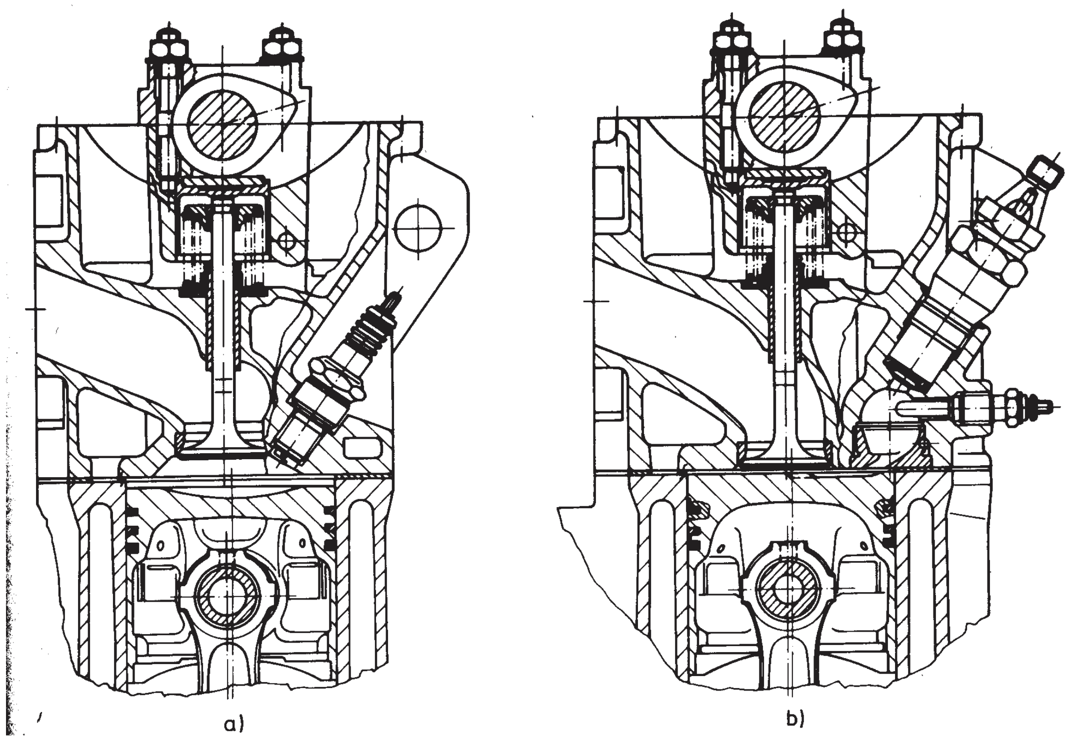


Fig. 9. DOHC valve arrangement



This two fig. shows cross section of VW GOLF engines.Fig. 10. - a SI engine, Fig. 11. - b CI engine

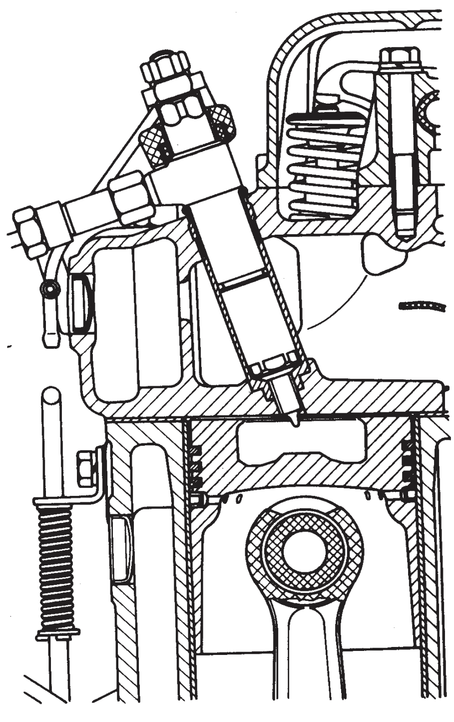
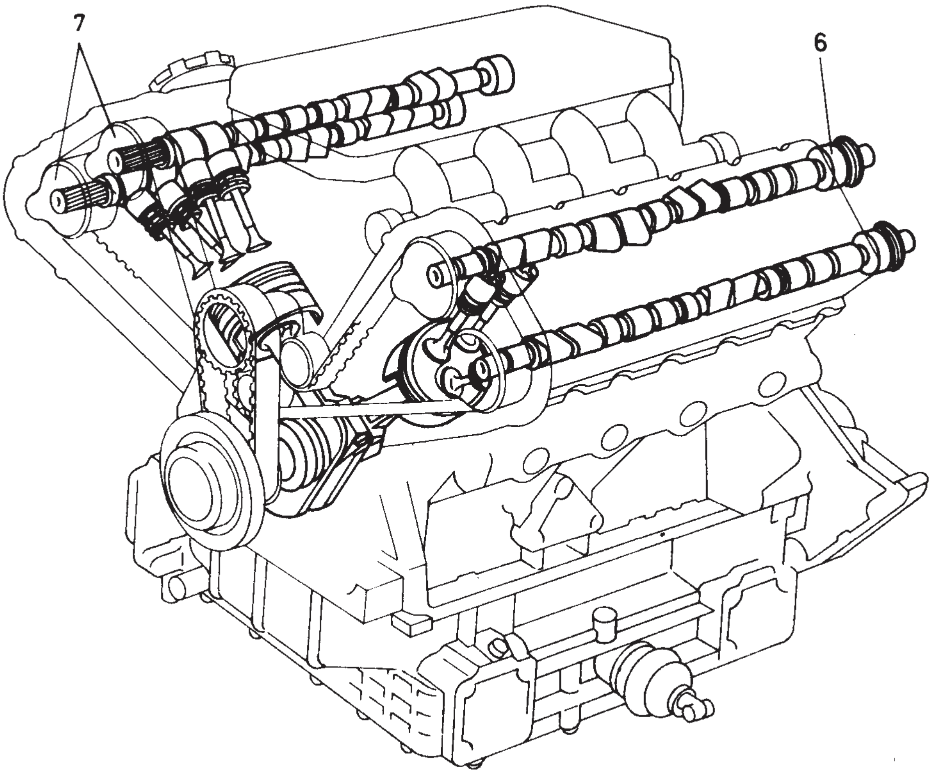


Fig. 12. shows a cross section of a CI engine.



b)

Fig.13.. shows a DOHC valve arrangement and four valve/cylinder at an engine with V cylinder arrangement

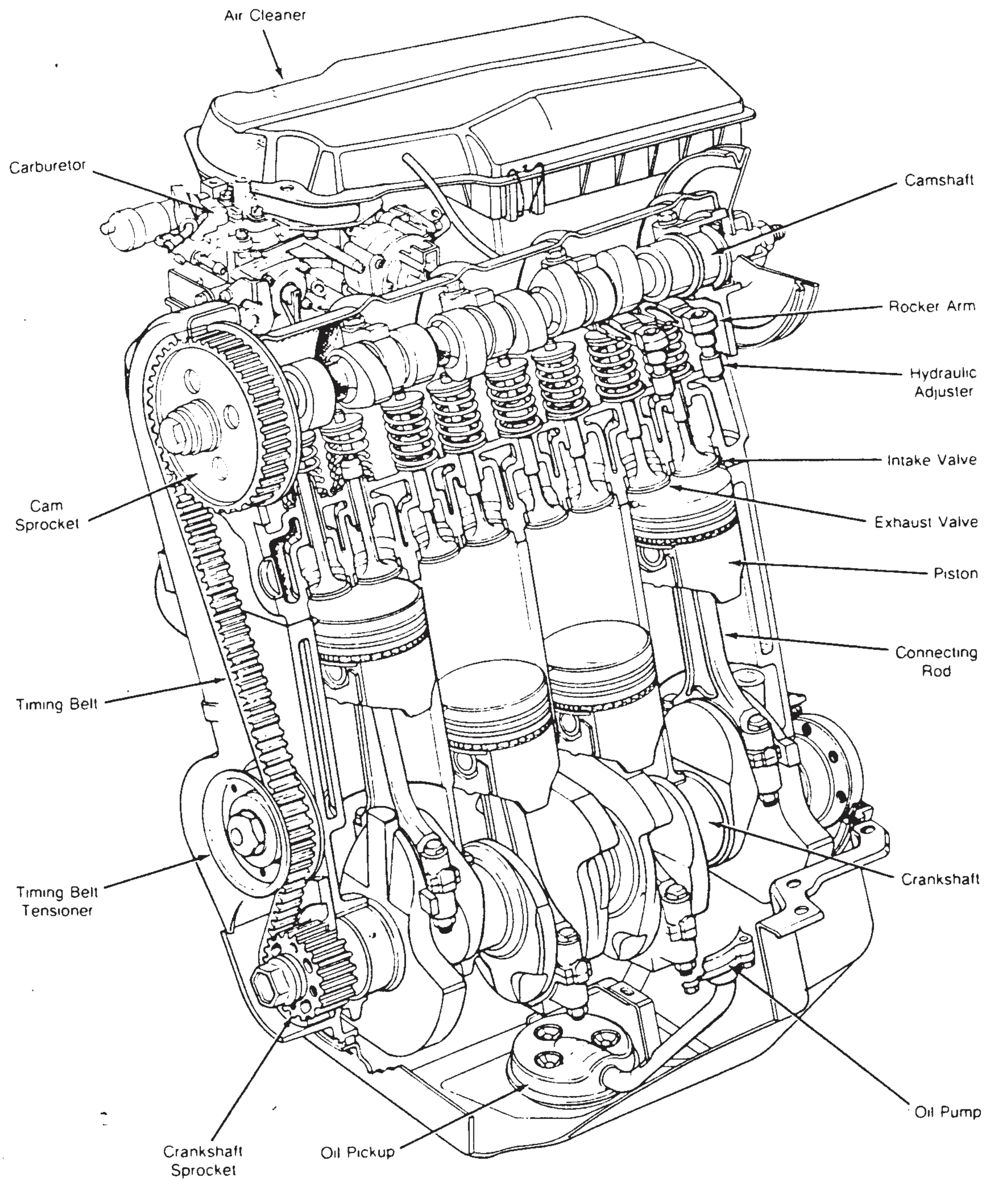


Fig. 14. Cross section of an up to date SI engine.

Wankel rotary engine

In order to reduce engine components and produce more compact engine, and to reduce losses wich caused by alternating movements in traditional engine, an egine with rotary piston (or pistons) were invented which is called Wankel engine.

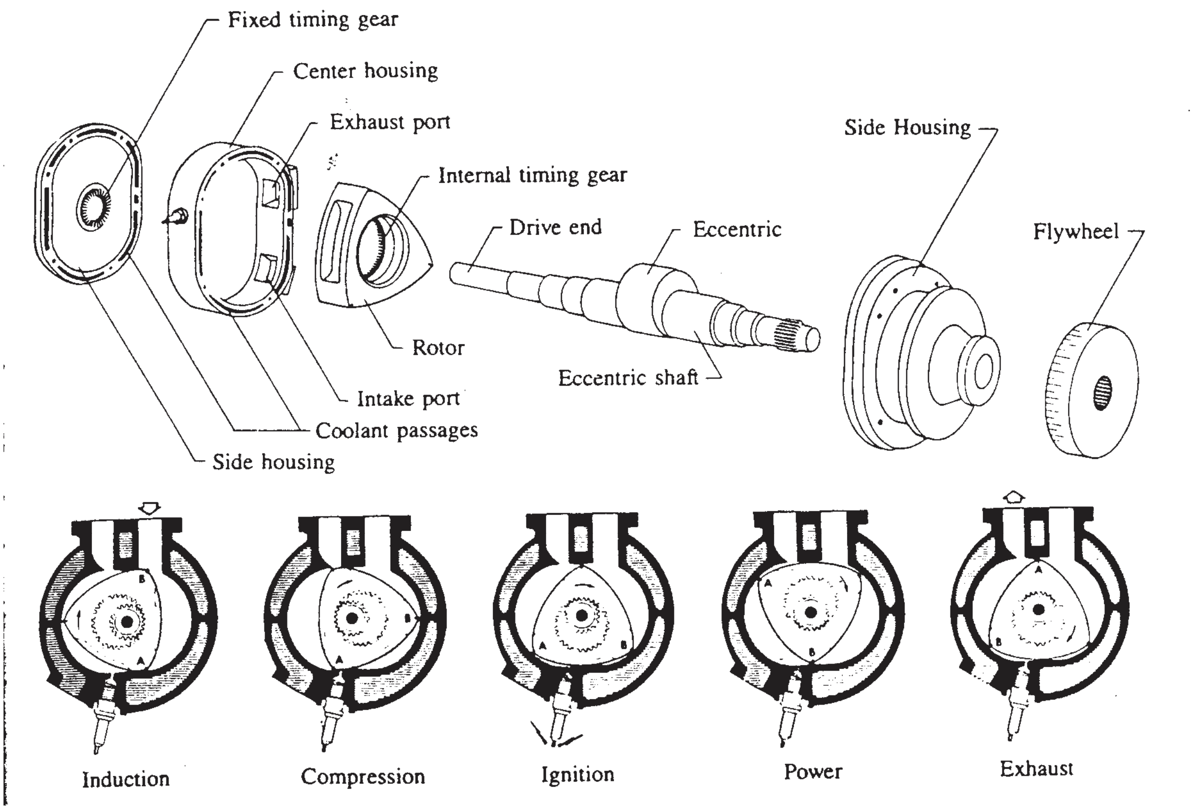
But now it is not used because its sealing and leakage problems.

There are two rotating parts, the triangular-shaped rotor and the output shaft with its integral eccentric. The rotor revolves directly on the eccentric. The rotor has an integral timing gear which meshes with the fixed timing gear on one side housing to maintain the correct phase relationship between the rotor and eccentric shaft rotations.

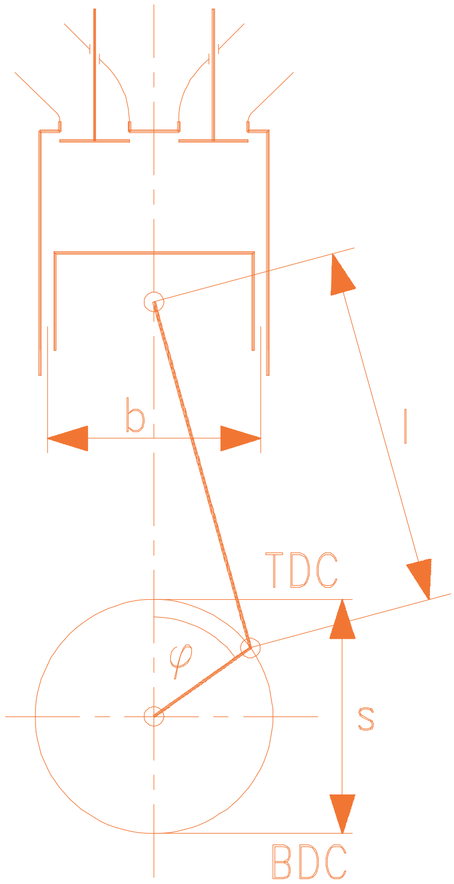
Thus the rotor rotates and orbits around the shaft axis. Breathing is through ports in the center housing. The combustion chamber lies between the center housing and rotor surface and is sealed by seals at the apex of the rotor and around the perimeters of the rotor sides. Wankel rotary engine operates with four stroke cycle.

As the rotor makes one complete rotation, during which the eccentric shaft rotates through three revolutions, each chamber produces one power stroke.

Three power pulses occur, therefore, for each rotor revolution; thus for each eccentric (output) shaft revolution there is one power pulse.



SUMMARY OF DEFINITIONS AND RELATED EQUATIONS IN FIELD OF INTERNAL COMBUSTION RECIPROCATING ENGINES



b: bore [mm] (1 in = 25.4 cm)

l: connecting rod length [mm]

s: stroke [mm]

φ: crank angle from the TDC

ABOUT THE GEOMETRY OF AN ENGINE CYLINDER

Displacement volume *(one)*

$$V_d = \frac{b^2 \cdot \Pi}{4} \cdot s \quad [\text{cm}^3]$$

Clearance volume

$$V_c \quad [\text{cm}^3]$$

Cylinder volume $V = V_d + V_c$

Compression ratio

$$\varepsilon = \frac{\text{max. volume}}{\text{min. volume}} = \frac{V}{V_c} = \frac{V_d + V_c}{V_c} \quad [-]$$

Criteria of performance:

Torque :

The torque measured by dynamometers. Obtained by reading off a net load (F [N]) at known radius (k [m]) from the axis of rotation. $M = F \cdot k$ [Nm]

Indicated power :

The rate of work done by the gas on the piston evaluated from the indicator diagram obtained from the engine. P_i [W]

Brake power :

The power delivered by the engine. $P_b = 2 \cdot \pi \cdot n \cdot M$ [W]

Indicated mean effective pressure (imep) :

It is defined as $p_i = \frac{P_i}{V_s \cdot n \cdot i}$ [bar]

n : engine revolution [rev/s]

i : 1 if two stroke engine

2 if four stroke engine

Break mean effective pressure (bmep) :

It is defined as $p_e = \frac{P_b}{V_s \cdot n \cdot i}$ [bar]

Friction power :

The difference between the indicated and the brake power i.e. the power required to overcome the frictional resistance of the engine parts.

$$P_f = P_i - P_b \qquad [\mathrm{W}]$$

Mechanical efficiency :

$$\eta_M = \frac{P_b}{P_i} = \frac{bmep}{imep} \qquad [\ -\ ]$$

Volumetric efficiency :

$$\eta_V = \frac{\dot{V}}{\dot{V}_s} = \frac{\dot{m}_a + \dot{B}}{V_s \cdot \rho_i \cdot n \cdot i} \qquad [-]$$

ρ_i : fuel-air mixture density in the intake manifold

Indicated efficiency :

$$\eta_i = \frac{P_i}{\dot{B} \cdot H_i} \qquad [\ -\ ]$$

H_i : available energy content of fuel [kJ/kg]

$\dot{B}$: mass flow rate of fuel [kg/s]

Brake thermal efficiency :

$$\eta_{eff} = \frac{P_b}{\dot{B} \cdot H_i} \qquad [\ -\ ]$$

Delivery ratio :

$$\lambda = \frac{\dot{m}_a}{V_s \cdot \rho_\infty \cdot n \cdot i} = \frac{p_\infty + \Delta p_i}{p_\infty} \cdot \frac{T_\infty}{T_\infty + \Delta T_i} \qquad [\ -\ ]$$

$p_\infty, \rho_\infty, T_\infty$: ambient density, pressure and temperature

$\Delta p_i, \Delta T_i$: pressure and temperature change through intake

Excess air factor :

$$\lambda_m = \frac{\dot{m}_a}{\dot{B} \cdot \mu} \qquad [\ -\ ]$$

$\dot{m}_a$: mass flow rate of air [kg/s]

Air-to-fuel ratio (A/F) :

$$A/F = \frac{\dot{m}_a}{\dot{B}} \qquad [\ -\ ]$$

Fuel-air equivalence ratio :

$$\phi = \frac{\dot{B} \cdot \mu}{\dot{m}_a} = \frac{1}{\lambda_m} \qquad [\ -\ ]$$

μ : stoichiometric air-fuel ratio

Brake specific fuel consumption (bsfc):

$$b_e = \frac{\dot{B}}{P_b} = \frac{1}{H_i \cdot \eta_{eff}} \qquad [\mathrm{g/kWh}]$$

Mean piston speed :

$$\bar{u}_p = 2 \cdot s \cdot n \qquad [\mathrm{m/s}]$$

Engine cooling and lubrication

Cooling

To prevent different parts of the engine from overheating, they need to be cooled.

The nearly only one fluid which can absorb the heat is ambient air.

At an engine operation every part of the engine has to be heated up quickly, up to the working temperature, and after it, the temperature of every detail has to be kept in a certain relatively narrow range. Meanwhile the load of the engine can be vary various from a minimum to maximum (100%) load, and can fluctuate in these load-range.

