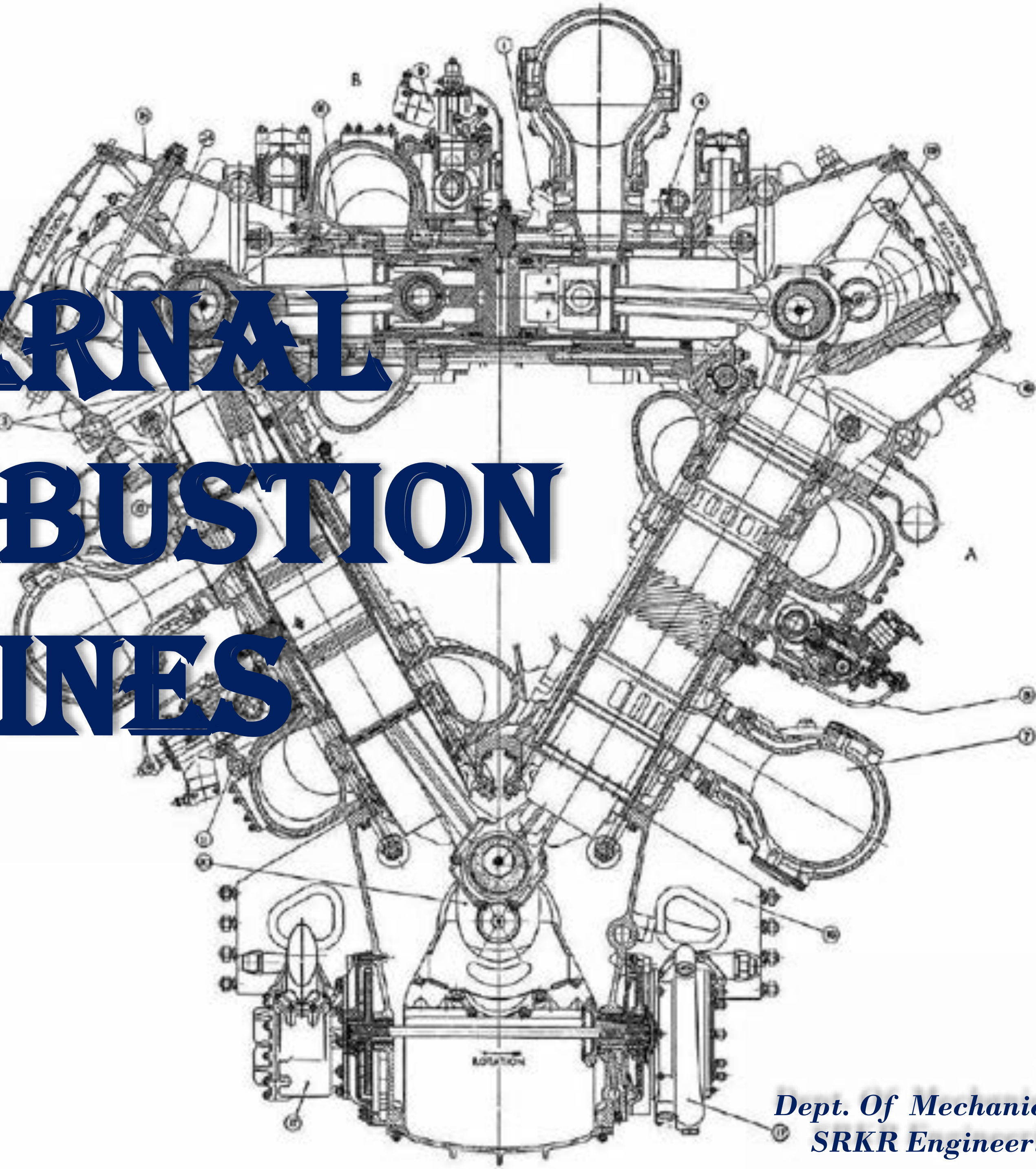
INTERNAL COMBUSTION ENGINES



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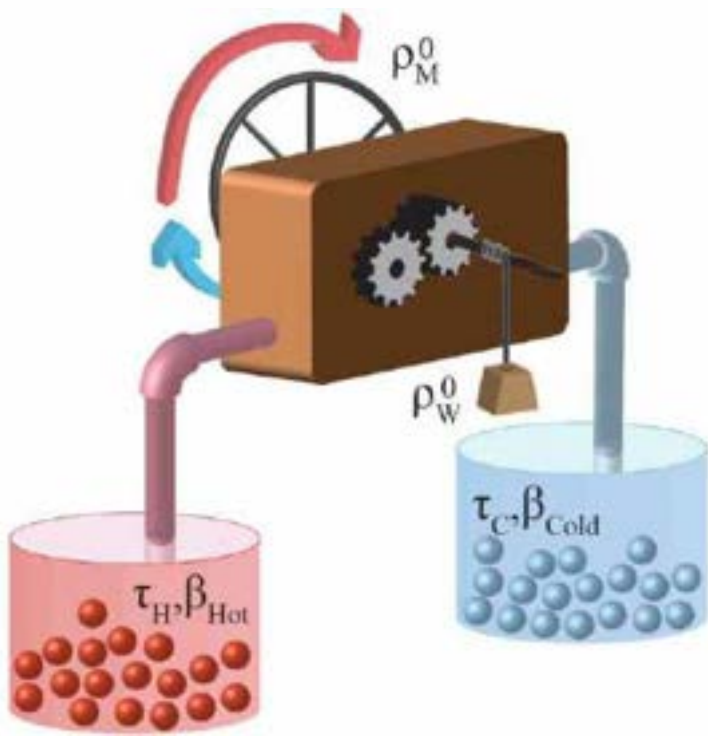
UNIT-I

I.C. engines: classification-comparison of two stroke and four stroke engines- comparison of S.I. and C.I. engines-Valve timing and port timing diagrams-Efficiencies- air standard efficiency, indicated thermal efficiency, brake thermal efficiency, mechanical efficiency, volumetric efficiency and relative efficiency-Testing and performances of I.C. engines -Basic principles of carburetion and fuel injection.

Definition of 'Engine'

An engine is a device which transforms one form of energy into another form. However, while transforming energy from one form to another, the efficiency of conversion plays an important role. Normally, most of the engines convert thermal energy into mechanical work and therefore they are called 'heat engines'.



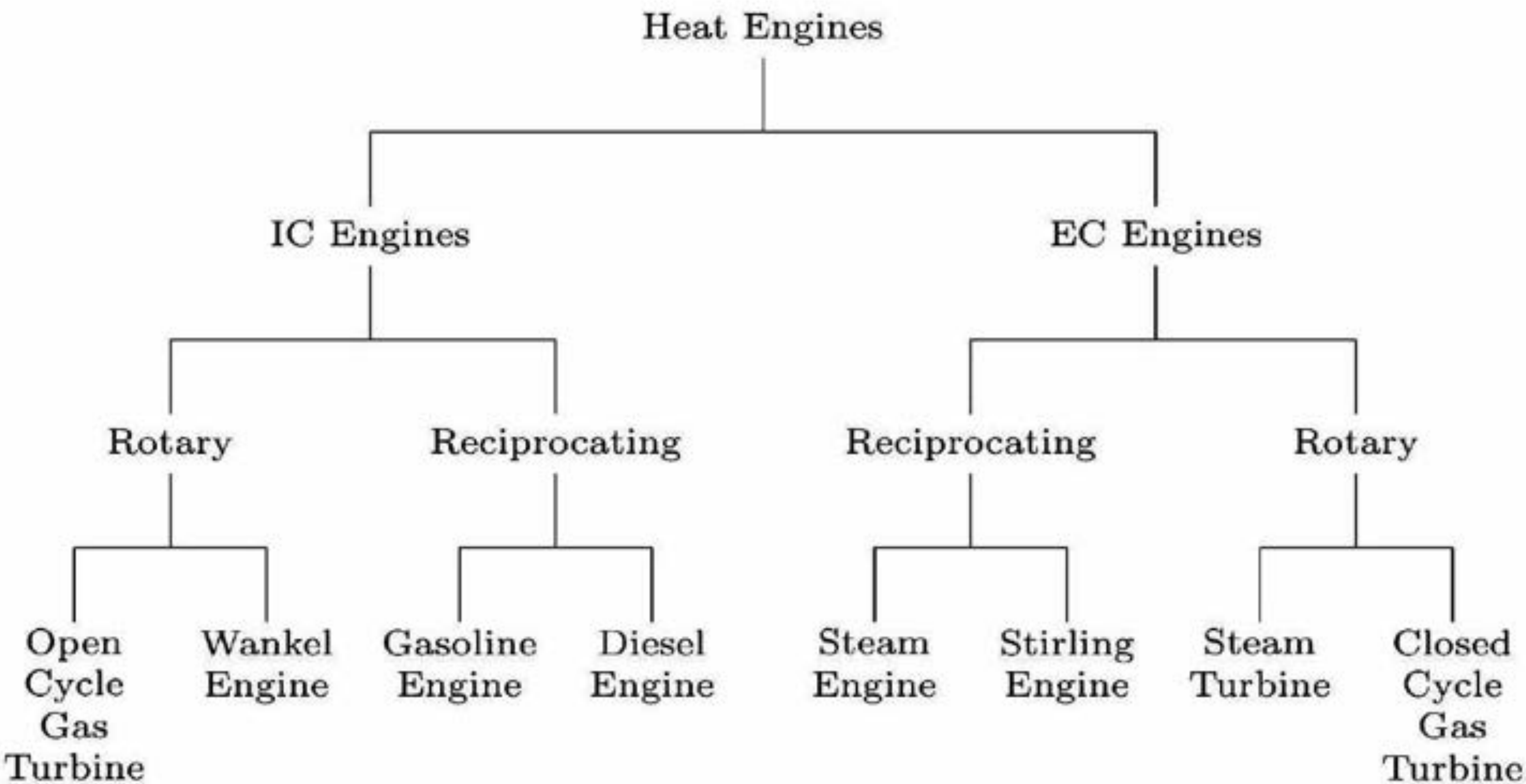
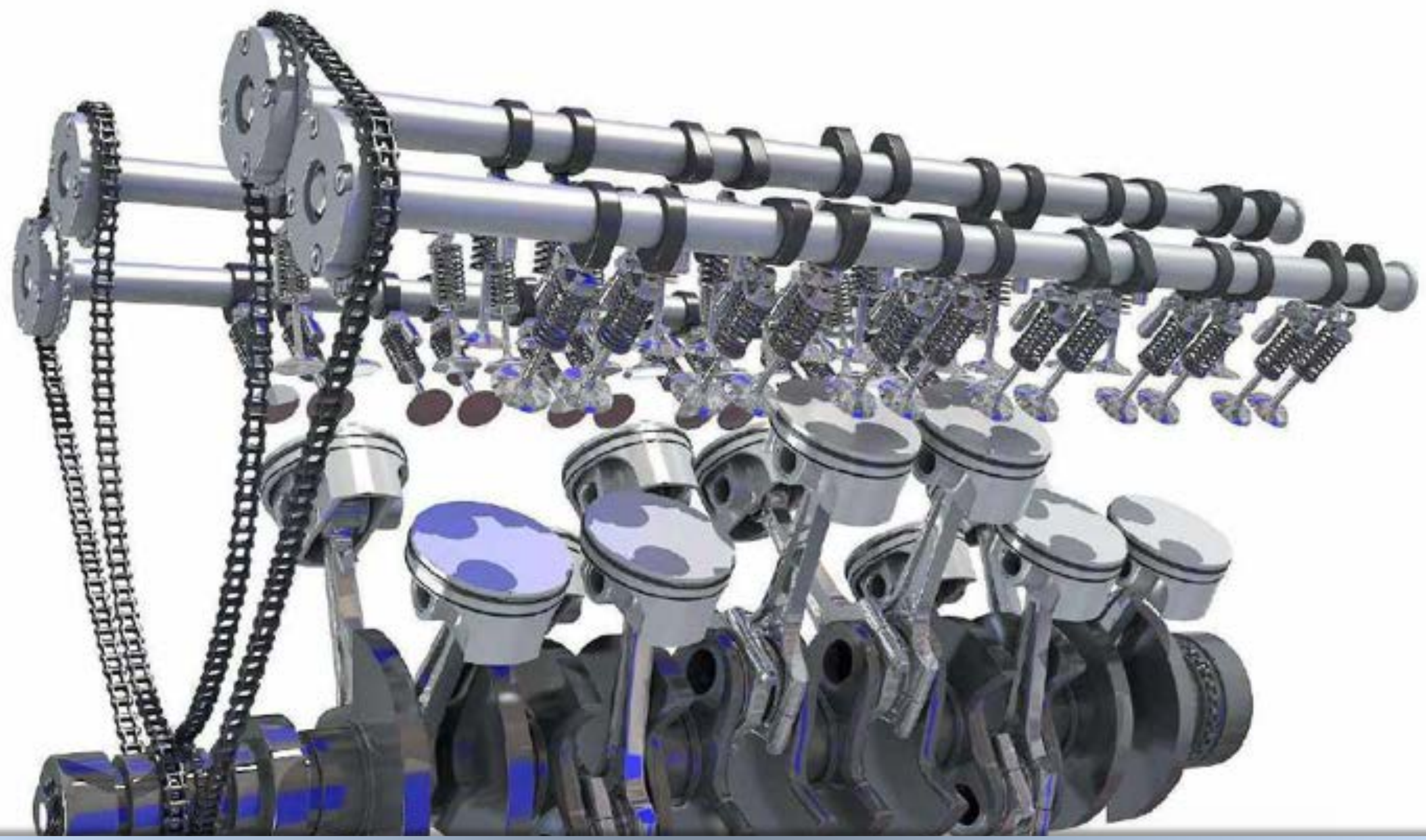


Fig. Classification of heat engines

Application of Engines

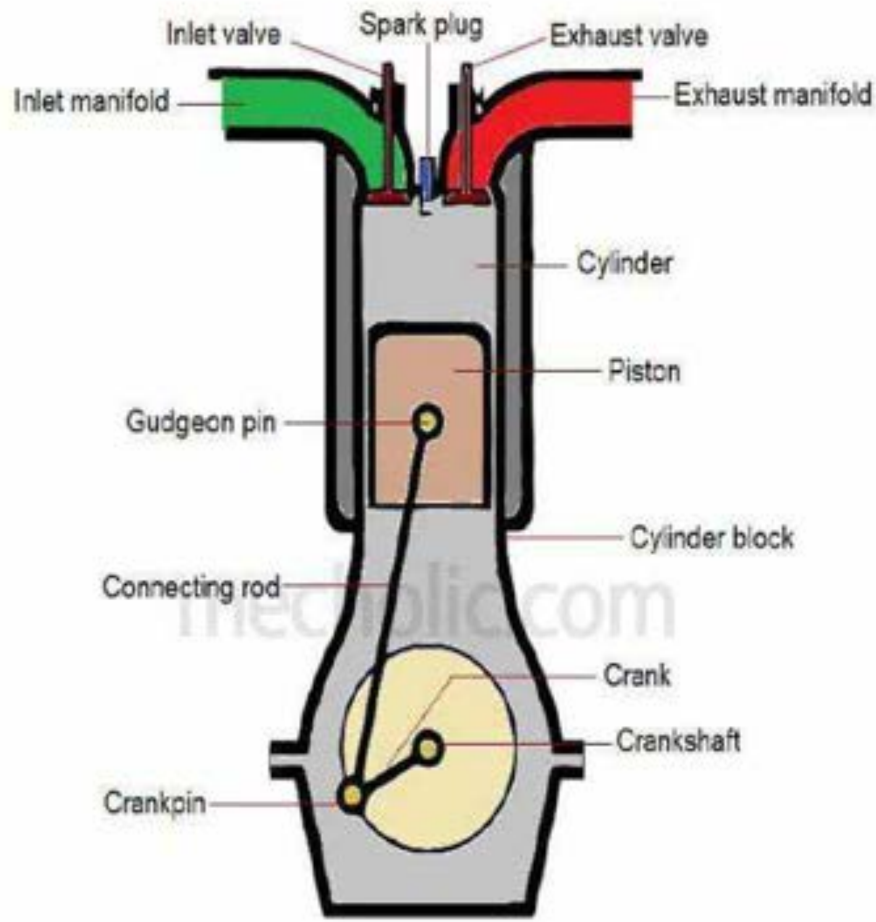
IC Engine		EC Engine	
Type	**Application**	**Type**	**Application**
Gasoline engines	Automotive, Marine, Aircraft	Steam Engines	Locomotives, Marine
Gas engines	Industrial power	Stirling Engines	Experimental Space Vehicles
Diesel engines	Automotive, Railways, Power,Marine	Steam Turbines	Power, Large Marine
Gas turbines	Power, Aircraft, Industrial, Marine	Close Cycle Gas Turbine	Power, Marine



Working of an
Internal Combustion Engine

Basic components of an IC engine

1.Cylinder Block
2.Cylinder
3.Piston
4.Combustion Chamber
5.Inlet Manifold
6.Exhaust Manifold
7.Inlet & Exhaust valve
8.Spark Plug
9.Connecting Rod
10.Crankshaft
11.Piston Ring
12.Gudgeon Pin
13.Camsaft



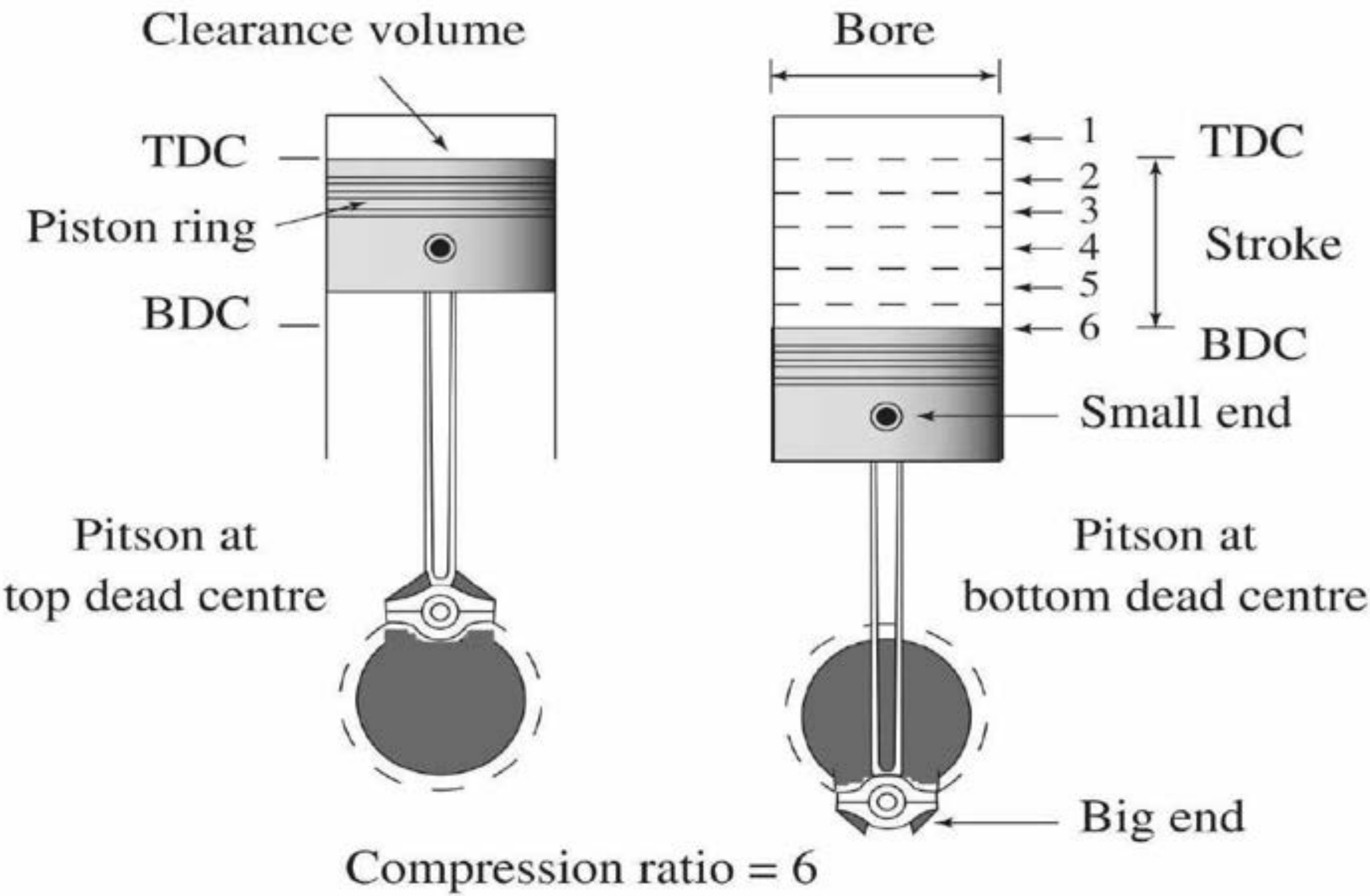


Fig. Top and bottom dead centres

Displacement or Swept Volume (V_s) **:** The nominal volume swept by the working piston when travelling from one dead centre to the other is called the displacement volume. It is expressed in terms of cubic centimeter (cc) and given by

$$V_s = A \times L \quad = \quad \frac{\pi}{4} d^2 L$$

Cubic Capacity or Engine Capacity : The displacement volume of a cylinder multiplied by number of cylinders in an engine will give the cubic capacity or the engine capacity. For example, if there are K cylinders in an engine, then

$$\text{Cubic capacity} = V_s \times K$$

Clearance Volume (V_C) **:** The nominal volume of the combustion chamber above the piston when it is at the top dead centre is the clearance volume. It is designated as V_C and expressed in cubic centimeter (cc).

Compression Ratio (r) **:** It is the ratio of the total cylinder volume when the piston is at the bottom dead centre, V_T, to the clearance volume, V_C. It is designated by the letter r.

$$r = \frac{V_T}{V_C} = \frac{V_C + V_s}{V_C} = 1 + \frac{V_s}{V_C}$$


And the camshafts are driven by the crank itself using a timing belt.

4-Stroke Petrol Engines

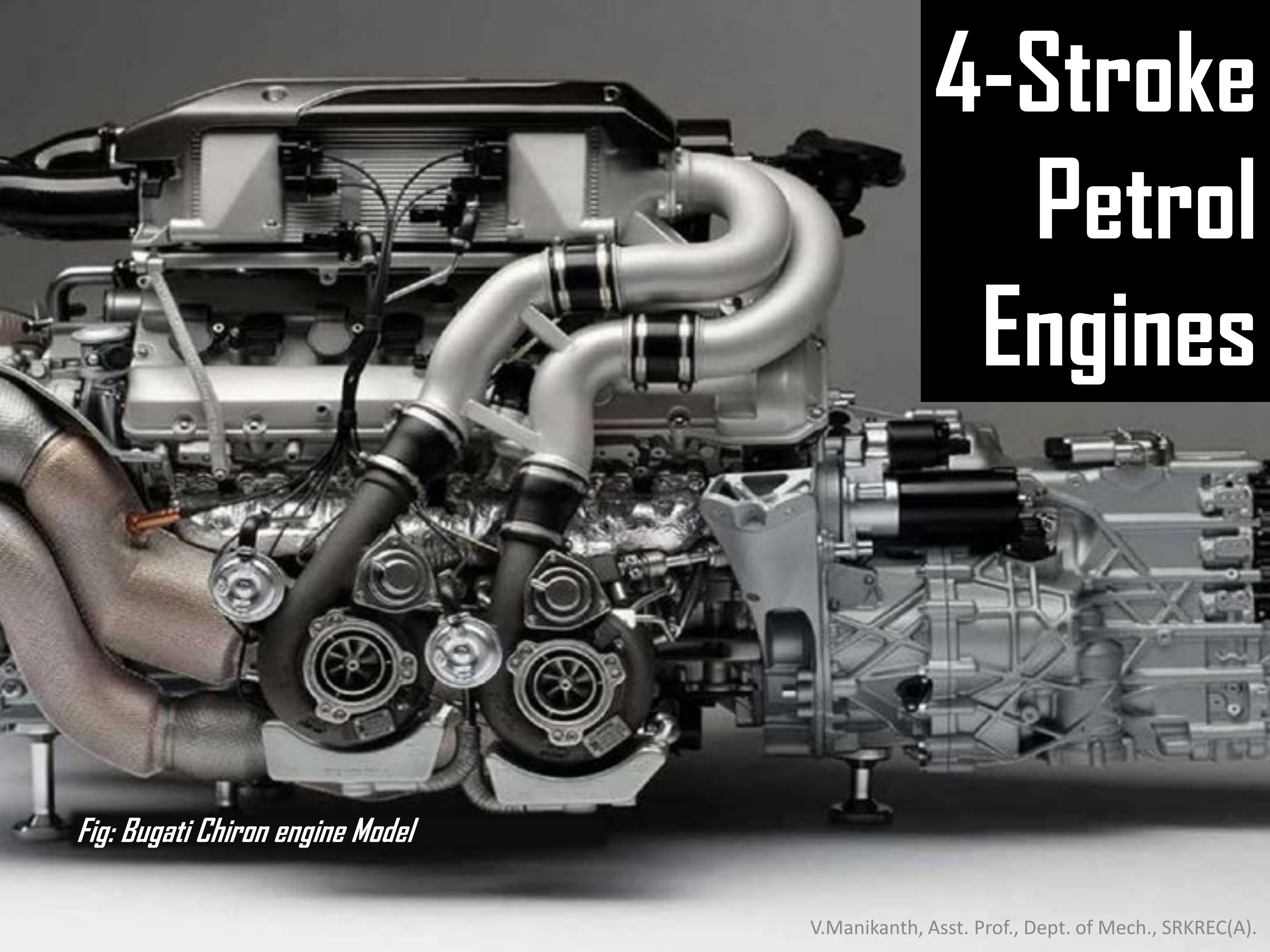
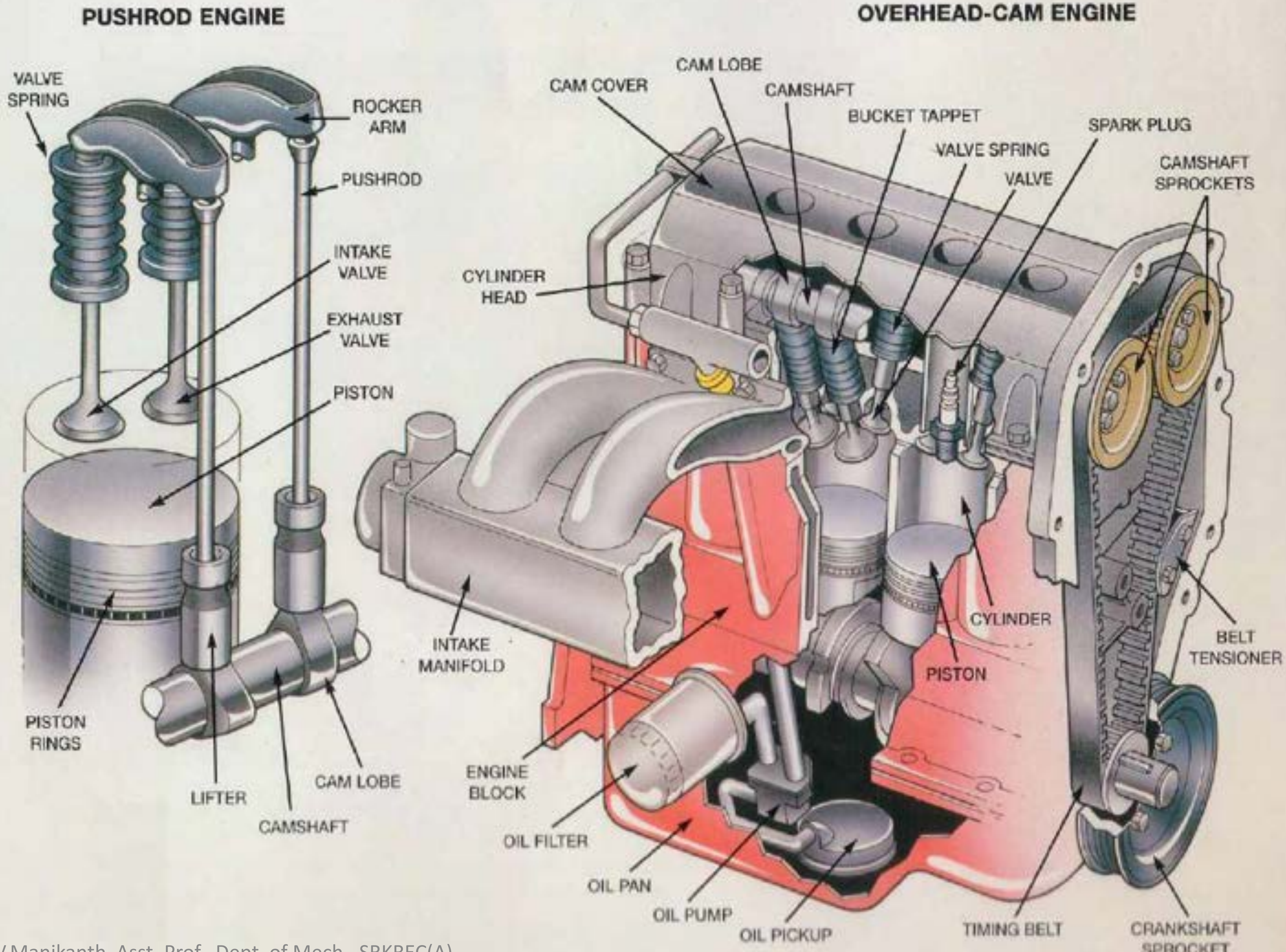


Fig: Bugati Chiron engine Model


PUSHROD ENGINE
OVERHEAD-CAM ENGINE
VALVE SPRING
ROCKER ARM
PUSHROD
INTAKE VALVE
EXHAUST VALVE
PISTON
PISTON RINGS
LIFTER
CAMSHAFT
CAM LOBE
CAM COVER
CAM LOBE
CAMSHAFT
BUCKET TAPPET
SPARK PLUG
VALVE SPRING
VALVE
CAMSHAFT SPROCKETS
CYLINDER HEAD
INTAKE MANIFOLD
ENGINE BLOCK
OIL FILTER
OIL PAN
OIL PUMP
OIL PICKUP
CYLINDER
PISTON
BELT TENSIONER
TIMING BELT
CRANKSHAFT SPROCKET

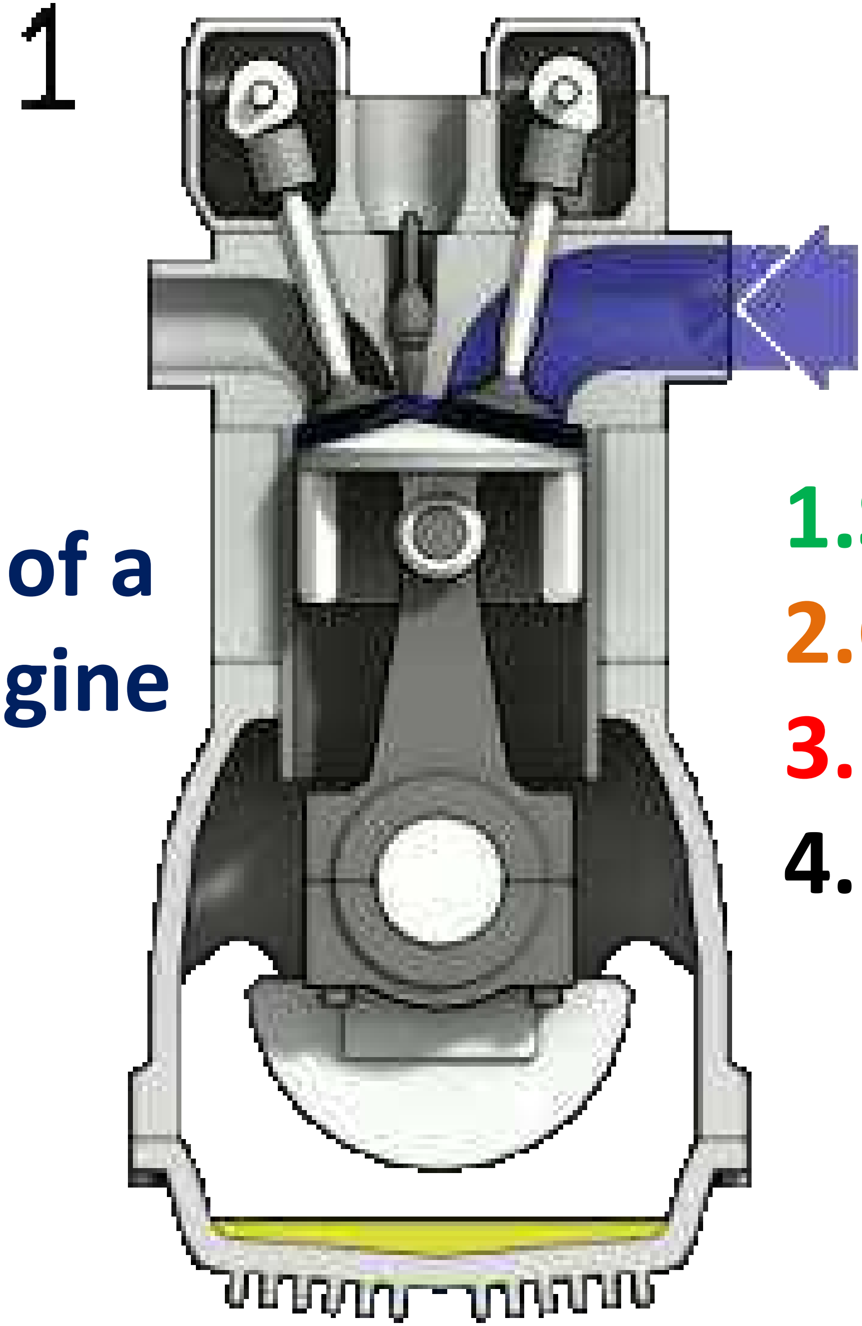
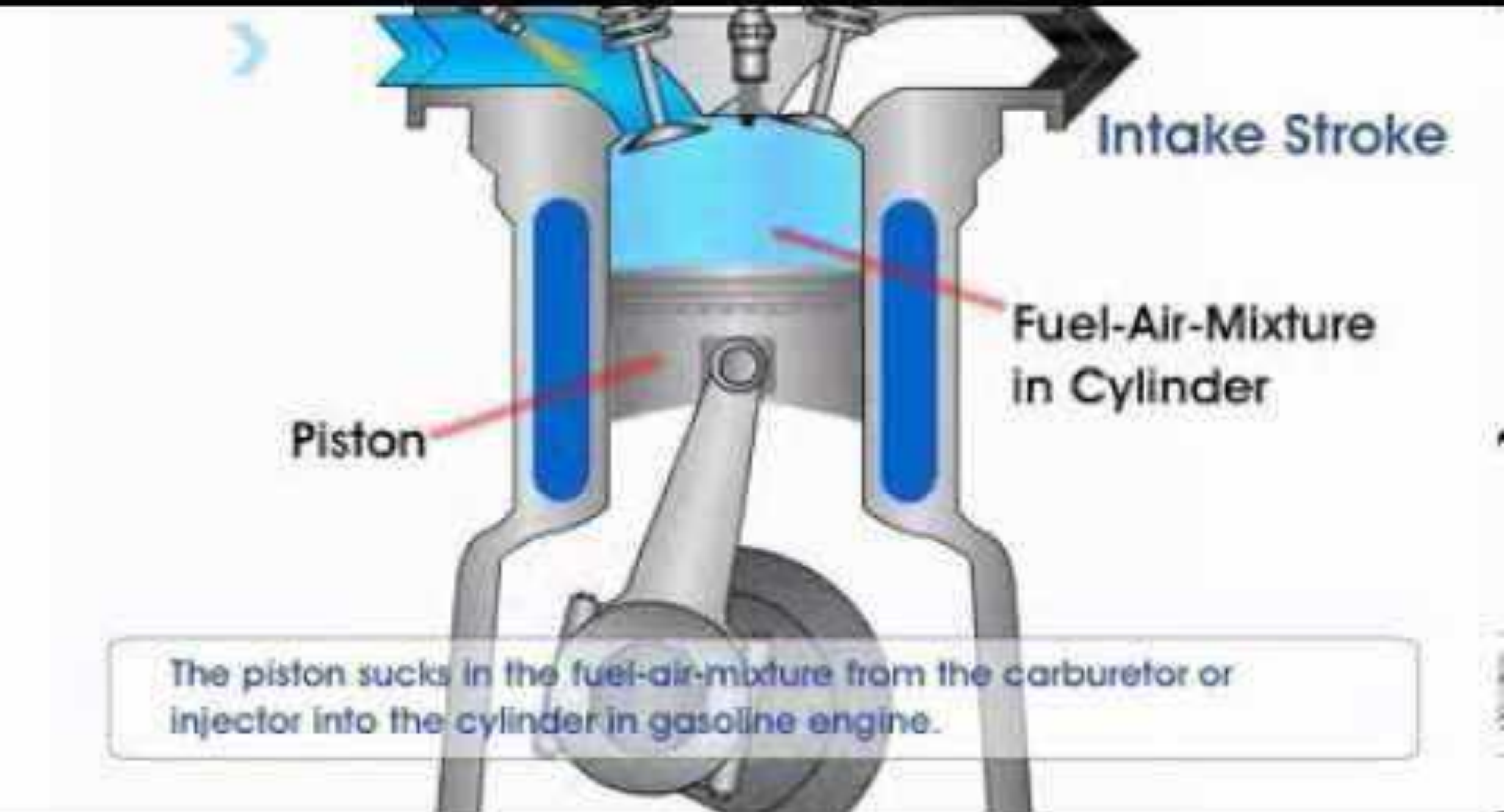


Fig: working of a
4-S Petrol Engine

1.Suction
2.Compression
3.Power
4.Exhaust


Intake Stroke
Fuel-Air-Mixture
in Cylinder
Piston
The piston sucks in the fuel-air-mixture from the carburetor or
injector into the cylinder in gasoline engine.

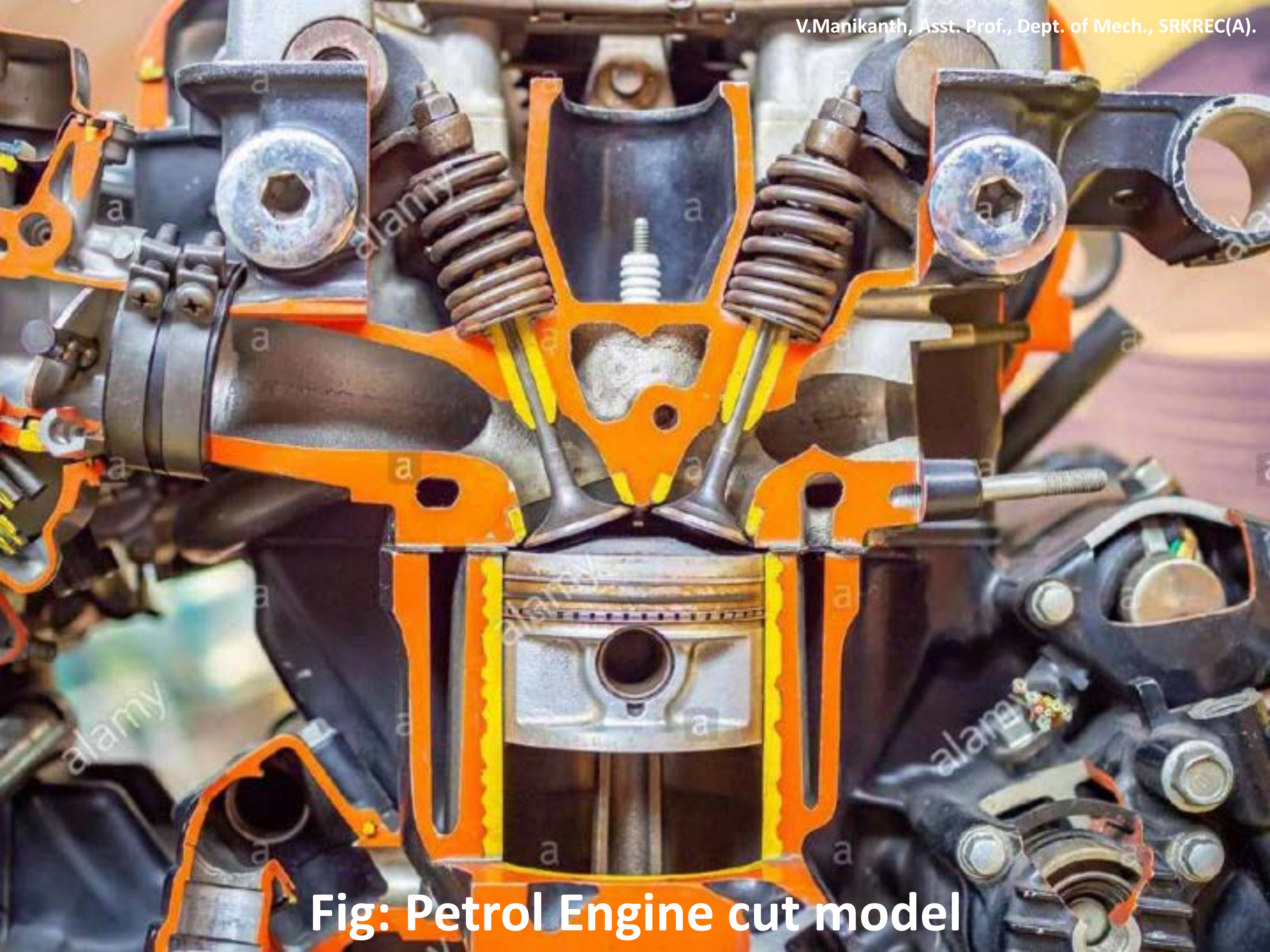
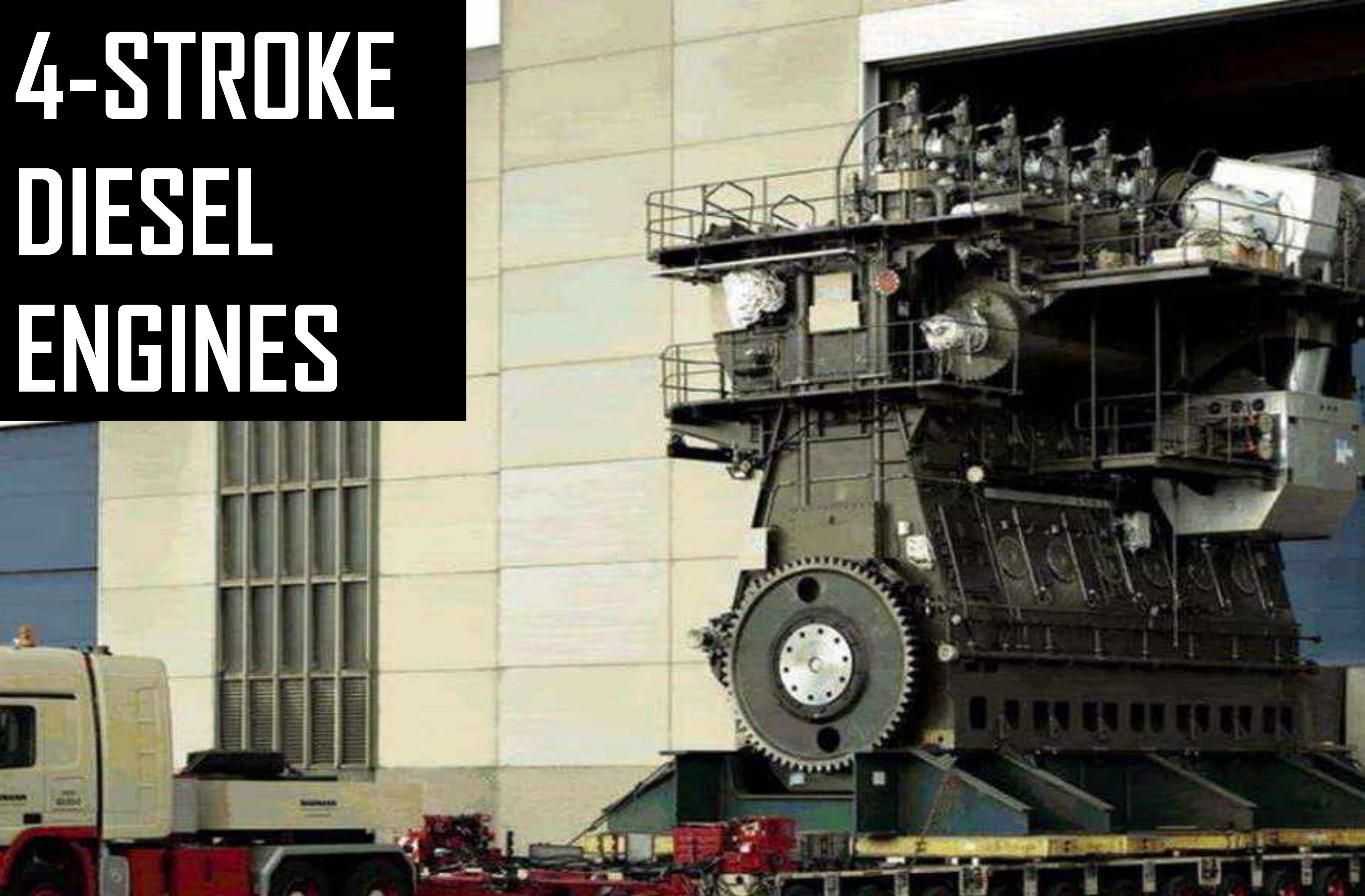


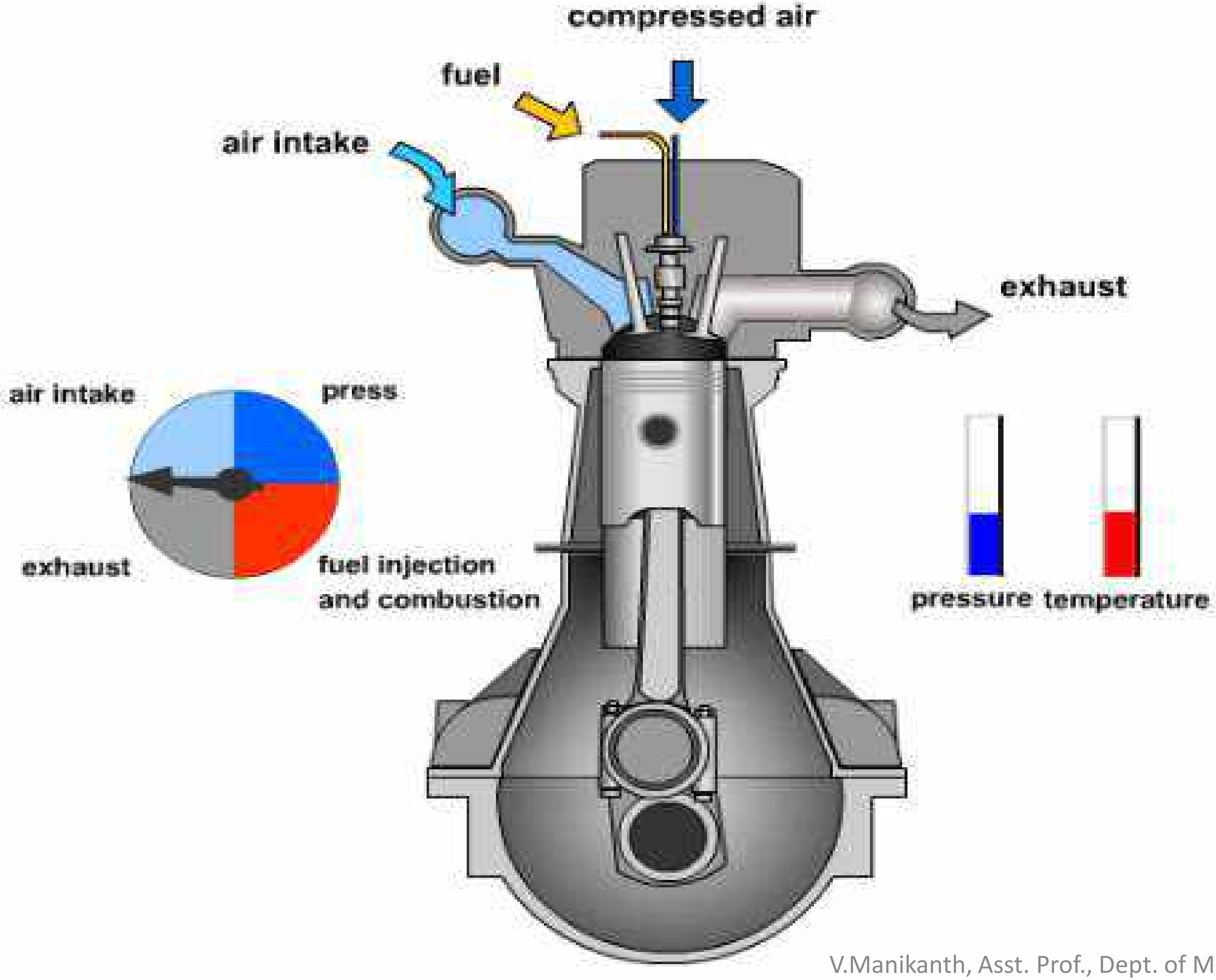
Fig: Petrol Engine cut model

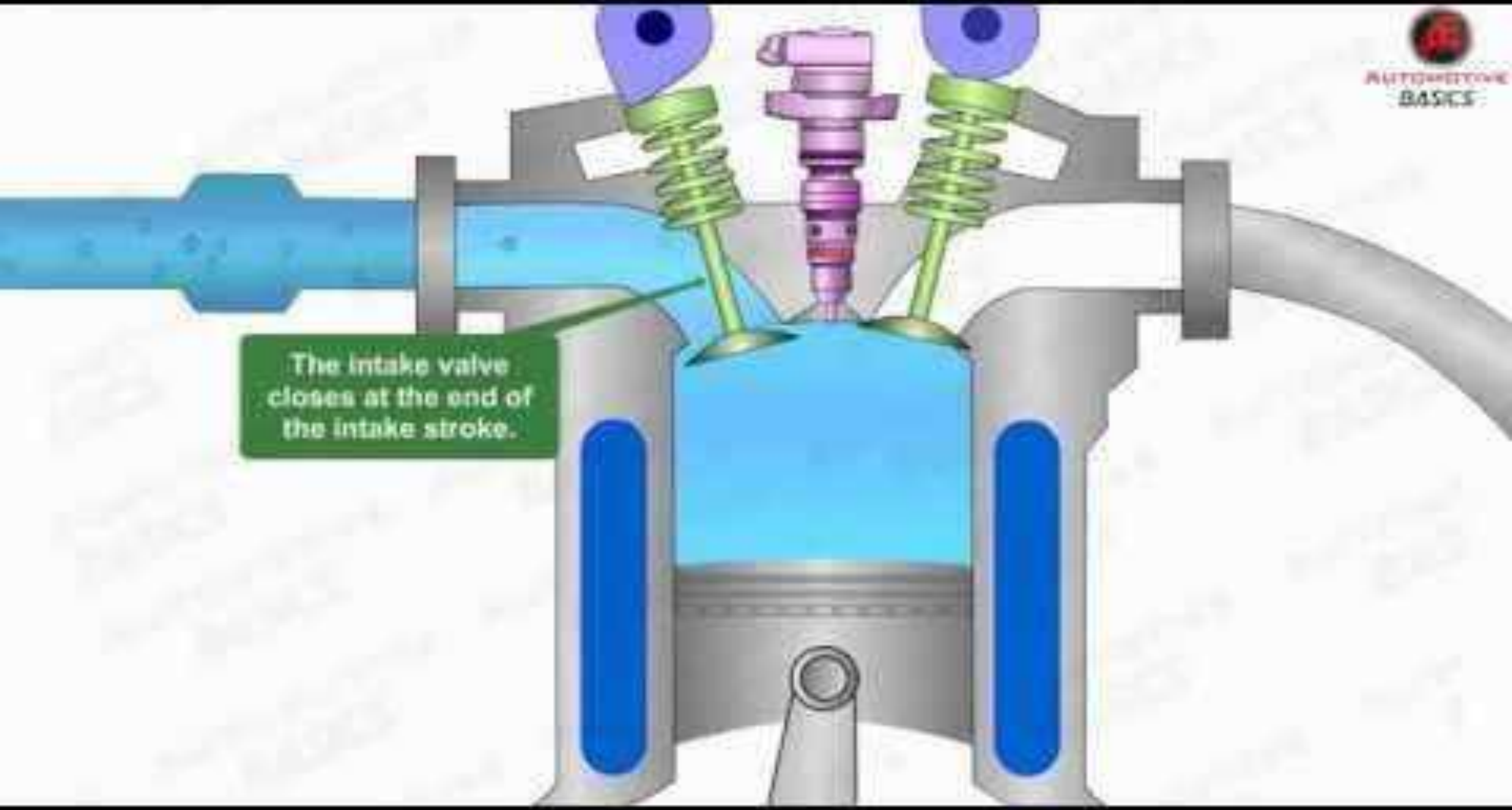
4-STROKE DIESEL ENGINES

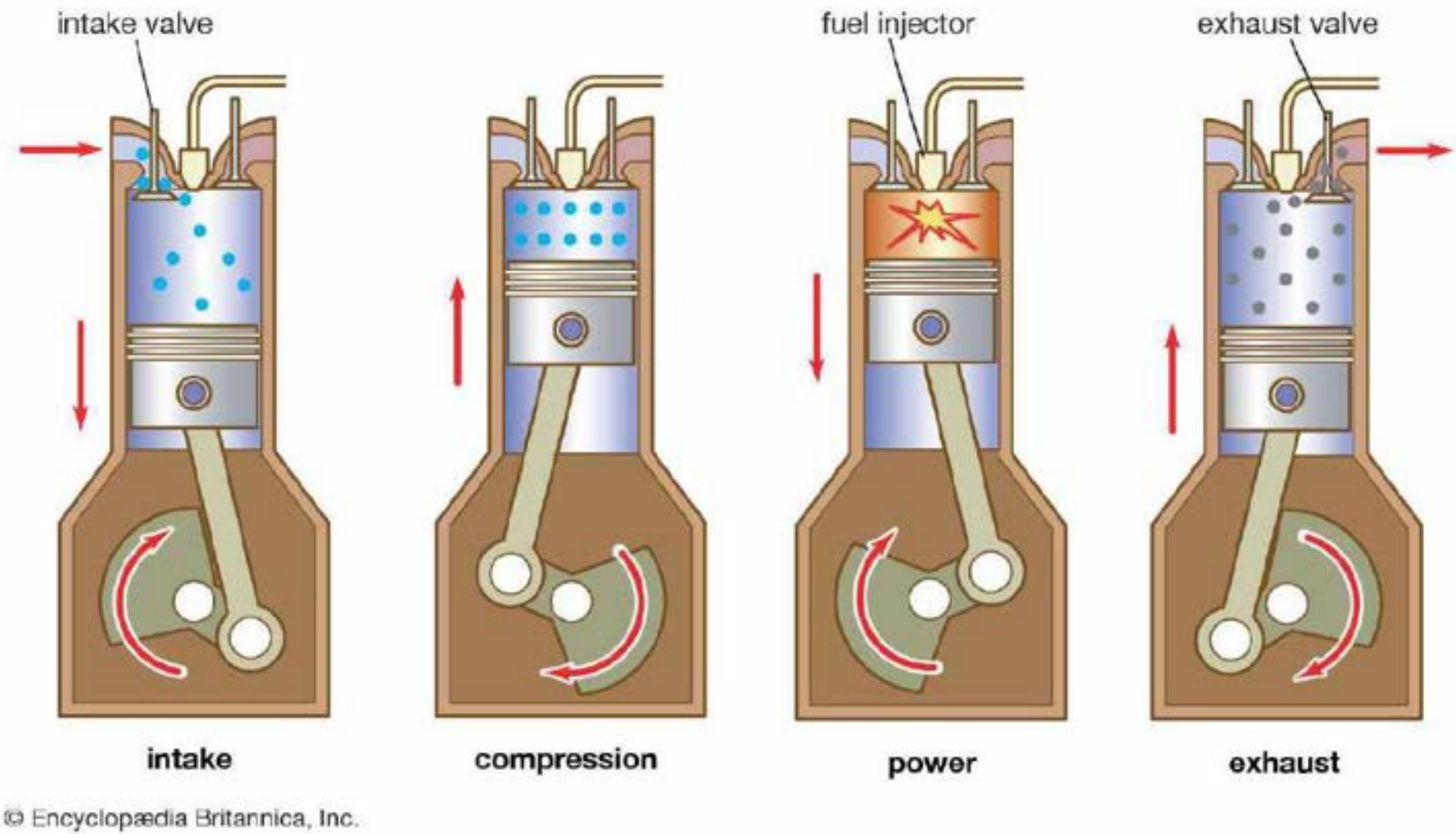


This jaw-dropper is the Wärtsilä RT-flex96C, the world's largest and most powerful **1,09,000 horsepower** ***diesel engine.***

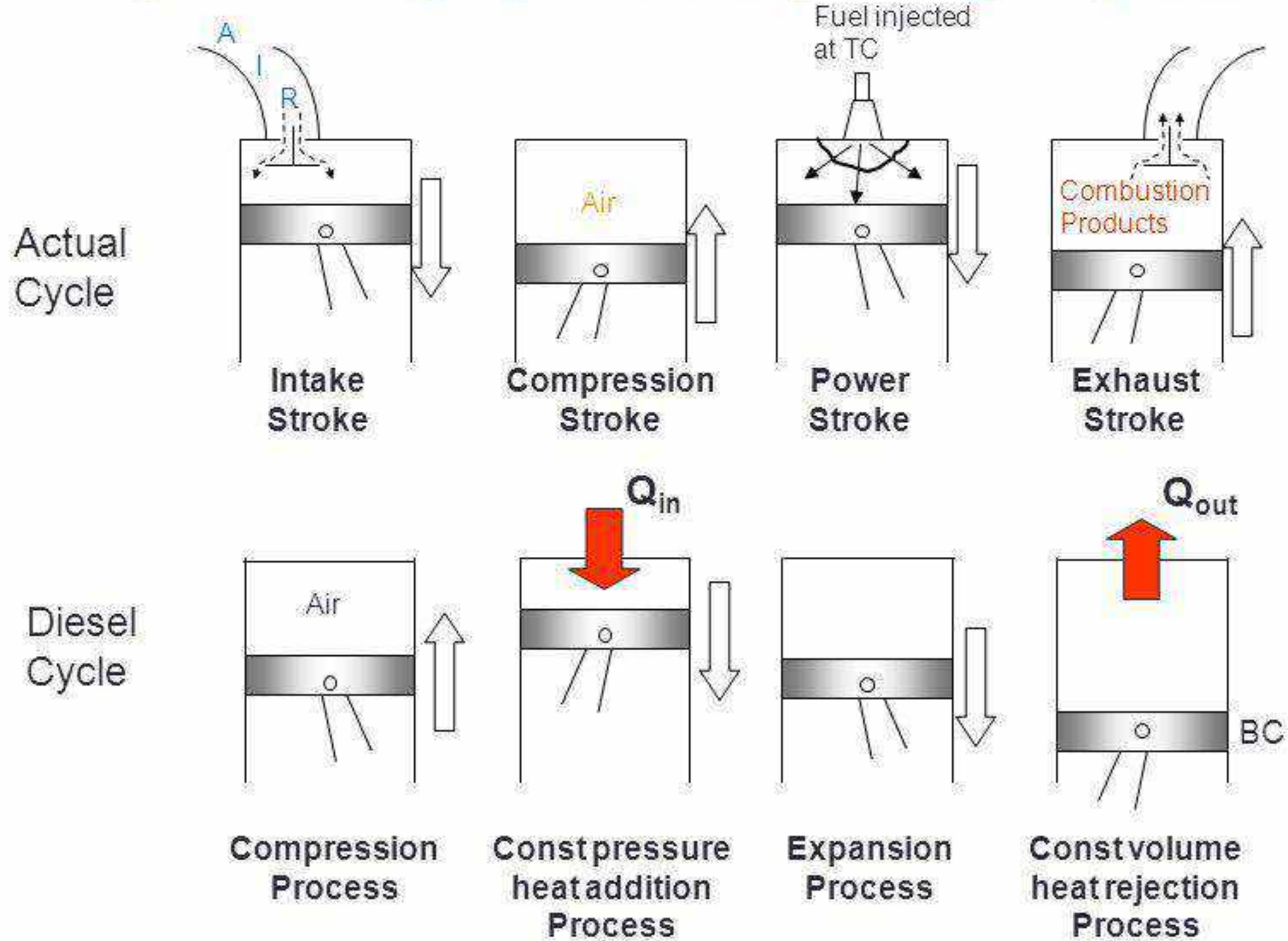
Diesel

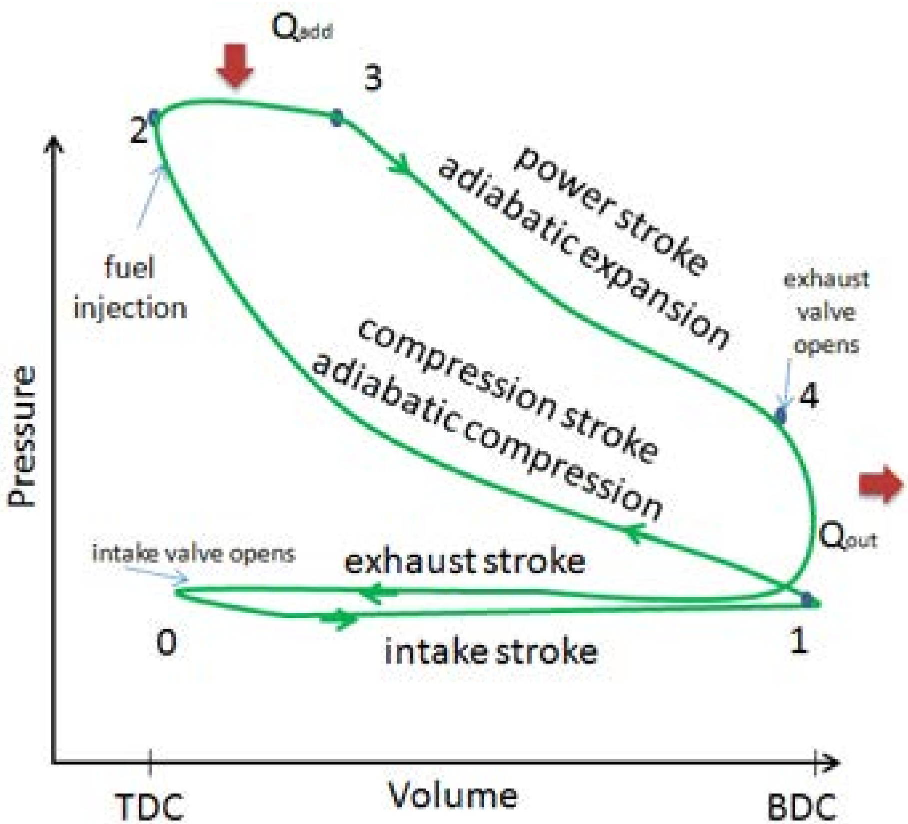

compressed air
fuel
air intake
exhaust
air intake
press
exhaust
fuel injection
and combustion
pressure
temperature


AUTOMOTIVE
BASICS
The intake valve closes at the end of the intake stroke.


intake valve
fuel injector
exhaust valve
intake
compression
power
exhaust
© Encyclopædia Britannica, Inc.

WORKING OF DIESEL CYCLE





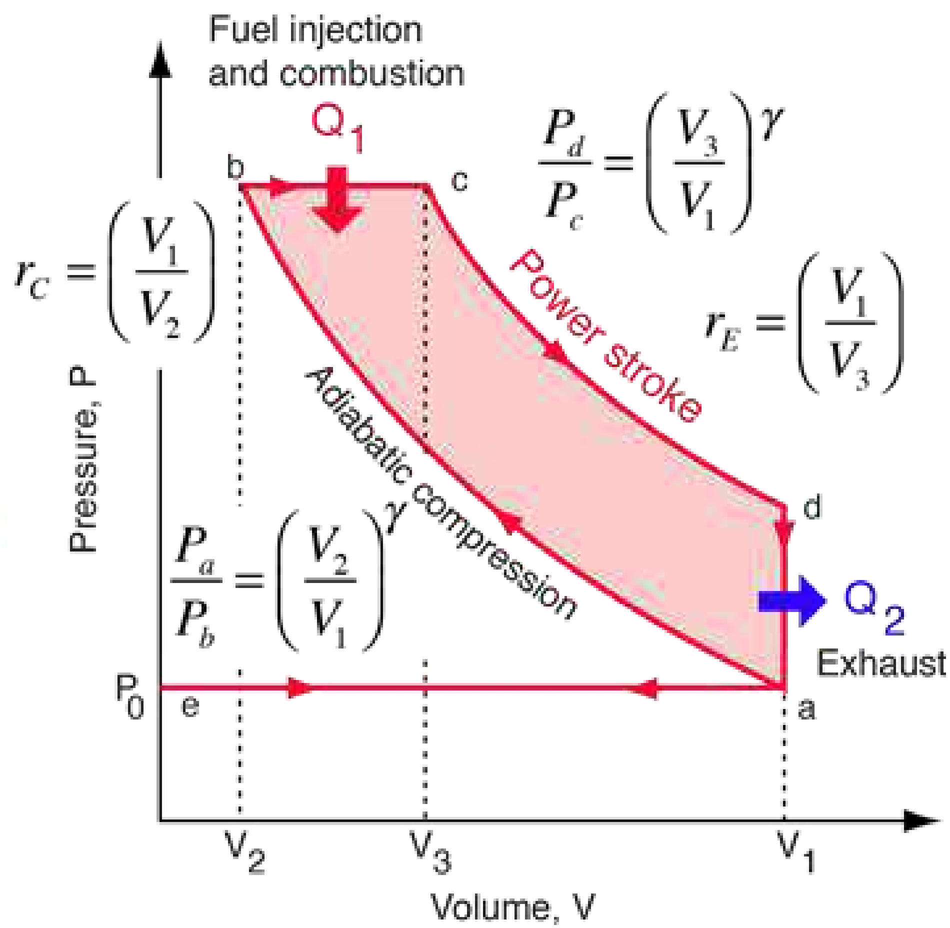
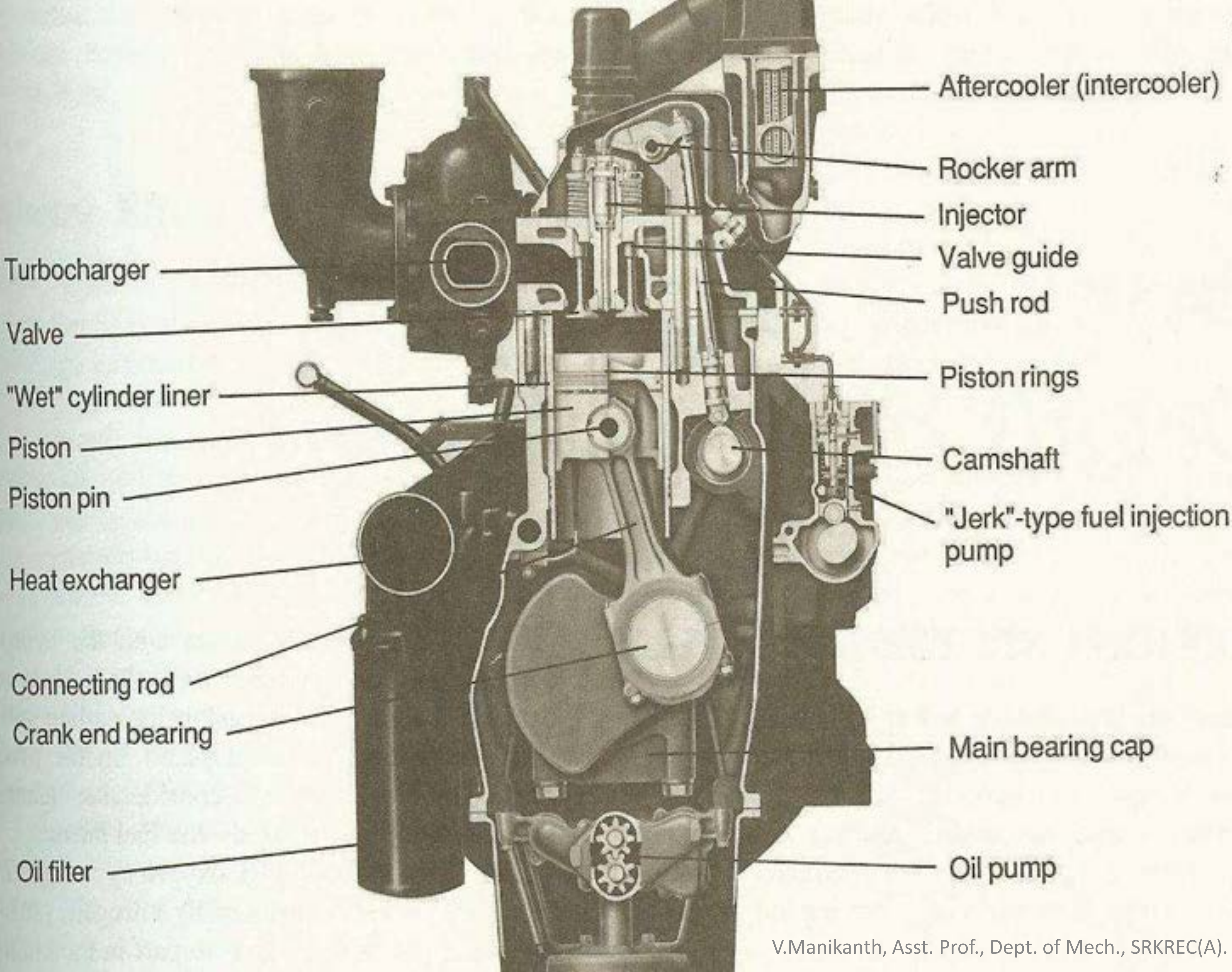


Fig: p-v dig. of practical diesel engine cycle

Fig: p-v diagram of Theoretical Diesel Cycle


Aftercooler (intercooler)
Rocker arm
Injector
Valve guide
Push rod
Turbocharger
Valve
Piston rings
"Wet" cylinder liner
Piston
Camshaft
Piston pin
"Jerk"-type fuel injection pump
Heat exchanger
Connecting rod
Crank end bearing
Main bearing cap
Oil filter
Oil pump