But the ambient air temperature can be very various, from -20°C up to +40°C.

Cooling system has to operate properly at that very wide and various range of circumstances.

There are two basic types of cooling systems: air cooling system
water cooling system

Air cooling system:

At this system the air directly cools the hot parts of the engine.
Generally that surface is finned to enlarge the heat transfer surface.

Benefits:
- It doesn't need any other fluid
- There is not freezing danger
- Easy to start the engine
- Warming up the engine is fast
- It need a few auxiliary equipment

Drawbacks:
- It is difficult to adjust it to very cold or to very hot circumstances, and ensure for every part of the engine not too much but enough cooling
- At composite engines which has more than one or two cylinders difficult to lead air everywhere when its needed, and distribute it to ensure the equal cooling of every part

Water cooling system:

At this system heat is transferred first to the water which delivers it, and then gives the heat to the ambient air by means of a heat exchanger (radiator).

The circulation between the engine and the radiator can be:
- natural circulation, or
- forced circulation

Benefits:
- Heat transfer coefficient between water and the engine parts, is greater than in case of air.
- Specific heat of the water is greater than in case of air, that is why it can absorb more heat at less warming up range
- It is much easier to lead water to every part of the engine than in case of air cooling, and ensure nearly equal cooling for every engine part.

Drawbacks:
- It needs more auxiliary equipment.
- There is freezing danger at cold circumstances
- There is a boiling danger at hot circumstances.

Regulation of cooling systems

Air cooling systems:

Regulation of air cooling systems can only be done by adjusting the cooling air quantity.

It can be done by means of: movable sheets, or
cooling fan
- driven by the crankshaft by means of V-belt, or
- driven by small electrical engine

But using a fan cooling depends on revolution not on power, that is why at some cases fan operation has to be regulated

It can be:
- switching on and off
- adjust the revolution of the fan
- adjust the blades of the fan

Water cooling systems:

All the method was mentioned for air cooling systems can be applied, but not for the engine cooling direct but for the regulation of heat exchange at the radiator between air and water.

But at water cooling systems there are other possibilities.

The nearly only one water side regulation method, is using thermostat.

Thermostat is an equipment which is filled up with a special fluid which has lower boiling point than the water, and it operates as an automatic valve which regulate the water flow streaming to the radiator, or going back directly to the engine.

Comparison of two systems

Air cooling system is generally used for small engines with one or two cylinder,
and at small power rate, because it is plain and cheap.
At high power rate engines and for composite engines water cooling system is used.

For heating of the passenger cabin generally the heat produced by the engine is used.
To solve this task is much more easier in case of water cooling system, than in case of air cooling system.

For water cooling systems distilled (desalinated) water is recommended, so as not to cause scale deposits either in the engine or in the radiator.
In case of cold weather some antifreeze liquid (which contains generally ethilene-glicol) is mixed to the water to reduce freezing point, and avoid freezing.

<u>**Lubrication**</u>

Nearly every moving part of the engine need lubrication.
Lubrication system: - prevent the engine parts from abrasion, erosion
- prevent the engine parts from corrosion
- help the cooling of the engine

Lubrication methods:

<u>Mixture lubrication</u> is only used at two stroke petrol engines, where oil is mixed with petrol, and when the petrol-oil-air mixture is streaming through the crank house, give lubrication to the moving parts.

<u>Splash/spray lubrication</u>, where oil takes place in the crank house, and the crankshaft during its revolution splashes into oil and sprays it up to bearings and cylinder walls.

<u>Pumping lubrication system</u>, where an oil pump driven by the crankshaft pumps the oil, and tubes and holes lead the oil to every part of the engine which has to be lubricated.

The plain splash system nowadays is not used. Generally combined system is used where the pumping and splash system are combined.
At very high power rate only pumping lubrication is used.

At a typical up to date engine, the oil is pumped from the oil pan to the filter. If the filter is plugged with dirt, then oil is bypassed. Oil is distributed from the main oil gallery to the valve train and crankshaft. Oil flows through the filters and pushrods to the rocker arms. That oil then dribbles back to the pan. Oil exists the main gallery through drilled holes to the timing chain and to each camshaft bearing. Oil then flows to each main bearing and then through passages in the crankshaft to the connecting rod bearings. The oil leaving the rod and main bearings is thrown off the crankshaft arms and counter weights in a spray. This spray provides oil to lubricate the cylinder and the piston pins. That oil and the oil sprayed on other surfaces then dribbles back to the pan.

Motor oil

Motor oil is to give adequate lubrication for every moving part of the engine. It is very difficult, because of the wide range of the engine-load, ambient conditions (temperature), different materials which are built in, and so on.

Basic demands against motor oil are the follows:

- Viscosity and other flowing properties have to be at a determined narrow range at the wide temperature range
- Ability to prevent engine details from abrasion in case of cold and hot engine too.
- Prevention from deposit layer arising on different parts of the engine.
- It has to be able to keep and to deliver dirts and soot to the oil filter.
- Foaming ability has to be very low
- It has to prevent engine-parts from corrosion
- It has to be chemically stable and long life
- It must not chemically agressive against different built in materials.

Oil consist of basic oil and additives.

Basic oil determine the basic properties of the oil, mainly at flowing, viscosity and chemical areas. It can be produced from crude oil by destillation and raffinary, or it can be produced by synthetic process, when it is called synthetic oil.

Additives modify the basic oil properties to reach all the demands which are determined.

Classification of motor oils

There are two method for classification of motor oils.

- According to the viscosity properties
- According to the power level

Viscosity classification

Different viscosity classes according to SAE J-300.
(SAE - Society of Automative Engineers)

SAE viscosity class	Viscosity maximum [mPa s]	pumpability limit temperature [°C]	Viscosity at 100°C [mm^2/s]
0W	3250 at -30°C	-35	min. 3.8
5W	3500 at -25°C	-30	min. 3.8
10W	3500 at -20°C	-25	min. 4.1
15W	3500 at -15°C	-20	min. 5.6
20W	4500 at -10°C	-15	min. 5.6
25W	6000 at - 5°C	-10	min. 9.3
20			min.5.6/max.9.3
30			min.9.3/max.12.5
40			min12.5/max16.3
50			min16.3/max21.9

An oil which is adequate one if these clasees is called monograde oil.
But the demand became higher and higher thet is why multigrade type motor oils has been invented so as to motor oil can be used for a long time over seasons.

Multigrade oils and their viscosity:

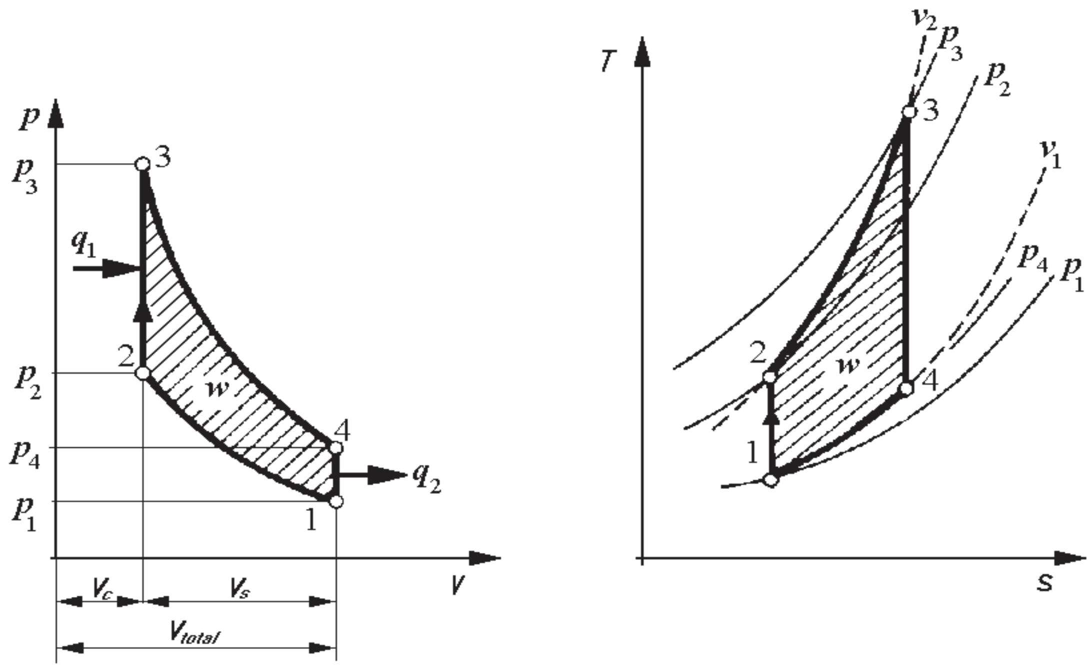
SAE class	Viscosity at -18°C maximum [mm^2/s]	Viscosity at 100°C minimum [mm^2/s]
5W - 20	1300	145
5W - 30	1300	205
10W - 20	2600	90
10W - 30	2600	145
10W - 40	2600	170
10W - 50	2600	190
15W - 40	4800	120
15W - 50	4800	150
20W - 30	10500	100
20W - 40	10500	115
20W - 50	10500	133

Multigrade oil has to be adequate for every including monograde parameter.

THE THEORETICAL OTTO CYCLE

The Otto cycle is the ideal air standard cycle for the petrol engine, the gas engine, and the high-speed oil engine. A special case of an internal combustion engine whose combustion is so rapid that the piston hardly moves during the combustion which therefore occurs at constant volume.

The cycle is shown on p-v and T-s diagrams below



The basic processes are: Process 1 to 2 is isentropic compression

Process 2 to 3 is reversible constant volume heat addition

Process 3 to 4 is isentropic expansion

Process 2 to 3 is reversible constant volume heat rejection

THERMODYNAMIC RELATIONS

Compression stroke	$\frac{v_2}{v_1} = \varepsilon^{-1}$	$\frac{p_2}{p_1} = \varepsilon^{\chi}$	$\frac{T_2}{T_1} = \varepsilon^{\chi-1}$	$W_{comp} = m \cdot c_v \cdot (T_2 - T_1)$
Heat addition	$v_3 = v_2$	$\frac{p_3}{p_2} = \frac{T_3}{T_2}$		$Q_{in} = m \cdot c_v \cdot (T_3 - T_2)$
Expansion stroke	$\frac{v_4}{v_3} = \varepsilon$	$\frac{p_4}{p_3} = \varepsilon^{-\chi}$	$\frac{T_4}{T_3} = \varepsilon^{1-\chi}$	$W_{exp} = m \cdot c_v \cdot (T_3 - T_4)$
Heat rejection	$v_4 = v_1$	$\frac{p_4}{p_1} = \frac{T_4}{T_1}$		$Q_{out} = m \cdot c_v \cdot (T_4 - T_1)$

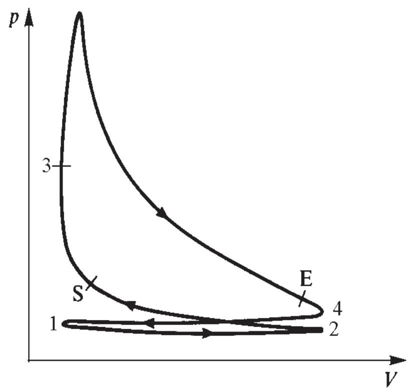
The theoretical efficiency of the cycle is determined as follows

$$\eta_{Ottoideal} = \frac{W_{exp} - W_{comp}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{1}{\varepsilon^{\chi-1}}$$

THE ACTUAL OTTO CYCLE

The four-stroke engine

The picture below shows a typical p-v diagram for a SI four-stroke pertol engine



The strokes are:

1-2 Induction stroke

The air-fuel charge is induced into the cylinder as the piston moves from TDC to BDC. Due to the movement of the piston the pressure in the cylinder is reduced to a value between the atmospheric pressure, and air flows through the induction system because of the pressure difference. In ideal case the inlet valve closes at point 2 , but in fact this does not occur until the piston has moved part of the way along the return stroke.

2-3 Compression stroke

With both valves closed the charge is compressed by the piston. At the TDC position the charge occupies the volume above the piston, which is called the clearance volume, and consists mainly of the volume of the combustion chamber. The spark is timed to occur at a point such as S, which is before TDC. There is a time delay between S and the actual commencement of combustion. The combustion process occurs mainly at almost constant volume, and there is large increase in pressure and temperature of the charge during this process.

3-4 Working stroke

The hot high-pressure gas expands, pushing the piston down the cylinder. It wouldwould appear that this expansion should proceed to completion at 4, but in order to assist in exhausting the gasous product the exhaust valve opens at some point E which is before TDC.

4-1 Exhaust stroke

The returning piston clears the swept volume of exhaust gas, and the pressure during this stroke is slightly higher than atmospheric pressure. In a normally aspirated engine the clearance volume cannot be exhausted, and at the commencement of the next cycle this volume is full of exhaust gas at about atmospheric pressure. The mixture which is compressed thus consists of fresh air and fuel mixture, diluted by exhaust gas from the previous cycle.

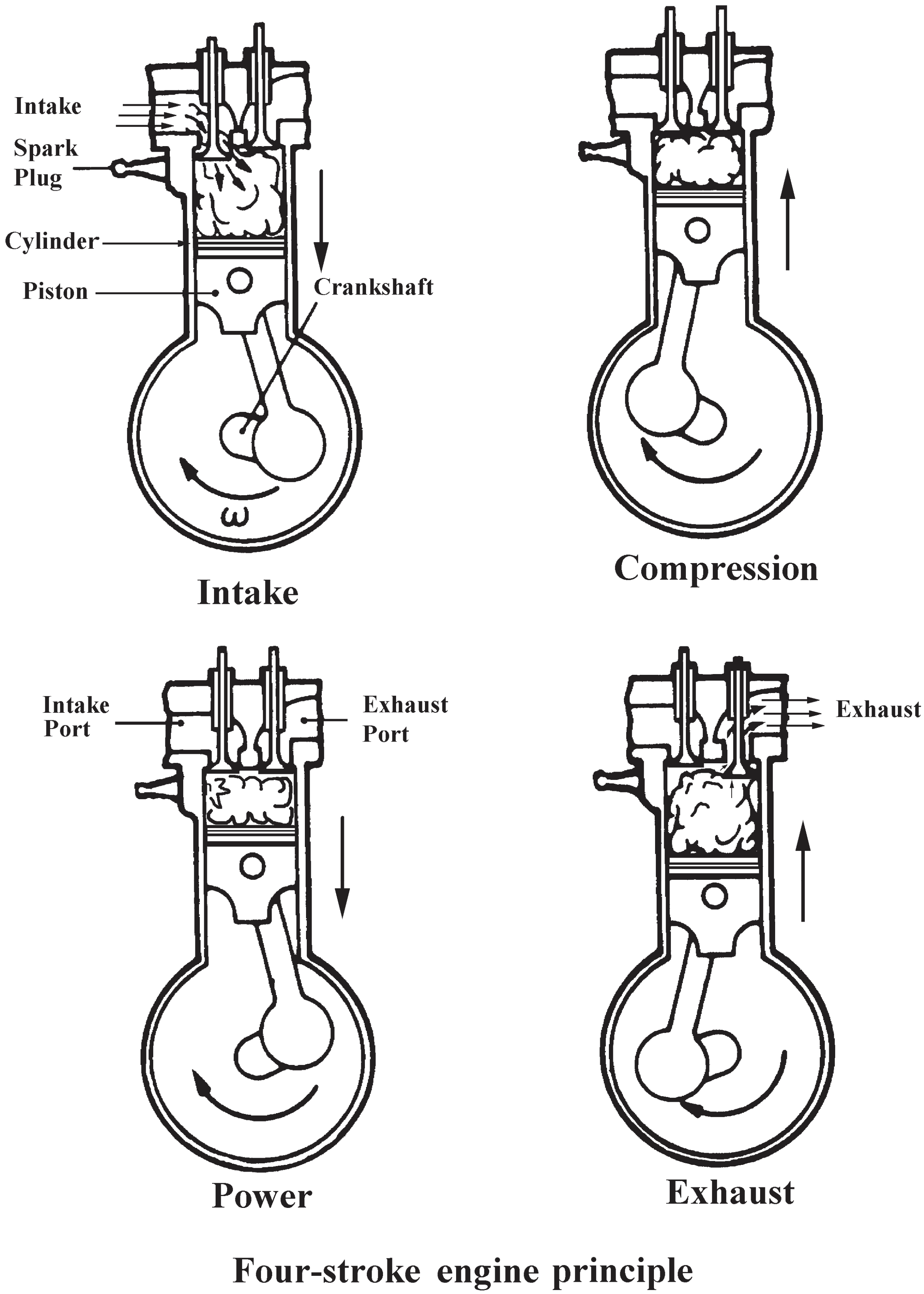
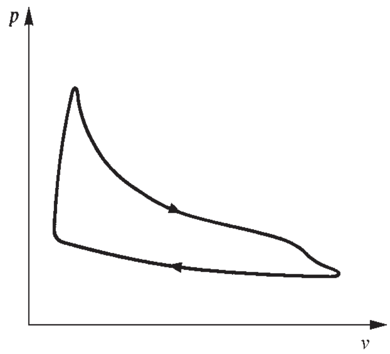

Intake
Spark
Plug
Cylinder
Piston
Crankshaft
ω
Intake
Compression
Intake
Port
Exhaust
Port
Exhaust
Power
Exhaust
Four-stroke engine principle

Fig. 15.

The two-stroke cycle

The picture below shows a typical p-v diagram for a SI two-stroke pertol engine



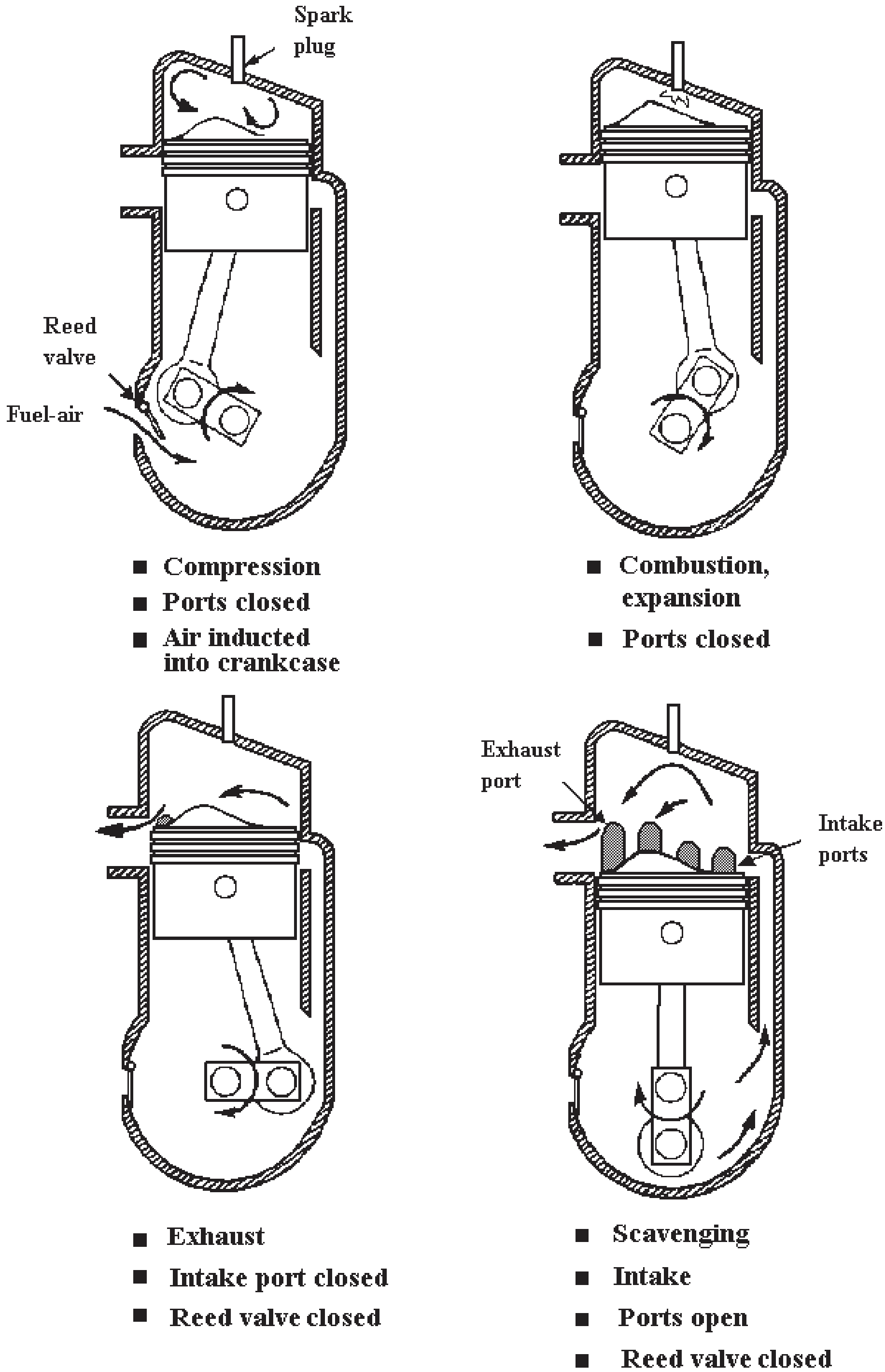
The figure on the following page shows the simplified work process of a two-stroke petrol engine. As the piston ascend on the compression stroke the next charge is drawn into the crankcase through a reed valve. Ignition occurs before TDC, and at TDC the working stroke begins.