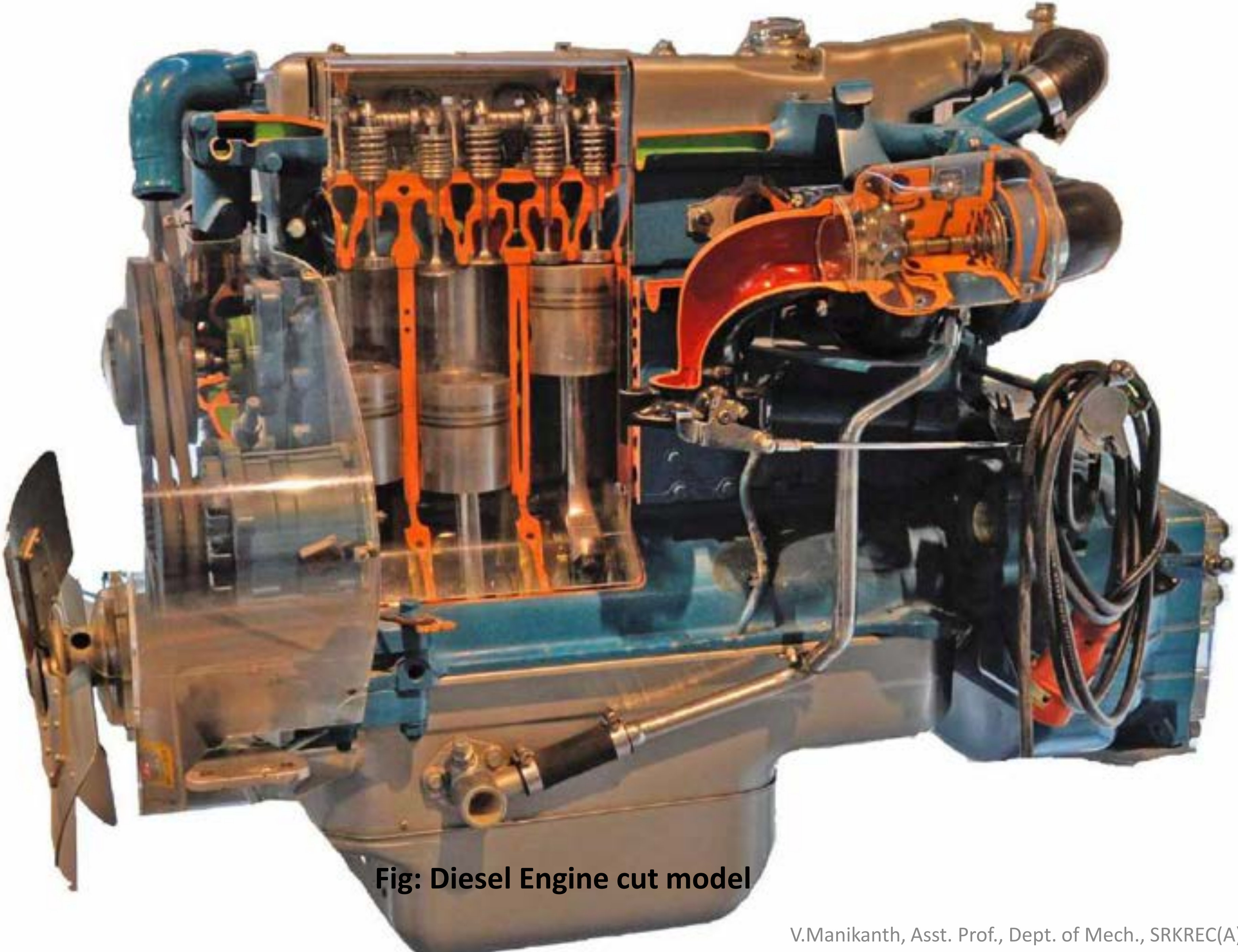
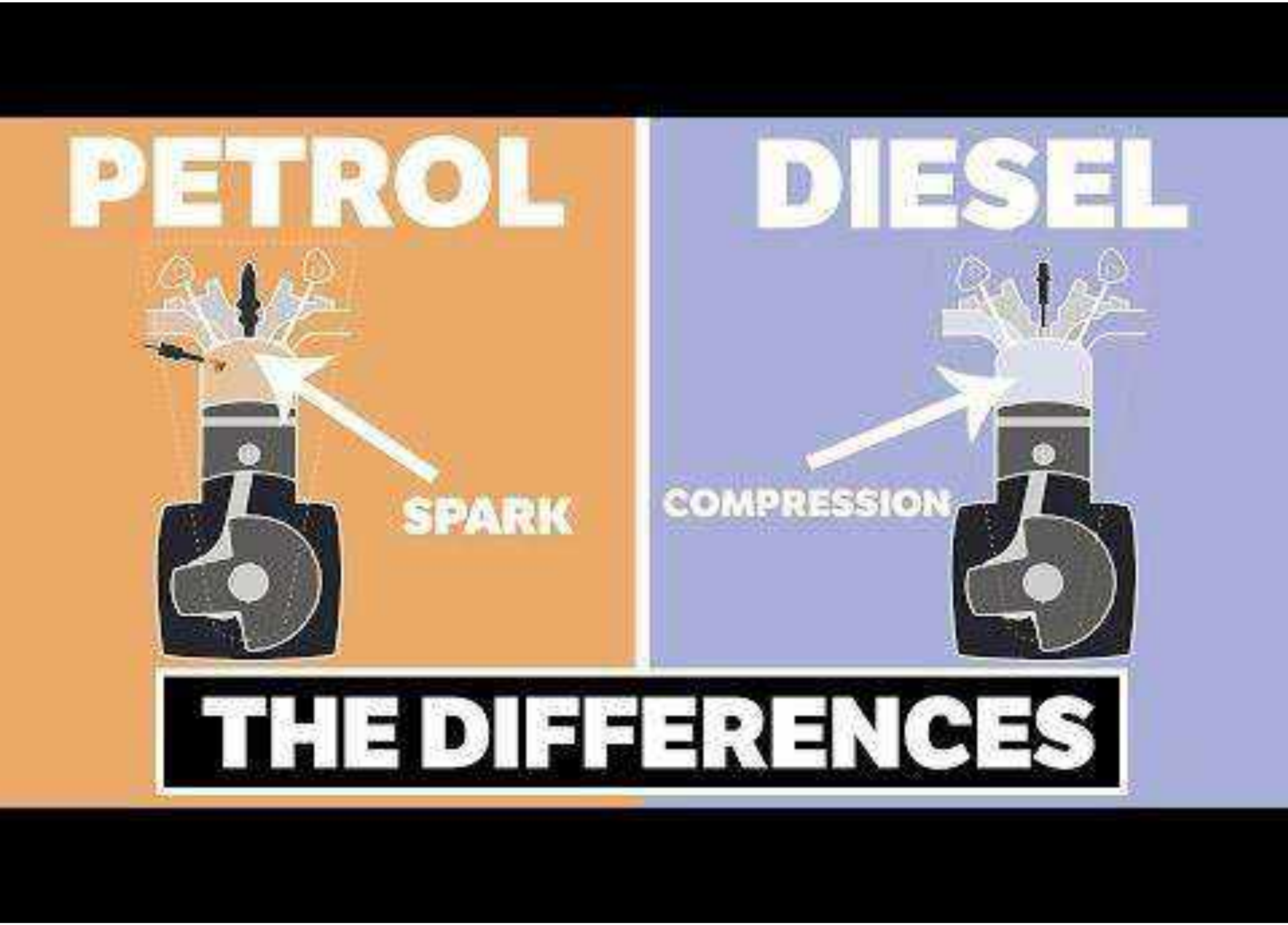


Fig: Diesel Engine cut model

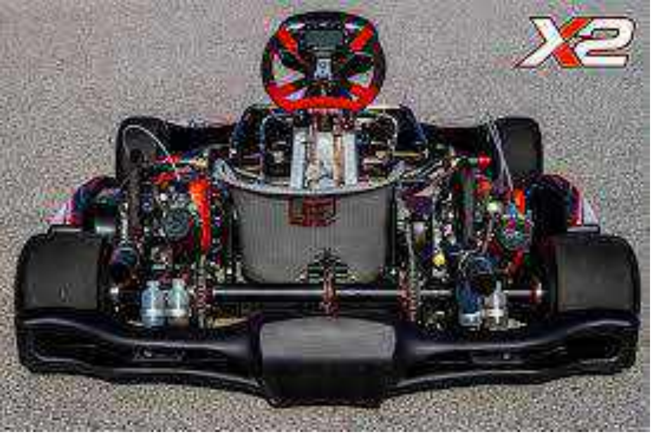
Comparison of SI and CI Engines

Description	SI Engine	CI Engine
Basic cycle	Works on Otto cycle or constant volume heat addition cycle.	Works on Diesel cycle or constant pressure heat addition cycle.
Fuel	Gasoline, a highly volatile fuel. Self-ignition temperature is high.	Diesel oil, a non-volatile fuel. Self-ignition temperature is comparatively low.
Introduction of fuel	A gaseous mixture of fuel-air is introduced during the suction stroke. A carburettor and an ignition system are necessary. Modern engines have gasoline injection.	Fuel is injected directly into the combustion chamber at high pressure at the end of the compression stroke. A fuel pump and injector are necessary.
Load control	Throttle controls the quantity of fuel-air mixture to control the load.	The quantity of fuel is regulated to control the load. Air quantity is not controlled.

Ignition	Requires an ignition system with spark plug in the combustion chamber. Primary voltage is provided by either a battery or a magneto.	Self-ignition occurs due to high temperature of air because of the high compression. Ignition system and spark plug are not necessary.
Compression ratio	6 to 10. Upper limit is fixed by antiknock quality of the fuel.	16 to 20. Upper limit is limited by weight increase of the engine.
Speed	Due to light weight and also due to homogeneous combustion, they are high speed engines.	Due to heavy weight and also due to heterogeneous combustion, they are low speed engines.
Thermal efficiency	Because of the lower CR, the maximum value of thermal efficiency that can be obtained is lower.	Because of higher CR, the maximum value of thermal efficiency that can be obtained is higher.
Weight	Lighter due to comparatively lower peak pressures.	Heavier due to comparatively higher peak pressures.

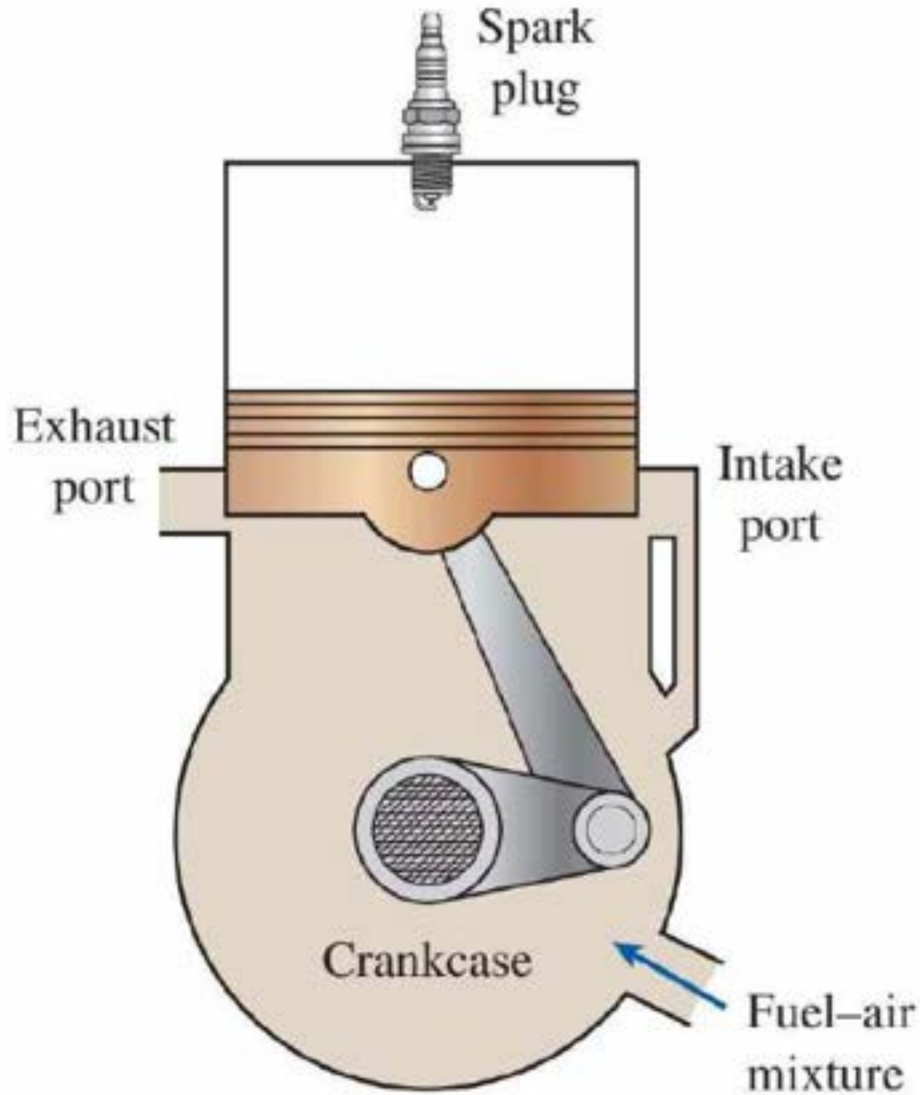

PETROL
DIESEL
SPARK
COMPRESSION
THE DIFFERENCES


DIESEL ENGINE
VS
PETROL ENGINE



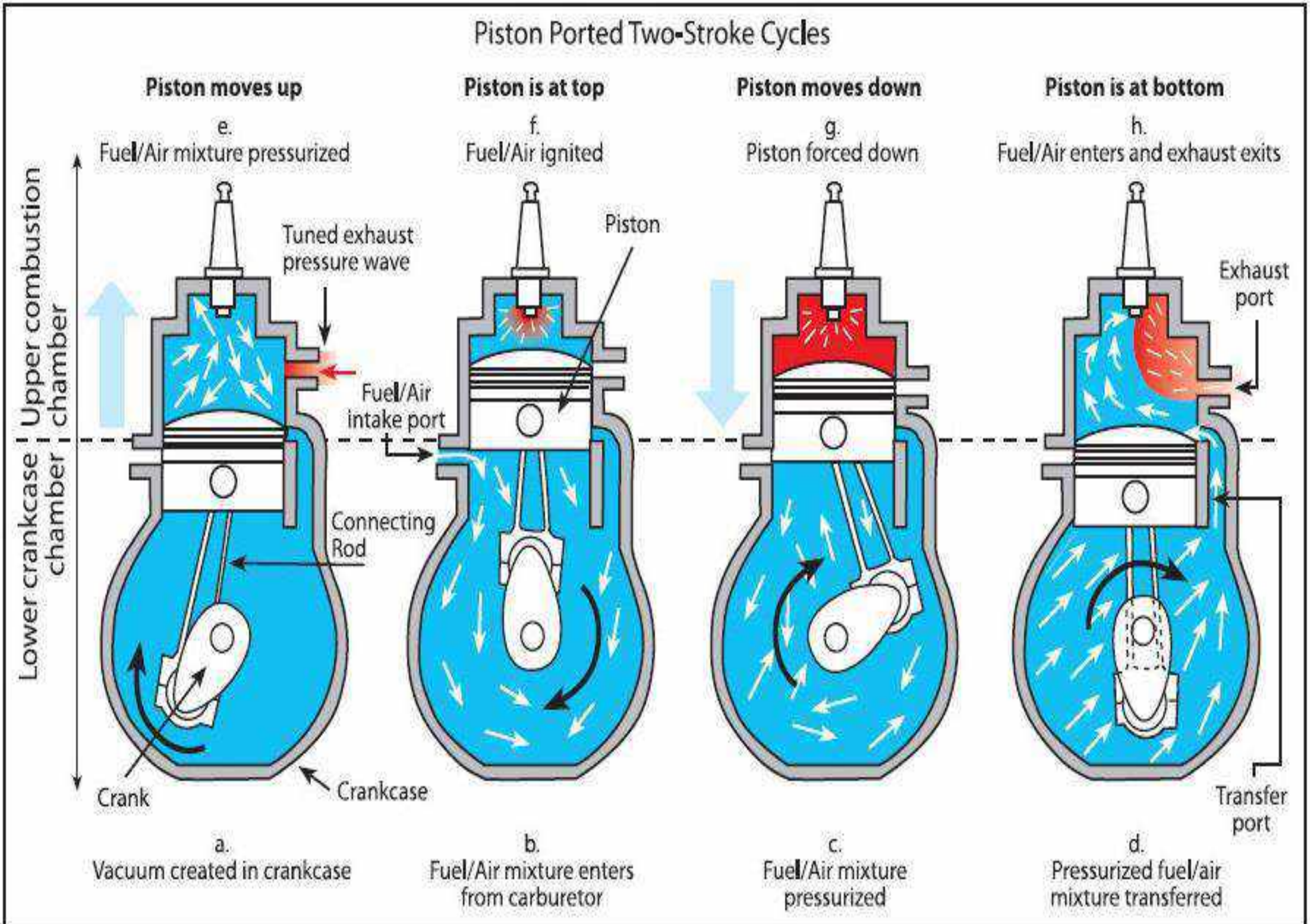
TWO STROKE ENGINES

V.Manikanth, Asst. Prof., Dept. of Mech., SRKREC(A).



FIGURE

Schematic of a two-stroke reciprocating engine.



Piston Ported Two-Stroke Cycles
Piston moves up
Piston is at top
Piston moves down
Piston is at bottom
e.
Fuel/Air mixture pressurized
f.
Fuel/Air ignited
g.
Piston forced down
h.
Fuel/Air enters and exhaust exits
Upper combustion chamber
Lower crankcase chamber
Tuned exhaust pressure wave
Piston
Exhaust port
Fuel/Air intake port
Connecting Rod
Crank
Crankcase
Transfer port
a.
Vacuum created in crankcase
b.
Fuel/Air mixture enters from carburetor
c.
Fuel/Air mixture pressurized
d.
Pressurized fuel/air mixture transferred

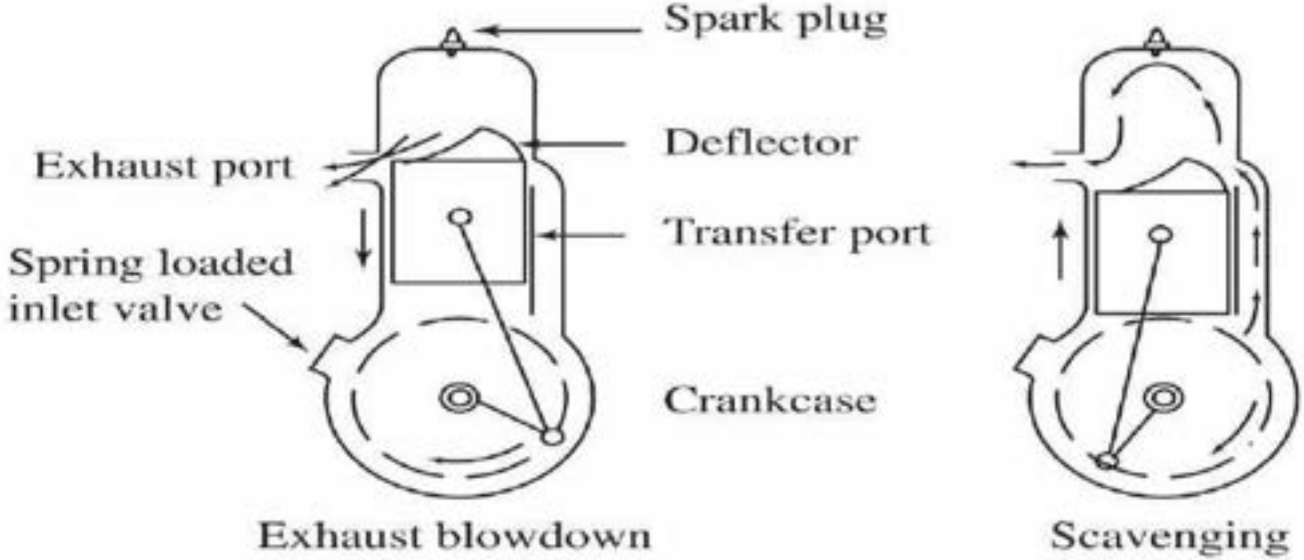


Fig. Crankcase scavenged two-stroke SI engine

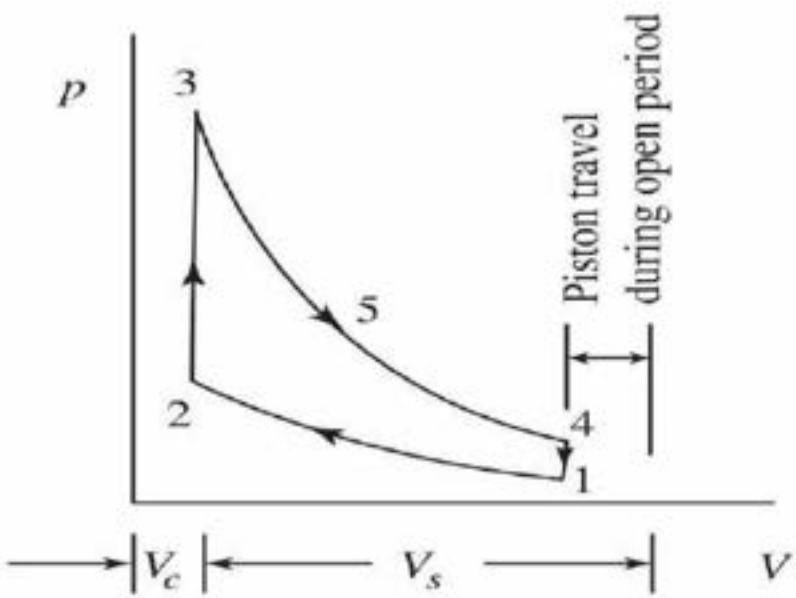


Fig. Ideal p-V diagram of a two-stroke SI engine

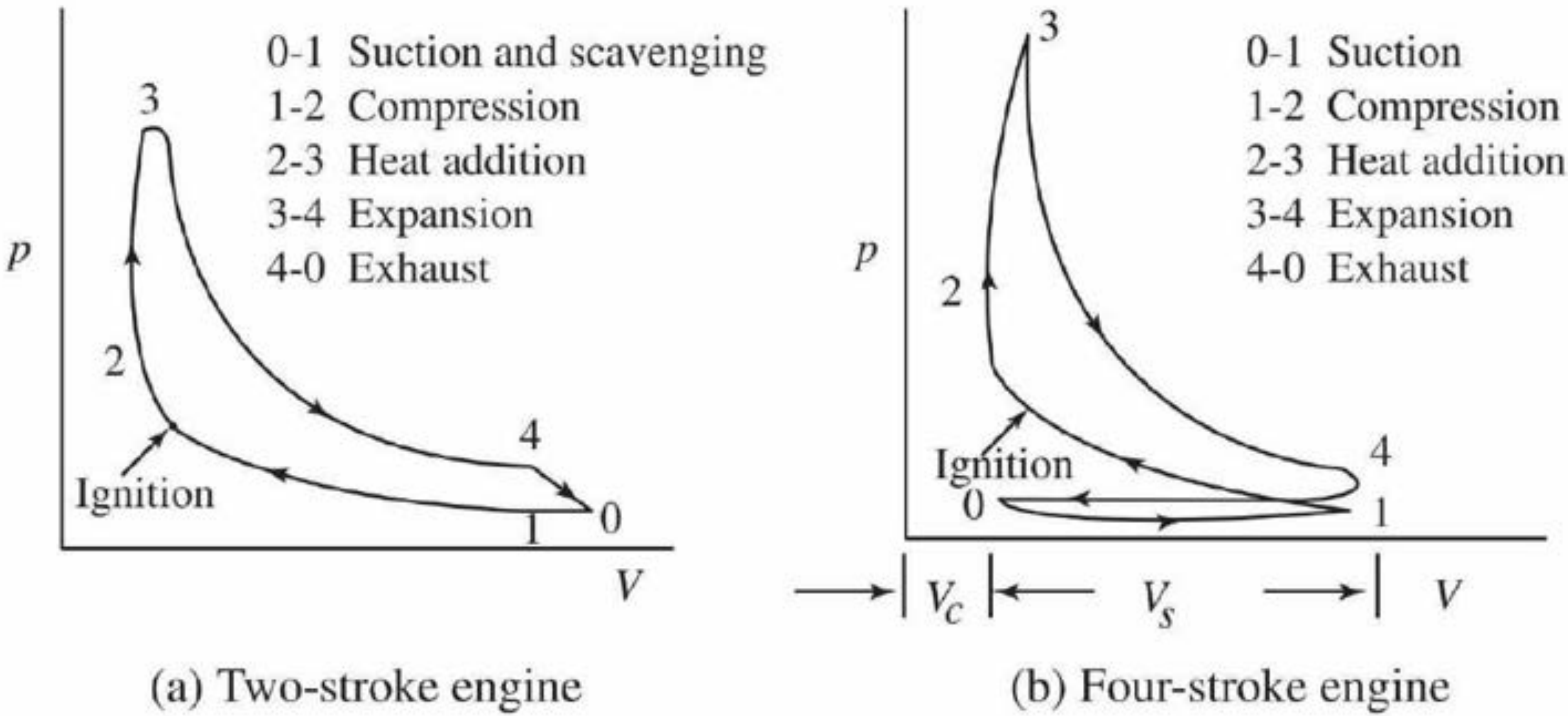


Fig. Actual p-V diagrams

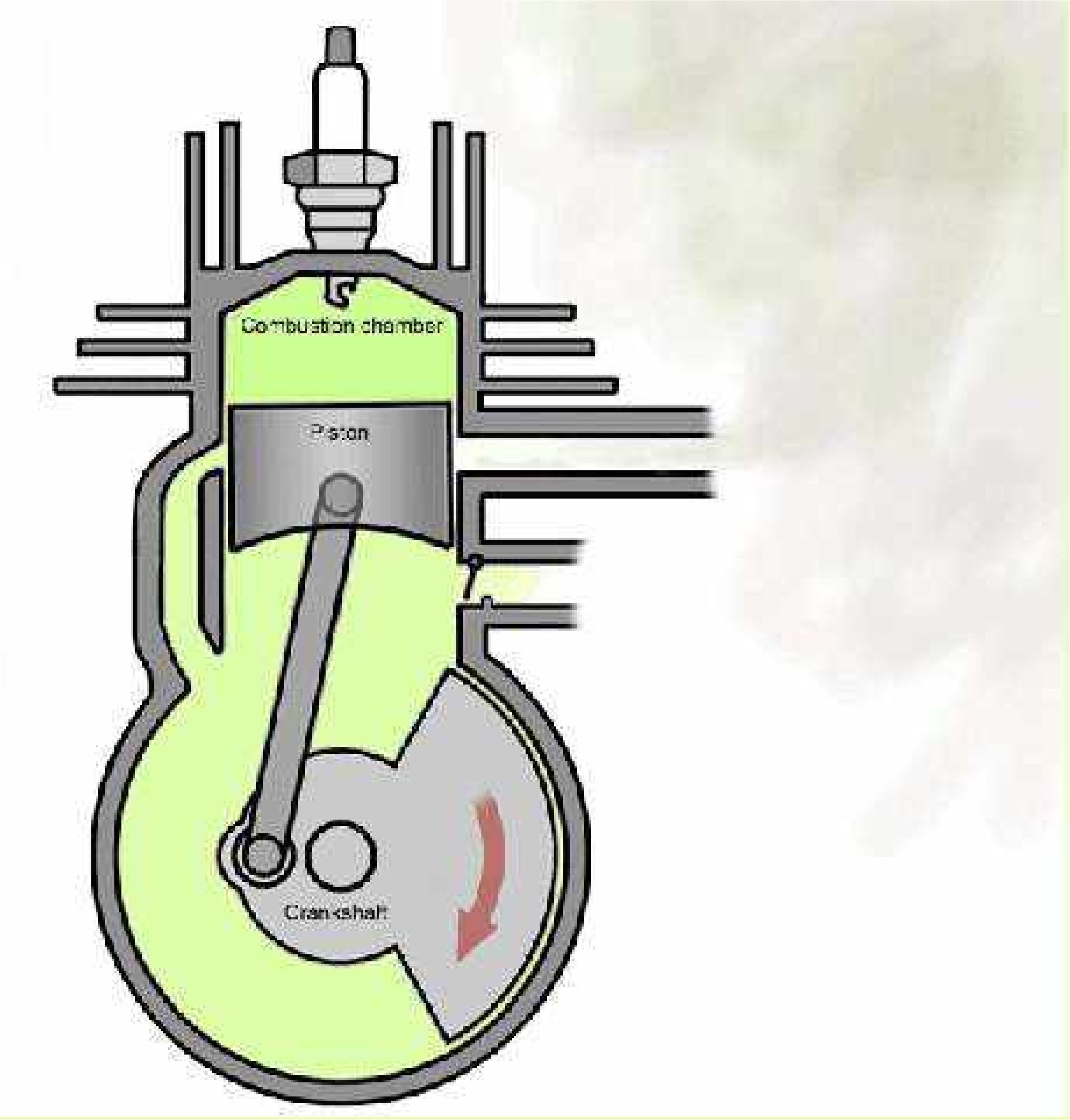
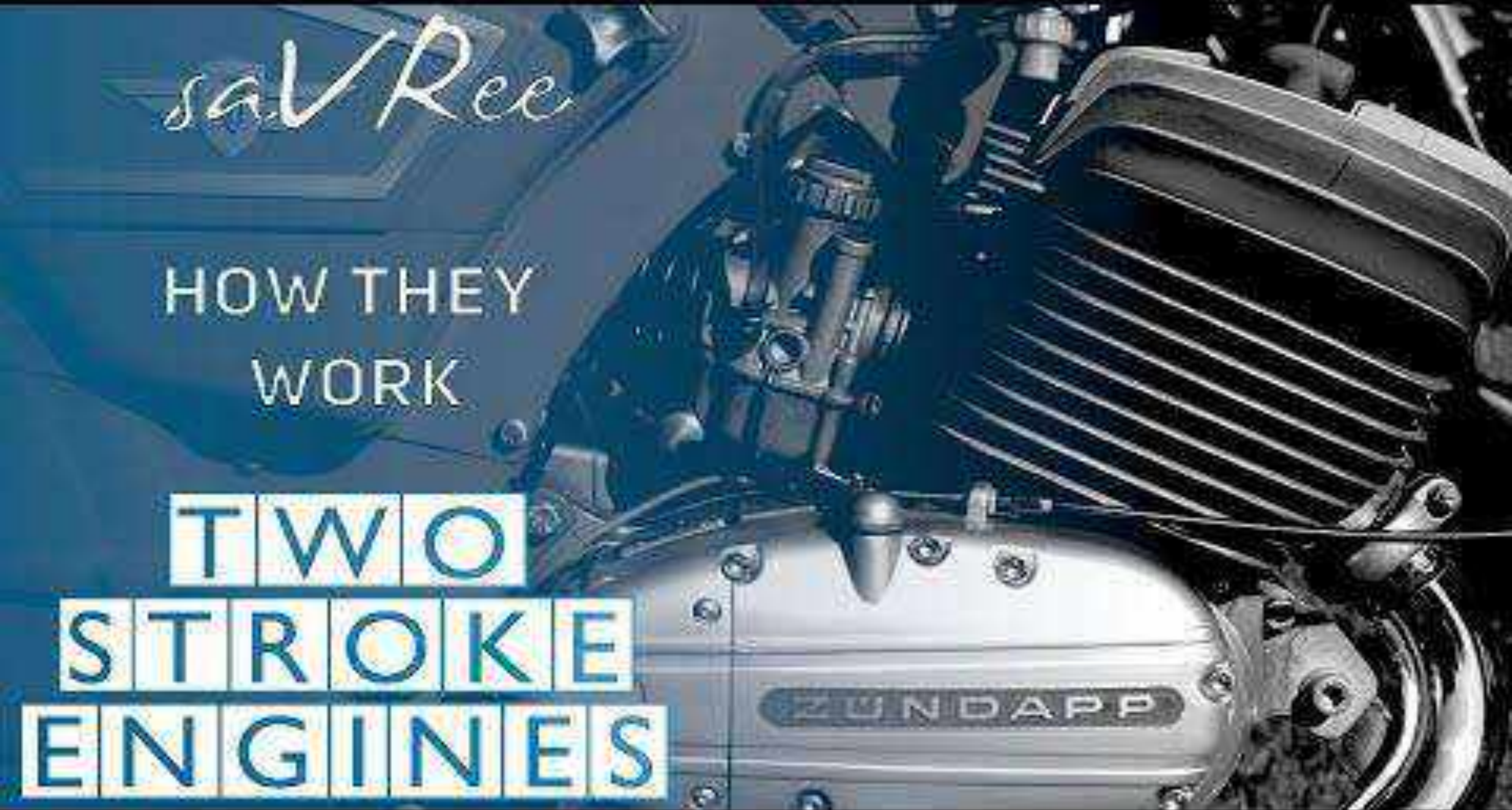
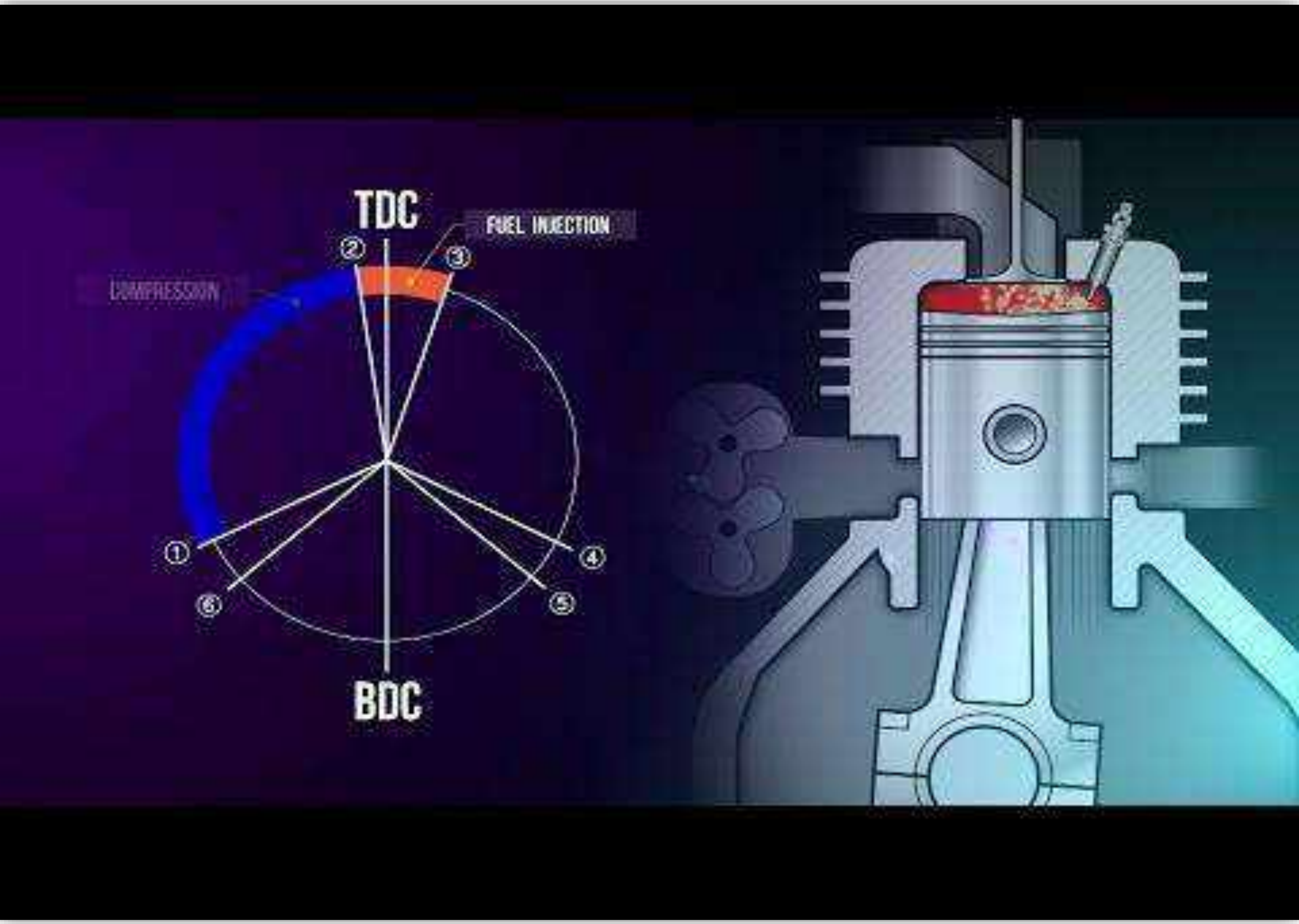


Fig: Working of Two Stroke Petrol Engine


Deutsche Animation


saVRee
HOW THEY WORK
TWO STROKE ENGINES
ZÜNDAPP


TDC
FUEL INJECTION
COMPRESSION
BDC

Comparison of Four and Two-Stroke Cycle Engines

Four-Stroke Engine	Two-Stroke Engine
The thermodynamic cycle is completed in four strokes of the piston or in two revolutions of the crankshaft. Thus, one power stroke is obtained in every two revolutions of the crankshaft.	The thermodynamic cycle is completed in two strokes of the piston or in one revolution of the crankshaft. Thus there is one power stroke for every revolution of the crankshaft.
Because of the above, turning moment is not so uniform and hence a heavier flywheel is needed.	Because of the above, turning moment is more uniform and hence a lighter flywheel can be used.
Again, because of one power stroke for two revolutions, power produced for same size of engine is less, or for the same power the engine is heavier and bulkier.	Because of one power stroke for every revolution, power produced for same size of engine is twice, or for the same power the engine is lighter and more compact.
Because of one power stroke in two revolutions lesser cooling and lubrication requirements. Lower rate of wear and tear.	Because of one power stroke in one revolution greater cooling and lubrication requirements. Higher rate of wear and tear.

Four-stroke engines have valves and valve actuating mechanisms for opening and closing of the intake and exhaust valves.	Two-stroke engines have no valves but only ports (some two-stroke engines are fitted with conventional exhaust valve or reed valve).
Because of comparatively higher weight and complicated valve mechanism, the initial cost of the engine is more.	Because of light weight and simplicity due to the absence of valve actuating mechanism, initial cost of the engine is less.
Higher volumetric efficiency due to more time for mixture intake.	Lower volumetric efficiency due to lesser time for mixture intake.
Thermal efficiency is higher; part load efficiency is better.	Thermal efficiency is lower; part load efficiency is poor.
Used where efficiency is important, viz., in cars, buses, trucks, tractors, industrial engines, aero planes, power generation etc.	Used where low cost, compactness and light weight are important, viz., in mopeds, scooters, motorcycles, hand sprayers etc.


Donut
2
STROKE
VS
4
STROKE

COMPARISON OF TWO–STROKE SI AND CI ENGINES

The two-stroke SI engine suffers from two big disadvantages—fuel loss and idling difficulty. The two-stroke CI engine does not suffer from these disadvantages and hence CI engine is more suitable for two-stroke operation.

If the fuel is supplied to the cylinders after the exhaust ports are closed, there will be no loss of fuel and the indicated thermal efficiency of the two-stroke engine will be as good as that of a four-stroke engine. However, in an SI engine using carburettor, the scavenging is done with fuel-air mixture and only the fuel mixed with the retained air is used for combustion. To avoid the fuel loss, fuel-injection just before the exhaust port closure may be used instead of a carburettor.

The two-stroke SI engine picks up only gradually and may even stop at low speeds when mean effective pressure is reduced to about 1.2 bar. This is because a large amount of residual gas (more than in four-stroke engine) mixing with small amount of charge. At low speeds there may be backfiring due to slow burning rate. Fuel-injection improves idling and also eliminates backfiring as there is no fuel present in the inlet system.

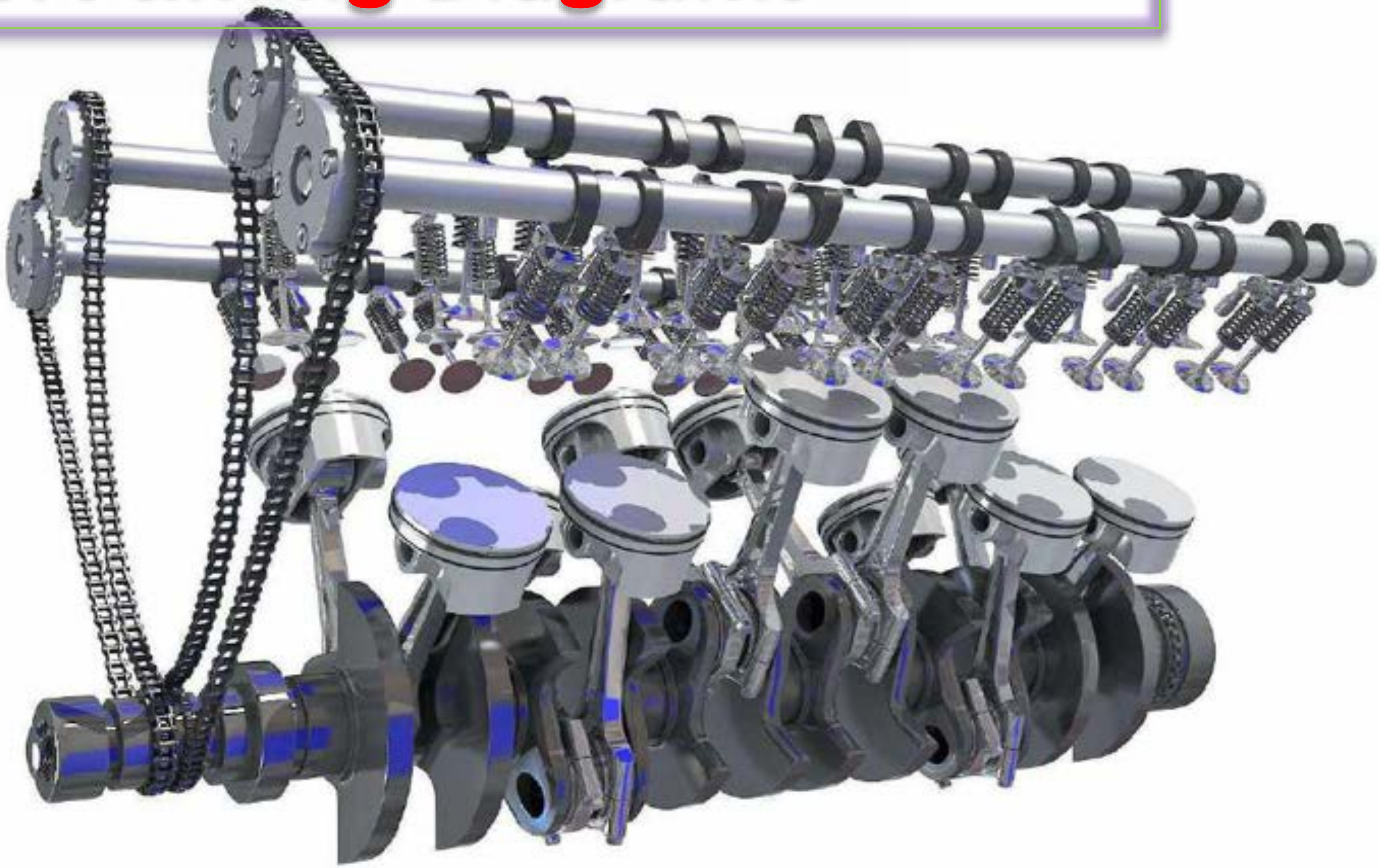
In CI engines there is no loss of fuel as the charge is only air and there is no difficulty at idling because the fresh charge (air) is not reduced.

HOW TO TELL A TWO-STROKE CYCLE ENGINE FROM A FOUR-STROKE CYCLE ENGINE ?

S. No.	Distinguishing features	Four-stroke cycle engine	Two-stroke cycle engine
1.	*Oil sump and oil-filter plug*	It has an oil sump and oil-filter plug.	It does not have oil sump and oil-filter plug.
2.	*Oil drains etc.*	It requires oil drains and refills periodically, just an automobile do.	In this type of engine, the oil is added to the gasoline so that a mixture of gasoline and oil passes through the carburettor and enters the crankcase with the air.
3.	*Location of muffler (exhaust silencer)*	It is installed at the head end of the cylinder at the exhaust valve location.	It is installed towards the middle of the cylinder, at the exhaust port location.
4.	Name plate	If the name plate mentions the type of oil and the crankcase capacity, or similar data, it is a four- stroke cycle engine.	If the name plate tells to mix oil with the gasoline, it is a two-stroke cycle engine.


2 STROKE
962
VS
4 STROKE

Valve Timing & Port timing Diagrams



Parts for Diesel engine only :

1. Fuel pump.
2. Injector.

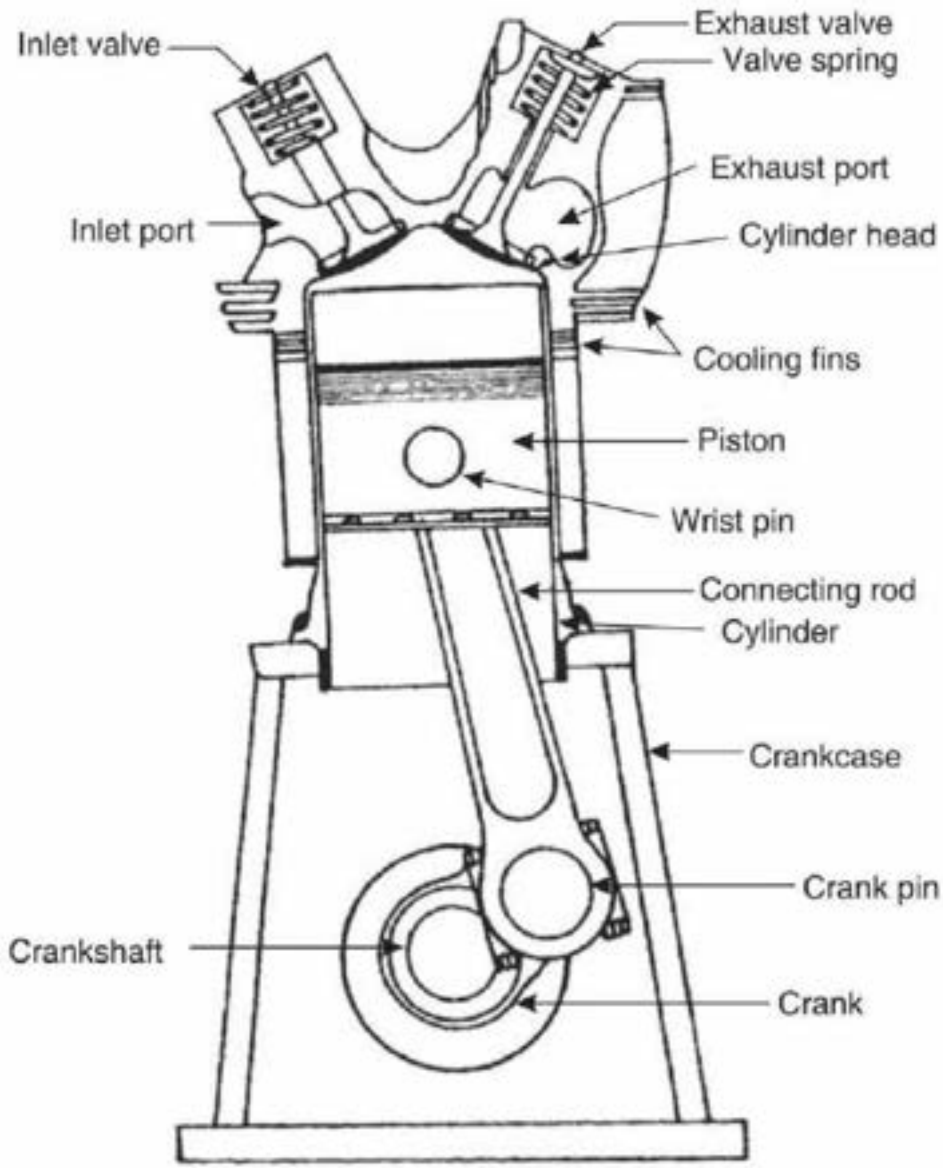
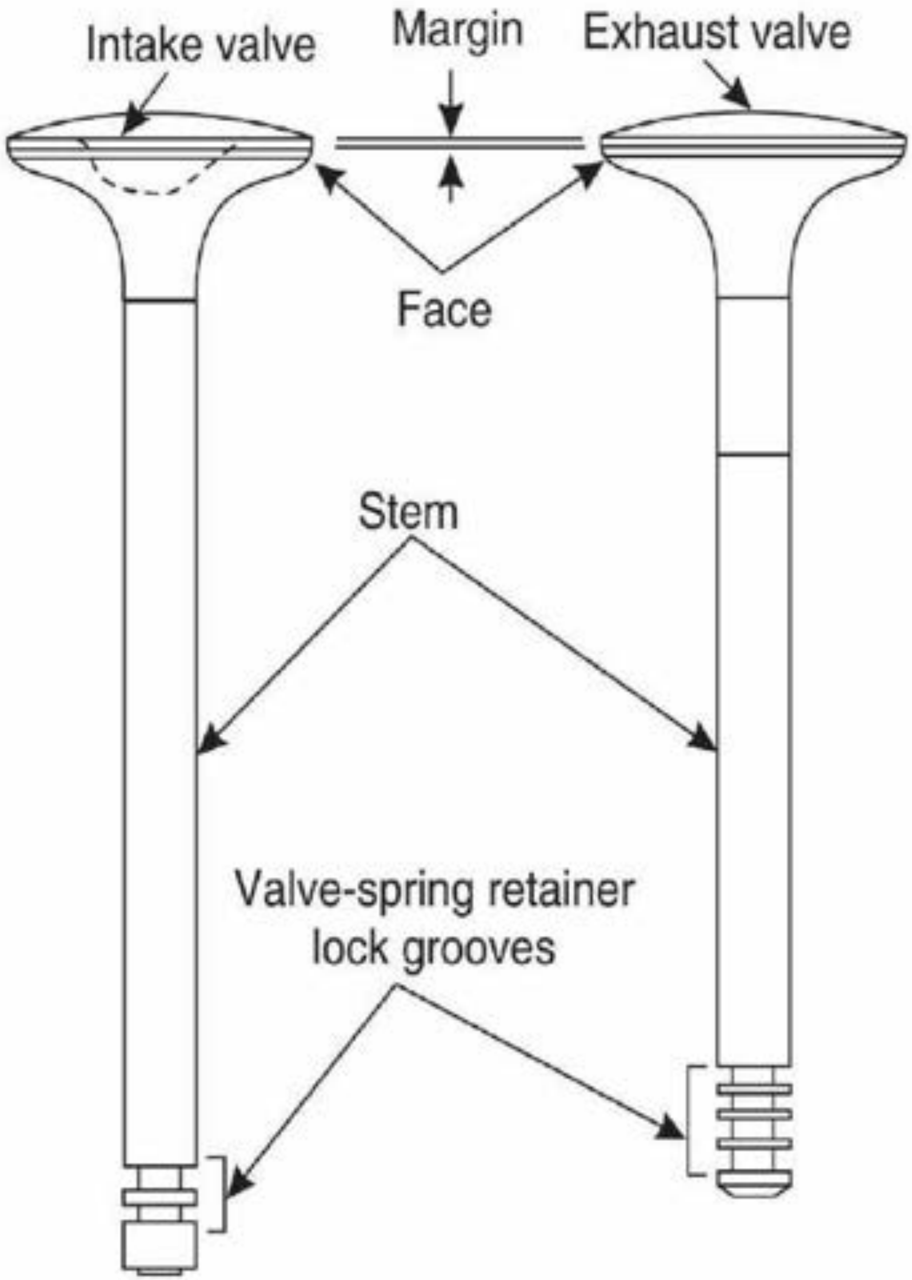
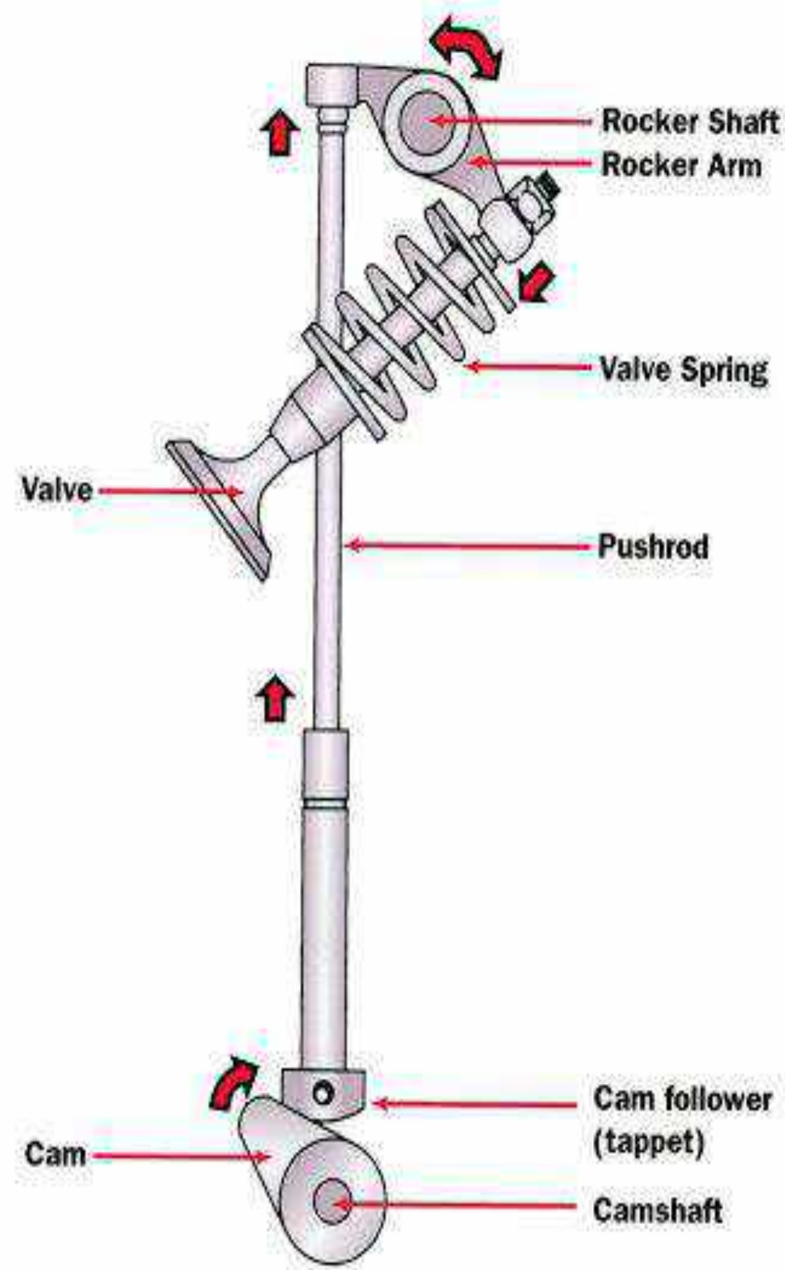


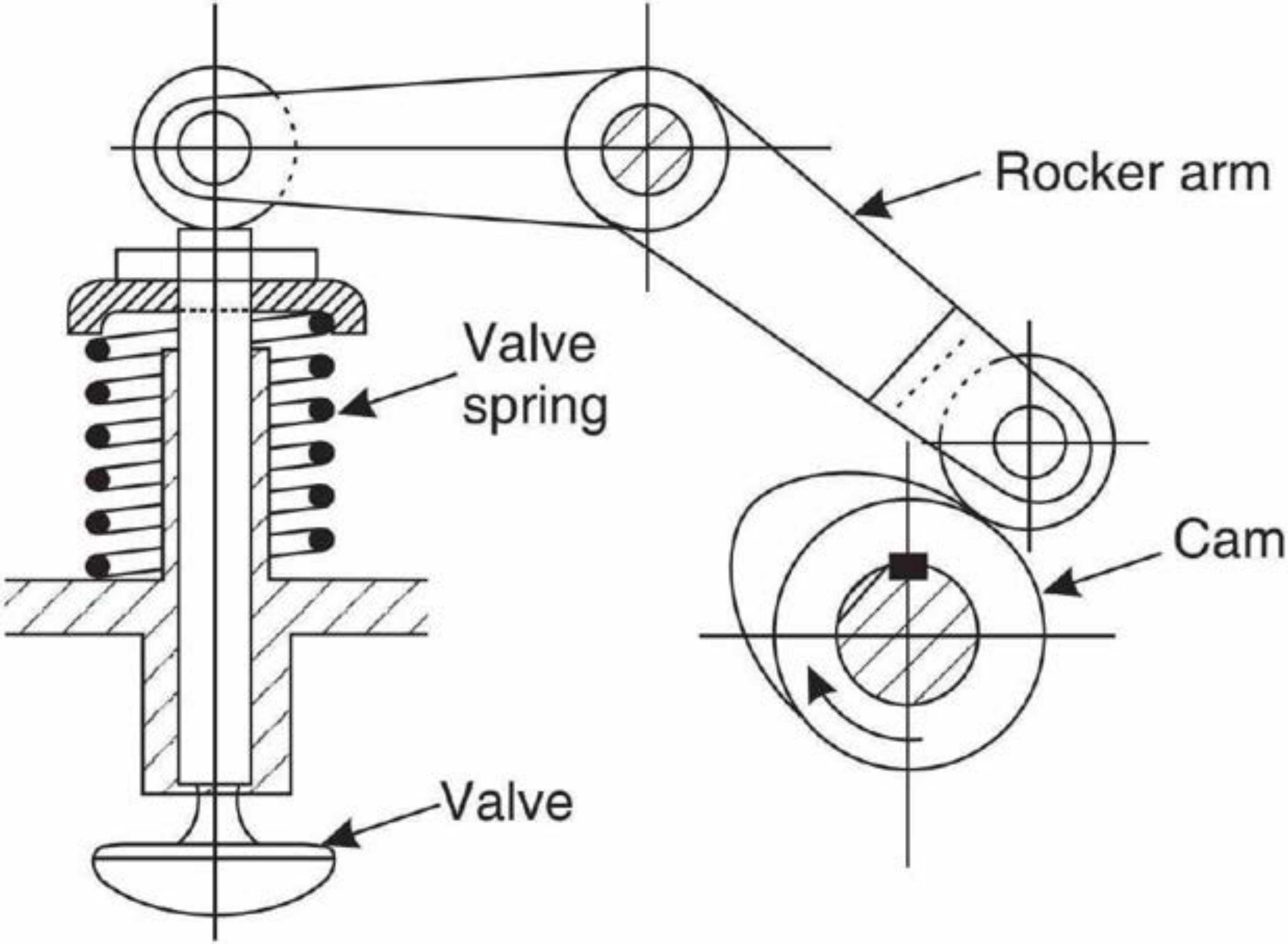
Fig. Air-cooled I.C. engine.


Rockerarm
Valve
Pushrod
Combustion Space
Piston
Camshaft
Connecting Rod
Crankshaft
Crankweb



(a) Inlet valve (b) Exhaust valve





Cam and rocker arm.

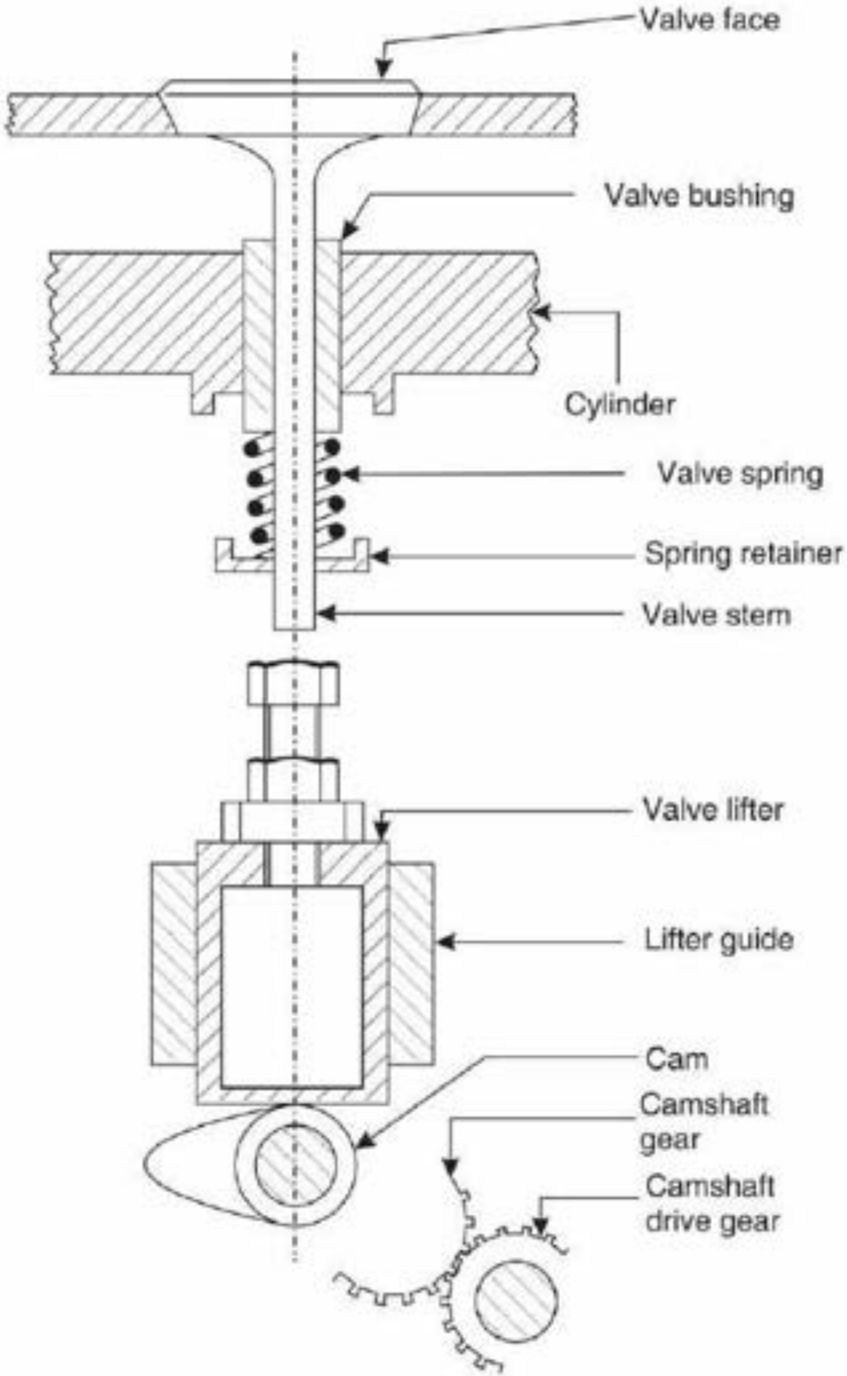


Fig. Valve gear for I.C. engine.

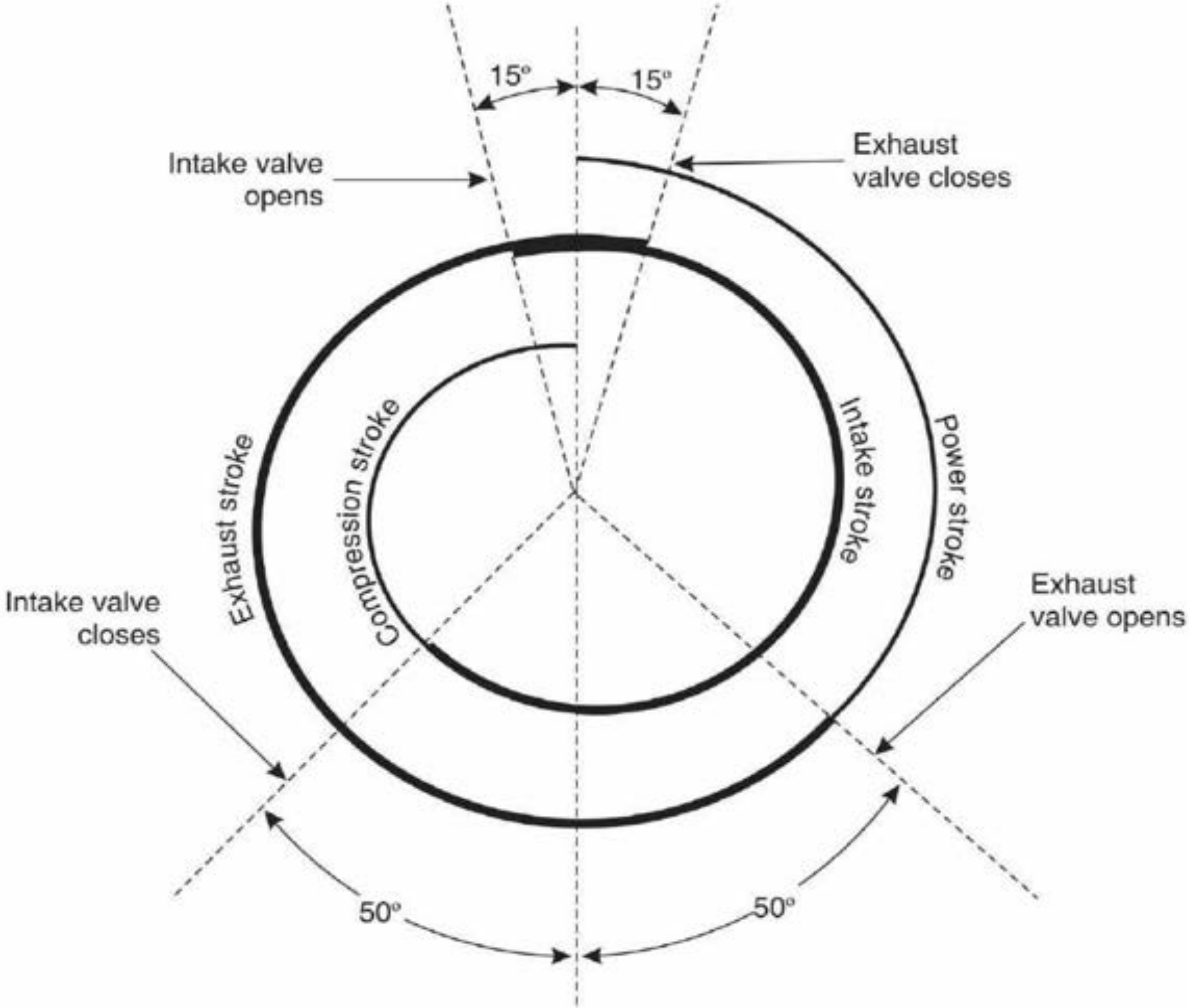


Fig. Typical valve timing diagram.

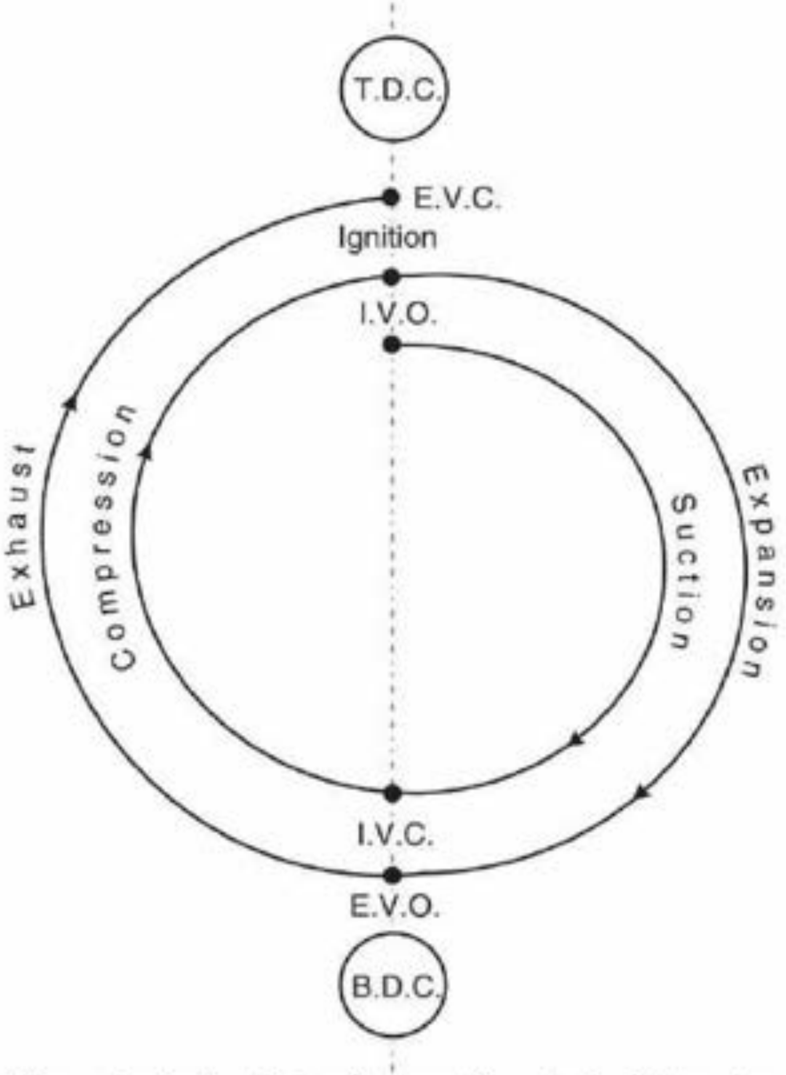


Fig. Theoretical valve timing diagram (four-stroke Otto cycle engin

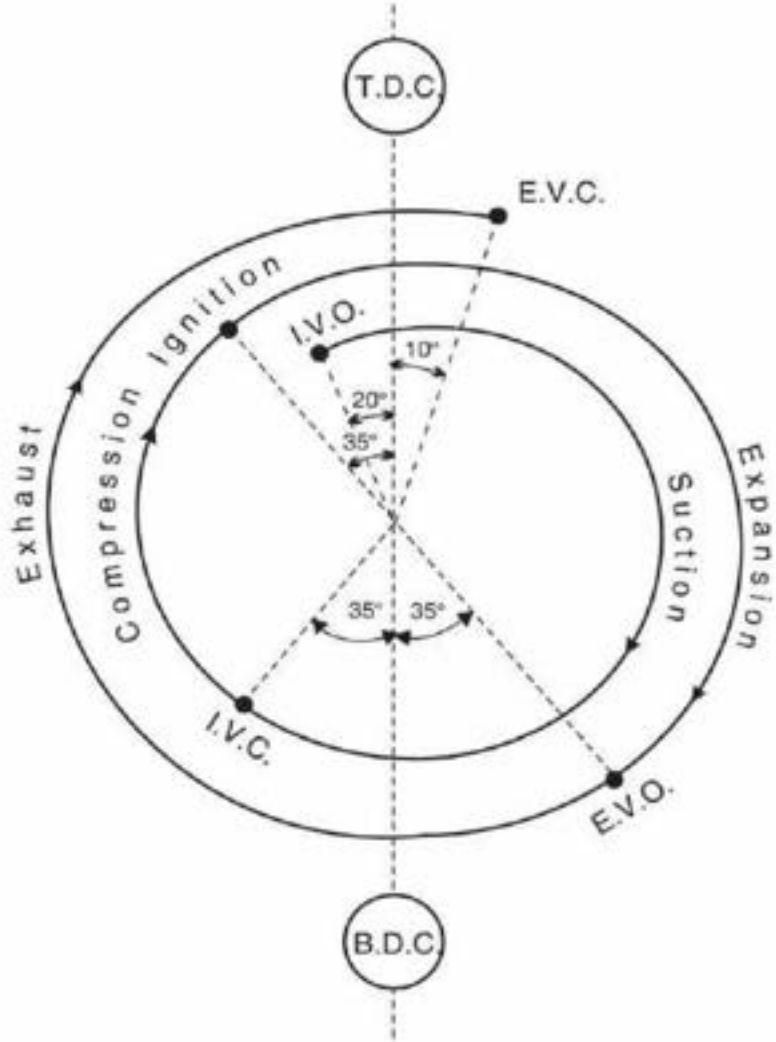
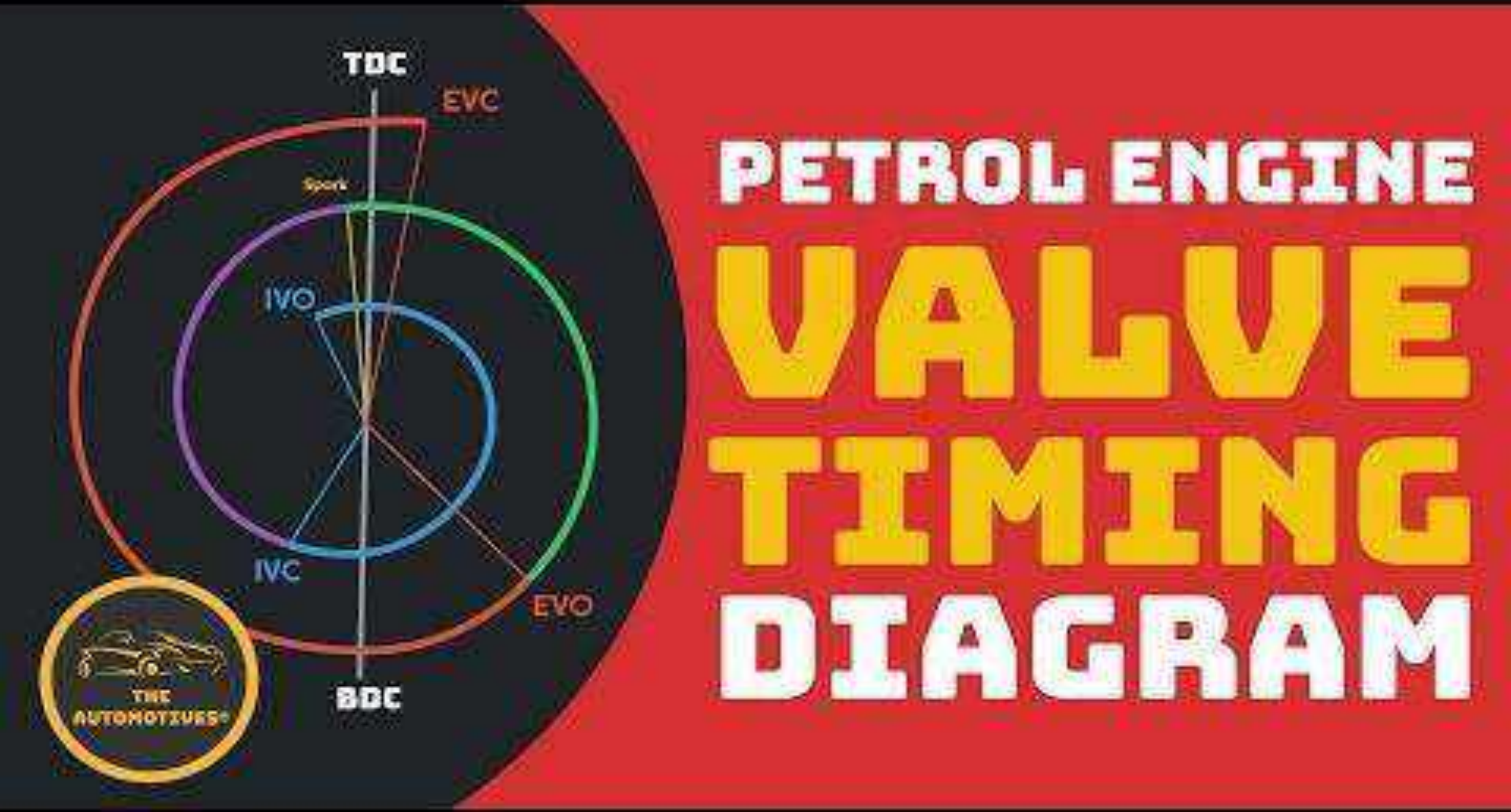
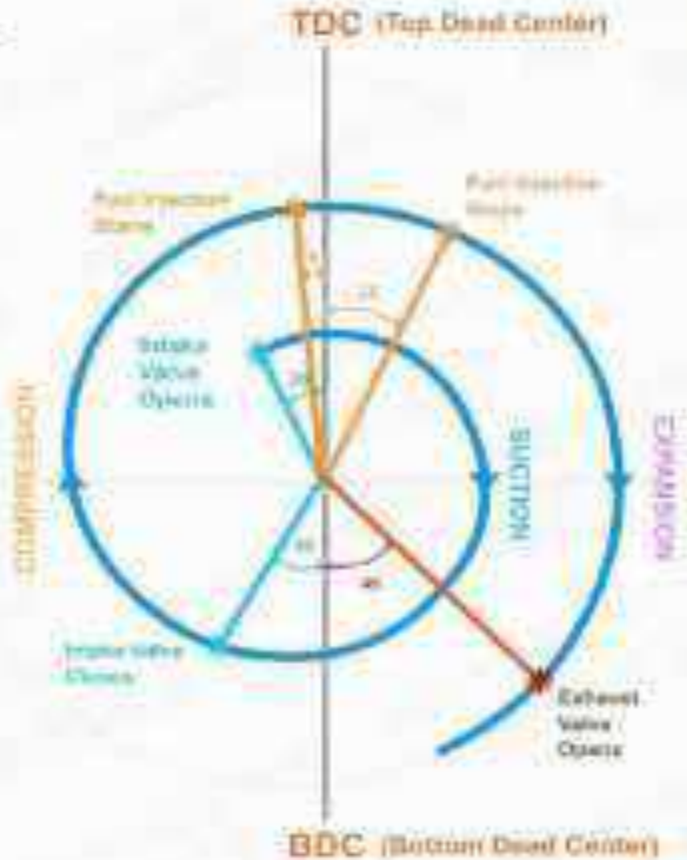


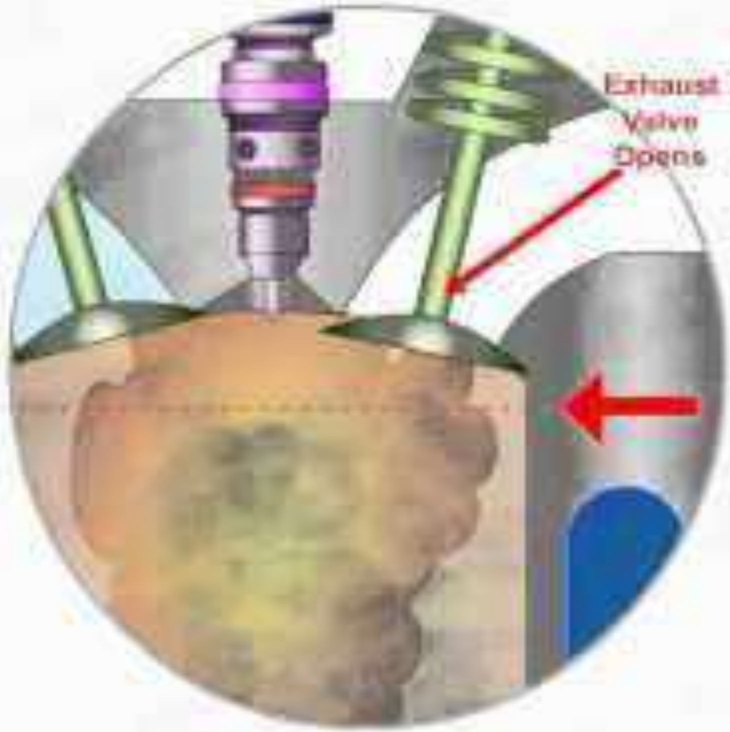
Fig. Actual valve timing diagram (four-stroke Otto cycle engines).