As the piston descends all the ports are closed until about 80% of the working stroke when the exhaust port will be uncovered by the piston and the exhaust begins. The intake ports are uncovered later in the stroke. When both of the ports are covered the descending piston starts to compress the fresh charge.

The piston can be shaped to deflect the fresh gas across the cylinder to assist the scavenging of the cylinder; this is called cross-flow scavenge.

Comparison of operating date of different engines

Engine size	Stroke/ ignition	Compr. ratio	Bore cm	Speed rev/min /max./	Power/ volume kW/dm^3	Power/ weight kW/kg	max. bsfc g/kWh
Small	2/Spark	6 - 11	4 - 8.5	4500-8500	20-70	0.25-0.4	380
Small	4/Spark	6 - 11	4 -8.5	4500-7500	20-60	0.2-0.35	350
Passenger car	4/Spark	8 - 10	7 - 10	4500-6500	30-50	0.5	270
Passenger car	4/Compr.	15-22	7-10	4000-5000	20-25	0.2-0.4	240
Truck	4/Spark	7 - 9	9 - 13	3500-5000	20-30	0.15-0.4	300
Truck	4/Compr.	16 - 22	10 - 15	2000-4000	15-22	0.15-0.25	210
Large stationary	2/Compr.	10 - 12	40-100	300-900	3-7	0.04 - 0.03	160-180



Two-stroke cross-scavenged engine principle

Fig. 16.

LOSSES IN INTERNAL COMBUSTION ENGINES

During the operation of the internal combustion engines only a fraction of the chemical energy is converted into mechanical work. The "lost work" can mainly be attributed to the following:

- Heat transfer

  Heat transfer occurs between the cylinder wall and working fluid. The most significant phenomenon is the heat loss of the hot burned gases, which occurs during combustion and expansion.

- Mass loss

  A fraction of the high pressure unburned gases flows from the combustion chamber into the crankcase (blowby) thus the cylinder pressure drops and the output work decreases. This mass loss is about one percent of the charge.

- Incomplete combustion

  The exhaust gases usually contain unburned particles (H_2, CO, CH) carrying a fraction of the fuel's chemical energy (SI engine : ~5%, CI engine : 1-2%).

- Limited combustion speed

  In an ideal SI engine the combustion time is zero i.e.: the combustion speed is infinitive. In a real case the combustion process requires certain time (order of milliseconds in passenger cars) therefore the ignition starts before the TC and complete after the TC. Thus the peak pressure will be less than the one of the perfect cycle and the extracted work will be less, too.

- Exhaust blowdown loss

  Considering that the blow down process takes time the exhaust valve must be opened before the BC thus the expansion stroke will be uncompleted and work will be lost.

- Pumping work

  The friction of the streaming gases and the aerodynamic losses during intake cause pressure drop in the cylinder before compression and sequentially lower peak pressure and less output work. The blowdown process of the exhaust gases requires work, too. The pumping loss is most superior in quantity governed (SI) engines at part load.

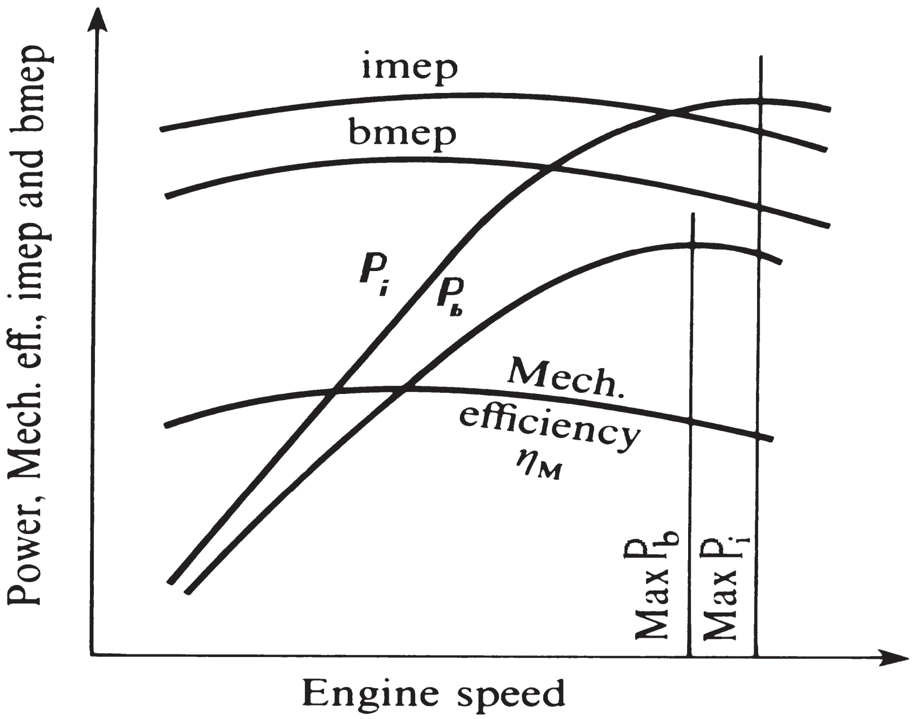
- Friction

  The most significant source of this loss is the friction between the piston skirt, rings and the cylinder (about 60-80% of the total frictional work). Usually it is higher in diesel engines, because of the stronger piston rings. The other sources of frictional losses are the crankshaft, camshaft, valve mechanism, gears, etc..

PERFORMANCE CHARACTERISTICS

The testing of IC engines consists of running them at different loads and speeds and taking sufficient measurements for the performance criteria to be calculated.

The most elementary power test is that which gives the power-speed and torque-speed characteristics, as shown in the figure below. The test is carried out at a constant throttle setting in the petrol engine. If the throttle area is the maximum then the test is called wide open throttle (WOT) test.



As the speed increases from the lower values the indicated and brake power curves are similar. The difference between these powers is the frictional power which increases with speed. Both curves show maximum values but they occur at different speeds. The indicated power falls after the maximum because of the reduction in volumetric efficiency with increased speed.

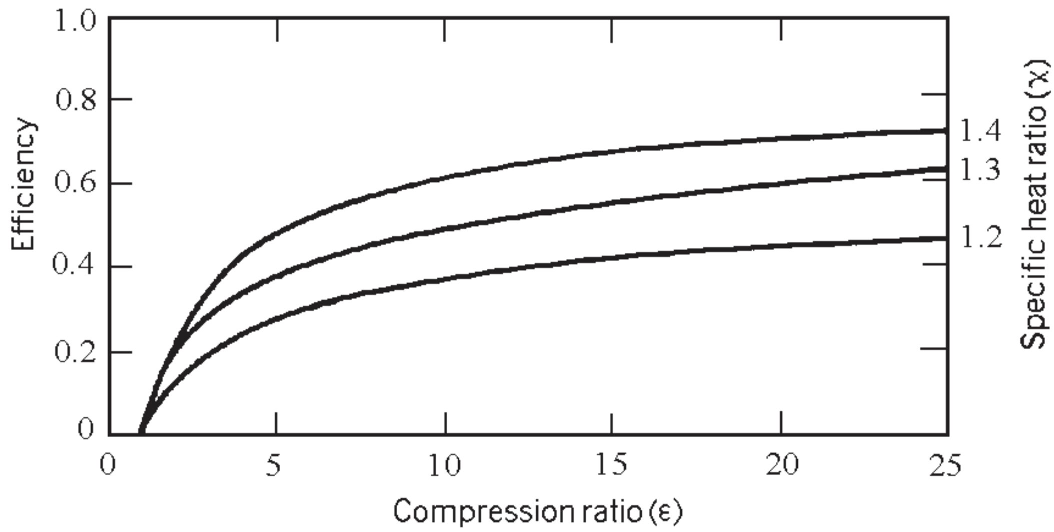
FACTORS INFLUENCING PERFORMANCE

Compression ratio

It has been proved that the theoretical efficiency of the Otto cycle depends on only the compression ratio and the specific heat ratio.

$$\eta_{Ottoideal} = 1 - \frac{1}{\varepsilon^{\chi - 1}}$$

The following graph shows that higher compression ratio indicates higher thermal efficiency. The ability to use higher compression ratio depend on the provision of better quality of fuel and of improved combustor chamber design to avoid the self ignition or pre ignition caused knocking.

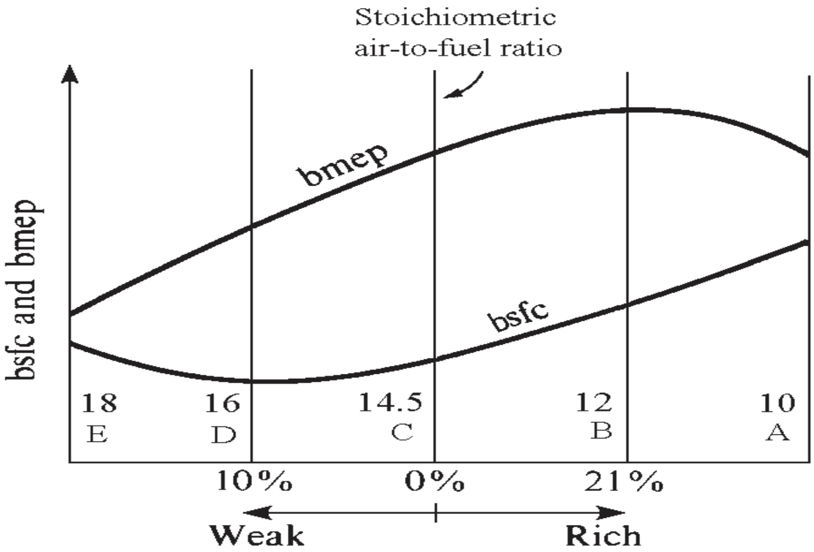


Excess air factor

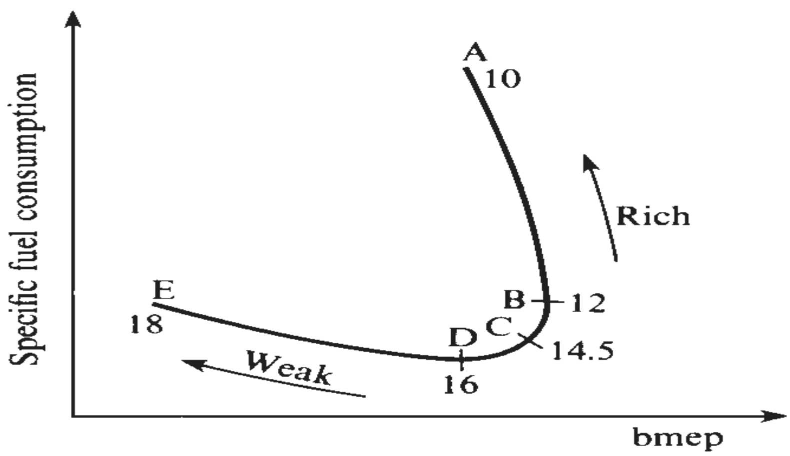
The petrol engines are quantity governed by opening or closing of throttle valve which regulates the mass of charge to the cylinders. The governed speed can be adjusted to select any value in its range.

The petrol engine operates on air-to-fuel ratio in the range 10-22, but not necessarily satisfactory at the extremes. An important test is to run the engine with the air-to-fuel ratio as the only variable. This is carried out at a constant speed, constant throttle opening and a constant ignition setting.

Bmep, bsfc may be plotted against air-fuel ratio as it is shown below.



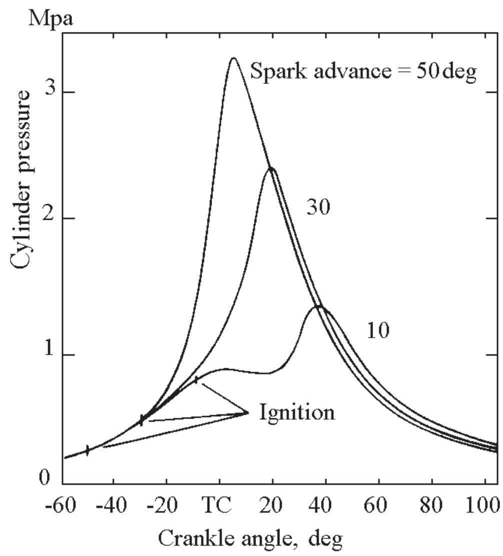
Also the bsfc can be plotted against the bmep. This is called consumption "loop".

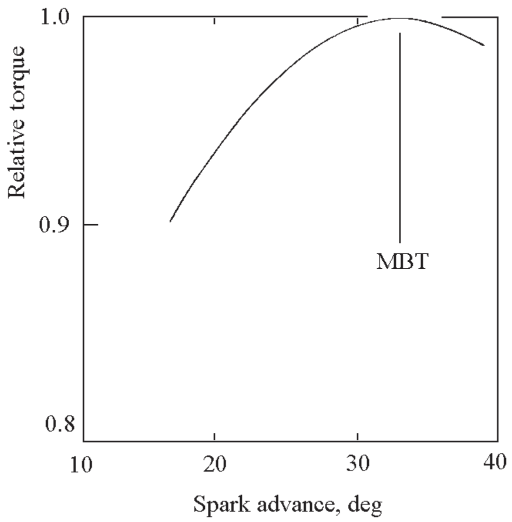


The engines maximum economy point is found in the lean (weak) mixture region meanwhile the maximum power is achieved with rich mixture.

<u>Ignition advance</u>

The figures below show the effect of spark timing on cylinder pressure and torque in a spark ignition engine.





Proper ignition timing is essential in SI engines. Small spark advance causes late occurrence of the pressure peak, reducing its magnitude and lessening the expansion work. High spark advance causes pressure peak occurrence before TC therefore higher compression work is needed.

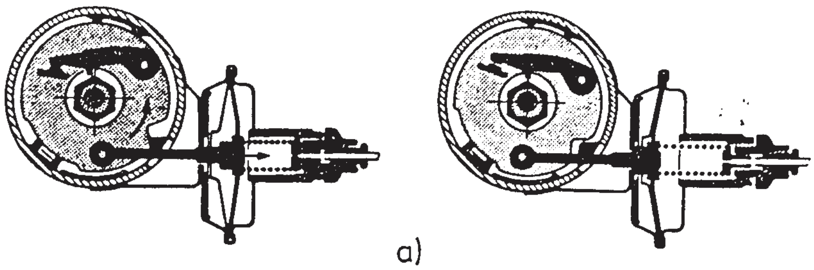
The optimum timing, which gives the maximum brake torque - called *maximum brake torque timing* (MBT timing) - occurs when the magnitudes of these opposing trends just offset each other. MBT can be expected to vary with engine speed and other parameters.

Spark timing

Spark timing has very imporatant rule at SI engine operation. Spark timing relative to top-center affected the pressure development in the cylinder. If combustion starts too early in the cycle, the work transfers from the piston to the gases in the cylinder at the end of the compression stroke is too large. If combustion starts too late, the peak cylinder pressure is reduced and the expansion stroke work transfer from the gas to the piston decreases. There exists a particular spark timing which gives maximum engine torque at fixed speed, and mixture composition and flow rate.This timing also gives maximum brake power and minimum brake specific fuel consumption.

Spark timing in a way depends on speed. As speed increases the spark must be advanced to maintain optimum timing because the duration of the combustion process in crank angle degrees increases. Optimum spark timing also depends on load. As load and intake manifold pressure are decreased, the spark timing must be further advanced to maintain optimum engine performance.

In another way spark timing is important, because NO and HC emissions vary significantly with spark timing.



a)

b)

SPARK IGNITION

In spark-ignition engines, the electrical discharge produced between the spark plug electrodes by the ignition system starts the combustion process close to the end of the compression stroke. The high-temperature plasma kernel created by the spark develops into a self-sustaining and propagating flame front - a thin reaction sheet where the exothermic combustion chemical reactions occur.
The function of the ignition system is to initiate this flame propagation process, in a repeatable manner cycle-by-cycle, over the full load and speed range of the engine at the appropriate point in the engine cycle.

A spark can arc from one electrode to another when a sufficiently high voltage is applied. Ignition systems commonly used to provide this spark are: battery ignition systems where the high voltage is obtained with an ignition coil (coil ignition systems); battery systems where the spark energy is stored in a capacitor and transferred as a high-voltage pulse to the spark plug by means of a special transformer (capacitive-discharge ignition systems); and magneto ignition systems where the magneto - a rotating magnet or armature - generates the current used to produce a high-voltage pulse.

Conventional Ignition Systems

The ignition system must provide sufficient voltage across the spark plug electrodes to set up the discharge and supply sufficient energy to the discharge to ignite the combustible mixture adjacent to the plug electrodes under all operating conditions.
It must create this spark at the appropriate time during the compression stroke.

Usually spark timing is set to give maximum brake torque for the specific operating condition, though this maximum torque may be constrained by emission control or knock control requirements. For a given engine design, this optimum spark timing varies as engine speed, inlet manifold preessure, and mixture composition vary.
Thus, in most applications, and especially the automotive applications, the system must have means for automatically changing the spark timing as engine speed and load vary.

With an equivalence ratio best suited for ignition and with homogeneous mixture distribution, spark energies of order 1 mJ and durations of a few microseconds would suffice to initiate the combustion process. In practice, circumstances are less ideal. The air, fuel, and recycled exhaust gas, and residual gas within each cylinder is not homogenous. Also the pressure, temperature, and density of the mixture between the spark plug electrodes at the time the spark is needed affect the voltage required to produce a spark. These vary significantly over the load and speed range of an engine. The spark energy and duration, therefore, has to be sufficient to initiate combustion under the most unfavorable conditions expected in the vicinity of the spark plug over the complete engine operating range. Usually if the spark energy exceeds 50 mJ and the duration is longer than 0.5 ms reliable ignition is obtained.

In addition to the spark requirements determined by mixture quality, pressure, temperature, and density, there are others determined by the state of the plugs. The erosion of the plug electrodes over extended mileage increases the gap width and requires a higher breakdown voltage. Also, spark plug fouling due to deposit buildup on the spark plug insulator can result in side-tracking of the spark.

When compounds formed by the burning of fuel, lubricating oil, and their additives are deposited on the spark plug insulator, these deposits provide an alternative path for the spark current. If the resistance of the spark plug deposits is sufficiently low, the loss of electrical energy through the deposits may prevent the voltage from rising to that required to break down the gas. The influence of side-tracking on spark generation decreases with lower source impedance of the high-voltage supply, and therefore with a higher available energy.