TDC
EVC
Spark
IVO
IVC
EVO
BDC
THE AUTOMOTIVES
PETROL ENGINE
VALVE
TIMING
DIAGRAM



To achieve maximum evacuation of exhaust gases, the exhaust valve is timed to open early.



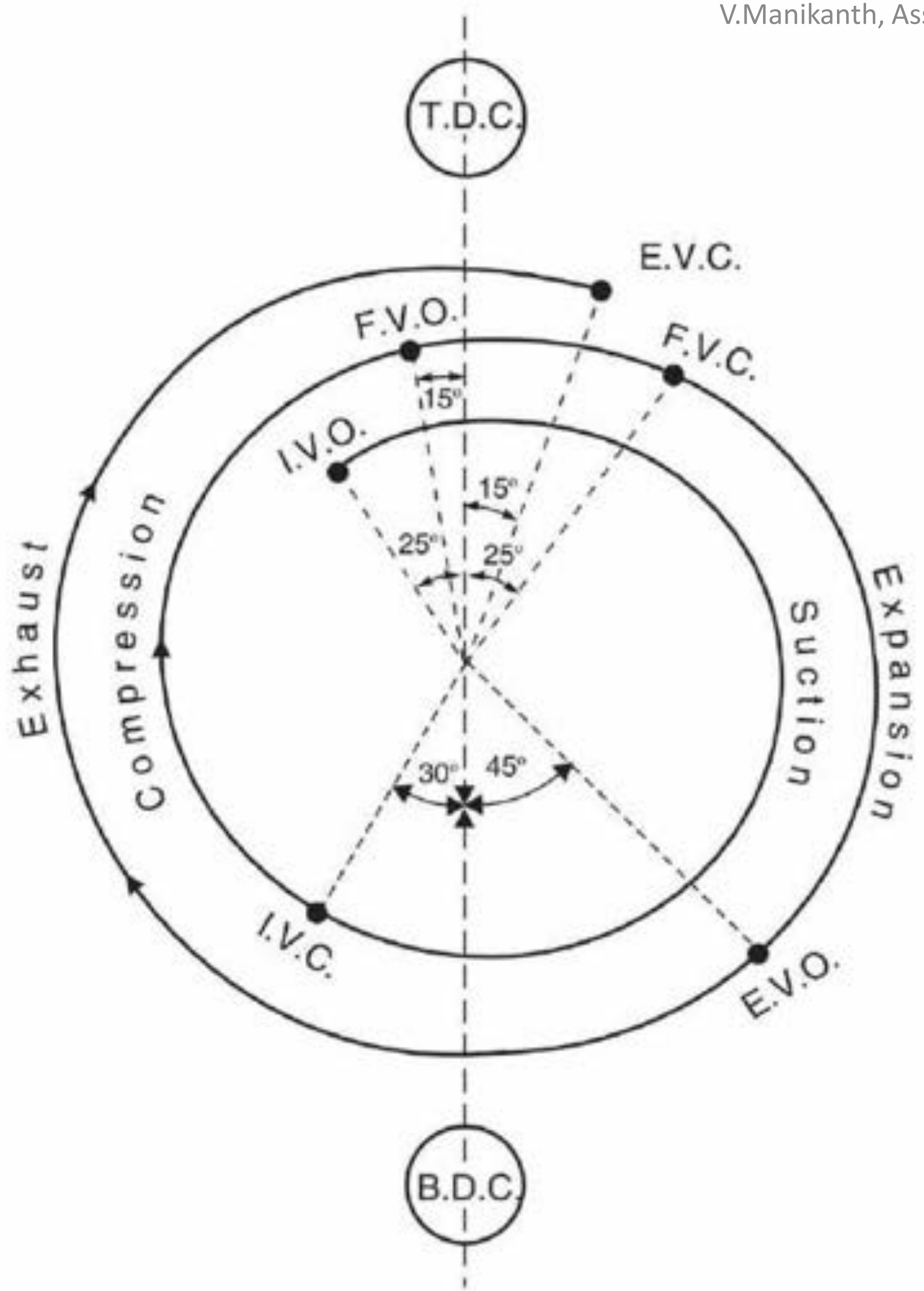
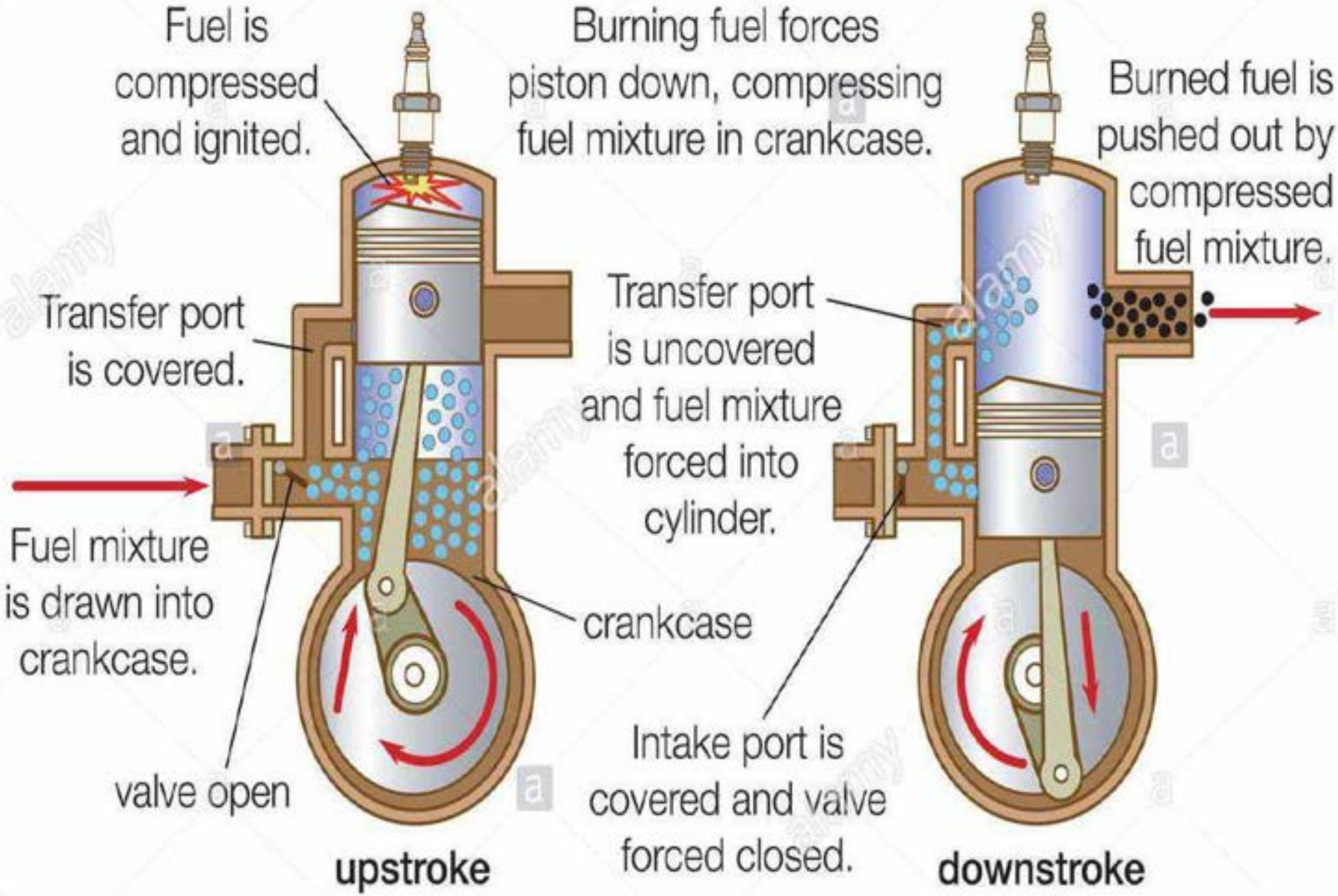
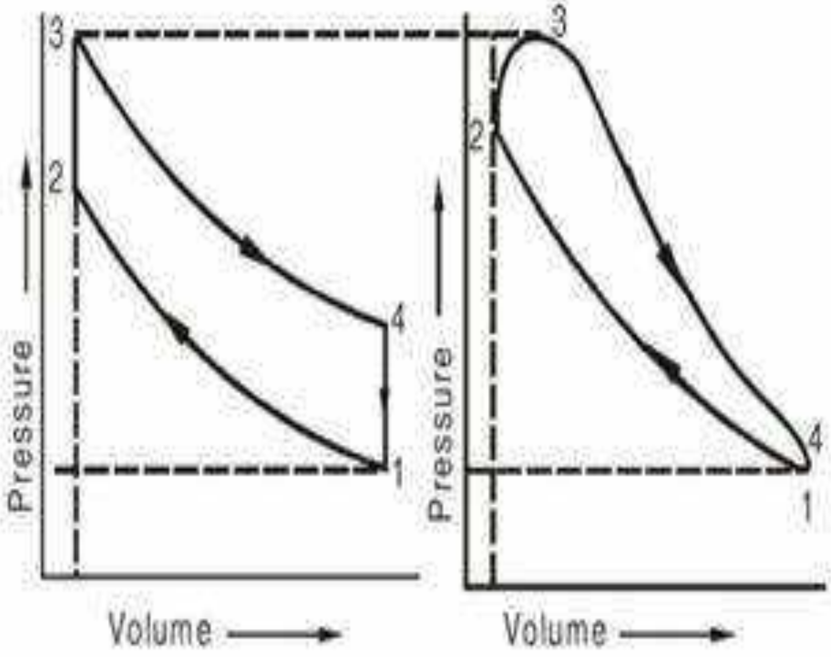
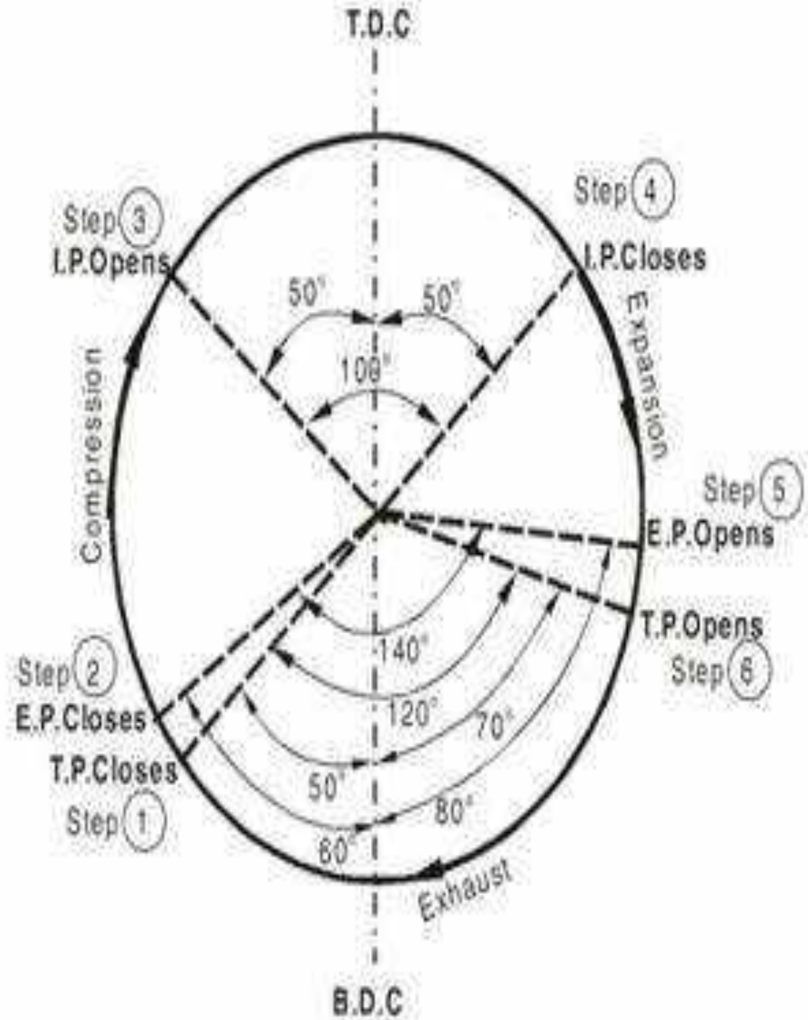


Fig. Actual valve timing diagram (four-stroke Diesel cycle engines).


Fuel is compressed and ignited.
Burning fuel forces piston down, compressing fuel mixture in crankcase.
Burned fuel is pushed out by compressed fuel mixture.
Transfer port is covered.
Transfer port is uncovered and fuel mixture forced into cylinder.
Fuel mixture is drawn into crankcase.
crankcase
valve open
Intake port is covered and valve forced closed.
upstroke
downstroke



(a) Theoritical cycle (b) Actual cycle



(c.) Port timing diagram of a two stroke petrol engine

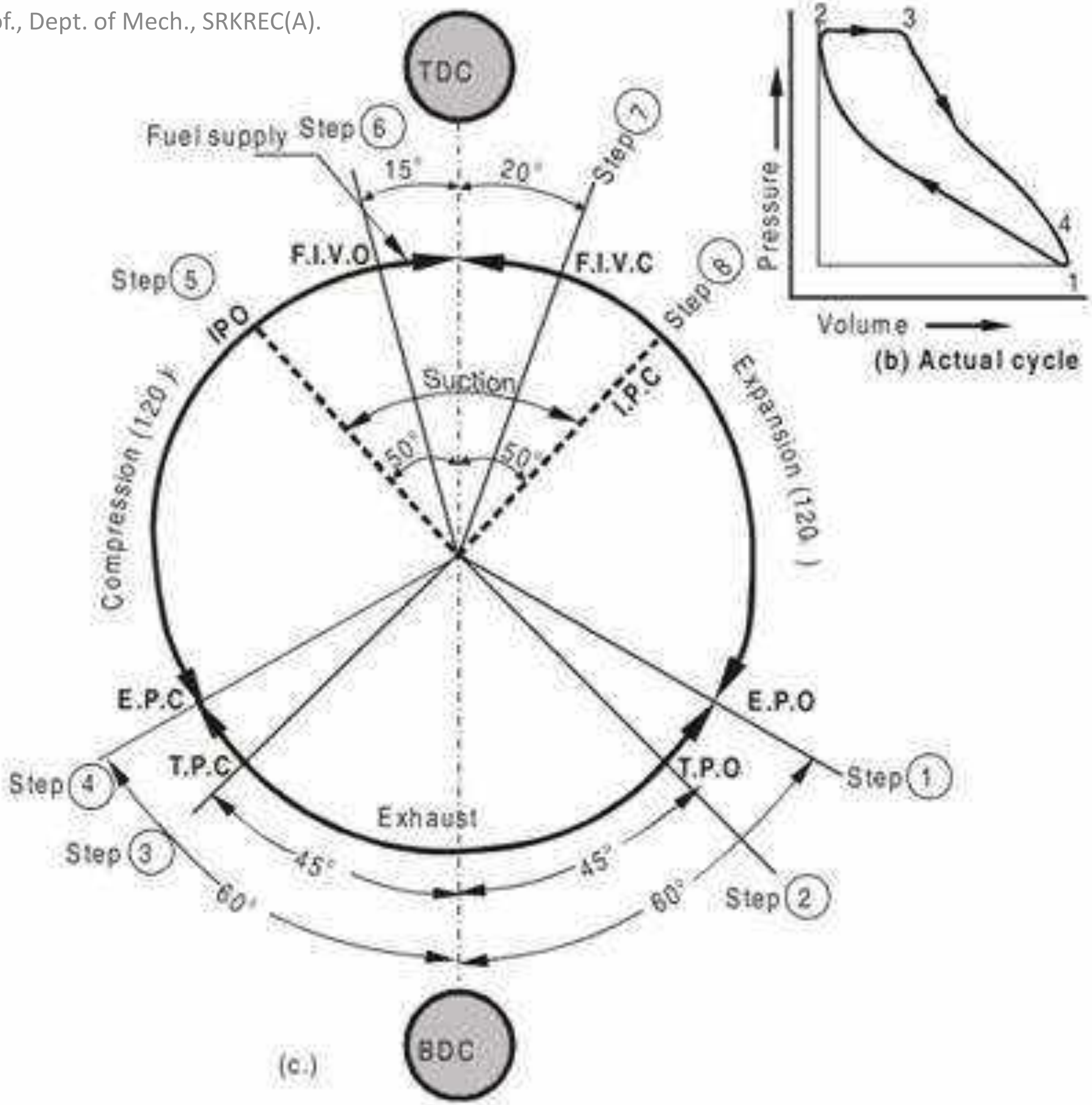
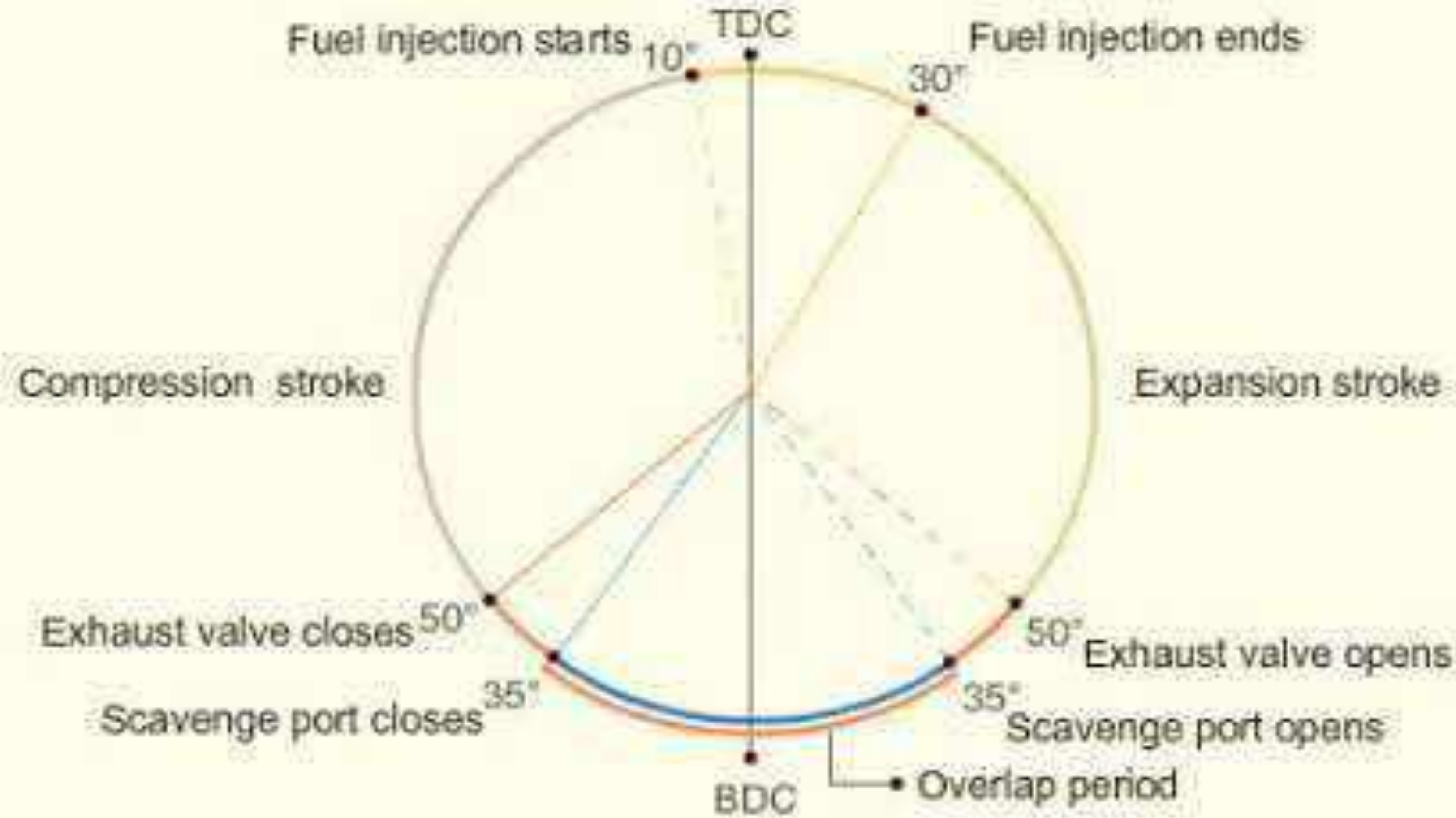
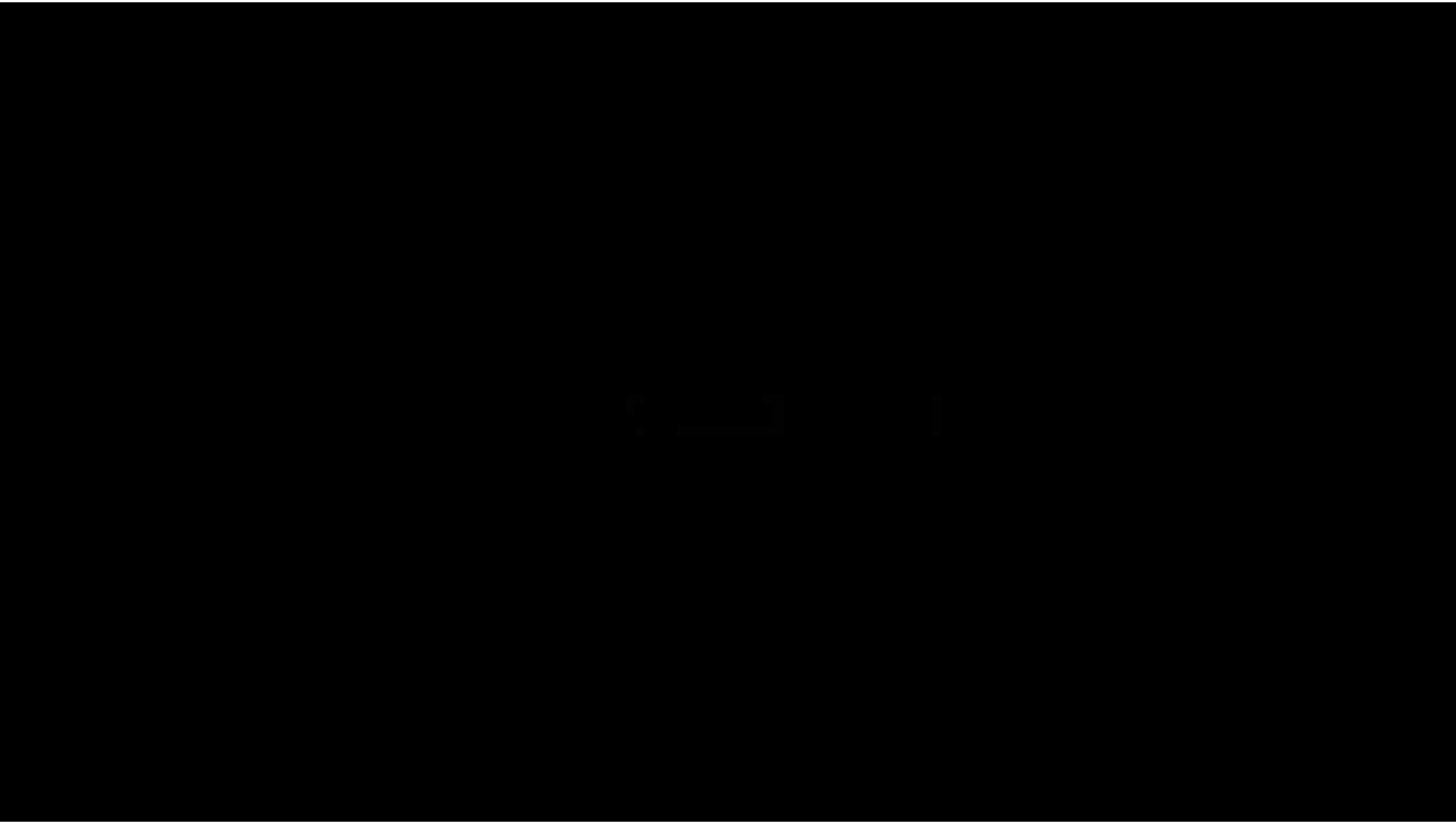


Fig. Port timing diagram for two stroke diesel engine

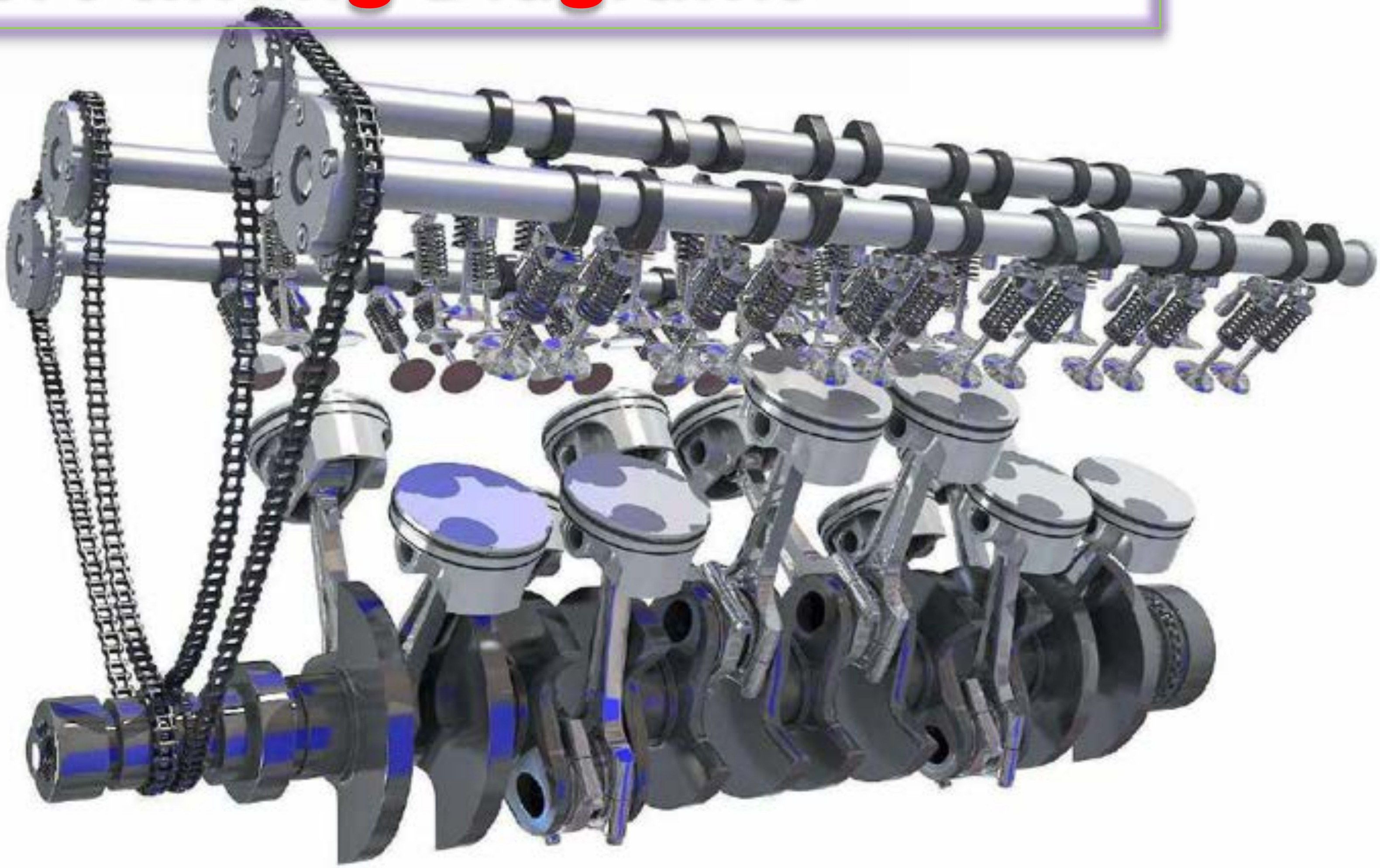


TDC
Fuel injection starts
10°
Fuel injection ends
30°
Compression stroke
Expansion stroke
Exhaust valve closes
50°
50°
Exhaust valve opens
35°
Scavenge port closes
35°
Scavenge port opens
BDC
Overlap period

How Gasoline Engine Works



Valve Timing & Port timing Diagrams



Parts for Diesel engine only :

1. Fuel pump.
2. Injector.

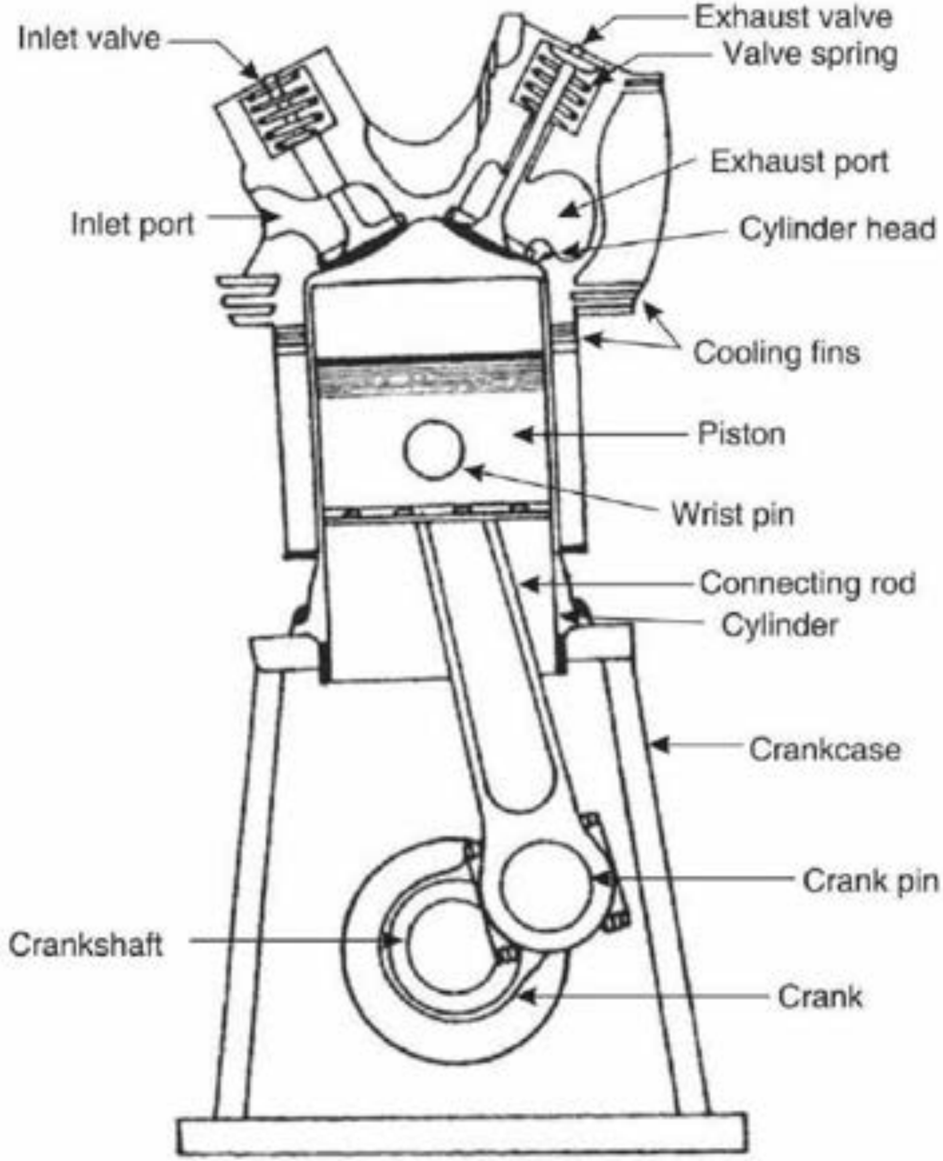
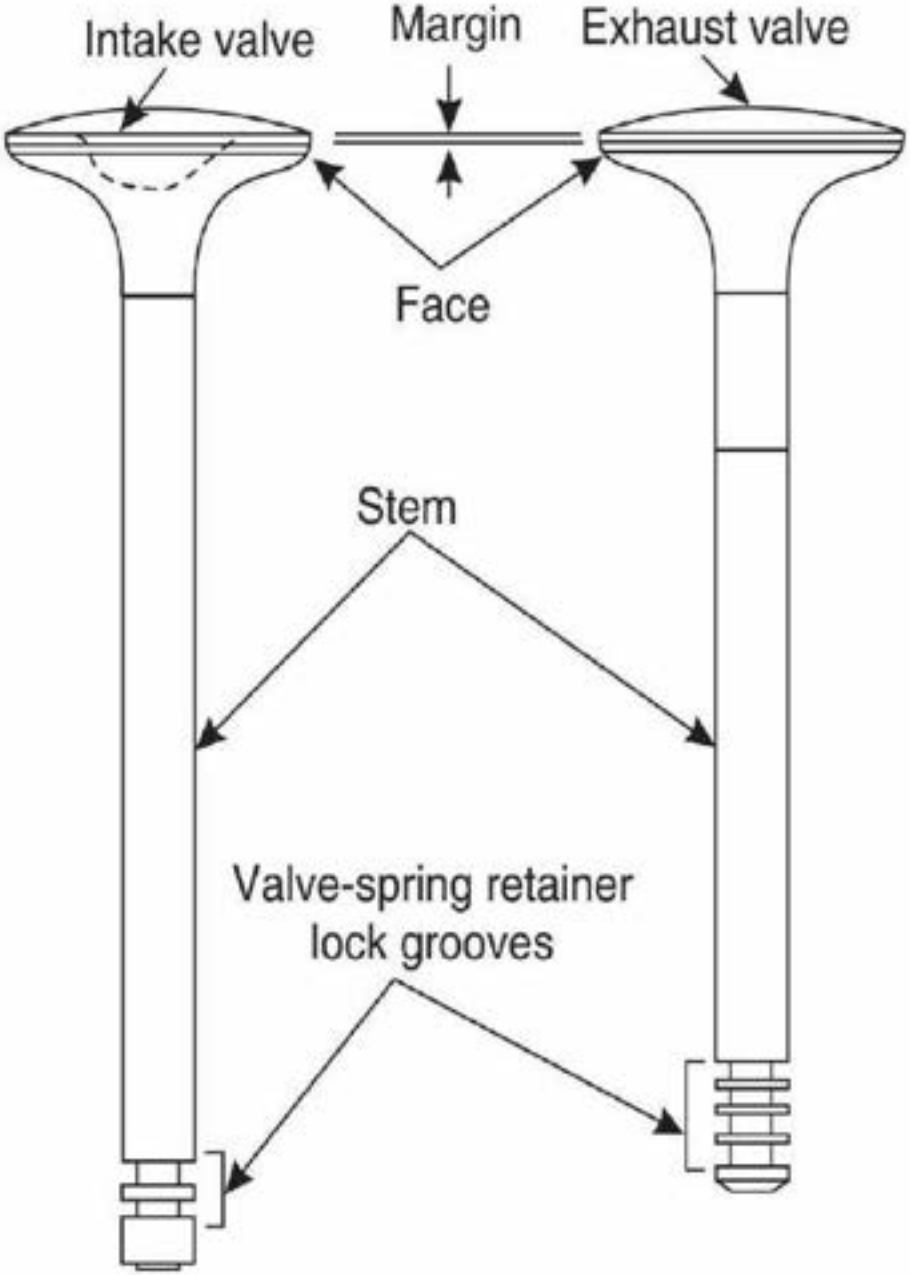


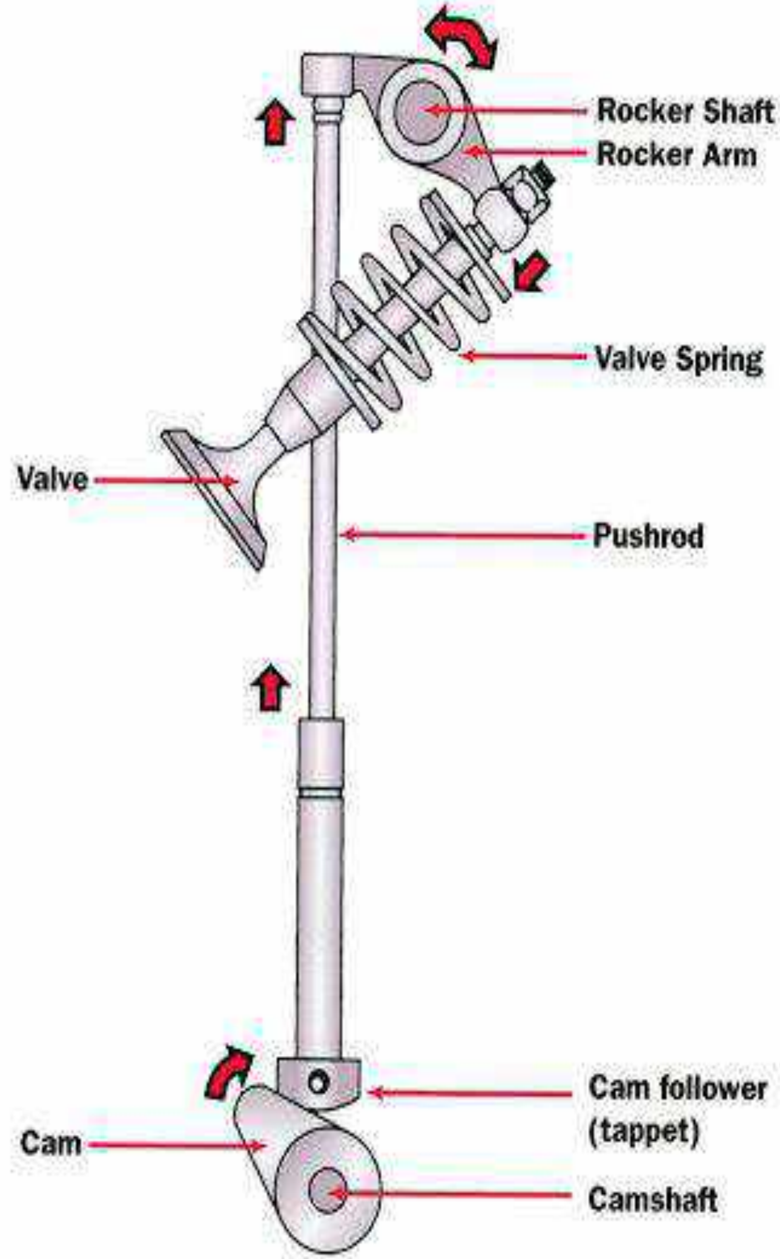
Fig. Air-cooled I.C. engine.