The fundamental requirements of the high-voltage ignition source can be summarized as:

- a high ignition voltage to break down the gap between the plug electrodes;
- a low source impedance or steep voltage rise;
- a high energystorage capacity to create a spark kernal of sufficient size;
- sufficient duration of the voltage pulse to ensure ignition.

There are several commonly used concepts that partly or fully satisfy these requirements.

Coil Ignition Systems

Breaker-operated inductive ignition systems have been used in automotive engines for many years. While they are being replaced with more sophisticated systems (such as transistorized coil ignition systems), they provide a useful introduction to ignition system design and operation.

The system includes a battery, switch, resistor, coil, distributor, spark plugs, and the necessary wiring. The circuit finctions as follows. If the breaker point is closed when the ignition is switched on, current flows from the battery, through the resistor, primary winding of the ignition coil, contacts, and back to the battery through ground.

This current sets up a magnetic field within the iron core of the coil. When ignition is required, the breaker points are opened by the action of the distributor cam, interrupting the primary current flow. The resulting decay of magnetic flux in the coil induces a voltage in both the primary and secondary windings. The voltage induced in the secondary winding is routed by the distributor to the correct spark plug to produce the ignition spark.

The primary current for any given time of contact closure t is given by:

$$I_P = \frac{V_0}{R} \cdot \left(1 - e^{-Rt/L_p}\right)$$

where I_P is the primary current, V_0 is the supply voltage, R is the total primary circuit resistance, and L_P is the primary circuit inductance. The primary current requires time to build up. At low speeds the time of contact closure is sufficient for the primary current to reach the maximum permitted by the circuit resistance; at high speeds the primary current may not reach its maximum. Thus, only at higher engine speeds does the term $e^{-Rt/Lp}$ become significant. When the points open the primary current falls to zero and a voltage of order 15 kV is induced in the secondary winding.

If the coil is not connected to a spark plug, this indiced voltage will have a damped sinusoidal waveform. The peak value of this voltage is the maximum voltage that can be produced by the system and is called the *available voltage* V_a of the system.

The maximum energy transferred to the secondary system is given by:

$$\boldsymbol{E}_{\boldsymbol{s},\max} = \frac{1}{2}\boldsymbol{C}_{\boldsymbol{s}}\boldsymbol{V}_{\boldsymbol{a}}^2$$

where C_S is the total capacitance of the secondary circuit. Hence, the available voltage of the system is given by

$$V_a = \left(\frac{2E_{s,\max}}{C_s}\right)^{1/2}$$

If all the energy stored in the primary circuit of the coil, $1/2\ L_P\ I_P^2$, is transferred to the secondary,

$$V_a = I_P\left(\frac{L_P}{C_s}\right)^{1/2}$$

When the coil is connected to a spark plug, the secondary voltage will rise to the breakdown potential of the spark plug, and a discharge between the plug electrodes will occur. After the spark occurs, the voltage is reduced to a lower value until all the energy is dissipated and the arc goes out. The value of this voltage which caused breakdown to occur is called the *required voltage* of the spurk plug. The interval during which the spark occurs is called the *spark duration*. The available voltage of the ignition system must always exceed the required voltage of the spark plug to ensure breakdown. The spark must then possess sufficient energy and duration to initiate combustion under all conditions of operation.

The major limitations of the breaker-operated induction-coil system are the decrease in available voltage as engine speed increases due to limitations in the current switching capability of the breaker system, and the decreasing time available to build up the primary coil stored energy. Also, because of the high source impedance

(about 500 kΩ) the system is sensitive to side-tracking across the spurk plug insulator. A further disadvantage is that due to their high current load, the breaker points are subject to electrical wear in addition to mechanical wear, which results in short maintenance intervals. The life of the breaker points is dependent on the current they are required to switch. Acceptable life is obtained with $I_P \approx 4$ A; increased currents cause a rapid reduction in breaker point life and system reliability.

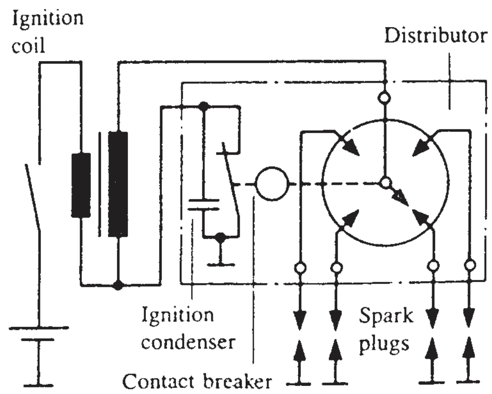


Fig.21.

Transistorized Coil Ignition (TCI) Systems

In automotive applications, the need for much reduced ignition system maintenance, extended spurk plug life, improved ignition of lean and dilute mixtures, and increased reliability and life has led to the use of coil ignition systems which provide a higher output voltage and which use electronic triggering to maintain the required timing without wear adjustment. These are called transistorized coil ignition (TCI) or high-energy electronic-ignition systems. The higher output voltage is required because spark plugs are now set to wider gaps (e.g., about 1 mm) to extend the ability to ignite the fuel mixture over a wider range of engine operation, and because during the extended mileage between spark plug replacement electrode erosion further increases the gap. In automotive applications an available ignition voltage of 35 kV is now usually provided. In addition to higher voltage, longer spark duration (about 2 ms) has been found to extend the engine operating conditions over which satisfactory ignition is achieved.

Most of the solid-state ignition systems now in use operate on the same basic principle. The distributor points and cam assembly of the conventional ignition system are replaced by a magnetic pulse generating system which detects the distributor shaft position and sends electrical pulses to an electronic control module. The module swithes off the flow of current to the coil primary windings, inducing the high voltage in the secondary windings which is distributed to the spark plugs as in the conventional breaker system. The control module contains timing circuits which then close the primary circuit so that buildup of primary circuit current can occur. There are many types of pulse generators that could trigger the electronic circuit of the ignition system. A magnetic pulse generator, where a gear-shaped iron rotor driven by the distributor shaft rotates past the stationary pole piece of the pickup, is usually used. The number of teeth on the rotor is the same as the number of cylinders.

A magnetic field is provided by a permanent magnet. As each rotor tooth passes the pole piece it first increases and then decreases the magnetic field strength ψ linked with the pickup coil, producing a voltage signal proportional to $d\psi/dt$. The electronic module switches off the coil current to produce the spark as the rotor tooth passes through alignment and the pickup coil voltage abruptly reverses and passes through zero. The increasing potion of the voltage waveform, after this voltage reversal, is used by the electronic module to establish the point at which the primary coil current is switced on for the next ignition pulse.

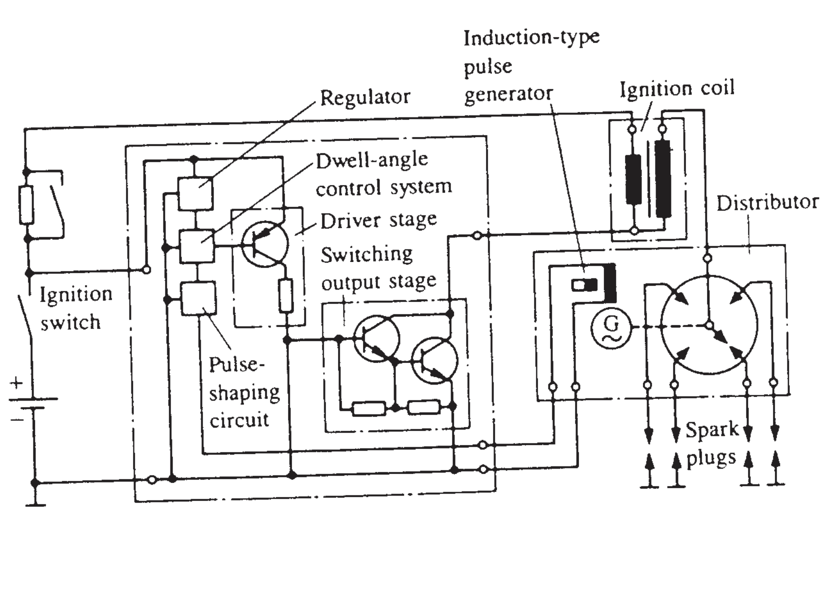


Fig.22.

Capacitive-Discharge Ignition (CDI) Systems

With this type of system a capacitor, rather than an induction coil, is used to store the ignition energy. The capacitance and charging voltage of the capacitor determine the amount of stored energy. The ignition transformer steps up the primary voltage, generated at the time of spark by the discharge of the capacitor through the thyristor, to the high voltage required at the spurk plug. The CDI trigger box containes the capacitor, thyristor power switch, charging device (to convert battery voltage to the charging voltage of 300 to 500 V by means of pulses via a voltage transformer),

pulse shaping unit, and control unit.

The principal advantage of CDI is its insensitivity to electrical shunts in the high-voltage ignition circuit that result from spark plug fouling. Because of the fast capacitive discharge, the spark is strong but short (0.1 to 0.3 ms). This can lead to ignition failure at operating conditions where the mixture is very lean or dilute.

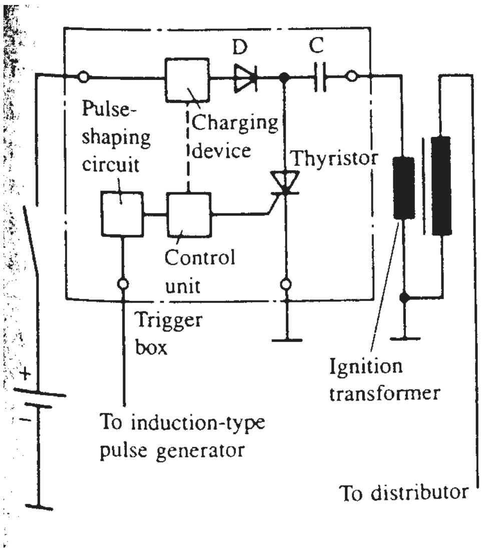


Fig.23.

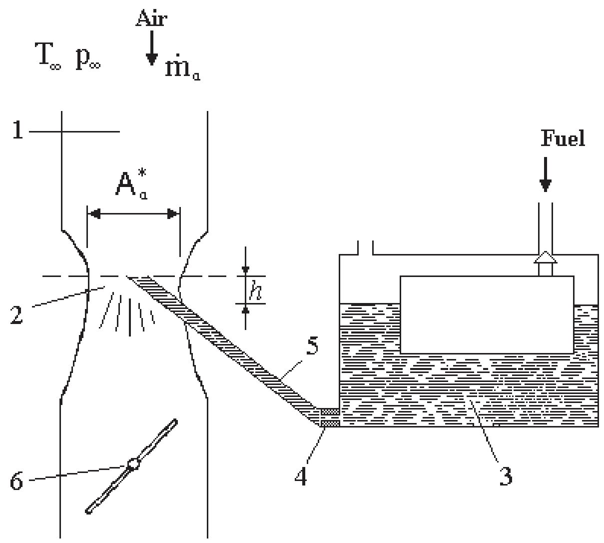
FUEL SYSTEMS FOR SPARK IGNITION ENGINES

The purpose of an engine fuel system is to provide the cylinder with a mixture of air and fuel in the correct proportion for the engine. There are two widely used methods, one is called *carburation* and the other one is the *fuel injection*.

THE CARBURETOR

Basic principle :

In a carburetor the air flows through a venturi nozzle. The presure difference between the carburetor inlet and the throat of the nozzle is used to meter the appropriate fuel flow. The fuel enters the air stream through the fuel discharge tube in the carburetor body, than it is atomized and convected by air stream past the throttle plate and into the intake manifold.



The elementary carburetor
1. Inlet section
2. Venturi nozzle
3. Float chamber
4. Metering orifice
5. Fuel discharge tube
6. Throttle plate

The mass flow rate of the air in the venturi nozzle is governed by the equation below:

$$\dot{m}_a = \rho_\infty \cdot c_\infty \cdot A_a^* \cdot \sqrt{\frac{2}{\kappa-1} \cdot \left[\left(\frac{p_2}{p_\infty}\right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_\infty}\right)^{\frac{\kappa+1}{\kappa}}\right]}$$

The maximum mass flow rate of the air :

this case $\frac{p_2}{p_\infty} = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$ hence $\dot{m}_a^{max} = \rho_\infty \cdot c_\infty \cdot A_a^* \cdot \sqrt{\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} = \frac{1}{1.728} \cdot \rho_\infty \cdot c_\infty \cdot A_a^*$

The mass flow rate of the fuel as the function of the pressure difference :

$$\dot{m}_f = A_a^* \cdot \sqrt{2 \cdot \rho_f \cdot (p_\infty - p_2)}$$

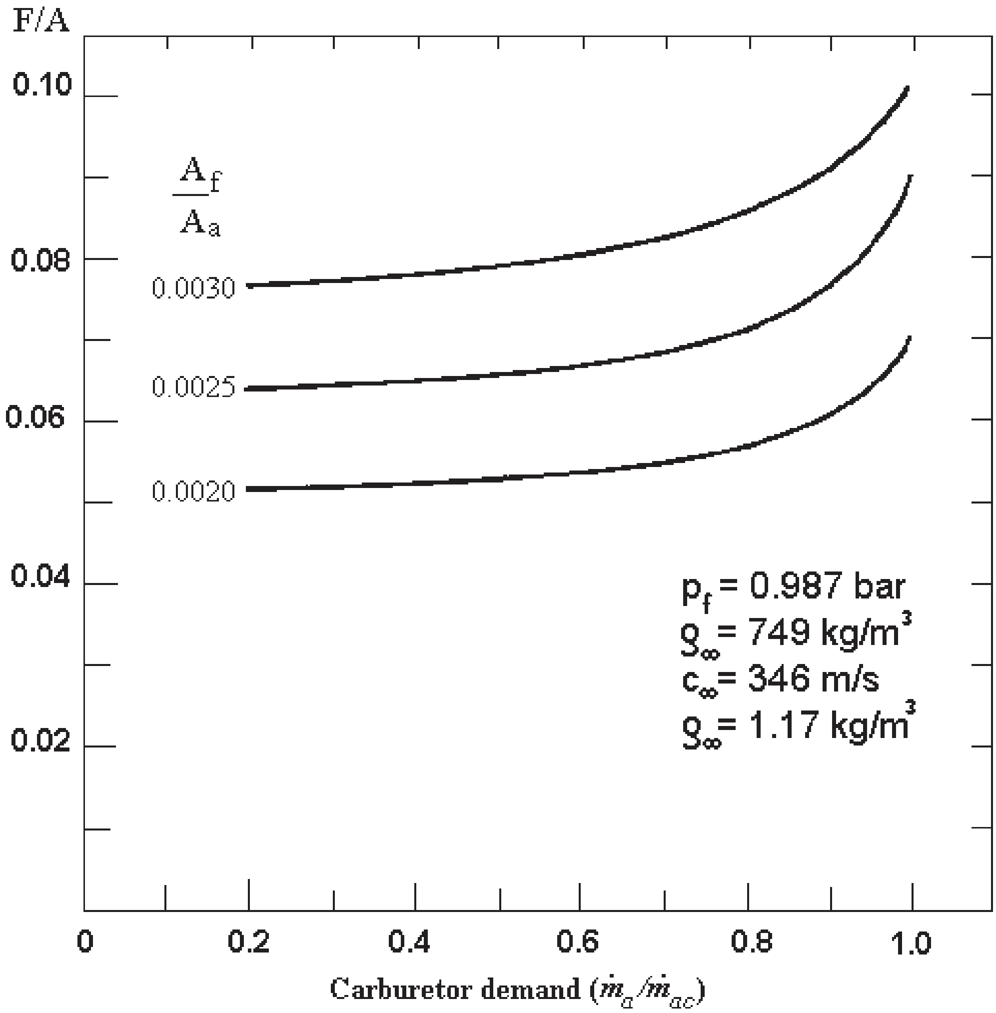
The carburetor demand :

$$D_c = \frac{\text{Actual air flow}}{\text{Critical air flow}} = 3.86 \cdot \left[\left(\frac{p_2}{p_\infty}\right)^{1.43} - \left(\frac{p_2}{p_\infty}\right)^{1.71} \right]^{1/2} \quad \text{thus } \dot{m}_a = \frac{D_c}{1.728} \cdot \rho_\infty \cdot c_\infty \cdot A_a^*$$

The fuel-to-air ratio :

$$F/A = \frac{1.728}{D_c} \cdot \frac{\rho_f}{\rho_\infty} \cdot \frac{A_f}{A_\infty} \cdot \left[\frac{2 \cdot (p_\infty - p_2)}{\rho_f \cdot c_\infty^2} \right]^{1/2}$$

The fuel-to-air ratio vs. the carburetor demand :



A typical carburettor operates over the range $0.2 < D_c < 0.8$. The D_c can be assumed constant in this region region and the F/A ratio is only the function of the pressure difference between the throat and the ambient air i.e. $F/A \sim \sqrt{(p_\infty - p_2)}$.

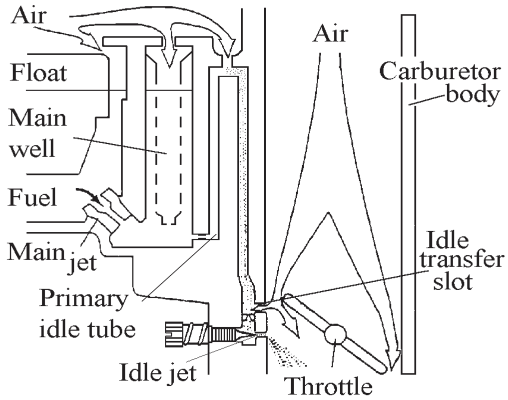
Cosequently the elementary carburettor cannot provide stabile mixture over a wide range of operation thus additional compensation circuits must be added to it.

The complex carburetor :

It was shown in the previous section that the fuel-air ratio is not a constant function of the engine's load. To compensate the bias of the fuel-air mixture the following additional circuits are added to the simple carburetor:

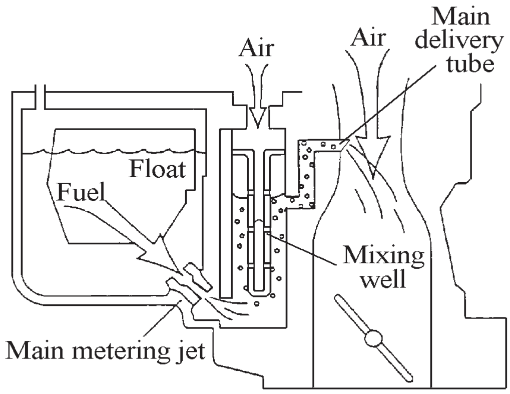
1. The idle circuit

Using simple carburetor the air-fuel mixture becomes lean close to idle engine speed therefore an idle fuel jet is added to the inlet flow.



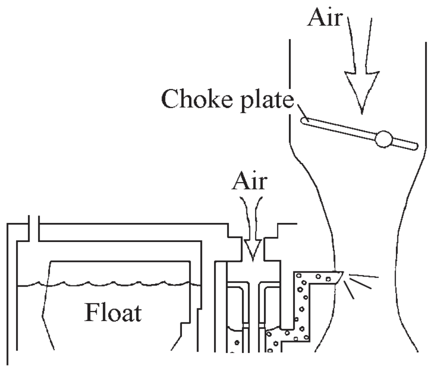
2. The main metering or high speed circuit

The mixture becomes rich near the wide open throttle (WOT) position. To compensate this bias the so called main metering circuit is applied. This circuit contains a main mixing well, which add air into the fuel delivery tube.



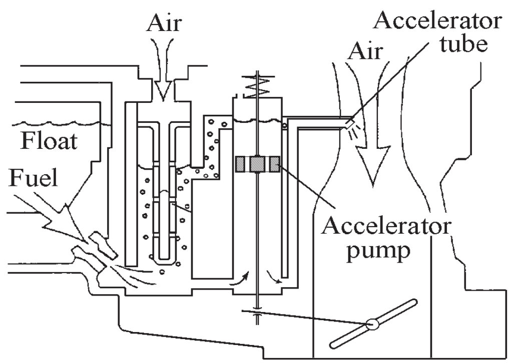
3. The choke

When the engine is cold or the temperature of the ambient air is low, the fuel does not evaporate easily thus it is difficult to ignite the mixture. The choke enrich the mixture when extra fuel is added to the inlet flow to promote the ignition process. The choke can be adjusted manually or automatically by thermostatic control.