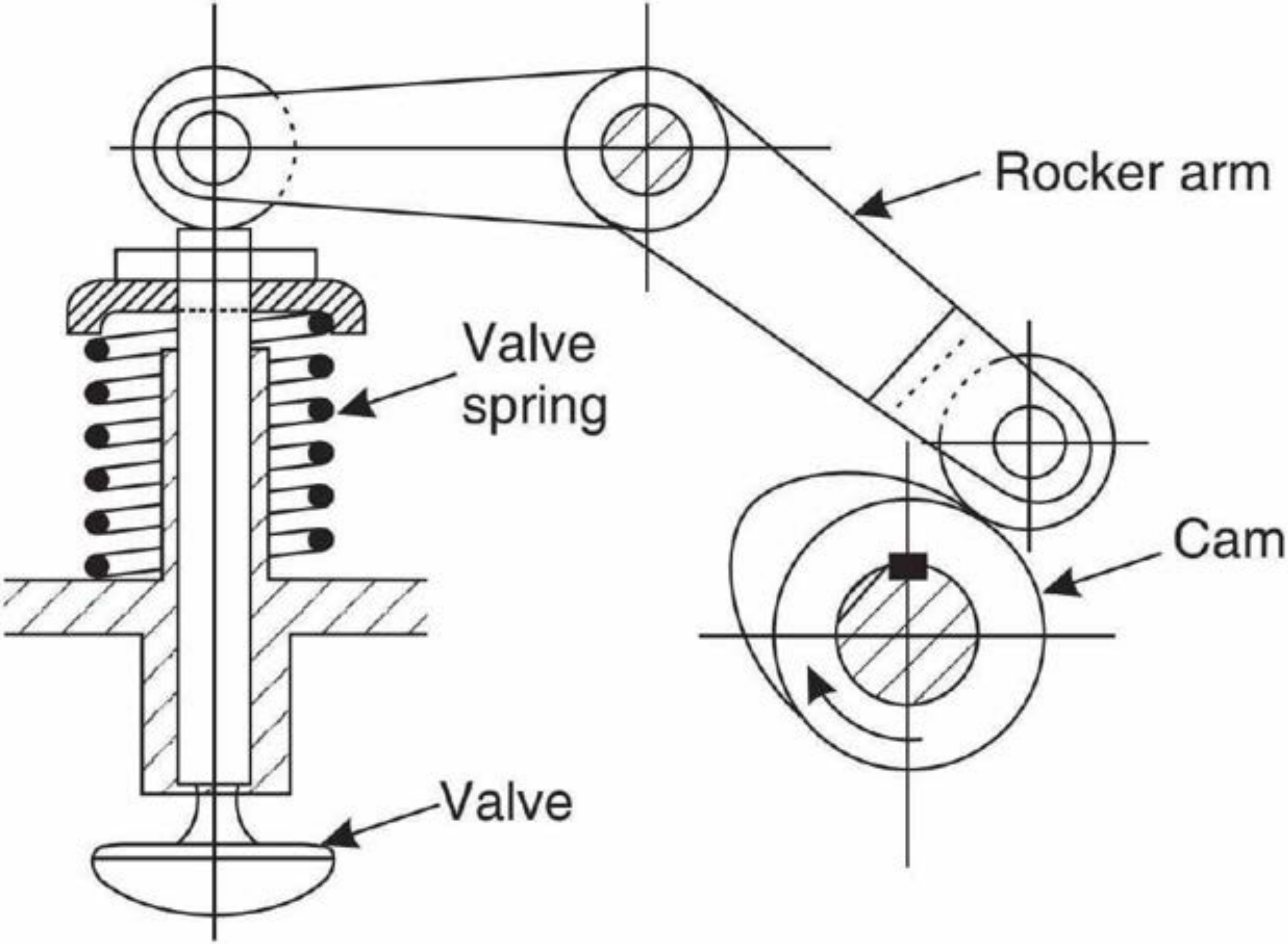

Rockerarm
Valve
Pushrod
Combustion Space
Piston
Camshaft
Connecting Rod
Crankshaft
Crankweb



(a) Inlet valve

(b) Exhaust valve





Cam and rocker arm.

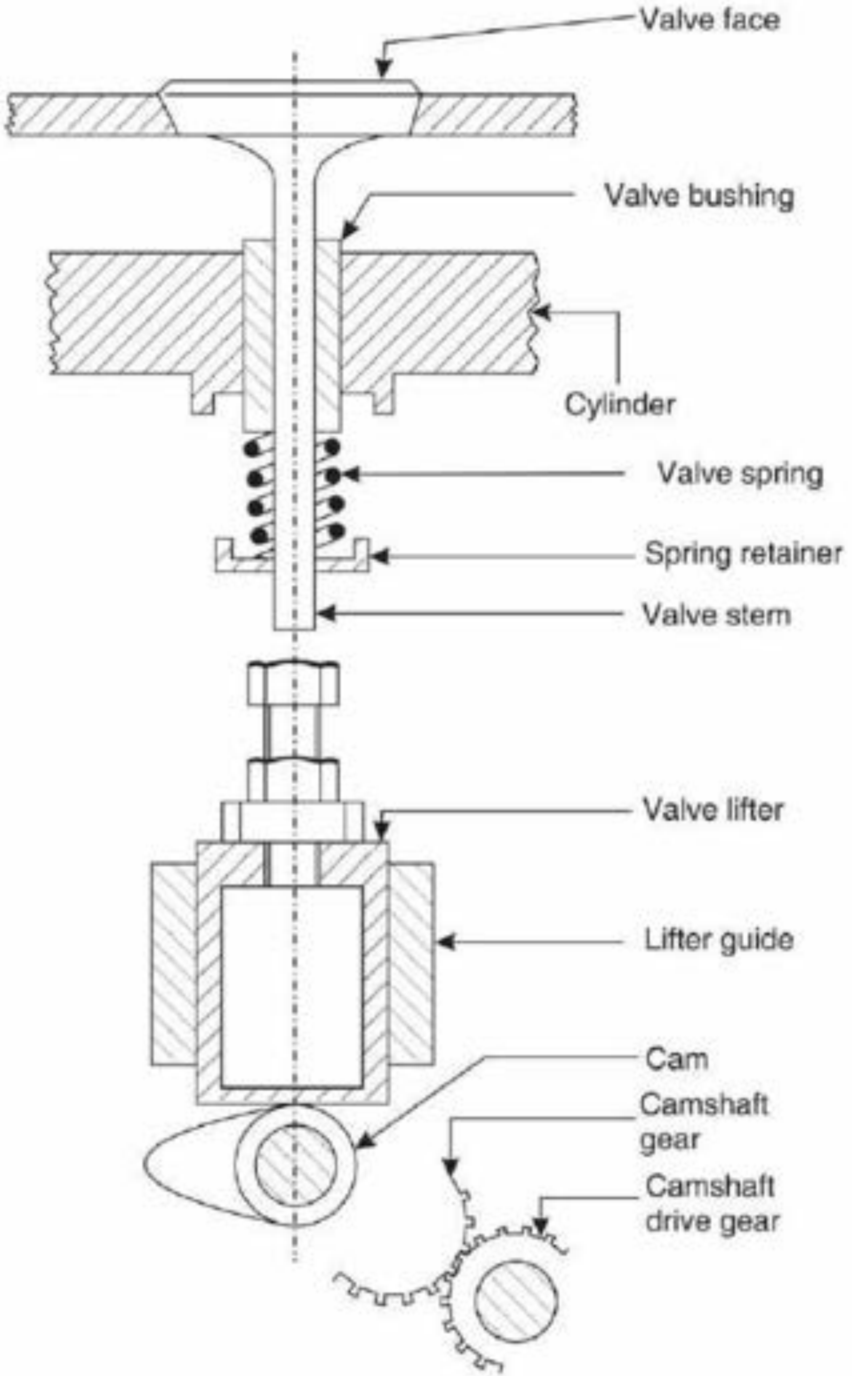
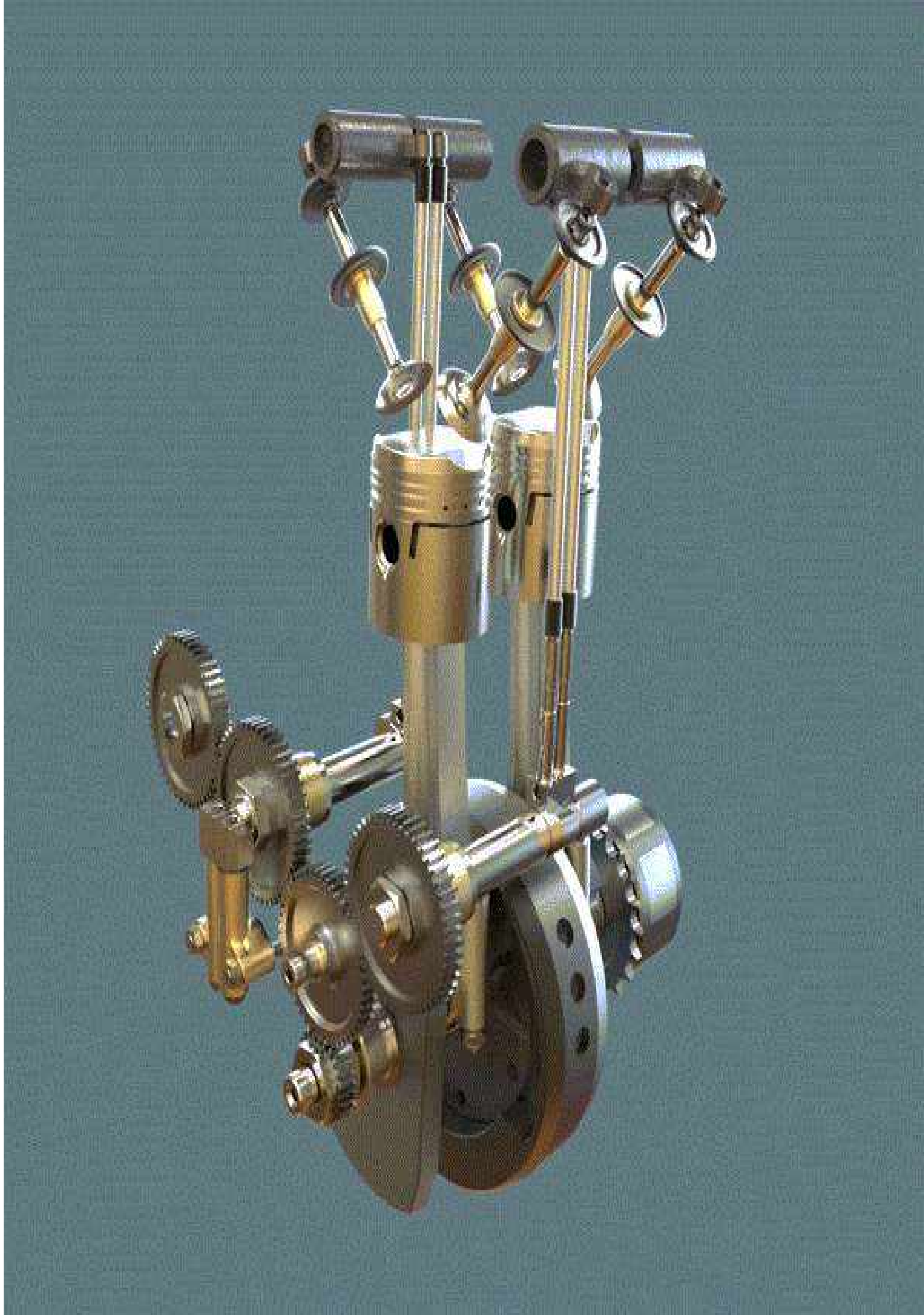


Fig. Valve gear for I.C. engine.



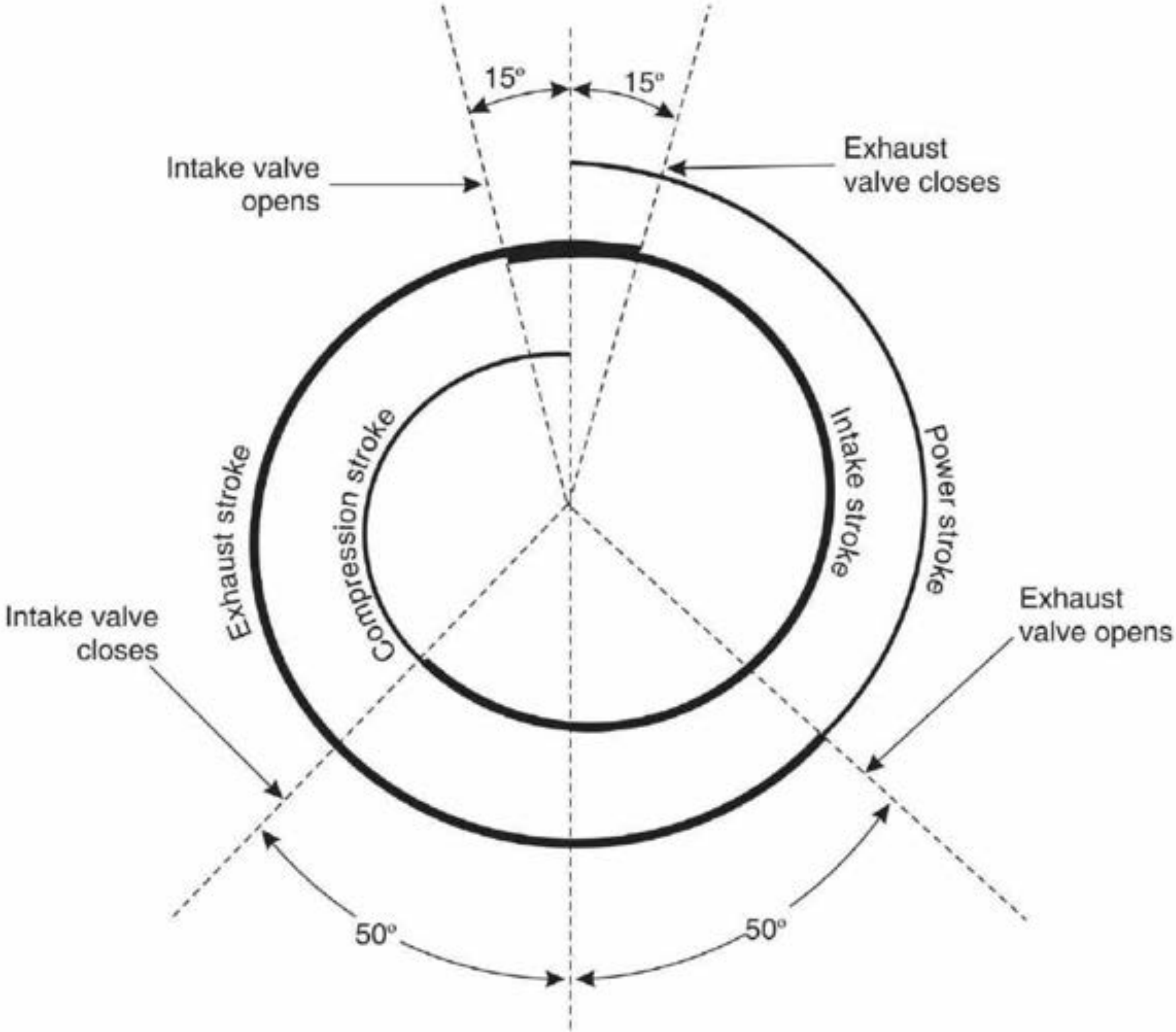


Fig. Typical valve timing diagram.

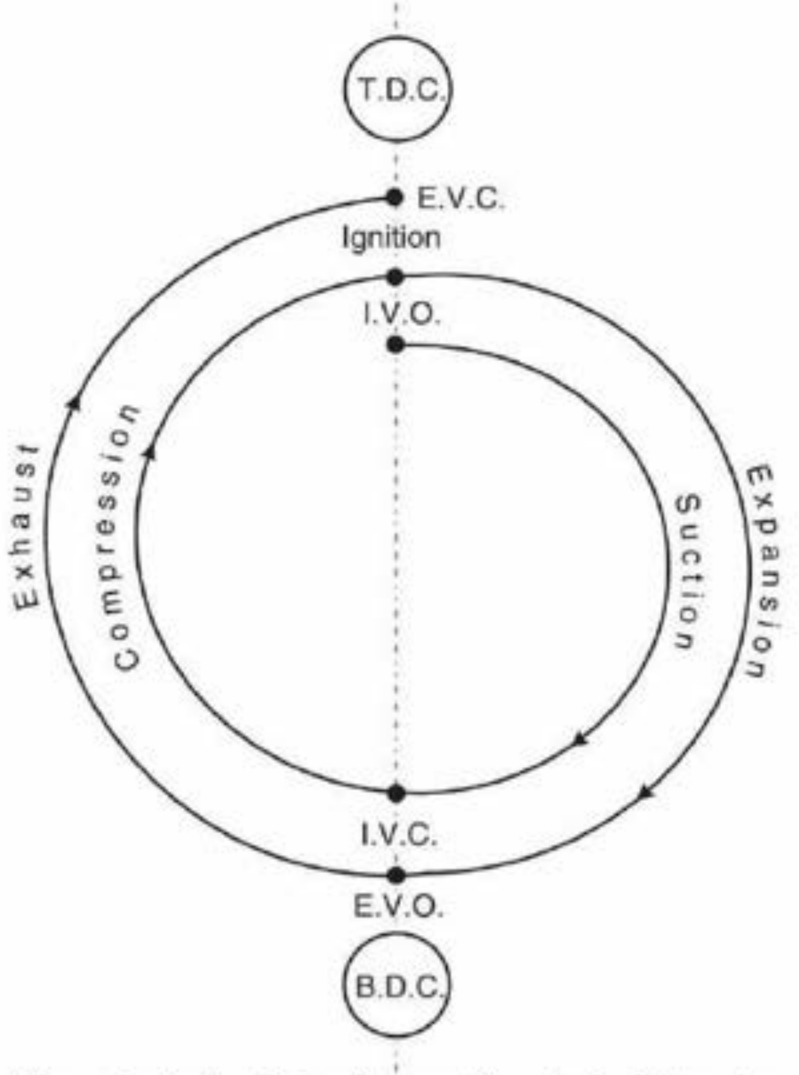


Fig. Theoretical valve timing diagram (four-stroke Otto cycle engin

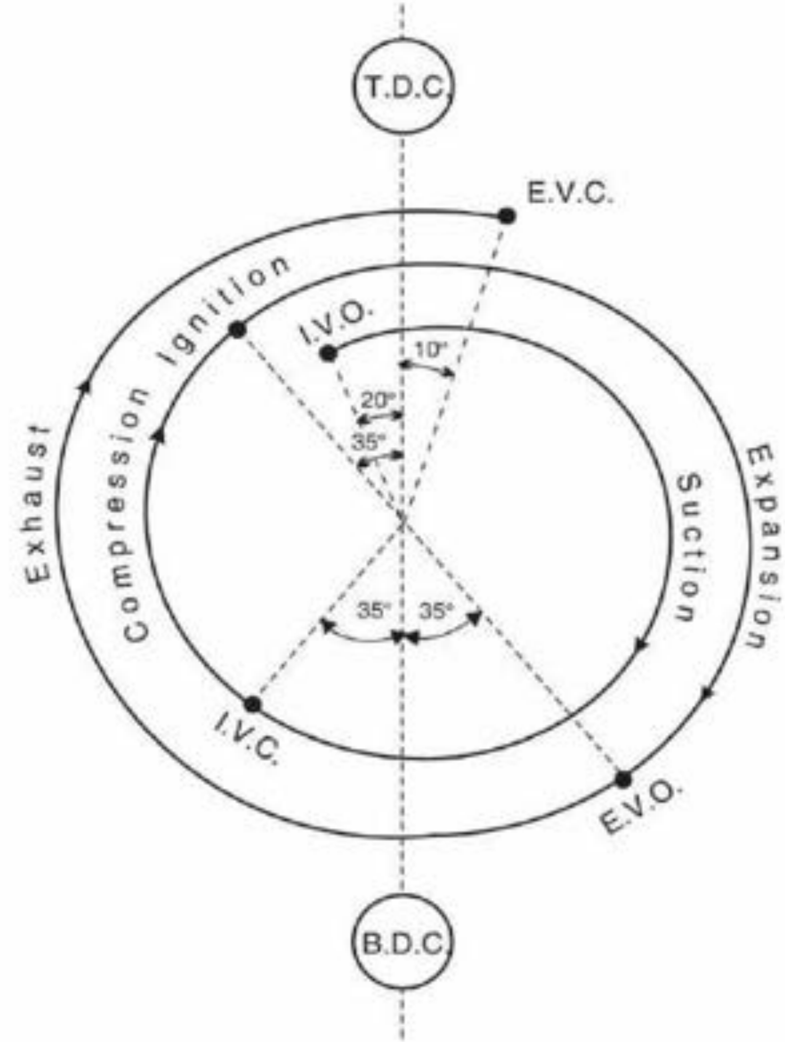


Fig. Actual valve timing diagram (four-stroke Otto cycle engines).

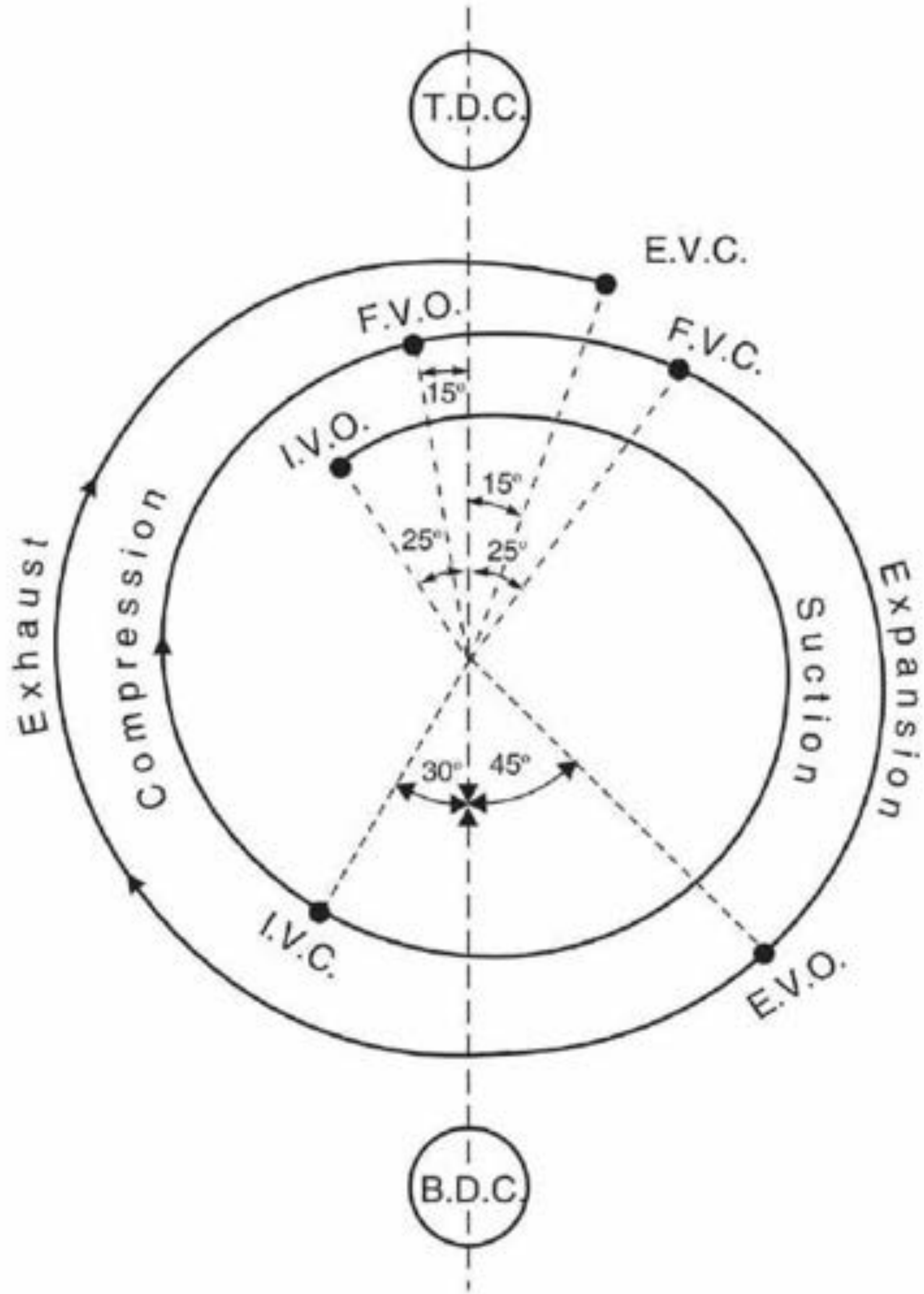


Fig. Actual valve timing diagram (four-stroke Diesel cycle engines).

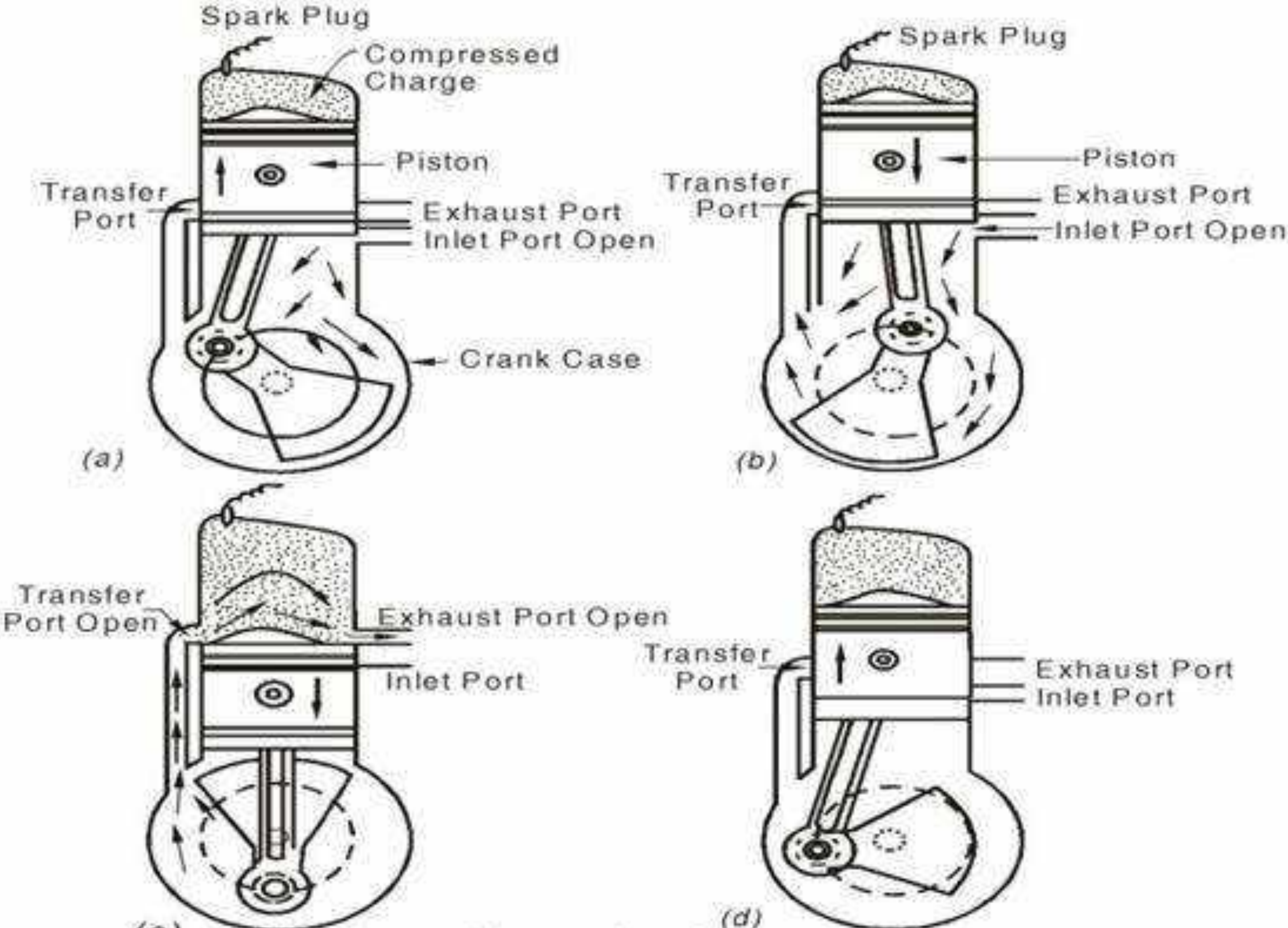
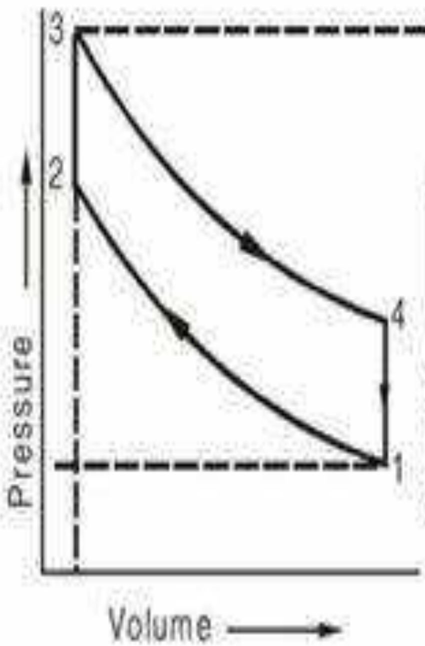
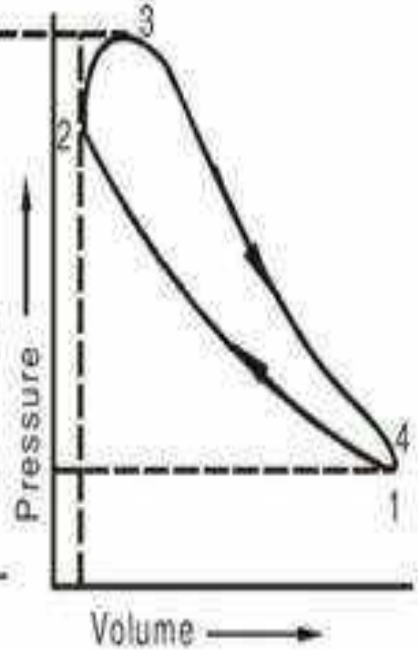


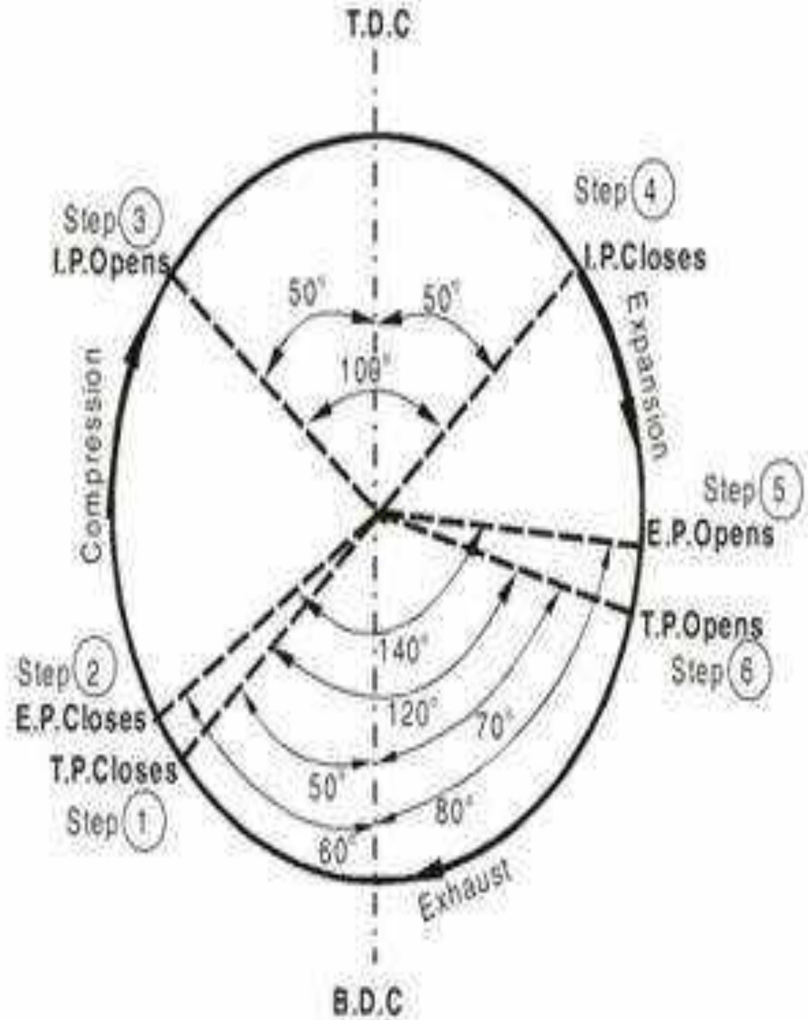
Fig. Two stroke petrol engine



(a) Theoritical cycle



(b) Actual cycle



(c.) Port timing diagram of a two stroke petrol engine

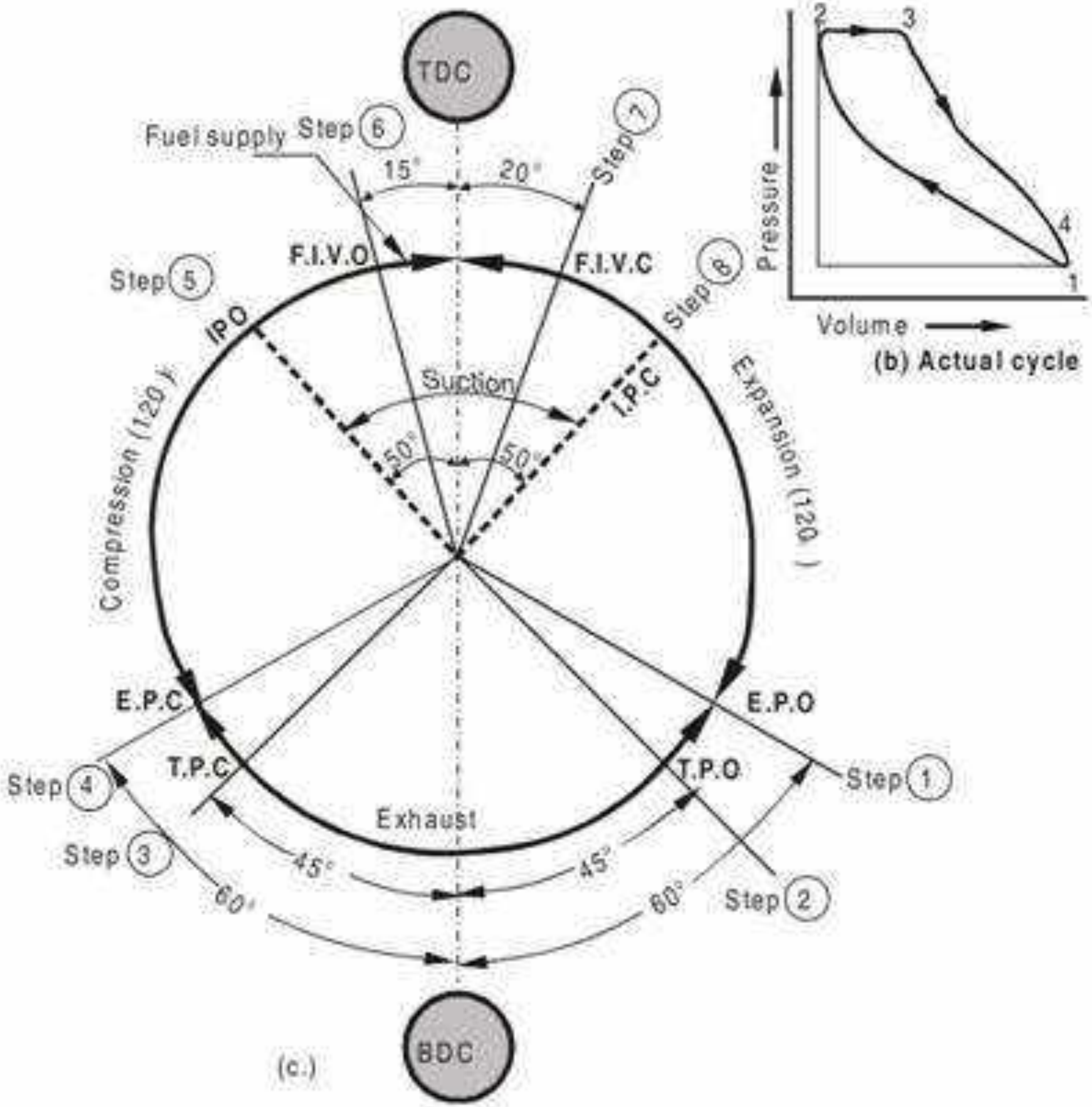
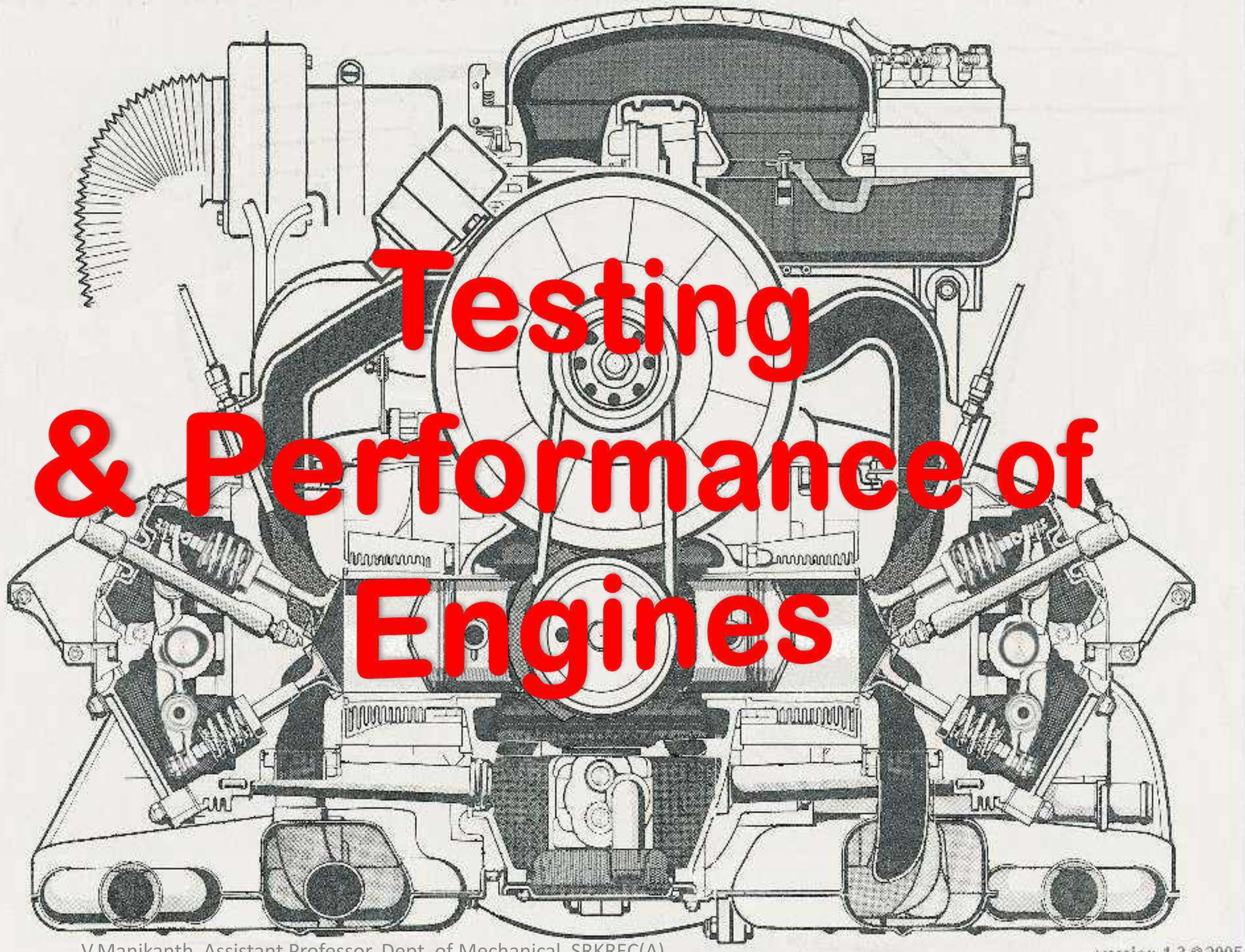


Fig. Port timing diagram for two stroke diesel engine

Testing & Performance of Engines



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INTRODUCTION

The basic task in the design and development of engines is to reduce the cost of production and improve the efficiency and power output. In order to achieve the above task, the development engineer has to compare the engine developed with other engines in terms of its output and efficiency. Towards this end he has to test the engine and make measurements of relevant parameters that reflect the performance of the engine. In general, he has to conduct a wide variety of engine tests. The nature and the type of the tests to be conducted depend upon a large number of factors.

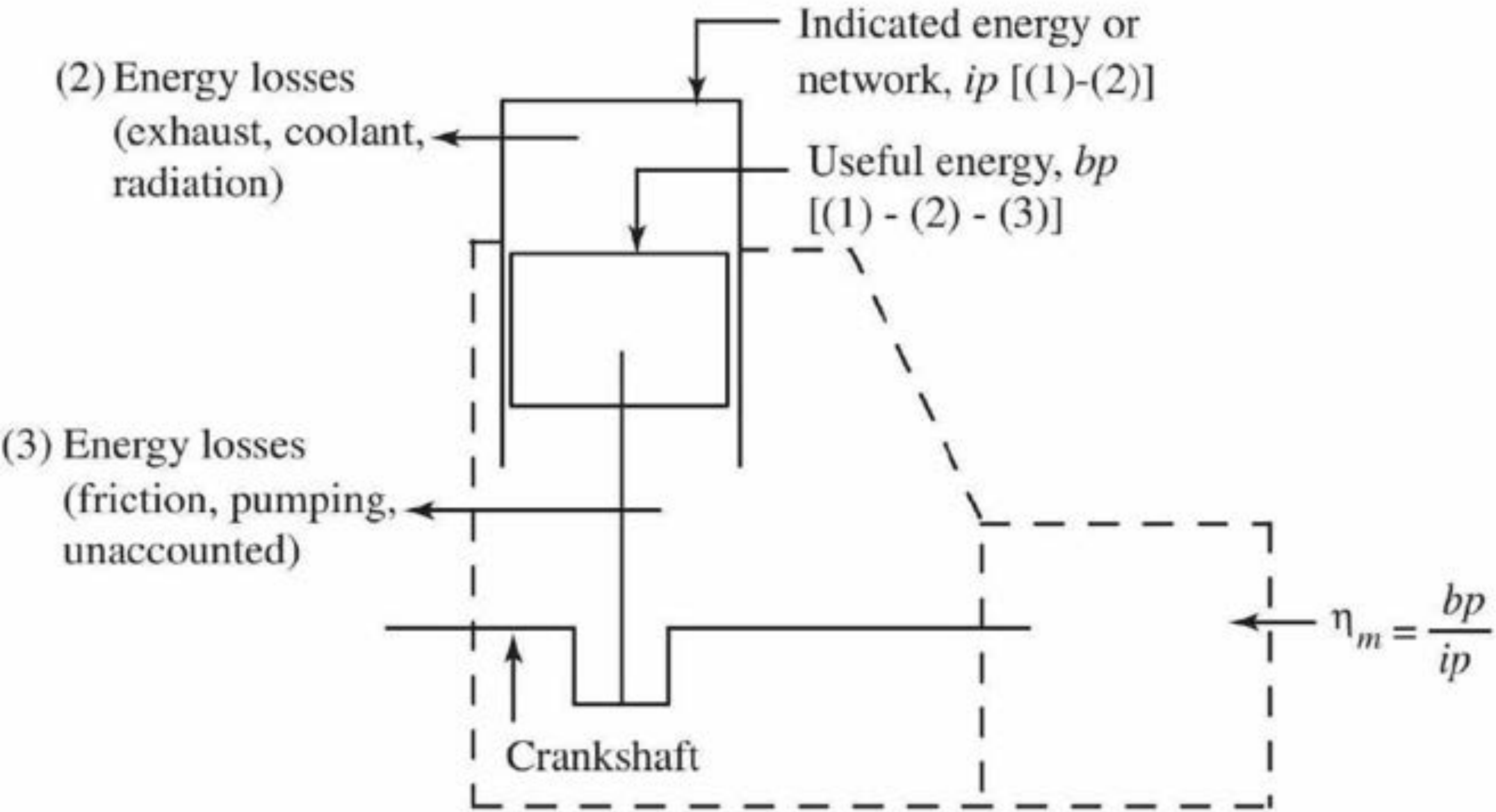


Fig. Energy flow through reciprocating engine

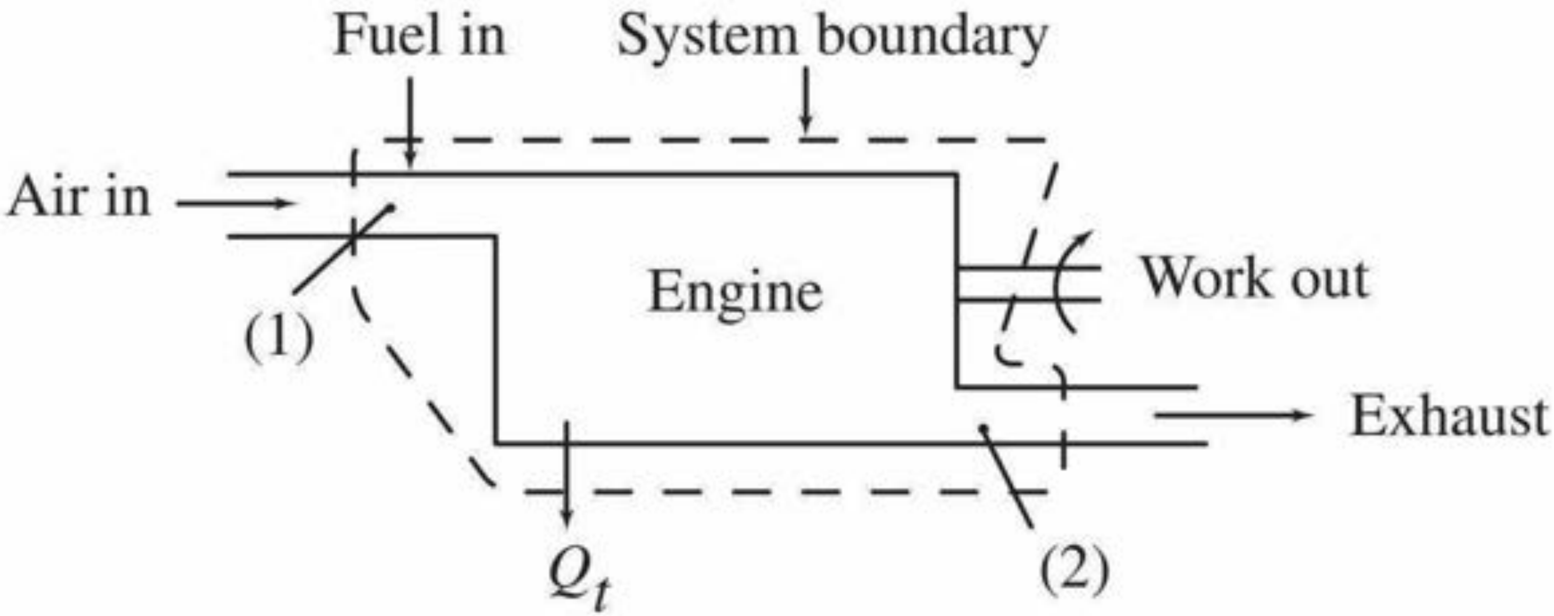


Fig. Reciprocating engine as an open system

What do we expect from an automotive engine?

- High torque
- Low weight and volume
- Good fuel economy
- Low emissions
- Ease of maintenance
- Good transient response
- Ability to adapt to changing driver requirements
- Ability to adapt to environmental conditions

PERFORMANCE OF I.C. ENGINES

Engine performance is an indication of the degree of success with which it does its assigned job *i.e.*, conversion of chemical energy contained in the fuel into the useful mechanical work.

In evaluation of engine performance certain basic parameters are chosen and the effect of various operating conditions, design concepts and modifications on these parameters are studied. The *basic performance parameters* are numerated and discussed below :

1. Power and mechanical efficiency
2. Mean effective pressure and torque
3. Specific output
4. Volumetric efficiency
5. Fuel-air ratio
6. Specific fuel consumption
7. Thermal efficiency and heat balance
8. Exhaust smoke and other emissions
9. Specific weight.

ENGINE PERFORMANCE PARAMETERS

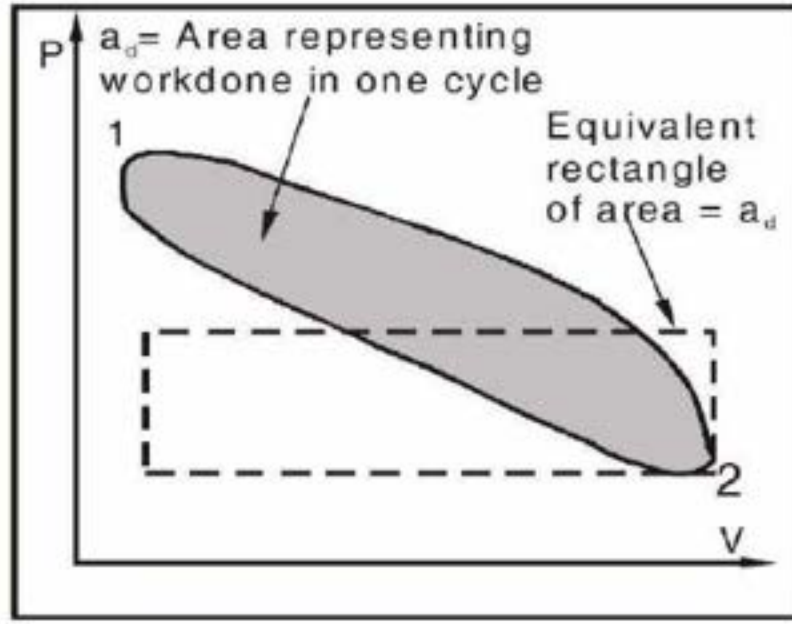
The engine performance is indicated by the term *efficiency*, η. Five important engine efficiencies and other related engine performance parameters are:

(i)	Indicated thermal efficiency	(η_{ith})
(ii)	Brake thermal efficiency	(η_{bth})
(iii)	Mechanical efficiency	(η_m)
(iv)	Volumetric efficiency	(η_v)
(v)	Relative efficiency or Efficiency ratio	(η_{rel})
(vi)	Mean effective pressure	(p_m)
(vii)	Mean piston speed	$(\overline{s}_p)$
(viii)	Specific power output	(P_s)
(ix)	Specific fuel consumption	(sfc)
(x)	Inlet-valve Mach Index	(Z)
(x)	Fuel-air or air-fuel ratio	$(F/A$ or $A/F)$
(xi)	Calorific value of the fuel	(CV)

Figure shows the diagrammatic representation of energy distribution in an IC engine.

Indicator Diagram

An indicator diagram is a P-V diagram traced by the indicator which is attached to the piston. The P-V diagram represents the work done by the engine in one cycle.



The power developed inside the engine cylinder is known as **indicated power**. This is measured by indicator diagram.

We can measure a_d (area of indicator diagram) by using planimeter. Now we can draw an equivalent rectangle whose area is equal to the area of indicator diagram. And the height of this rectangle gives the **Mean effective pressure P_m**.

Mean Effective Pressure (P_m)

The mean effective pressure 'P_m' can be calculated from the following formula.

$$P_m = \frac{a_d}{l_d} \times S$$

where a_d = Area of indicator diagram (or) rectangle in m^2.

l_d = length of the diagram in m.

S = Spring constant (or) Spring number used in engine indicator - unit in $(N/m^2)/m$ or bar/m.

INDICATED POWER

Indicated power of an engine tells about the health of the engine and also gives an indication regarding the conversion of chemical energy in the fuel into heat energy. Indicated power is an important variable because it is the potential output of the cycle. Therefore, to justify the measurement of indicated power, it must be more accurate than motoring and other indirect methods of measuring frictional power. For obtaining indicated power the cycle pressure must be determined as a function of cylinder volume. It may be noted that it is of no use to determine pressure accurately unless volume or crank angle can be accurately measured.

In order to estimate the indicated power of an engine the following methods are usually followed.

(i) using the indicator diagram

(ii) by adding two measured quantities viz. brake power and friction power

MEASUREMENT OF CYLINDER PRESSURE

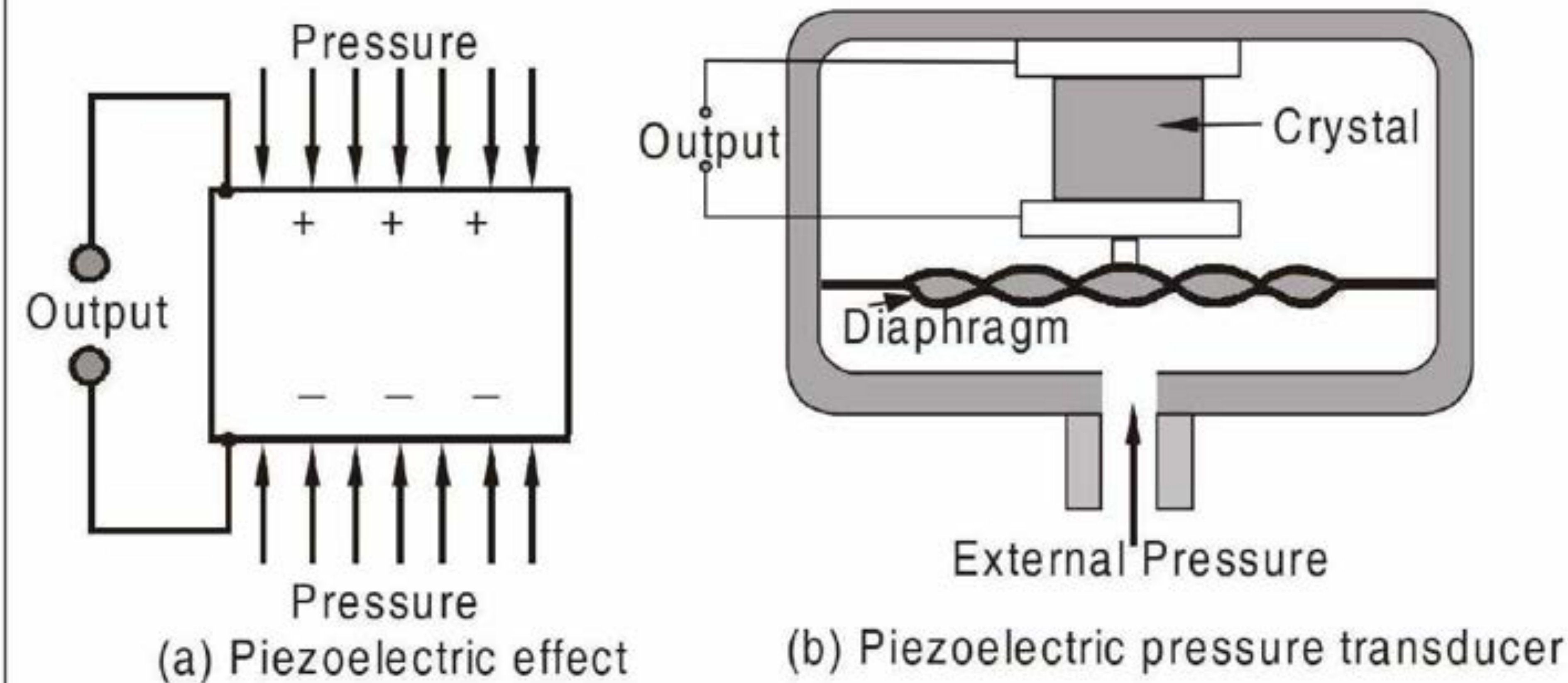


Fig. Piezoelectric transducer

Indicated Power (IP)

$$IP = \frac{P_m A\, L(N \text{ or } N/2)\, k}{60} \text{ kW}$$

where P_m = Mean effective pressure in kN/m^2 or KPa

A = Area of piston in m^2

L = Length of stroke in m

N = Speed of the engine in r.p.m.

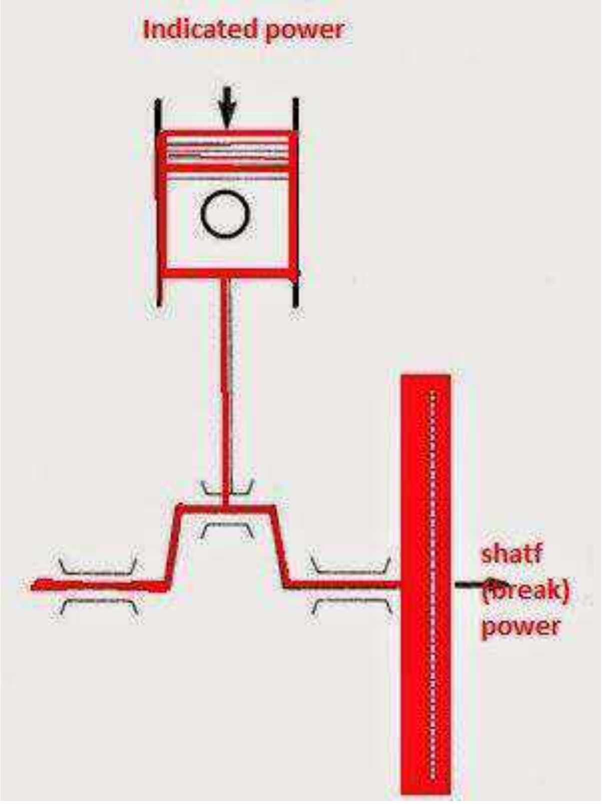
N for 2 stroke engine [$\because$ In two stroke engine, the cycle is completed in two strokes of the piston or in one revolution of the crankshaft.]

$N/2$ for 4 stroke engine [$\because$ In four stroke engine, the cycle is completed in 4 strokes of the piston (or) in two revolutions of the crankshaft.]

and k= No. of cylinders in the engine.

Brake Power (BP)

The power available at the crankshaft of the engine is known as brake power. The brake power is measured by some brake mechanism, hence the name brake power.



The B.P. of an engine can be determined by a brake of some kind applied to the brake pulley of the engine. The arrangement for determination of B.P. of the engine is known as *dynamometer*. The dynamometers are classified into following two classes :

(*i*) Absorption dynamometers (*ii*) Transmission dynamometers.

(*i*) **Absorption dynamometers.** *Absorption dynamometers are those that absorb the power to be measured by friction*. The power absorbed in friction is finally dissipated in the form of heat energy.

Common forms of absorption dynamometers are :

- Prony brake
- Rope brake
- Hydraulic brake
- Fan brake
- Electrical brake dynamometers
  - Eddy current dynamometer
  - Swinging field d.c. dynamometer.

(*ii*) **Transmission dynamometers.** These are also called *torquemeters*. These are very accurate and are used where continuous transmission of load is necessary. There are used mainly in automatic units.

Here we shall discuss *Rope brake dynamometer* only :

Fig: Mechanical belt type Dynamometer test rigs

Rope brake dynamometer

Refer Fig. A rope is wound round the circumference of the brake wheel. To prevent the rope from slipping small wooden blocks (not shown in the Fig. are laced to rope. To one end of the rope is attached a spring balance (S) and the other end carries the load (W). The speed of the engine is noted from the tachometer (revolution counter).

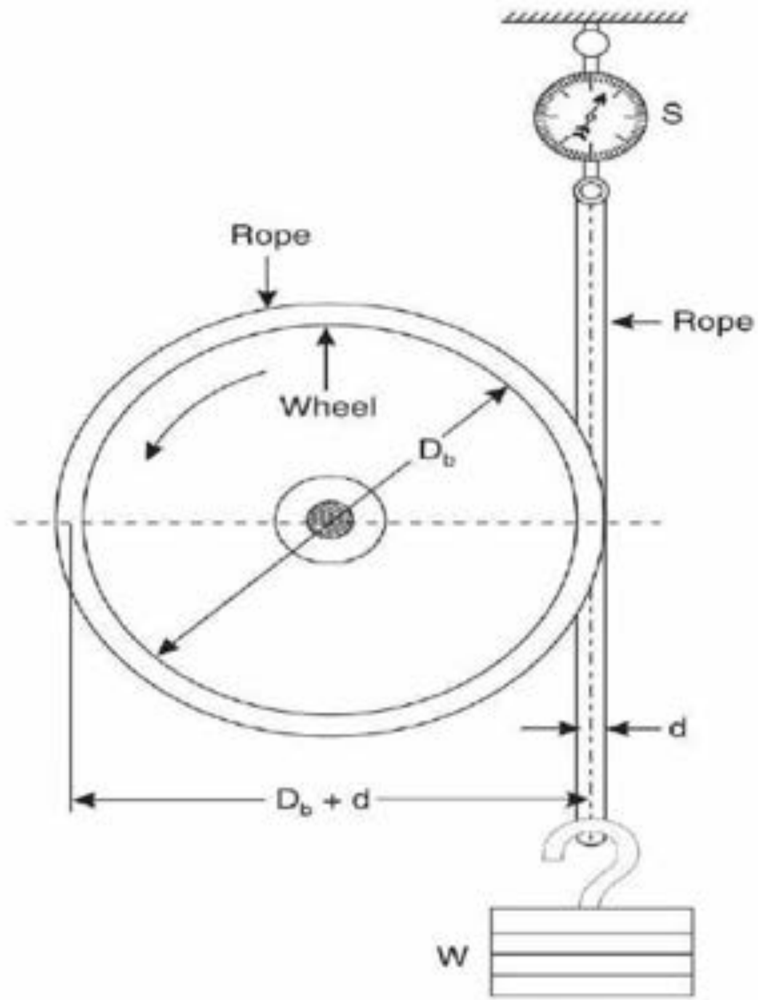


Fig. Rope brake dynamometer.

If, W = weight at the end of the rope, N,

S = spring balance reading, N,

N = engine speed, r.p.m.,

D_b = diameter of the brake wheel, m,

d = diameter of the rope, m., and

$(D_b + d)$ = effective diameter of the brake wheel,

Then work/revolution $= \text{Torque} \times \text{angle turned per revolution}$

$$= (W - S) \times \left(\frac{D_b + d}{2}\right) \times 2\pi = (W - S)(D_b + d) \times \pi$$

Work done/min $= (W - S)\,\pi\,(D_b + d)\,N$

Work done/sec $$= \frac{(W - S)\,\pi\,(D_b + d)\,N}{60}$$

$\therefore$ $$\text{B.P.} = \frac{(W - S)\,\pi\,(D_b + d)\,N}{60 \times 1000}\ \text{kW}$$

$$= \frac{(W - S)\,\pi\,D_b N}{60 \times 1000} \quad \text{...... if } d \text{ is neglected}$$

or $$\left(= \frac{T \times 2\pi N}{60 \times 1000}\ \text{kW}\right)$$

Rope brake is cheap and easily constructed but not very accurate because of changes in friction co-efficient of the rope with temperature.

FRICTIONAL POWER (FP):

Friction generally refers to forces acting between surfaces in relative motion. In engines, frictional losses are mainly due to sliding as well as rotating parts. Normally, engine friction, in its broader sense, is taken as the difference between the indicated power, ip, and the brake power, bp. Usually engine friction is expressed in terms of frictional power, fp. Frictional loss is mainly attributed to the following mechanical losses.

(i) direct frictional losses

(ii) pumping losses

(iii) power loss to drive the components to charge and scavenge

(iv) power loss to drive other auxiliary components

A good engine design should not allow the total frictional losses to be more than 30% of the energy input in reciprocating engines. It should be the aim of a good designer to reduce friction and wear of the parts subjected to relative motion. This is achieved by proper lubrication. In this section the various losses associated with friction is enumerated.

Measurement of frictional power (F.P.) :

The frictional power of an engine can be determined by the following methods :

1. Willan's line method (used for C.I. engines only)
2. Morse test
3. Motoring test
4. Difference between I.P. and B.P.

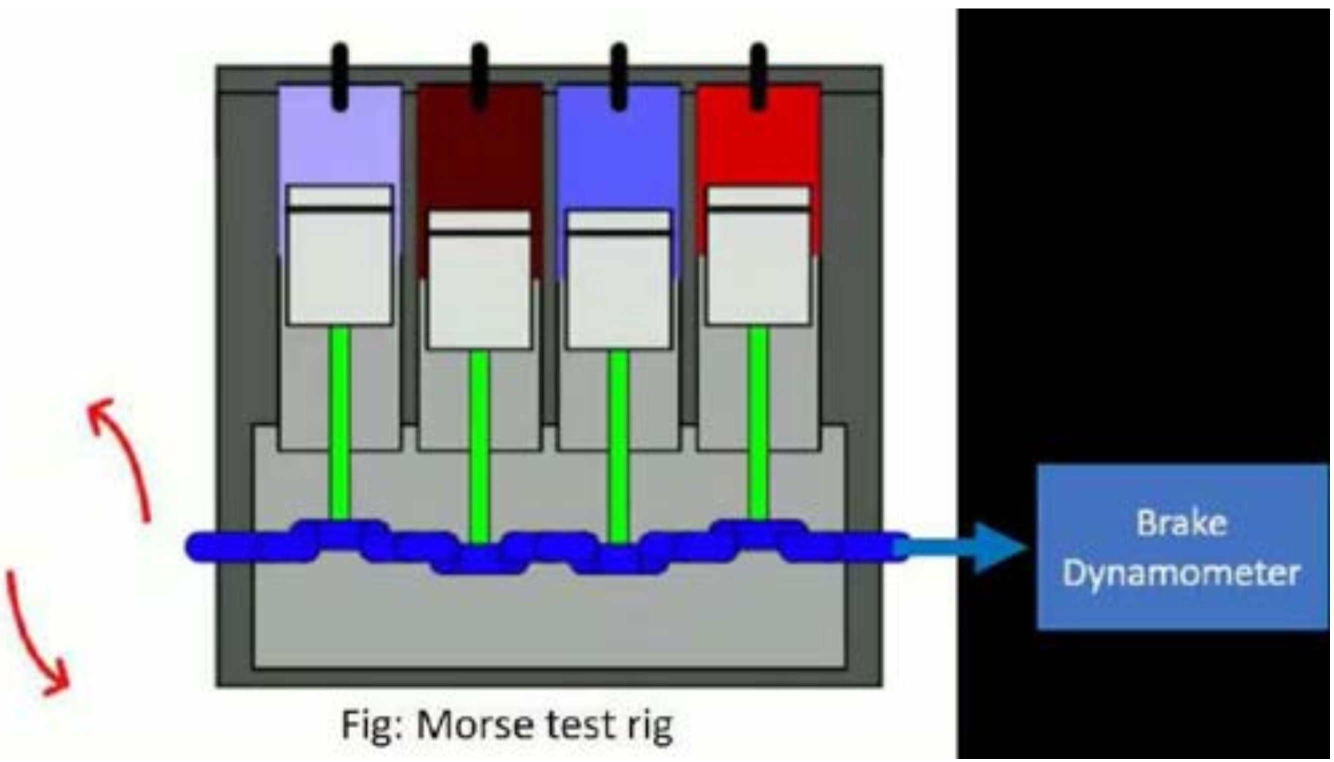


Fig: Morse test rig

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1. ***Willan's line method***

At a constant engine speed the load is reduced in increments and the corresponding B.P. and gross fuel consumption readings are taken. A graph is then drawn of fuel consumption against B.P. as in Fig. The graph drawn in called the Willan's line (analogous to Willan's line for a steam engine), and is extrapolated back to cut the B.P. axis at the point L. The reading OL is taken as the power loss of the engine at that speed. The fuel consumption at zero B.P. is given by OM ; and if the relationship between fuel consumption and B.P. is assumed to be linear, then a fuel consumption OM is equivalent to a power loss of OL.