4. The power or accelerator pump circuit

Air is very light in weight meanwhile the fuel is much heavier. When the throttle of the carburettor is opened very quickly - on a running engine - the air almost immediately rushes through the carburetor. The fuel reacts slower, the mixture becomes lean.



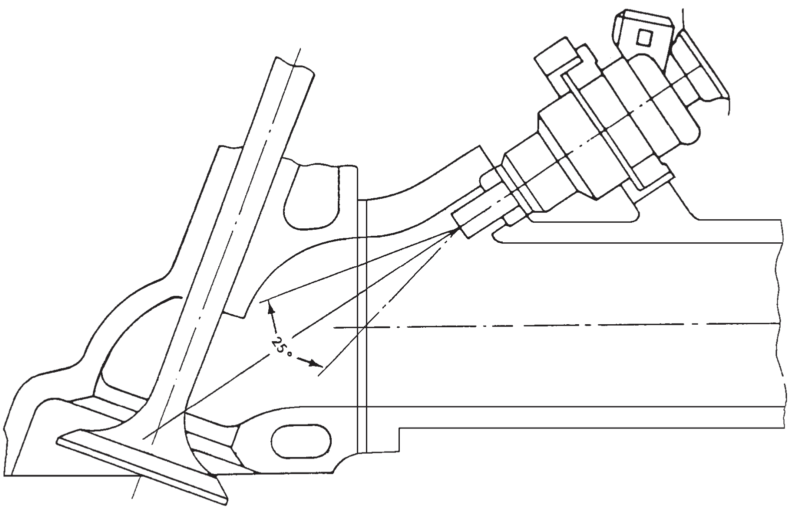
FUEL INJECTION

Fuel systems is SI engines:	**Carburation**	**Injection**
The principles:	The differences :	
- to meter the fuel flow /the propelling force/	pressure difference in venturi nozzle	pressure difference generated by fuel pump
- to mix the fuel with air	air velocity > fuel velocity	fuel velocity > air velocity

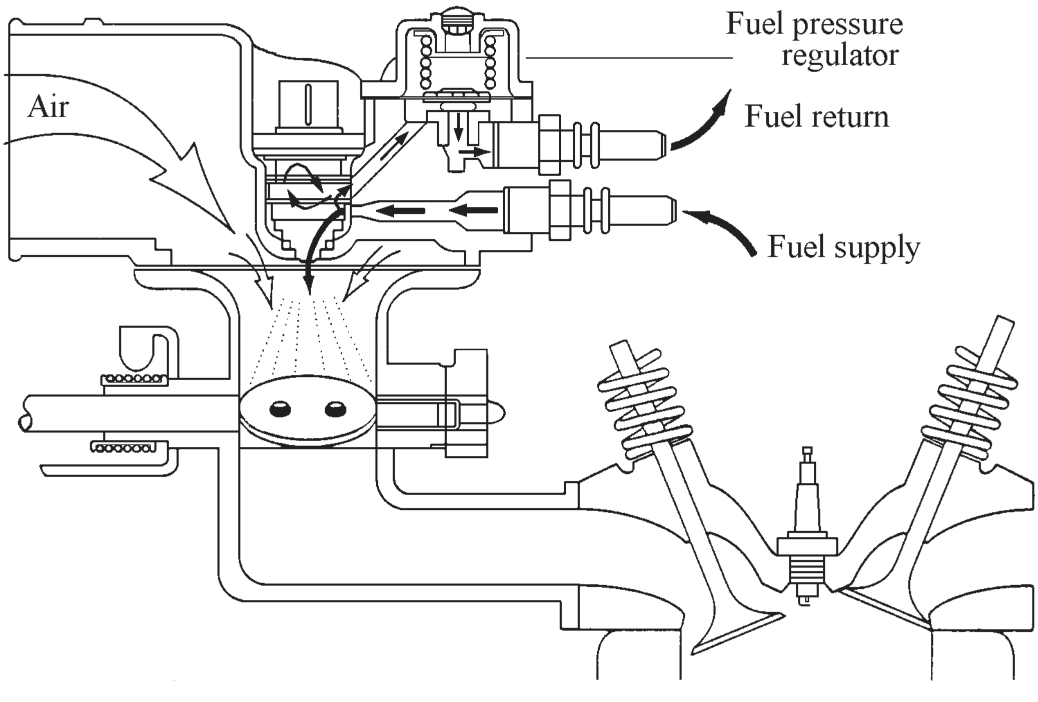
INJECTION SYSTEMS

Classification :

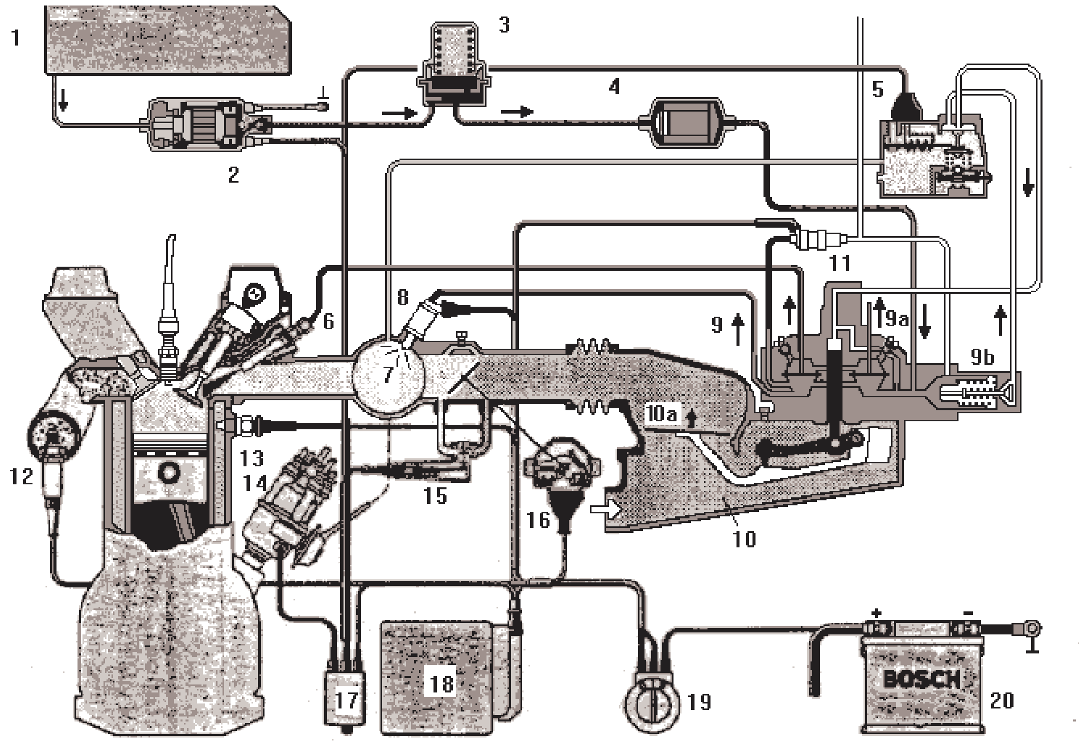
Indirect injection The fuel injector sprays into the air stream. Types: - port ~ - throttle body ~	vs.	Direct injection The fuel injector sprays directly into the combustor.
Continous injection The fuel flows continously into the air stream. Used in indirect injection systems.	vs.	Pulsed injection Each ignition has a finite duration. The system can be direct or indirect, too.
Mechanical injection Mechanical devices are applied to measure and deliver the fuel to the intake manifold or into the cylinder.	vs.	Electronic injection Electronic fuel meter and fuel delivery devices are used.
Multipoint injection One injector serves one cylinder.	vs.	Single point injection One injector serves more than one cylinder.



Port fuel-injector

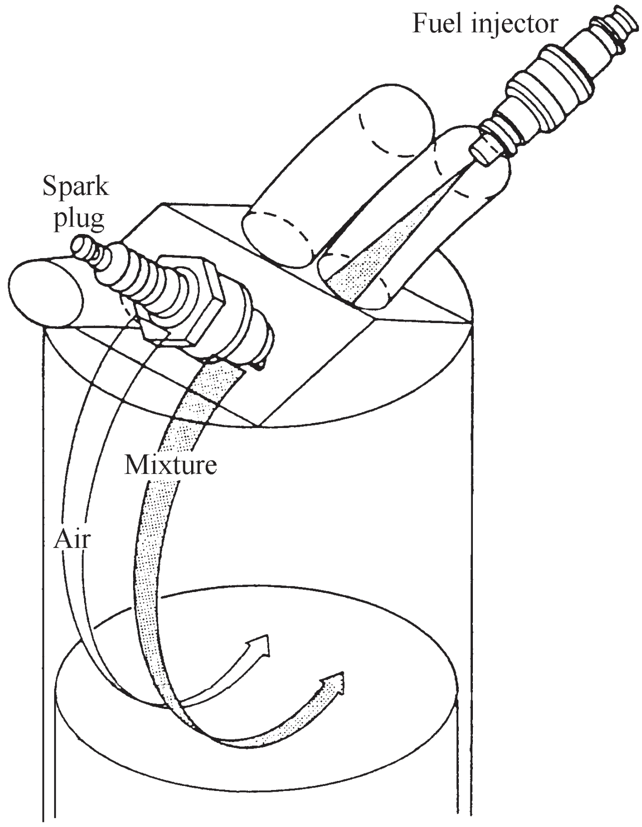


Throttle body fuel-injector



The Bosch K-Jetronic system

1. fuel tank
2. electric fuel pump
3. fual accumulator
4. fuel filter
5. warm up regulator
6. fuel-injection valve
7. intake manifold
8. start valve
9. mixture control unit
9a. fuel distributor
9b. primary pressure regulator
10. air-flow sensor
10a. sensor plate
11. frequency valve
12. Lambda sensor
13. thermo-time switch
14. ignition distributor
15. auxiliary-air device
16. throttle-valve switch
17. control relay
18. control unit
19. ignition and starting switch
20. battery



Concept of startified charge lean combustion system

The advantages of the fuel injection over the carburation

#1 Increased volumertic efficiency

There is no choking caused by the Ventury nozzle.

#2 Higher thermal efficiency

Improved mixture control because of the better conditioned fuel-air ratio

#3 Lower exhaust emission

See #2.

#4 More fuel tolerant

The distribution is independent of vaporization.

#5 Charge stratification can be achieved

Charge stratification means that the fuel air ratio is the function of the position within the combustion chamber. Therefore rich mixturte can be achieved within the vicinity of the spark plug to stabilize the ignition and the mixture is lean in the rest of the volume to reduce the "knock" phenomenon and excel higher thermal efficiency.

The disadvantages of the injection

#1 More complicated construction, consequently it is more expensive

#2 More complicated service

Elementary combustion processes

There are three modes of combustion significant in internal combustion engine work. These are:

1. *Simultaneous explosion,* which occurs when the whole of a mixture ignites as it is above its self ignition temperature. This can also occur in the unburnt portion of the mixture in a progressive combustion when the temperature rises by radiation or compression to a value in excess of the self ignition temperature.
2. *Progressive combustion* is the name given to the combustion process in which a flame front advances through a fuel air mixture. Provided that the unburned mixture does not become overheated the combustion will proceed smoothly until all the mixture is consumed and the flame is extinguished.
3. *Diffusive combustion* of the mixture produced by an evaporating droplet will occur if the mixture temperature is greater than the self ignition temperature. The rate of combustion depends on the droplet size, the sorrounding air temperature and the volatility of the fuel.

Combustion in spark ignition engines

Experiments shows that air fuel mixture can only be ignited by a spark when the mixture strength is within certain limits. These limits correspond to a condition when the energy release by combustion is not great enough to initiate combustion in the neighbouring mixture. Petrol engines will operate with air-fuel ratios which lie between 10:1 (rich) and 20:1 (weak). The stoichiometric ratio is about 14.5:1.

Normal combustion

Combustion processes may be illustrated on a pressure crank angle diagram. The central point is known as top dead centre and corresponds to the point when the piston is at rest at the end of the compression stroke. In the next figure two curves are shown, one illustrating the pressure changes when the engine is motored with no ignition and the other shows the situation when the fuel ignites and burns normally.

After the spark is made there is a delay period before any noticeable rise in pressure. The delay period is completed when the flame has become established so that progressive combustion can occur during the combustion period. In this period the flame front moves at 30-35 m/s across the chamber. It is found by experiment that the maximum work transfer occurs with a slightly rich mixture when the peak pressure occurs about 12 degrees after top dead centre. The delay period is approximately constant in time for given conditions and is of the order 0.0014 seconds, which corresponds to 17 degrees of crank angle at 2000 rev/min. The combustion period is approximately constant in crank angle. (Or rather the angle between the end of the delay period and the peak pressure is approximately constant, for the identification of the exact end of the combustion period is so difficult that the peak pressure is often used as the criterion.) These factors enable the crank angle at which the spark should occur to be estimated. It is found that the rate of rise of pressure for good combustion in an engine with moderate compression ratio is of the order 250 kPa per degree which gives a combustion period of about 20 degrees of crank angle.

Factors affecting normal combustion

- Induction pressure. Delay period increases as pressure falls and the ignition must be earlier at low pressures. A vacuum control may be fitted.
- Engine speed. As speed rises the constant time delay period needs more crank angle and the ignition must be earlier. A centrifugal control may be used.
- Ignition timing. If ignition is too early the peak pressure will occur too early and work transfer falls. If ignition is too late the peak pressure will be low and work transfer falls. Combustion may not be complete by the time exhaust valve opens and the valve may burn.
- Mixture strength. Although the stoichiometric ratio should give the best results, the effect of dissociation is to make a slightly rich mixture necessary for maximum work transfer.
- Compression ratio. An increase in compression ratio increases the maximum pressure and the work transfer.
- Combustion chamber. This should be designed to give a short flame path to avoid knock (see abnormal combustion) and it should promote optimum turbulence.
- Fuel choice. The induction period of the fuel will affect the delay period. The calorific value and the enthalpy of vaporisation will affect the temperatures achieved.

Abnormal combustion

Combustion may be slow due to excessively weak mixtures or may be mistimed. These are however obvious. There are two combustion abnormalities which are less obvious, the first of these is pre or post ignition of the mixture by incandescent carbon particles in the chamber. This will have the effect of reducing the work transfer.

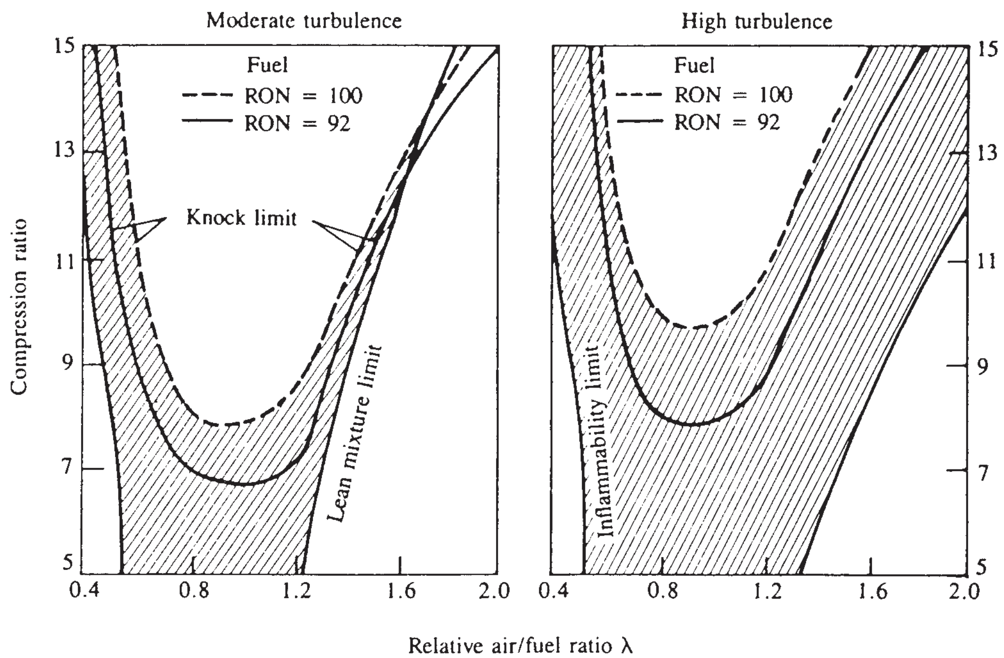
(See effect of ignition timing above.) The second abnormality is generally known as knock and is a complex condition with many facets. A simple explanation shows that knock occurs when the unburnt portion of the gas in the combustion chamber is heated by compression and radiation so that its temperature becomes greater than the self ignition temperature. If normal progressive combustion is not completed before the end of the induction period then a simultaneous explosion of the unburnt gas will occur. The explosion is accompanied by a detonation (pressure) wave which will be repeatedly reflected from the cylinder walls setting up a high frequency resonance which gives an audible noise. The detonation wave causes excessive stress and also destroys the thermal boundary layer at the cylinder walls causing overheating.

Factors affecting knock

Any reduction in the induction period of the combustion and any reduction in the progressive explosion flame velocity will increase the likelihood of knock.

- Induction pressure. Increase of pressure decreases the self ignition temperature and the induction period. Knock will tend to occur at full throttle.
- Engine speed. Low engine speeds will give low turbulence and low flame velocities (combustion period is constant in angle) and knock may occur at low speed.
- Ignition timing. Advanced ignition timing increases peak pressures and promotes knock.
- Mixture strength. Optimum mixture strength gives high pressures and promotes knock.
- Compression ratio. High compression ratios increase the cylinder pressures and promote knock.
- Combustion chamber design. Poor design gives long flame paths, poor turbulence and insufficient cooling all of which promote knock.
- Cylinder cooling. Poor cooling raises the mixture temperature and promotes knock.
- Fuel choice. A low self ignition temperature promotes knock. The constituents of a fuel affect its knock resistance and certain additives help to delay the onset of the phenomenon.

From the notes above it would appear that knock will occur if many desirable features such as optimum mixture strength and a high compression ratio are used. Knock can be reduced by avoiding extremes such as low speeds at heavy loads and full throttle running, however it is largely a matter of choosing the correct fuel for a particular engine.



THE DIESEL CYCLE

The theoretical cycle is the ideal air standard cycle for the diesel engine. The cycle's p-v and T-s diagrams are introduced in the first chapter.

THE THERMODYNAMIC RELATIONS OF THE IDEAL DIESEL CYCLE :

Compression stroke	$\frac{v_2}{v_1} = \varepsilon^{-1}$	$\frac{p_2}{p_1} = \varepsilon^{\kappa}$	$\frac{T_2}{T_1} = \varepsilon^{\kappa-1}$	$W_{comp} = m \cdot c_v \cdot (T_2 - T_1)$
Heat addition	$\frac{v_3}{v_2} = \frac{T_3}{T_2}$	$p_3 = p_2$		$Q_{in} = m \cdot c_p \cdot (T_3 - T_2)$
Expansion stroke	$\frac{v_4}{v_3} = \varepsilon$	$\frac{p_4}{p_3} = \varepsilon^{-\kappa}$	$\frac{T_4}{T_3} = \varepsilon^{1-\kappa}$	$W_{exp} = m \cdot c_v \cdot (T_3 - T_4)$
Heat rejection	$v_4 = v_1$	$\frac{p_4}{p_1} = \frac{T_4}{T_1}$		$Q_{out} = m \cdot c_v \cdot (T_4 - T_1)$

The **theoretical efficiency** of the cycle is determined as follows

$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^{\kappa} - 1}{\kappa \cdot (\rho - 1)}$$

where: ε - the compression ratio

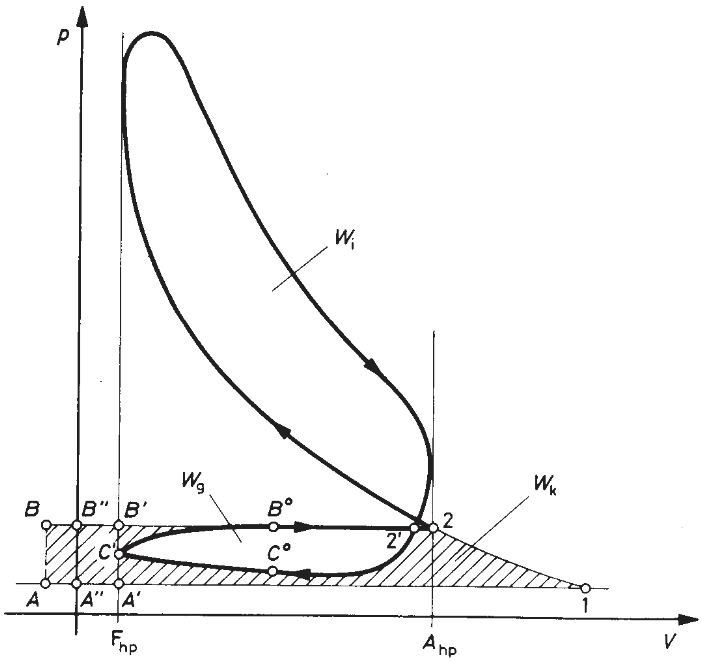
$\rho = \frac{v_3}{v_2}$ - cut off ratio

Differences between theoretical and real processes

The main differences are the following:

- These cycles are reciprocating cycles, not steady flow cycles, like the Joule cycle in gas turbine, which means that in every cycle a certain amount of fluid takes part.
- In the theoretical cycle there is not chemical changes, the heat is exchanged from externally to the cycle. In the real process the heat is supplied by combustion in the engine.
- Because the combustion takes place inside the engine, fuel and air have to induct into the engine, and combustion products must be exhausted. So the real process consists of induction and exhaust processes together with the processes which take place in the theoretical cycles.
- Compression and expansion processes are irreversible
- Combustion running not exactly at constant volume or pressure
- There is heat exchange from the working fluid to the walls
- There are fluid flow friction losses at the induction and exhaust process.

Next Fig shows a p-V chart of a real Compression Ignition cycle.



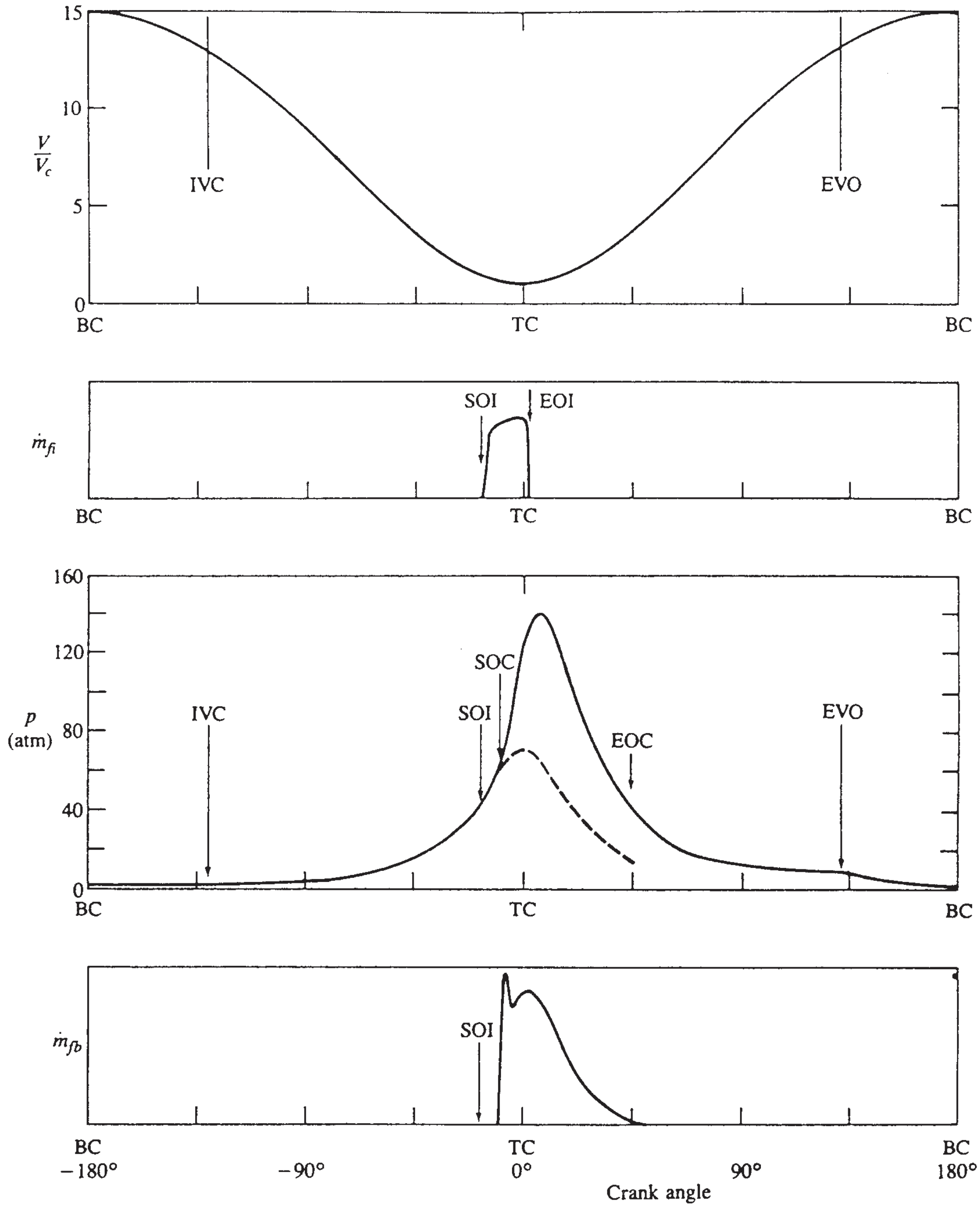
Compr ession-ignition engine operation

In Compression-ignition engines, air alone is inducted into the cylinder.

The fuel is injected directly into the engine cylinder just before the combustion process is required to start. Load control is achieved by varying the amount of fuel injected each cycle. The air flow at a given engine speed is essentially unchanged.

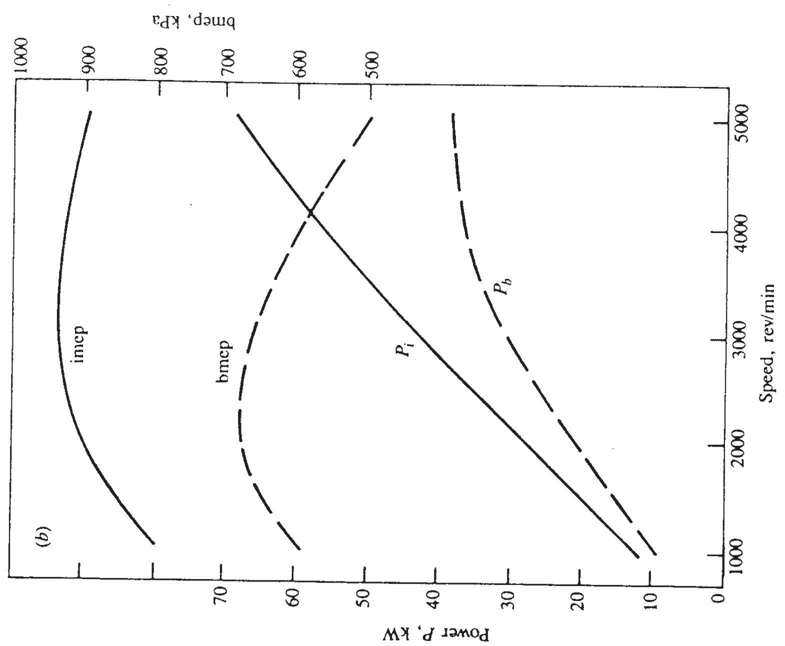
There are a great variety of IC engine designs in use in a wide range of applications, as automobile, truck, locomotive, marine, power generation.

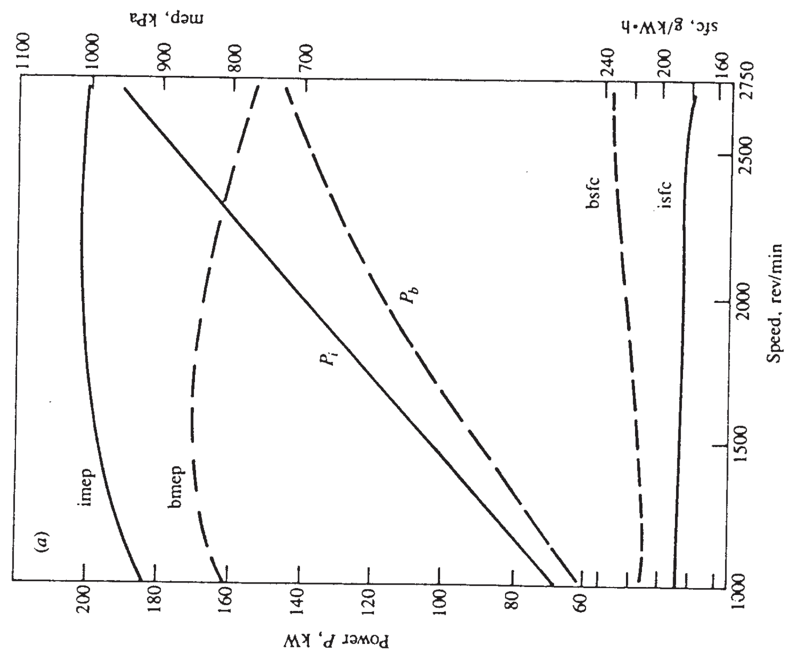
Next figure shows sequence events duringcompression, combustion, and expansion processes of a naturally aspirated compression-ignition engine operating cycle.



Next two figs show characteristic curves of two different diesel engine:

1. V_d = 8400 cm^3 , six-cylinder naturally aspirated direct injection diesel engine
2. V_d = 1800 cm^3 , four-cylinder naturally aspirated indirect injection swirl-chamber diesel engine





Combustion in compression ignition engines

In the compression ignition engine a stoichiometric air-fuel ratio cannot be used. This is because the fuel and air are not premixed. At a constant speed the mass of air taken in to the cylinder of a compression ignition engine is substantially constant irrespective of the power output and the fuel which is sprayed into the cylinder must find oxygen in order to react. A great deal of effort has been made to design combustion chambers to enable as much fuel as possible to be burned and it is unlikely that it will ever be possible to design a chamber that will allow all the air in a cylinder to react. The richest air-fuel ratio that a compression ignition engine can burn is about 18:1. At the other end of the scale there is no problem in burning small amounts of fuel and air-fuel ratios as low as 100:1 may be used under no load conditions.

The compression ratio of the engine must be sufficient to raise the temperature of the air to a value above the self ignition temperature of the fuel. The minimum useful compression ratio is about 12:1 and the maximum for engines of moderate design 20:1.

Normal combustion

The combustion process is shown on a pressure crank angle diagram with a motoring curve superimposed. After injection commences there is a delay period which is longer than the corresponding period in the spark ignition engine. During the delay period the fuel evaporates and mixes with the air. The induction period of the fuel is the prime factor in determining the length of the delay period. At the end of the delay, simultaneous explosion of the fuel air mixture in the cylinder occurs and there is a rapid pressure rise (500 kPa per degree). Injection has not yet ceased and there follows a period of diffusive combustion when the fuel droplets burn. Diffusive combustion does not stop at the instant that injection finishes for there will be a number of unburned droplets of fuel in the cylinder.

Factors affecting combustion

The length of the delay period is the major factor in compression ignition combustion. The period serves a useful purpose in that it allows the fuel jet to penetrate well into the combustion space. If there were no delay the fuel would burn at the injector and there would be an oxygen deficiency around the injector resulting in incomplete combustion. If delay is too long the amount of fuel available for simultaneous explosion is too great and the resulting pressure rise too rapid. Delay is reduced by a high charge temperature, a high fuel temperature, good turbulence, and a fuel with a short induction period.

Combustion chamber design.

As mentioned earlier much effort has been made to produce good combustion chamber designs. The basic requirement is to enable the fuel to find the air. The shape of the chamber is critical so that the fuel is introduced with swirl and the pistons are also shaped to produce a 'squish' effect, the whole concept of design being to give controlled motion of the air and fuel rather than rely on haphazard turbulence.The fuel is sometimes injected into a separate chamber from which it rushes into the combustion space in a controlled direction already partly mixed with air. The topic of compression ignition engine combustion chamber design is too complex for discussion here.

Engine load.

In most engines injection starts at a fixed crank angle and continues as long as the rack setting allows. at light loads all the fuel will be injected in the delay and simultaneous explosion periods, or at very light loads when temperatures are lower and the delay period longer injection may be completed in the delay period.

Engine speed.

The effect of speed is allied to the design of the combustion chamber and varies with individual designs. Delay may be reduced or increased.

Fuel choice.

The choice of fuel has a large effect on combustion since the induction period of the fuel contributes a major part of the delay period.

Abnormal combustion

This is not as great a problem in compression ignition engines as in spark ignition engines. The only abnormality is diesel knock. This occurs when the delay period is excessively long so that there is a large amount of fuel in the cylinder for the simultaneous explosion phase. The rate of pressure rise per degree of crank angle is then so great that an audible knocking sound occurs. Running is rough and if allowed to become extreme the increase in mechanical and thermal stresses may damage the engine. Knock is function of the fuel chosen any may be avoided by choosing a fuel with characteristics that do not give too long a delay period.

Fuels for Diesel engines

Diesel engines are compression ignition engines, which means that fuel is injected to the cylinder where fuel is ignited by hot air. Air is heated up because of the compression.

Inclination to ignition

Inclination to ignition very important feature of diesel fuels.
To measure this feature were lead the cetan number. This number is very similar to octan number at petrol.

The comparision mixture consist of cetan and α-metil-naphtaline.

The cetan number shows that the fuel inclination to ignition is the same as at a mixture of cetan and α-metil-naphtaline where the cetan number shows the volume percentage of cetan.

The relation between the self-ignition temperature and the cetan number is the following.

$$t_{self\text{-}ignition} = 310 - 0.75 \times \text{cetan-number} \quad [°C]$$

Influence of the cetan number to the engine operation:

- pressure increasing speed at burning
- peak pressure of burning
- specific fuel consumption
- exhaust gas temperature
- soot contain of exhaust gas
- deposit inside the cylinder and piston surface

There has been invented that there are relation between cetan number and other phisical properties of the fuel. That is why cetan number can be determined by means of the following iequation:

$$\text{cetan number} = 454.74 - 1641.41 * \rho + 774.74 * \rho^2 - 0.554 * t_b + 97.803 * (\lg t_b)^2$$

where: ρ - density of the fuel at 15 °C

t_b - mean boiling temperature of the fuel

Other important feature the fraction distribution of the fuel.

Fractions which have lower boiling point produce sufficient fuel-air mixture easy, but too muvch of them cause too high pressure peak and knocking operation.

Fractions which have too high boiling point (over 350 °C) cause deposit in the cylinder and soot in the exhaust gas.

Viscosity of the fuel

Atomization and mixture formation need low viscosity, on the other hand fuel supply as lubricating oil of the injection system, that is why the viscosity of the fuel has to be in a certain range.

Environment protection

To reduce pollutant in the exhaust gas the sulfur content of the fiuel has to be as low as possible. Sulfur-dioxid and -trioxid can form sulfur-acid with water.

This acid not only pollut the air, but also demage the engine.

The Conradson-number of the fuel, which shows the ash content and inclination to soot formation, has to be at low level.

Ambient temperature conditions

Diesel engines have to be started and operated at low ambient temperature as well.

At that time very important that the solidification temperature has to be under the ambient temperature.

To reach this generally appropriate additives are given to the fuel.

Rated data of diesel fuel accordint to the hungarian satandard

Features	**A quality**	**B quality**	**C quality**
Density	0.815 - 0.860 kg/l		
Cetan number	45	42	42
Boiling point - where 50% evaporated	300 °C	290 °C	280 °C
- where 90% evaporated	370 °C	370 °C	-
- where 96% evaporated	-	-	350 °C
Kinematic viscosity at 20°C	2.5 - 8.0 mm^2/s	2.5 - 8.0 mm^2/s	2.3 - 6.0 mm^2/s
Solidification temperature max.:	0 °C	- 10 °C	- 20 °C
Sulfur content max.:	0.2 % (at reduced Sulfur content) 0.5 % (at normal Sulfur content)		
Conradson number max.:	0.2 %		
Watercontent max.:	0.025 %		

INJECTION SYSTEMS IN COMPRESSION-IGNITION ENGINES

In compression-ignition engines air is introduced into the cylinder. After compression fuel is injected into the combustion chamber by the fuel injection system.

The injection can be direct (the fuel is injected into a single open combustion chamber) and indirect (the combustion chamber consists of regions, a main chamber and prechamber where the fuel is injected into).

The injection system must fulfil the following requirements:

1. The diesel fuel injection systems must be high pressure for two reasons:
   - the fuel pressure must be greater than the compression pressure
   - the fuel should be evaporated quickly therefore small droplet size is desirable which is the function of the injection pressure
2. The injection system must be able to time both the start and the length of the injection process.
3. The injection system must be able to meter the fuel flow. It is essential to supply the right proportion of fuel to the engine for the sake of the best achievable performance.
4. The injection process should be free of pulsation avoiding pressure fluctuation.
5. The injection of bulk fuel should not be followed by subsequent small secondary injections. This phenomenon causes high hydrocarbon, soot and particle emission and efficiency decrease.

To accomplish these tasks, fuel is usually drawn from the fuel tank by a supply pump and forced through a filter to the injection pump. The injection pump sends fuel under pressure to the nozzle pipes which carry fuel to the injector nozzles located in each cylinder head.

The injection pressure is between 300 and 1200 bar.

THE INJECTION PUMPS

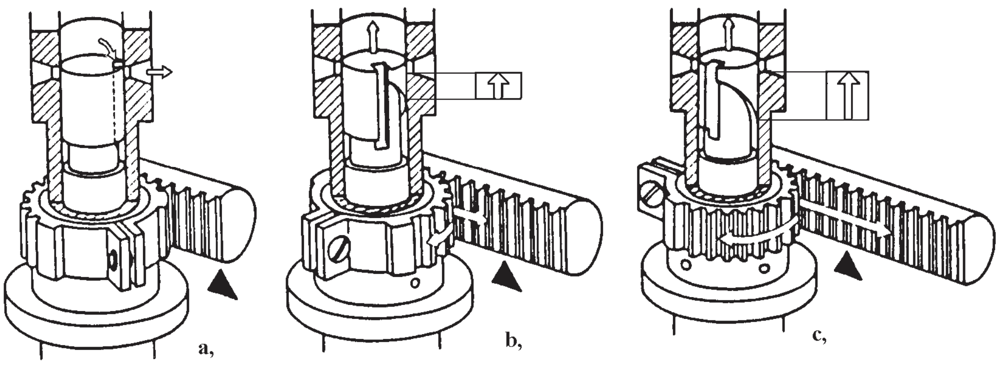
The injector pumps can be classified as:

1. In-line injection pump (a set of cam driven plungers, one for each cylinder). They are used in engines in the 40kW to 100kW per cylinder power range.
2. Distributor-type fuel-injection pump (only one pump plunger and barrel)
   They are used in engines with less than 30kW per cylinder power. These pumps are usually smaller.
3. Single-barrel injection pumps are used in small (one or two cylinder) and large (over 100kW power/cyl.) engines. The injection pump and the injector nozzle are built into a singe unit.

Each type must supply fuel
- at the right time
- at a proper pressure
- in a right proportion

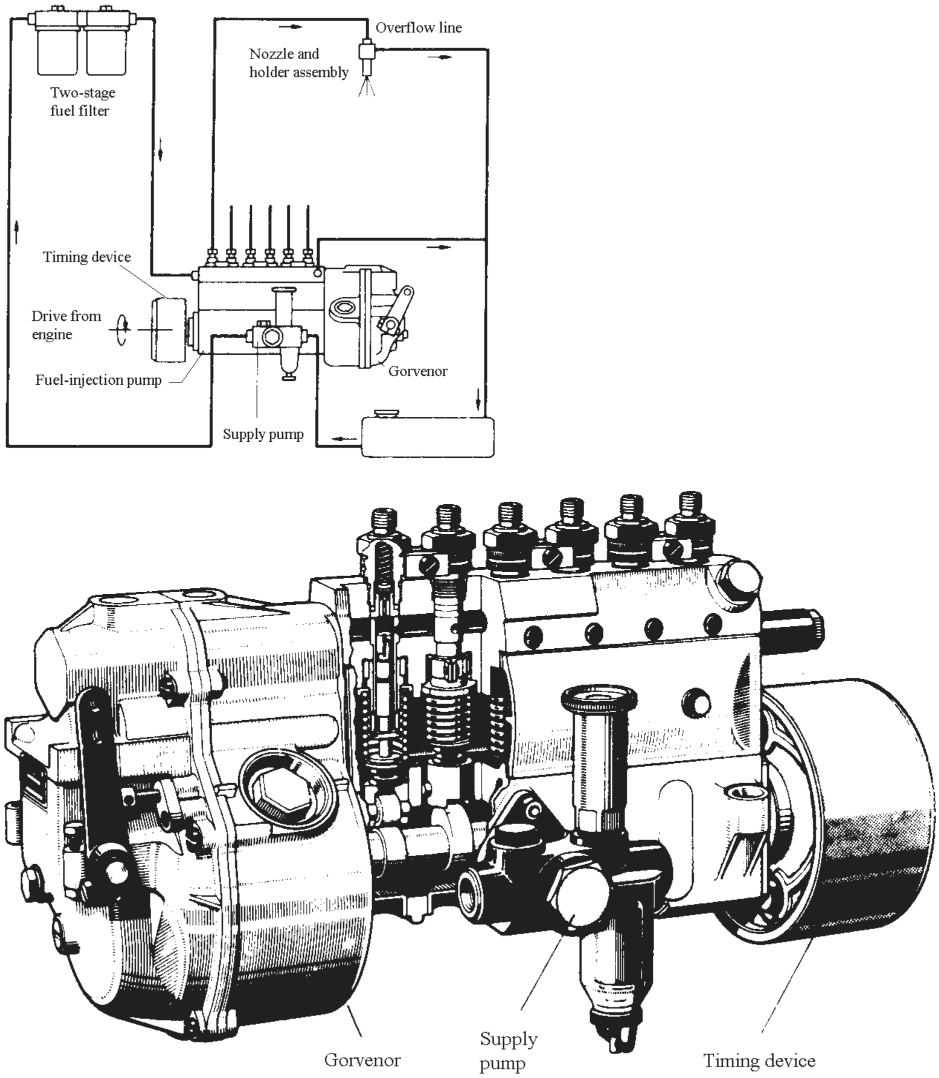
The injection is timed by timing device which is governed by the engine speed. Higher engine speed requires higher injection-advance angle similarly to the spark advance in SI engines.
The injection pressure is provided by the injection pump and the opening pressure limit is determined by the resistance of the spring in the nozzle.
The amount of fuel is controlled by the injection pump. The stroke of the pump's plunger can be constant or adjustable.

An example for the constant plunger stroke is the Bosch's injection system shown below.

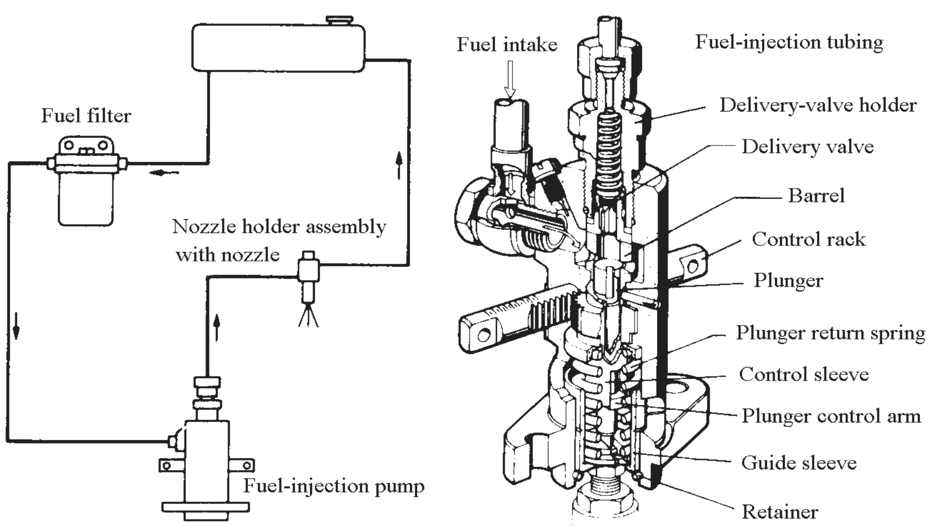


a, stop   b, part load   c, full load

Here the amount of fuel is controlled by a helical groove which opens the overflow (and inlet) port.



Diesel fuel system with in-line fuel-injection pump

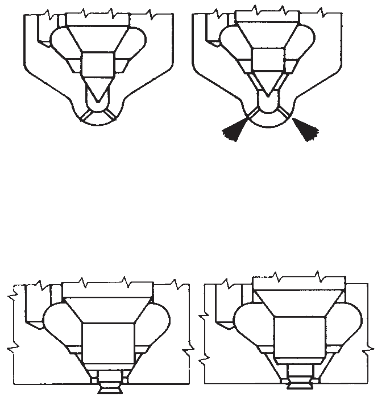


Fuel injection system with single-barrel pump

THE INJECTOR NOZZLE

This is the most essential part of the injection system.

The needle valve which controls the injection usually driven hydraulically by the pressurised fuel. The injection nozzle has one or more holes through which the fuel sprays into the cylinder. These holes are closed by a needle valve. When the pressure of the fuel exceeds the minimum injection pressure limit within the pressure chamber, the spring-loaded valve opens and fuel spray penetrates into the chamber. When the pressure drops the spring closes the valve.



Injector nozzles closed and open

The main tasks of the injector nozzle :

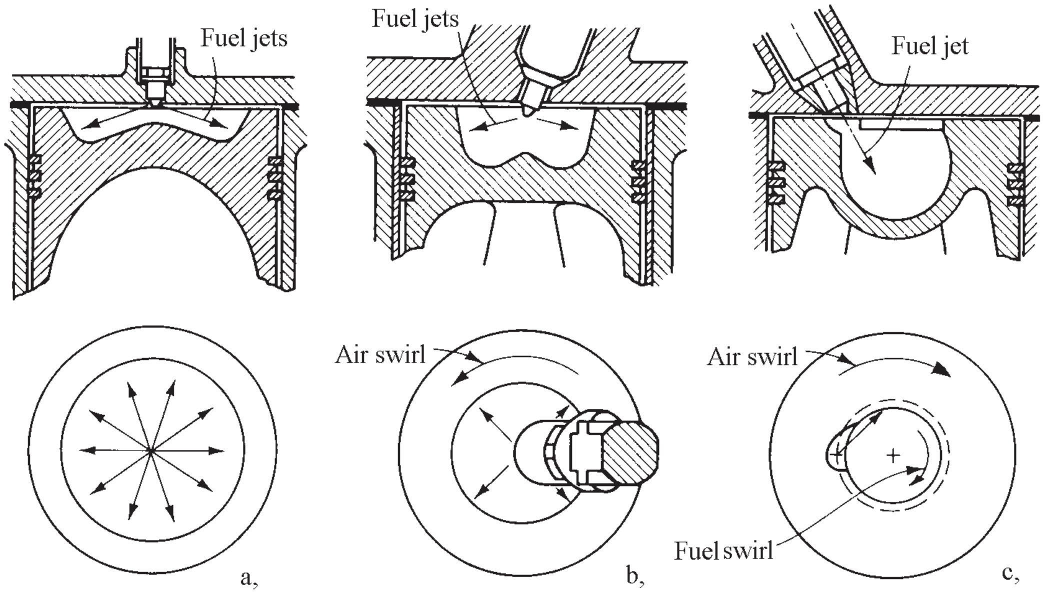
1. Provide homogeneous fuel jet
2. Guaranty satisfactory droplet size
3. Promote fuel jet development (structure, shape, penetration)
4. Adjustable opening and closing pressure
5. Prevent from secondary injection
6. Avoid of back flow

THE COMBUSTION CHAMBER

There are two different family of combustion chamber designs, the single open (direct injection) and the divided chamber (indirect injection).

The single open combustion chamber

This chamber design is usually used in large-size engines, where sufficient mixing rate is achievable without using special chamber pattern. In smaller, medium-size engines, where faster burning is required tangential air and/or fuel inlets promote the formation of the high swirl rate within the chamber.



Quiescent chamber design with multihole nozzle

Bowl-in-piston chamber design with multihole nozzle

Bowl-in-piston chamber with multihole nozzle and air and fuel swirl

HESSELMAN CHAMBER (b)

Shallow combustion chamber with multihole (5-8) injector nozzle

Mixture formation and combustion are governed mainly by the air swirl - inlet ducts and special chamber shape

MAN-M (MAURER) CHAMBER (c)

Deep combustion chamber with single-hole injector nozzle

Most of the fuel is injected on the wall of the piston bowl

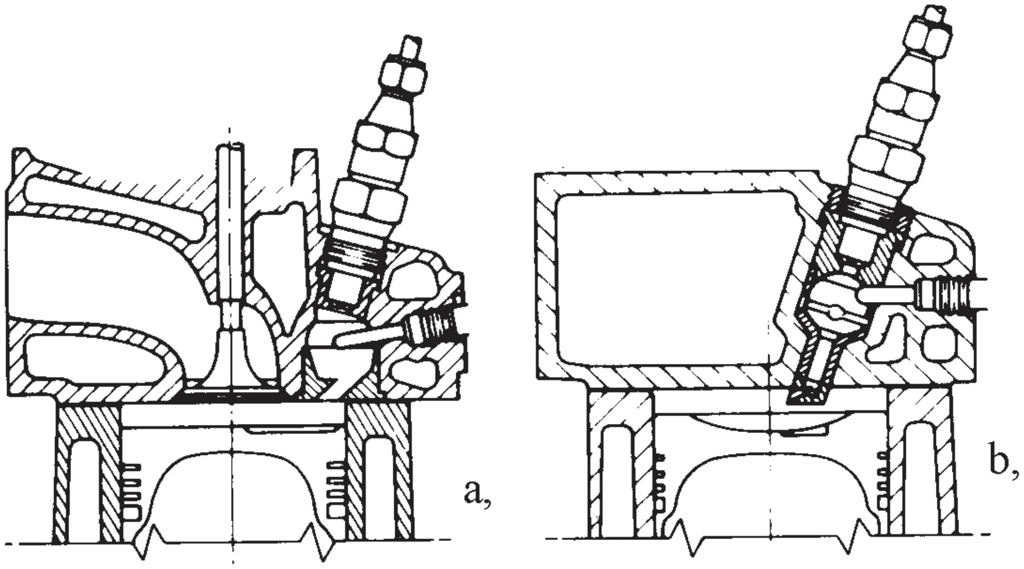
Mixture is formed from the evaporating fuel film and the high swirl air

Reduced hydrocarbon and carbon particle emissions are achieved

The divided combustion chamber

In medium class high speed diesel engines the air swirl cannot provide sufficient mixing rate. Divided combustion chamber or indirect injection stystem is applied for these engines for the benefiit of maximum power extraction. This chambers can be classified as:

1. swirl chamber systems (a)
2. prechamber systems (b)



In both systems the fuel injected into an auxiliary chamber where fast combustion is achieved by burning rich mixture. The ignition and combustion within the auxiliary chamber followed by the expansion of the burned and unburned gases which stream into the main chamber where secondary combustion takes place. In swirl chamber systems the air within the auxiliary chamber rotates rapidly, thus more intense combustion is attained.

Comparison of the divided combustion chamber to the single-open chamber

Advantages

- higher power/volume ratio
- lower maximum pressure
- easier ignition
- reduced emission

Disadvantages

- lower economy
- cold start problems at low ambient temperature

GAS EXCHANGE PROCESSES

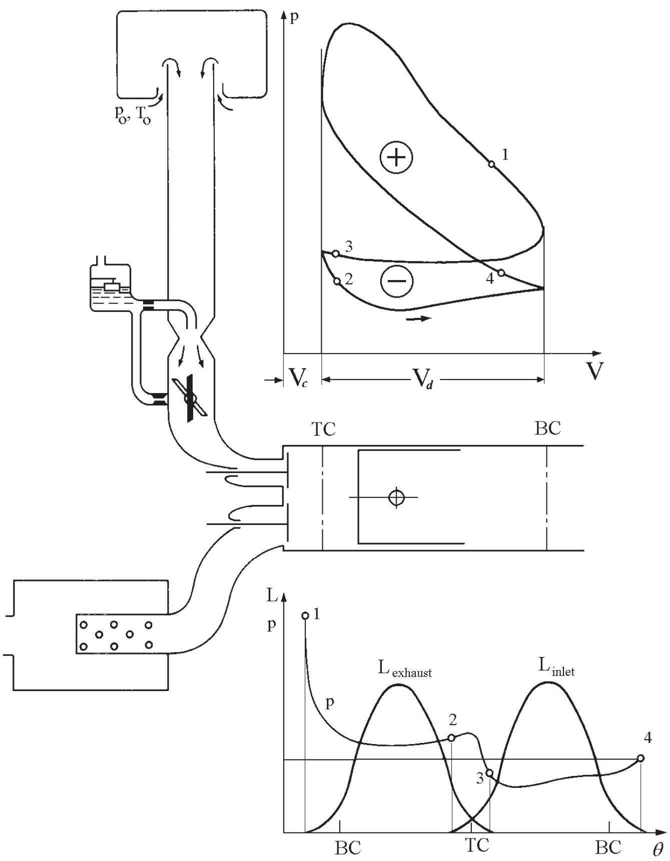
NATURAL ASPIRATION

In a spark-ignition engine, the intake system typically consists of an air filter, a carburettor and throttle or fuel injector and a throttle of throttle with individual fuel injectors in each intake port, and intake manifold.

Each element of the aspiration system causes pressure drop of the air stream. This drop along the intake system depends on the engine speed, the flow resistance of the elements in the system, the cross-sectional area through which the the fresh charge moves, and the charge density. The pressure drop in the exhaust system depends on the engine speed, the flow resistance of the exhaust manifold, exhaust pipe, silencer and the catalytic converter.

Intake and exhaust process in a fou
cycle spark-ignition engine.

p_o	ambient pressure
T_o	ambient temperature
L	valve lift
BC	bottom center
TC	top center
1	exhaust valve opens
2	exhaust valve closes
3	inlet valve opens
4	inlet valve closes
⊕	extracted work
⊖	pumping loss



The usual practice is to extend the valve open phases beyond the intake and exhaust strokes to improve emptying and charging of the cylinders and make the best use of inertia of the gases in the intake and exhaust system. The exhaust process usually starts about 40 to 60° before BC (1) and closes between 15 and 30° after TC (2). The inlet valves opens 10 to 20° before TC (3) and closes 50 to 70° after BC (4). When both of the inlet and exhaust valves are open i.e. period between (3) and (2) called the *overlap period.* The advantage of the overlap period occures when the engine operates at high speed and the longer valve-open period improve volumetric efficiency.

THE VOLUMETRIC EFFICIENCY

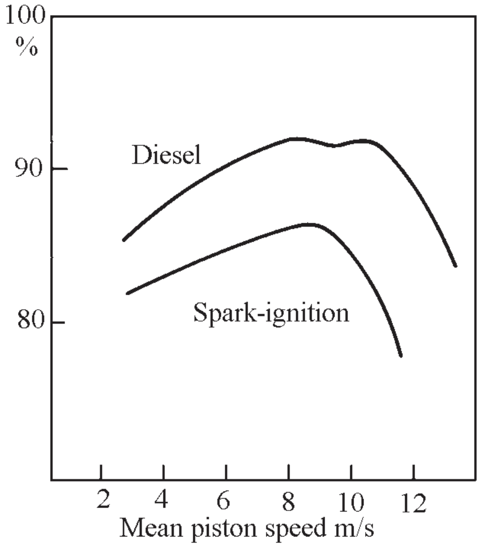
Volumetric efficiency is used as an overall measure of the effectiveness of a four stroke cycle engine and its intake and exhaust system as an air pumping device. It has been defined as :

$$\eta_V = \frac{\dot{V}}{\dot{V}_s} = \frac{\dot{m}_a + \dot{B}}{V_s \cdot \rho_i \cdot n \cdot i} \quad [-]$$

Volumetric efficiency is affected by the following variables :

1. Fuel type, fuel-to-air ratio, fraction of fuel vaporized in the intake system, and fuel heat of vaporization
2. Mixture temperature as influenced by heat transfer
3. Ratio of exhaust to inlet manifold pressures
4. Compression ratio
5. Engine speed
6. Intake and exhaust manifold and port design
7. Intake and exhaust valve geometry, size, lift and timings

The values of the volumetric efficiency vary typically between 80 and 90%. Usually it is higher for compression-ignition engines than in the spark-ignition ones (see the figure below).



Volumetric efficiency vs. mean piston speed for a four-cylinder automobile indirect-injection diesel and a six-cylinder spark-ignition engine.

THE SUPERCHARGING

The power that can be extracted from engine is limited by the amount of fuel can be burned efficiently within the combustion chamber. This is limited by the amount of air being introduced into the cylinder each cycle. If the inducted air compressed to a higher density than ambient, prior to entry into the cylinder, the maximum power of the pressure of the inlet air comparing to the ambient one the delivered power of the engine will be increased.

The term supercharging refers to increasing the air density by increasing its pressure prior to entering the engine cylinder. There are three basic methods used to accomplish this task:

1. **mechanical supercharging**

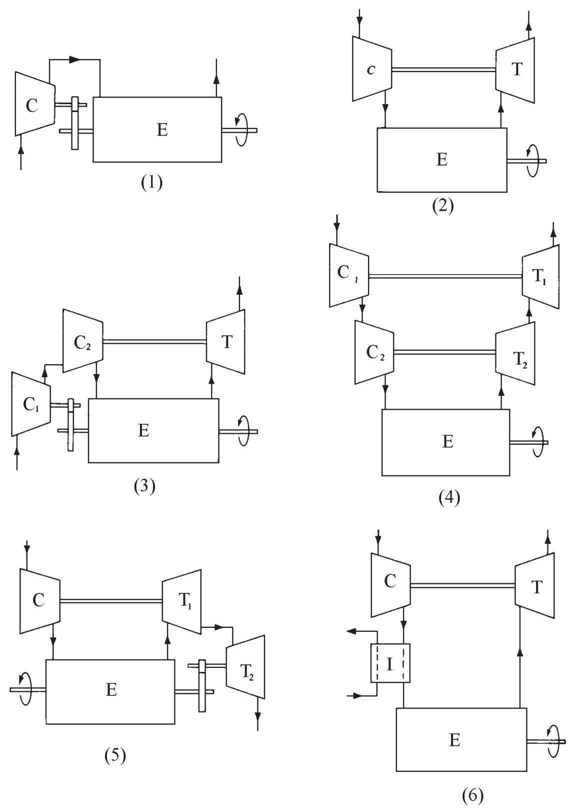
   The compressed air is provided by a compressor or a blower or a pump driven by power taken from the engine.

2. **turbocharging**

   The compressed air is supplied by a compressor powered by a turbine. The turbine is built into the exhaust system and extracts the available energy of the exhaust gases.

3. **pressure wave supercharging**

   This method uses the wave action in the intake and the exhaust pipe system to increase the volumetric efficiency and compress the intake mixture. (COMPREX)



Supercharging configurations :

(1) mechanical supercharging

(2) turbocharging

(3) engine-driven compressor and turbocharger

(4) two-stage turbocharging

(5) turbocharging with turbocompounding

(6) turbocharger with inter cooler

C Compressor

E Engine

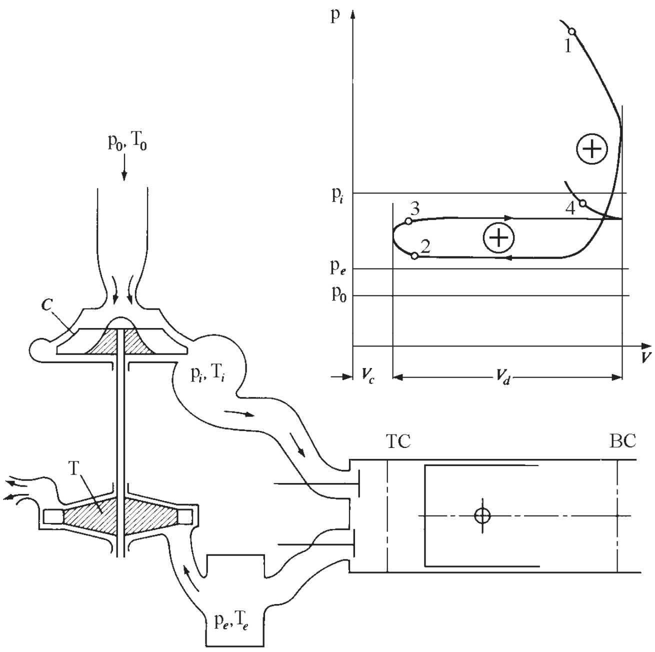
I Inter-cooler

T Turbine

TURBOCHARGING

Intake and exhaust process for four-strok engine.

The turbocharger compressor C raises air pressure and temperature from ambient p_o,T_o to p_i,T_i. Cylinder pressure during intake is less than p_i. During exhaust, the cylinder gases flow through the exhaust manifold to the turbocharger turbine T. Manifold pressure may vary during the exhaust process and lies between cylinder pressure and ambient.



TC top center

BC bottom center

⊕ extracted work

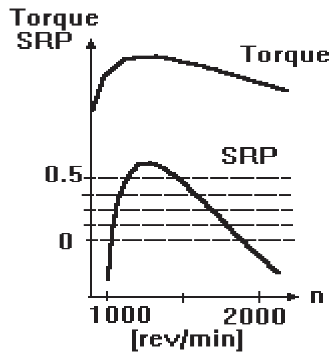
The most common turbocharger is a single shaft free turbine where centrifugal (radial) compressor is powered by radial turbine (or axial turbine in large engines). This arrangement is the most compact, and able to utilize high mass flow rates.* The turbine rotates at high speed to achieve high efficiency thus direct drive of the compressor is more beneficial than mechanical coupling through gearbox.

There are two way of extracting available energy from the exhaust gases :

1). constant pressure turbocharging (used in passenger cars)

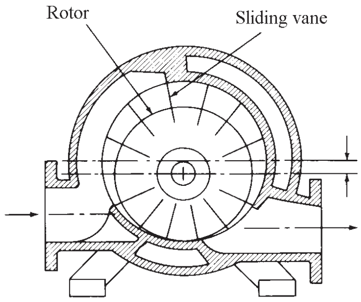
2). pulse turbocharging (used in large diesel engines)

The so called *scavenging pressure ratio* (SPR) is the parameter that refers to the effectiveness of the turbocharger. The SPR is the difference of the pressure ratio of the compressor and the turbine. When it is bigger than zero i.e. the pressure is higher in the inlet manifold than in the exhaust the pumping work is positive (extractable work). Where it reaches the maximum, the additional torque derived from the turbocharger is the greatest, too.

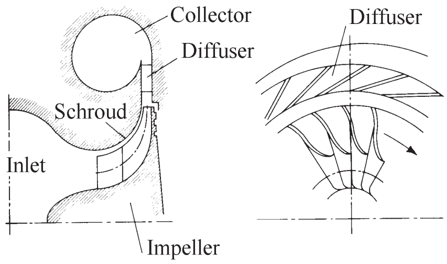


Note, that the turbocharging described above works efficiently only at a certain engine speed. To match the compressor and the turbine engine the operating conditions must be known. For example the turbine maximum SPR should be close to the maximum power point in a race car engine and below the half load point in a city-bus engine.

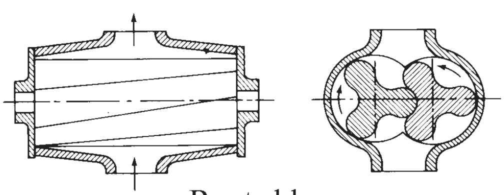
* Detailed description on gas turbines can be found in the Gas Turbines lecture note.



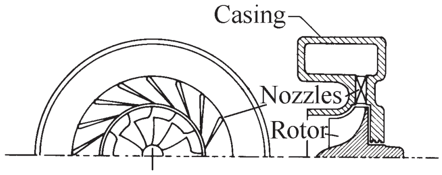
Sliding vane compressor



Centrifugal compressor

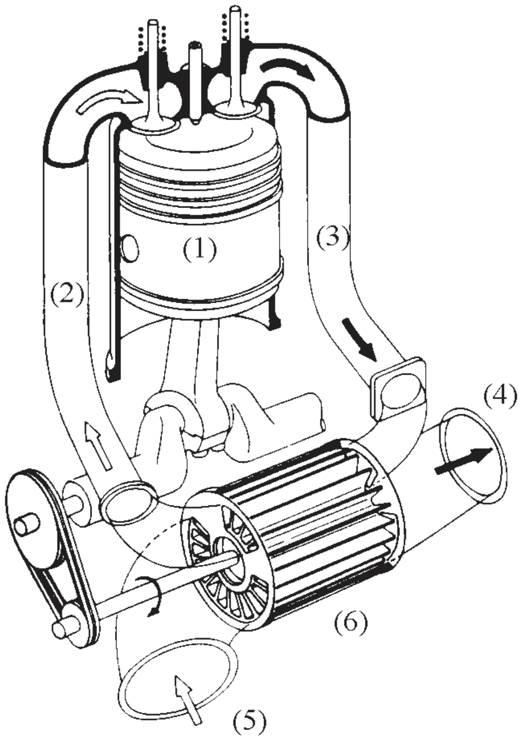


Roots blower



Rad

ial flow turbine



Schematic of COMPREX supercharger

(1) Engine

(2) High-pressure air

(3) High-pressure exhaust gas

(4) Low-pressure exhaust gas

(5) Low-pressure air

(6) Cell wheel or rotor

THE POLLUTANTS AND EMISSION CONTROL

THE POLLUTANTS

The SI engines have several pollution emitting sources (eg: exhaus pipe, crank shaft, brake system, etc..). The exhaust pipe emission is the most important.

The major polluting materials within the exhaust geases are the following:

#1 Unburned hidrocarbons (CH)

It damages the brain cells and the breathing organs.

It is the result of the lack of oxigen or the uncompleted combustion process.

#2 Carbonmonoxide (CO)

The inhalation of this gas can cause suffocation.

It is the result of the lack of oxigen or the uncompleted combustion process.

#3 Nitric oxides (NOx)

They attack the breathing organs and are responsible for the allergy.

There are two dominant nitric oxides the nitrogen-monoxide (NO) and the nitrogen-dioxide (NO2) which is more agressive. The NO2 is created of NO by the sun light. Mainly NO is produced during the combustion process. The classification of the NO is the following:

- thermal NO

Nitric oxide produced by the dissociation. This is a slow reaction on low temperatures and speeds up in the higher region.

- prompt NO

This is the result of a fast reaction in rich mixtures where the hydrocarbons react with the nitrogen.

- fuel NO

This product originates from the nitrogen carried by the fuel.

#4 Carbon dioxide

This is a so called "hot house gas", experts state that this gas is one of those which are responsible for the global warming.

#5 Heavy metals

The lead is responsible for several deseases, damage the brain and lethal in big dose.

Most of the fuels contain lead originated from the crude oil and some fuels containing lead as an additive, which reduces the knock phenomenon.

EMISSION CONTROL

The following techniques are applied for reducing the exhaust pipe emission:

#1 Design of the combustion process

Combustion chamber

The surface of the combustion chamber cools the flame, which causes quenching (i.e.: interrupted reactions). Thus the smaller surface to volume ratio the combustion chamber has the lower CH emission occurs.

Charge stratification

A common way to alter the combustion process is the usage of the charge stratification.. Its principle and advantages have been discussed before.

Swirl

The higher turbulence causes faster flame propagation and reduced combustion time. Therefore less thermal NO is produced.

#2 Optimisation the operating parameters

The following parameters have the strongest influence on the exhaust emissions:

- fuel - air ratio

- spark advance

Their influences are shown in the diagrams.

#3 After treatment

The catalytic conversion is the most common after treatment. The converters are called catalysts. Within the catalyst the NO is converted to N_2 and O_2 by CO and CH which are transformed to CO_2 and H_2O. The sufficient conversion requires the right proportion of the NO, CO and CH gases in the exhaust pipe. This can be achieved by operating the spark ignited engine at about 0.995 - 0.994 air to fuel equivalence ratio (slightly rich mixture).

Classification of catalyst systems:

- Closed loop system

  There is an oxygen sensor built in the exhaust pipe which controls the injection to keep the air to fuel equivalence ration within the limits for satisfactory conversion.

- Open loop system

  There in no feedback in the system. This is cheaper but its conversion efficiency generally is significantly lower than using the closed loop system.

- Two way converter

  It reduces the CO and CH emission.

- Three way converter

  It reduces the NO, CO and CH emission.

The preferred combination is the closed loop three way catalytic system. It converges the 99% of NO, 95% of CO and 70% of CH emitted by the engine.

#4 Special techniques

The exhaust gas recycling (EGR) is one of these techniques. It is found that about 15% recycled exhaust gas gives the best achievable emission reduction.

Diesel engines
Problem solving

1. Determine the effective power and specific fuel consumption at a Diesel engine where given data are following:

Number of cylinders:	$z = 8$
Four-stroke engine:	$i = 0.5$
Indicated mean pressure:	$p_i = 7.5$ bar
Compression ratio:	$\varepsilon = 16.5$
Clearance volume of a cylinder:	$V_c = 150$ cm^3
Revolution:	$n = 2100$ 1/min
Mechanical efficiency:	$\eta_m = 80$ %
Mass flow rate of fuel:	$B = 10.2$ g/s

Solution:

Effective mean pressure: $p_e = p_i \cdot \eta_m = 7.5 \cdot 0.8 = 6$ bar

Displacement volume of a cylinder:

$$V_d = (\varepsilon - 1) \cdot V_c = (16.5 - 1) \cdot 150 = 2325 \text{ cm}^3 = 2.325 \cdot 10^{-6} \text{ m}^3$$

Effective power:

$P_e = p_e \cdot V_d \cdot n \cdot z \cdot i = 6 \cdot 10^5 \cdot 2.325 \cdot 10^{-6} \cdot 2100/60 \cdot 8 \cdot 0.5 = 1.562 \cdot 10^5$ W =
= 156.2 kW

Effective specific fuel consumption:

$$b_e = \frac{B \cdot 3.6}{P_e} = \frac{10.2 \cdot 3.6}{156.2} = 0.235 \text{ kg / kWh}$$

2. Determine the indicated power and specific fuel consumption at a Diesel engine where given data are following:

Number of cylinders:	$z = 6$
Four-stroke engine:	$i = 0.5$
Effective mean pressure:	$p_e = 6.2$ bar
Bore:	$d = 110$ mm
Stroke:	$s = 140$ mm
Piston mean velocity:	$c_m = 8.4$ m/s
Mechanical efficiency:	$\eta_m = 82$ %
Mass flow rate of fuel:	$B = 5.53$ g/s

Solution:

Effective mean pressure: $p_e = p_i \cdot \eta_m = 7.5 \cdot 0.8 = 6$ bar

Displacement volume of a cylinder:

$$V_d = \frac{d^2 \cdot \pi}{4} \cdot s = \frac{11^2 \cdot \pi}{4} \cdot 14 = 1330 \ \text{cm}^3 = 0.00133 \ \text{m}^3$$

Revolution:

$$n = \frac{c_m}{2 \cdot s} = \frac{8.4}{2 \cdot 0.14} = 30 \ 1/\text{s} = 1800 \ 1/\text{min}$$

Indicated mean pressiure:

$$p_i = \frac{p_e}{\eta_m} = \frac{6.2}{0.82} = 7.56 \ \text{bar}$$

Indicated power:

$P_i = p_i \cdot V_d \cdot n \cdot z \cdot i = 7.561 \cdot 10^5 \cdot 0.00133 \cdot 30 \cdot 6 \cdot 0.5 = 9.0505 \cdot 10^5$ W =
= 90.505 kW

Indicated specific fuel consumption:

$$b_i = \frac{B \cdot 3.6}{P_i} = \frac{5.53 \cdot 3.6}{90.505} = 0.22 \ \text{kg / kWh}$$

3. Determine the effective and indicated specific fuel consumption at a Diesel engine where given data are following:

Number of cylinders:	z = 4
Four-stroke engine:	i = 0.5
Indicated mean pressure:	p_i = 6.8 bar
Compression ratio:	ε = 15
Total volume of cylinders:	V = 3750 cm^3
Revolution:	n = 1500 1/min
Mechanical efficiency:	η_m = 84 %
Mass flow rate of fuel:	B = 5.95 g/s

Solution:

Effective mean pressure: $p_e = p_i \cdot \eta_m = 6.8 \cdot 0.84 = 5.712$ bar

Displacement volume of a cylinder:

$$V_d = V - \frac{V}{\varepsilon} = 3750 - \frac{3750}{15} = 3500 \ \text{cm}^3 = 0.0035 \ \text{m}^3$$

Indicated power:

$P_i = p_i \cdot V_d \cdot n \cdot z \cdot i = 6.8 \cdot 10^5 \cdot 0.0035 \cdot 1500/60 \cdot 4 \cdot 0.5 = 1.19 \cdot 10^5$ W =
= 119 kW

Indicated specific fuel consumption:

$$b_i = \frac{B \cdot 3.6}{P_i} = \frac{5.95 \cdot 3.6}{119} = 0.18 \ \text{kg / kWh}$$

Effective power:

$P_e = p_e \cdot V_d \cdot n \cdot z \cdot i = 5.712 \cdot 10^5 \cdot 0.0035 \cdot 1500/60 \cdot 4 \cdot 0.5 = 9.996 \cdot 10^4$ W =
= 99.96 kW

Effective specific fuel consumption:

$$b_e = \frac{B \cdot 3.6}{P_e} = \frac{5.95 \cdot 3.6}{99.96} = 0.214 \ \text{kg / kWh}$$

4. Determine the effective specific fuel consumption at an Otto and a Diesel engine where given data are following:

Both indicated power:	P_i = 148 kW
Indicated efficiency of the Otto engine:	η_{iO} = 34 %
Indicated efficiency of the Diesel engine:	η_{iD} = 45 %
Lower heating value of petrol:	H_{LP} = 43500 kJ/kg
Lower heating value of diesel fuel:	H_{LD} = 42800 kJ/kg
Both mechanical efficiency:	η_m = 84 %

Solution:

Effective specific fuel consumption of the Otto engine:

$$b_{eO} = \frac{3600}{\eta_{iO} \cdot \eta_m \cdot H_{LP}} = \frac{3600}{0.34 \cdot 0.84 \cdot 43500} = 0.29 \ \ \mathrm{kg/kWh}$$

Effective specific fuel consumption of the Diesel engine:

$$b_{eD} = \frac{3600}{\eta_{iD} \cdot \eta_m \cdot H_{LD}} = \frac{3600}{0.45 \cdot 0.84 \cdot 42800} = 0.223 \ \ \mathrm{kg/kWh}$$

Ratio of the specific fuel consumptions: $\frac{b_{eD}}{b_{eO}} = \frac{0.223}{0.29} = 0.768$

Effective fuel consumption of the Otto engine at full load:

$$B_O = b_{eo} \cdot P_i = 0.29 \cdot 148 = 42.92 \ \ \mathrm{kg/h}$$

Effective fuel consumption of the Diesel engine at full load:

$$B_D = b_{eD} \cdot P_i = 0.223 \cdot 148 = 33.0 \ \ \mathrm{kg/h}$$

5. Determine the heat balance of a Diesel engine where given data are following:

Number of cylinders:	$z = 6$
Four-stroke engine:	$i = 0.5$
Effective mean pressure:	$p_e = 9.2$ bar
Compression ratio:	$\varepsilon = 17$
Clearance volume of a cylinder:	$V_c = 107.8$ cm^3
Stroke:	$s = 150$ mm
Revolution:	$n = 2100$ 1/min
Lower heating value of diesel fuel:	$H_{LD} = 42500$ kJ/kg
Effective specific fuel consumption:	$b_e = 0.23$ kg/kWh
Delivery ratio:	$\lambda = 0.83$
Exhaust gas temperature:	$t_{out} = 650$ °C
Ambient temperature:	$t_{amb} = 30$ °C
Ambient pressure:	$p_{amb} = 1.017$ bar
Specific heat of flue gas:	$c_{pfg} = 1.08$ kJ/kg°K

Solution:

Displacement volume of a cylinder:

$$V_d = (\varepsilon - 1) \cdot V_c = (17 - 1) \cdot 107.8 = 1725 \text{ cm}^3 = 1.725 \cdot 10^{-3} \text{ m}^3$$

Effective power:

$P_e = p_e \cdot V_d \cdot n \cdot z \cdot i = 9.2 \cdot 10^5 \cdot 1.725 \cdot 10^{-3} \cdot 2100/60 \cdot 6 \cdot 0.5 = 1.667 \cdot 10^5$ W =
= 166.7 kW

Indicated specific fuel consumption:

$$b_i = \frac{B \cdot 3.6}{P_i} = \frac{5.95 \cdot 3.6}{119} = 0.18 \text{ kg / kWh}$$

Input heat: $$Q_{in} = b_e \cdot P_e \cdot \frac{H_{LD}}{3600} = 0.23 \cdot 166.6 \cdot \frac{42500}{3600} = 452.41 \text{ kW}$$

Molar mass of air: $M = 29.4$ kg/kmol

Universal gas constant: $R_u = 8315$ kJ/kg°K

Density of the ambient air:

$$\rho_0 = \frac{p_{amb} \cdot 10000}{273.15 + t_{amb}} \cdot \frac{M}{R_u} = \frac{101700}{273.15 + 30} \cdot \frac{29.4}{8315} = 1.186 \text{ kg / m}^3$$

Inducted air to the engine:

$G_{in} = V_d \cdot n \cdot z \cdot i \cdot \rho_o \cdot \lambda = 1.725 \cdot 10^{-3} \cdot 2100/60 \cdot 6 \cdot 0.5 \cdot 1.186 \cdot 0.83 =$
$= 0.178$ kg/s

Heat going out by means of flue gas:

$$Q_{fg} = G_{in} \cdot c_{pfg} \cdot (t_{out} - t_{amb}) = 0.178 \cdot 1.08 \cdot (650 - 30) = 119.4 \text{ kW}$$

Heat going out by means of cooling water:

$$Q_{cool} = Q_{in} - P_e - Q_{fg} = 452.4 - 166.6 - 119.4 = 166.4 \text{ kW}$$

Ratios against input power:

$$\frac{P_e}{Q_{in}} = \frac{166.6}{452.4} = 0.368 \qquad \frac{Q_{fg}}{Q_{in}} = \frac{119.4}{452.4} = 0.264 \qquad \frac{Q_{cool}}{Q_{in}} = \frac{166.4}{452.4} = 0.368$$