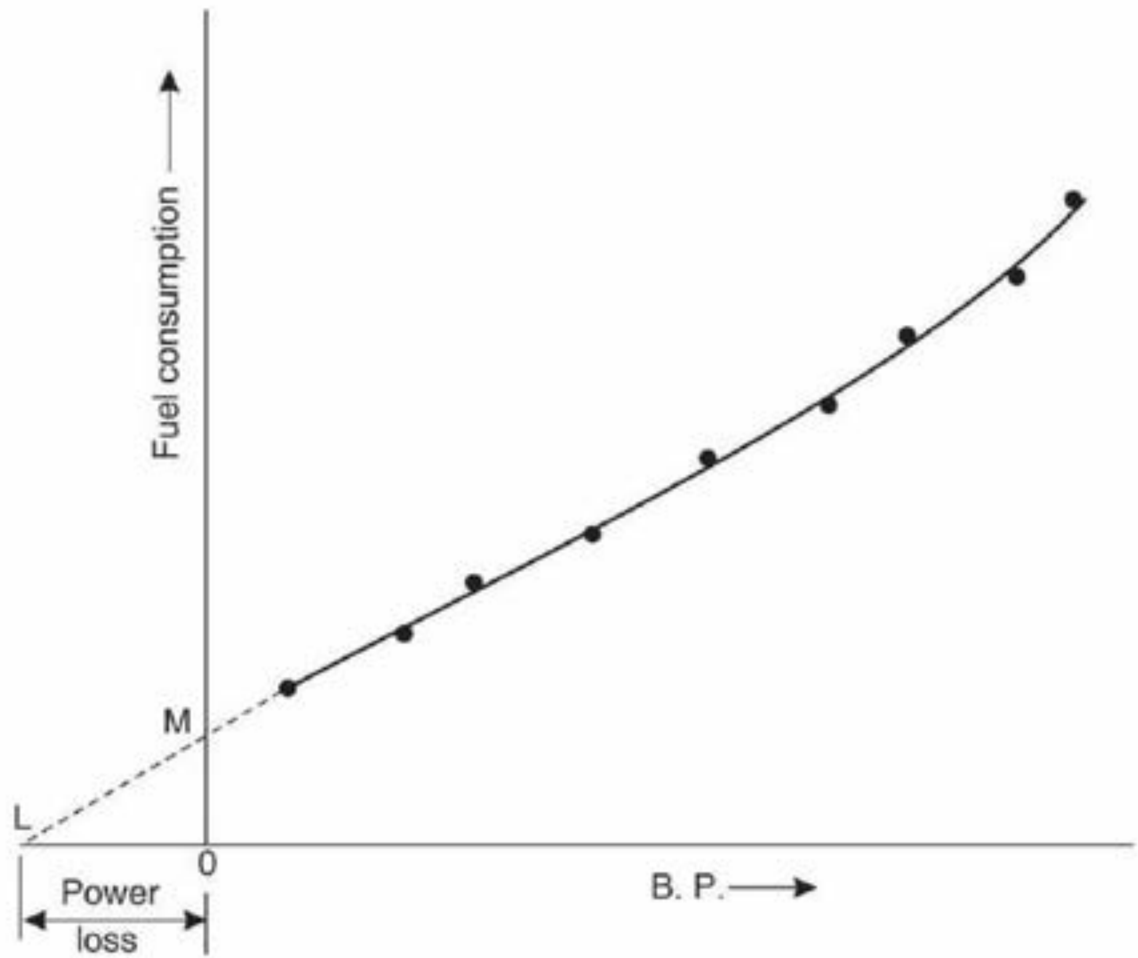


Fig. Willan's line method.

MORSE TEST

This method is used to measure the indicated power without the use of indicator diagram in **multicylinder engines**. The brake power of the engine is measured by cutting off each cylinder in turn. The cylinder of a petrol engine is cut off by shorting the spark plug and in case of diesel engine, this is done by cutting off the diesel supply to the required cylinder.

For example, consider a 4 cylinder engine. First of all, measure the brake power of the engine when all the cylinders are in operation. Then cylinder 1 is cut-off so that it does not develop any power. The speed of the engine decreases. In order to attain the initial speed back, the load on the engine is reduced. Now, the brake power is measured with this new condition which gives the brake power of the remaining three cylinders.

Similar way, we can cut-off each cylinder one by one and measure the brake power of the remaining three cylinders by maintaining the engine speed as original speed.

Let I_1, I_2, I_3 and I_4 = Indicated power of cylinder 1, 2, 3 and 4 respectively.

F_1, F_2, F_3 and F_4 = Frictional power of cylinder 1, 2, 4 and 4 respectively.

When all the cylinders are in operation the total brake power B.P. simply B is given as follows.

$$B = \text{Total indicated power} - \text{Total Friction Power} \quad \dots (1)$$

$$B = (I_1 + I_2 + I_3 + I_4) - (F_1 + F_2 + F_3 + F_4)$$

when cylinder 1 is cut off, $I_1 = 0$, but the frictional losses of the cylinder 1 remain the same

$\therefore$ Brake power of the remaining three cylinders $= B_1$

$$B_1 = (0 + I_2 + I_3 + I_4) - (F_1 + F_2 + F_3 + F_4) \quad \dots (2)$$

Subtracting the equation (2) from equation (1), we get

$$B - B_1 = I_1 \text{ (or)}$$

Indicated power of the first cylinder, I_1

$$I_1 = B - B_1$$

Similarly, I.P. of 2nd cylinder I_2

$$I_2 = B - B_2$$

IP of 3rd cylinder, I_3

$$I_3 = B - B_3$$

I.P. of 4th cylinder, I_4

$$I_4 = B - B_4$$

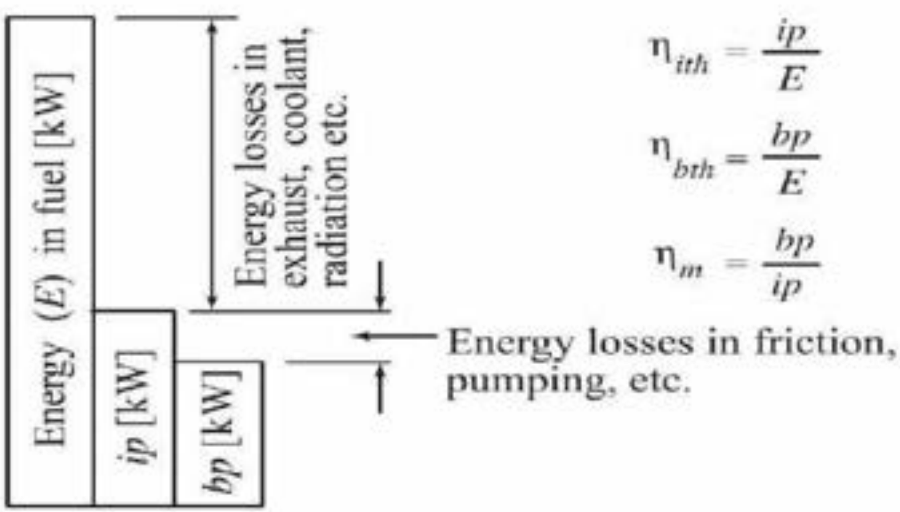
and the total indicated power $IP = I_1 + I_2 + I_3 + I_4$

3 Mechanical Efficiency (η_m)

Mechanical efficiency is defined as the ratio of brake power (delivered power) to the indicated power (power provided to the piston) or can be defined as the ratio of the brake thermal efficiency to the indicated thermal efficiency.

$$\eta_m = \frac{bp}{ip} = \frac{bp}{bp + fp}$$

$$fp = ip - bp$$



$$\eta_{ith} = \frac{ip}{E}$$

$$\eta_{bth} = \frac{bp}{E}$$

$$\eta_m = \frac{bp}{ip}$$

Fig. 1.15 Energy distribution

$$= \frac{ip}{\text{mass of fuel/s} \times \text{calorific value of fuel}}$$

CALORIFIC OR HEATING VALUES OF FUELS

The "calorific value or heating value" of the fuel is defined as the energy liberated by the complete oxidation of a unit mass or volume of a fuel. It is expressed in kJ/kg for solid and liquid fuels and kJ/m^3 for gases.

If a fuel contains hydrogen water will be formed as one of the products of combustion. If this *water is condensed, a large amount of heat will be released* than if the water exists in the vapour phase. For this reason *two heating values* are defined ; the *higher or gross heating value* and the *lower* or *net heating value.*

The *higher heating value*, HHV, is obtained when the *water formed by combustion is completely condensed.*

The *lower heating value*, LHV, is obtained when the water formed by combustion exists *completely in the vapour phase.*

Thus :

$$(HHV)_p = (LHV)_p + m\, h_{fg}$$

$$(HHV)_v = (LHV)_v + m(u_g - u_f)$$

where m = Mass of water formed by combustion,

h_{fg} = Enthalpy of vaporisation of water, kJ/kg,

u_g = Specific internal energy of vapour, kJ/kg, and

u_f = Specific internal energy of liquid, kJ/kg.

In almost all practical cases, the water vapour in the products is vapour, the lower value is the one *which usually applies.*

Thermal Efficiency

The ratio of B.P (or) I.P to the energy supplied by fuel during the same interval of time is known as thermal efficiency.

If it is based on I.P, then it is known as **Indicated thermal efficiency.**

If it is based on B.P, then it is known as **Brake thermal efficiency.**

Indicated Thermal Efficiency ($\eta_{\text{indicated}}$)

$$\eta_{\text{indicated}} = \frac{I.P \text{ in kW} \times 3600}{\dot{m}_f \times C.V.}$$

where $C.V$ = Calorific value of fuel in kJ/kg

$\dot{m}_f$ = Mass of fuel in kg/hr

If C.V is given in kJ/m^3, then

$$\eta_{\text{indicated thermal}} = \frac{I.P \text{ in kW} \times 3600}{\dot{V}_f \times C.V.}$$

where $\dot{V}_f$ = Volume of gas fuel supplied in m^3/hr

Brake Thermal Efficiency ($\eta_{\mathbf{Brake}}$)

$$\eta_{\mathbf{Brake}} = \frac{\boldsymbol{B.P} \textbf{ in kW} \times \mathbf{3600}}{\dot{\boldsymbol{m}}_{\boldsymbol{f}} \times \boldsymbol{C.V.}}$$

where $C.V$ in kJ/kg.

If C.V is in kJ/m^3, then $\eta_{\text{Brake}} = \dfrac{B.P \text{ in kW} \times 3600}{\dot{V}_f \times C.V.}$

$$\eta_{\mathbf{Brake}} = \eta_{\mathbf{indicated}} \times \eta_{\mathbf{mech}}$$

Brake thermal efficiency is also known as overall efficiency.

i.e., $\eta_{\text{Brake}} = \eta_{\text{overall}}$

Relative Efficiency or Efficiency Ratio

The ratio of the indicated thermal efficiency or the brake thermal efficiency to the air standard efficiency is known as relative efficiency or efficiency ratio.

Relative efficiency,

$$\eta_{relative} = \frac{\text{Indicated (or) Brake thermal efficiency}}{\text{Air standard efficiency}}$$

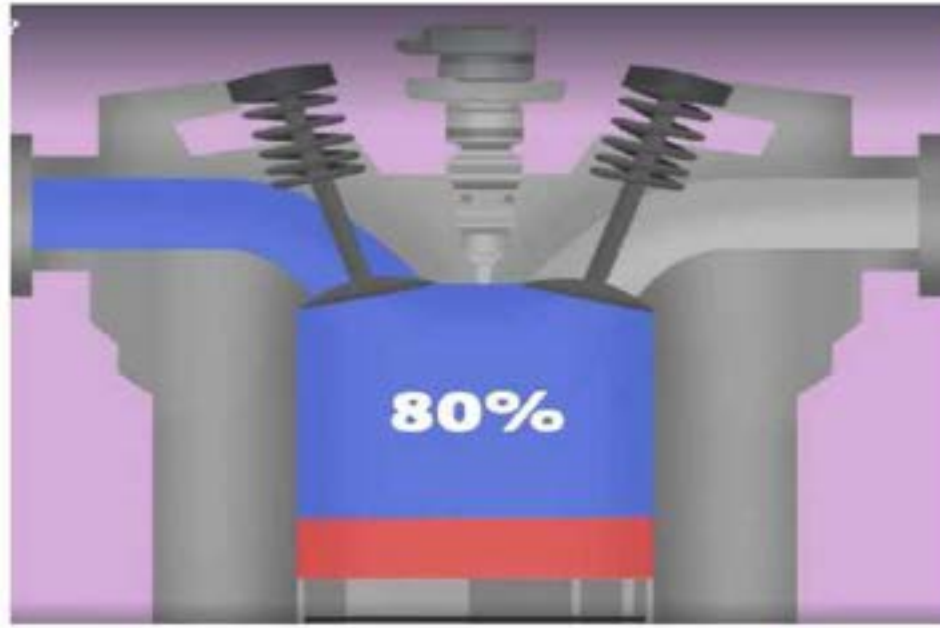
Volumetric Efficiency ($\eta_{(\text{volumetric})}$)

The ratio of the actual volume of the charge admitted into the cylinder to the swept volume of the piston is known as volumetric efficiency.

$$\eta_{\text{volumetric}} = \frac{\text{Volume of charge admitted (at NTP condition)}}{\text{Sweptvolume}}$$

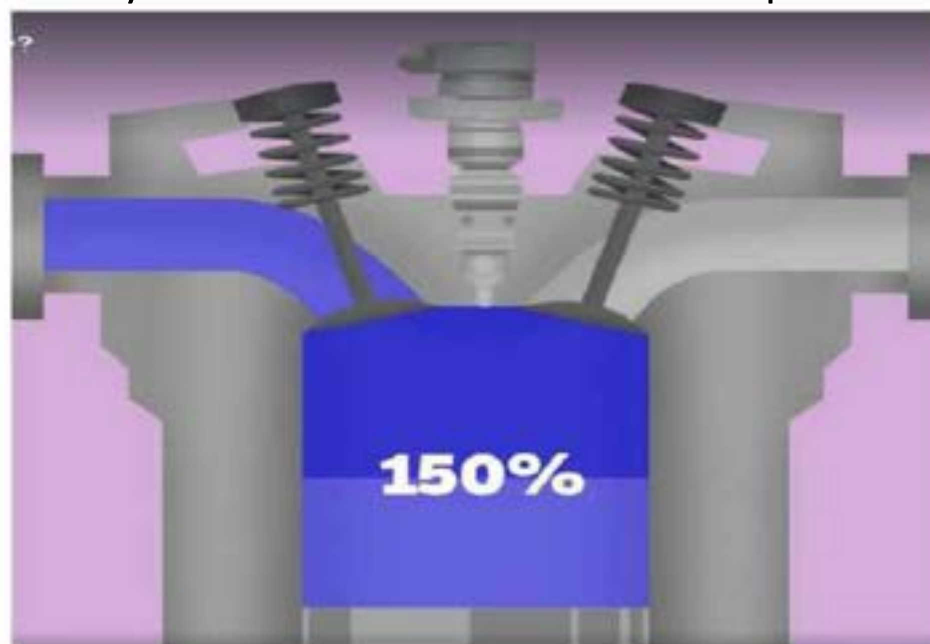
NTP = Normal temperature 0°C and pressure (1.01325 bar) condition.

Actual vol. available(Vs) inside a cylinder



If only 80% of the vol. is filled means η_{vol}=80%

vol. wasted due to incomplete filling



Why super chargers and turbochargers are used

Fuel-Air (F/A) or Air-Fuel Ratio (A/F)

A mixture that contains just enough air for complete combustion of all the fuel in the mixture is called a chemically correct or stoichiometric fuel-air ratio. A mixture having more fuel than that in a chemically correct mixture is termed as rich mixture and a mixture that contains less fuel (or excess air) is called a lean mixture. For most of the hydrocarbon fuels, the stoichiometric air-fuel ratio is around 15:1. SI engines operate around this ratio during normal operation. The air-fuel ratio for CI engines vary from 18:1 to 80:1 from full load to no load.

The ratio of actual fuel-air ratio to stoichiometric fuel-air ratio is called equivalence ratio and is denoted by ϕ.

$$\phi = \frac{\text{Actual fuel-air ratio}}{\text{Stoichiometric fuel-air ratio}}$$

Accordingly, $\phi = 1$ means stoichiometric (chemically correct) mixture, $\phi < 1$ means lean mixture and $\phi > 1$ means rich mixture.

Specific Fuel Consumption (S.F.C) in kg/kW hr

It is defined as the amount of fuel consumed per unit of power developed per hour.

The ratio $\frac{\dot{m}_f}{B.P \text{ or } I.P}$ is known as specific fuel consumption per kW per hour. [Here $\dot{m}_f$ = mass of fuel consumed in kg/hr.]

$$\text{BSFC} = \frac{\dot{m}_f}{\text{B.P}} \text{ kg/kW} - \text{hr}$$

$$\text{ISFC} = \frac{\dot{m}_f}{\text{I.P}} \text{ kg/kW} - \text{hr}$$

where, BSFC - Brake Spepcific fuel consumption.

ISFC - Indicated Specific fuel consumption)

HEAT BALANCE TEST

A heat balance sheet is an account of heat supplied, heat utilized and dissipated in different ways in a system. The performance of the engine is obtained from the heat balance sheet. Energy supplied to an engine is the heat value of the fuel consumed. As has been repeatedly pointed out, only a part of this energy is transformed into useful work. The rest of it is either wasted or utilized in special application like turbo compounding. The two main parts of the heat not available for work are the heat carried away by the exhaust gases and the cooling medium.

To give sufficient data for the preparation of a heat balance sheet, a test should include a method of determining the friction power and the measurement of speed, load, fuel consumption, air consumption, exhaust temperature, rate of flow of cooling water and its temperature rise while flowing through the water jackets. Besides, the small losses, such as radiation and incomplete combustion, the above enumerated data makes it possible to account for the heat supplied by the fuel and indicates its distribution.

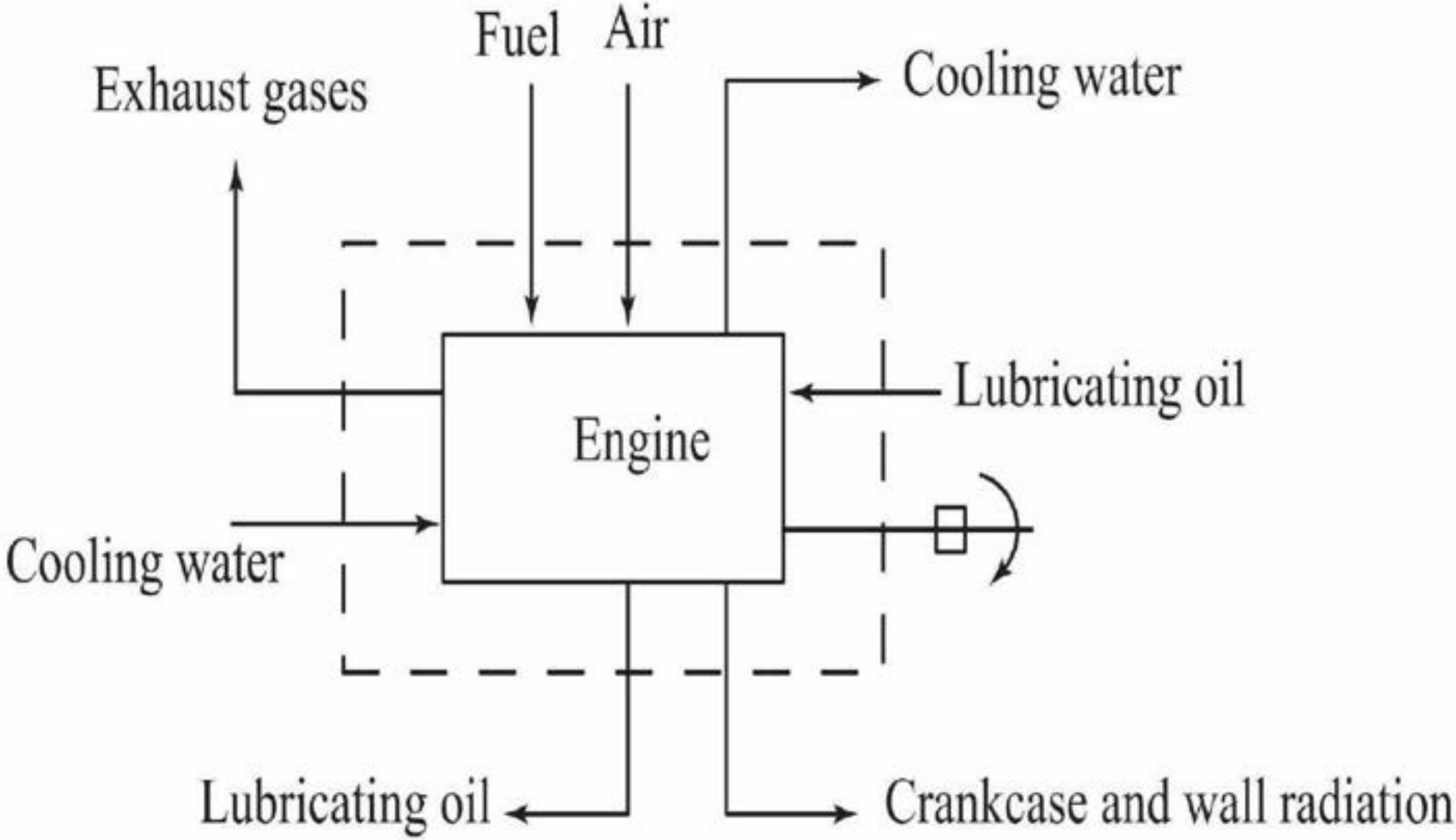


Fig. External heat balance

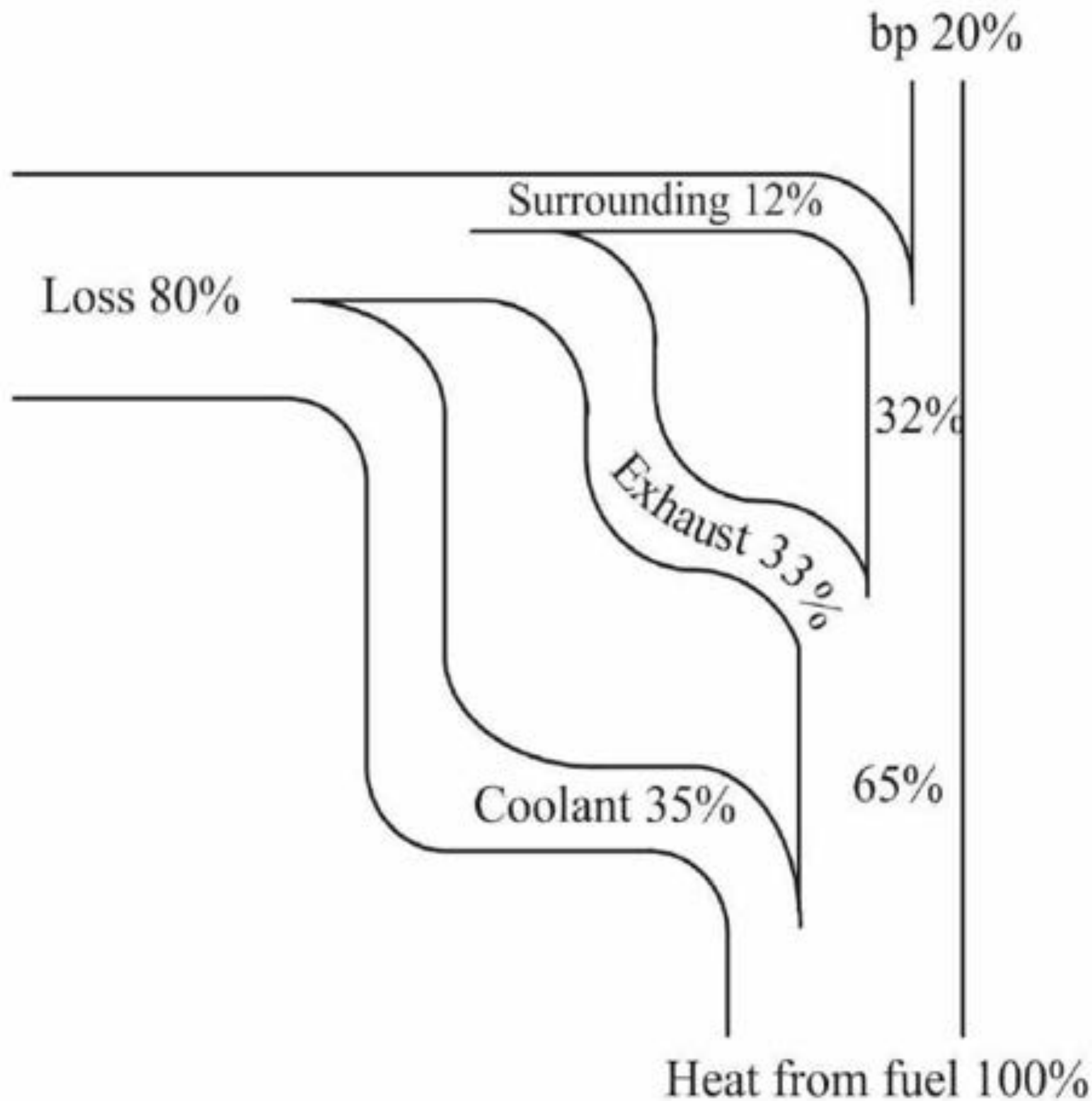


Fig. Sankey diagram

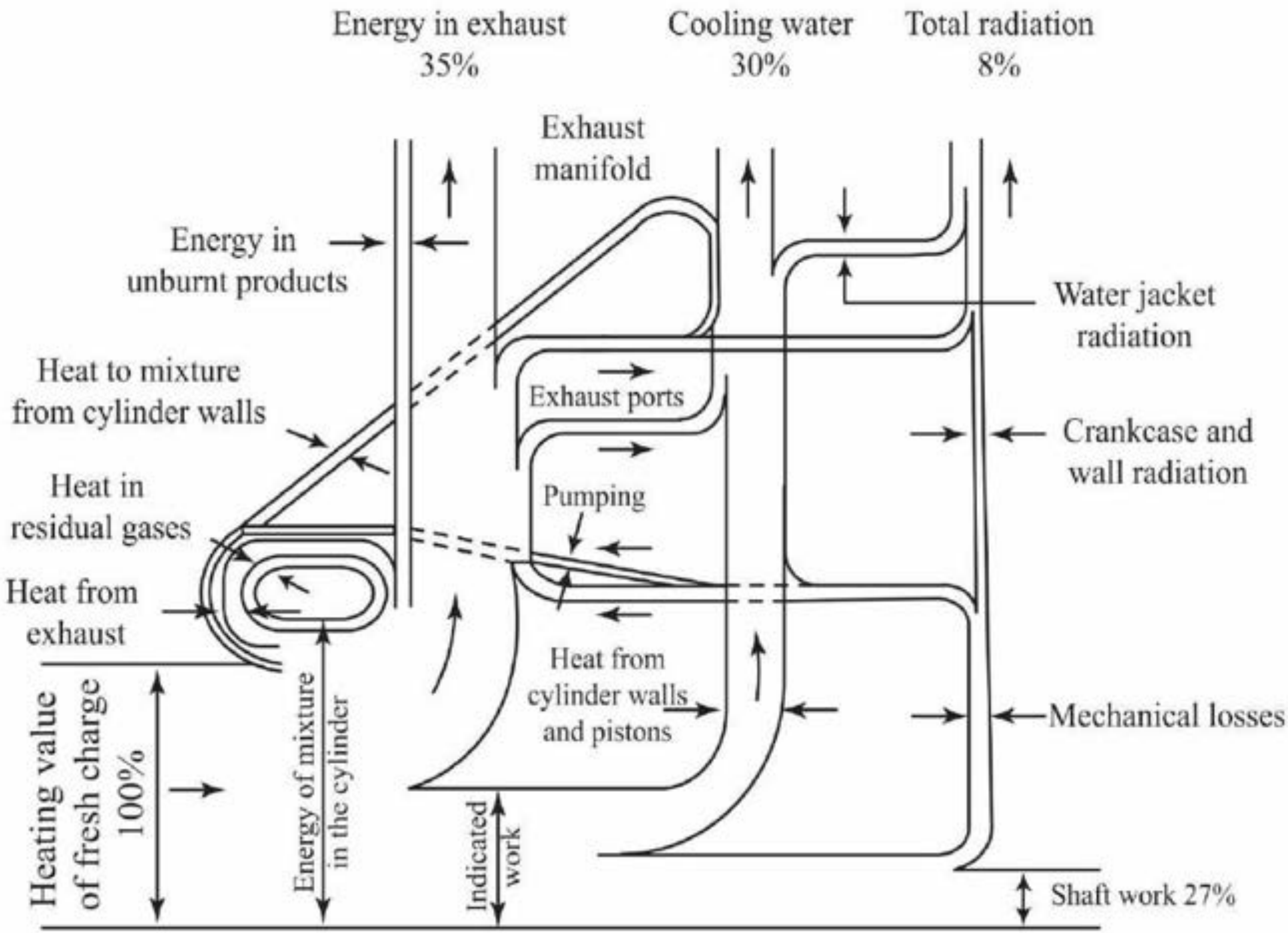


Fig. Sankey diagram for an SI engine

A heat balance account includes the following items.

Heat balance sheet on minute basis:

Sl. No.	Description	Load = 8 amp (1/2 load)		Load = 12 amp (3/4 load)	
		Heat (in KJ/min)	Heat (in %)	Heat (in KJ/min)	Heat (in %)
1	Heat input				
2	Heat converted to B.P.				
3	Heat carried by Jacket cooling water				
4	Heat carried by exhaust gases				
5	Heat unaccounted for				
	Total of 2,3,4,5				

Heat supplied by the fuel to the engine $= \dot{m}_f \times L.C.V$

where $\dot{m}_f$ is the mass of fuel supplied per minute and L.C.V is the lower calorific value of the fuel.

1. **Heat equivalent of brake power** =

   Brake power $\times$ 60 kJ/min. where Brake power is in kW.

2. **Heat lost to jacket cooling water**

$$= m_w \, C_{pw} \, (T_o - T_i) \text{ kJ/min}$$

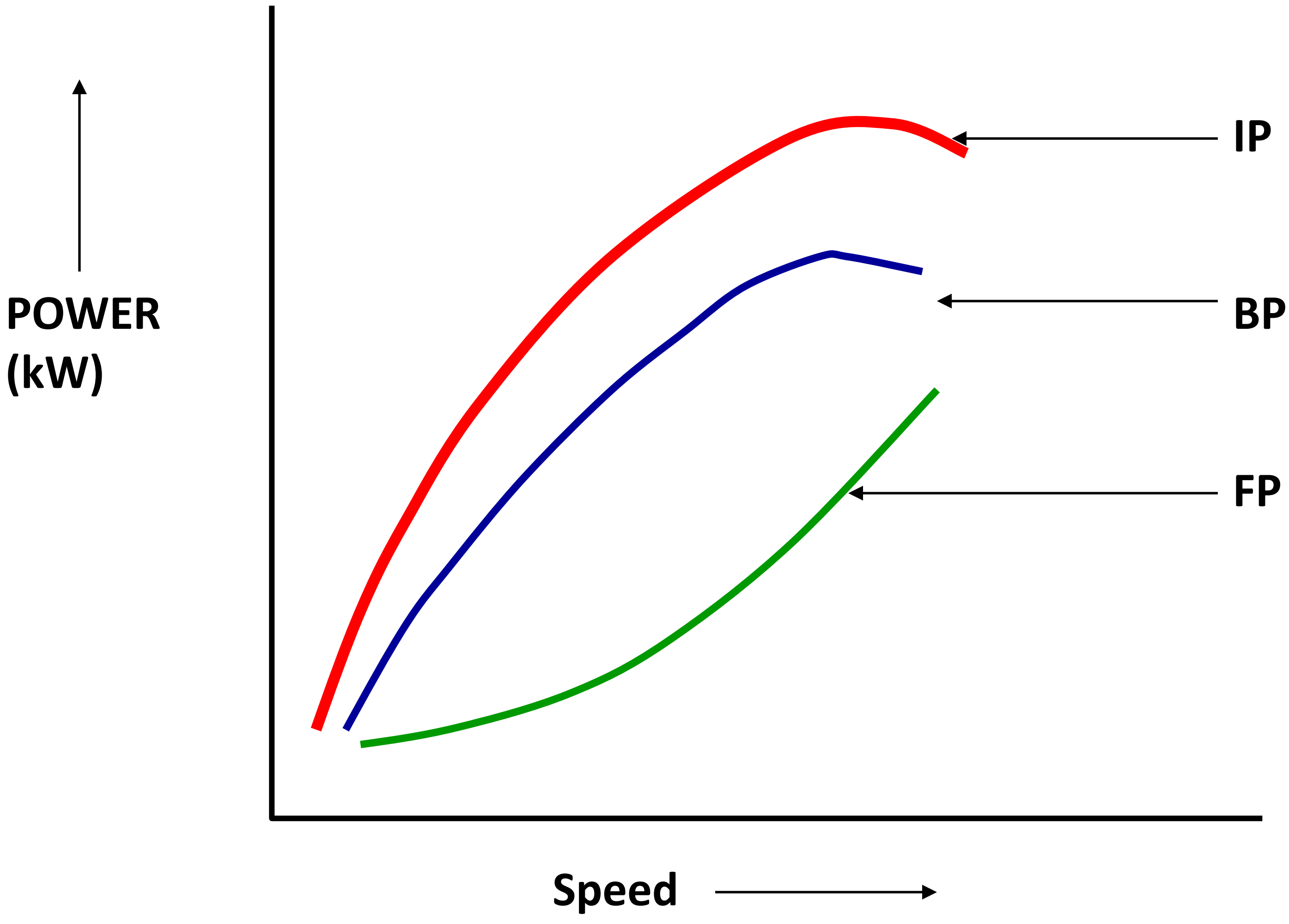
3. **Heat lost to exhaust gases**

$$= m_g \, C_{pg} \cdot (T_g - T_a) \text{ kJ/min}$$

4. The remaining heat is lost by convection and radiation. This cannot be measured and so this is known as **unaccounted loss**. This is calculated by the difference of heat supplied and the sum of (1) + (2) + (3). i.e $Q_{ua} = Q_s - [Q_{\text{I.P (or) B.P}} + Q_w + Q_g \text{ kJ/hr}]$

Engine performance curves

SI Engine Characteristic Curves



SI Engine Characteristic Curves

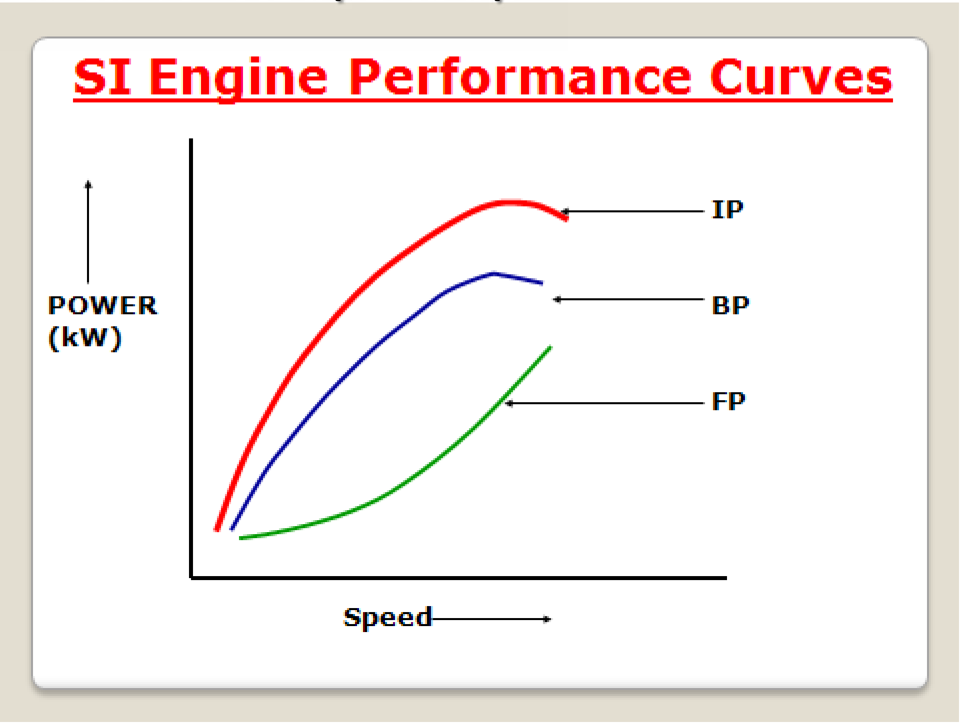
- Lab tests carried out to determine eng performance
- During tests, throttle is kept full (full /rated load, max fuel consumption) and speed is varied by adjusting the brake load
- IP, BP, FP, bsfc, mechanical & volumetric efficiencies etc are worked out
- Same tests can be repeated at half load

$$hp = \frac{Tx2\pi N}{60,000} kW$$

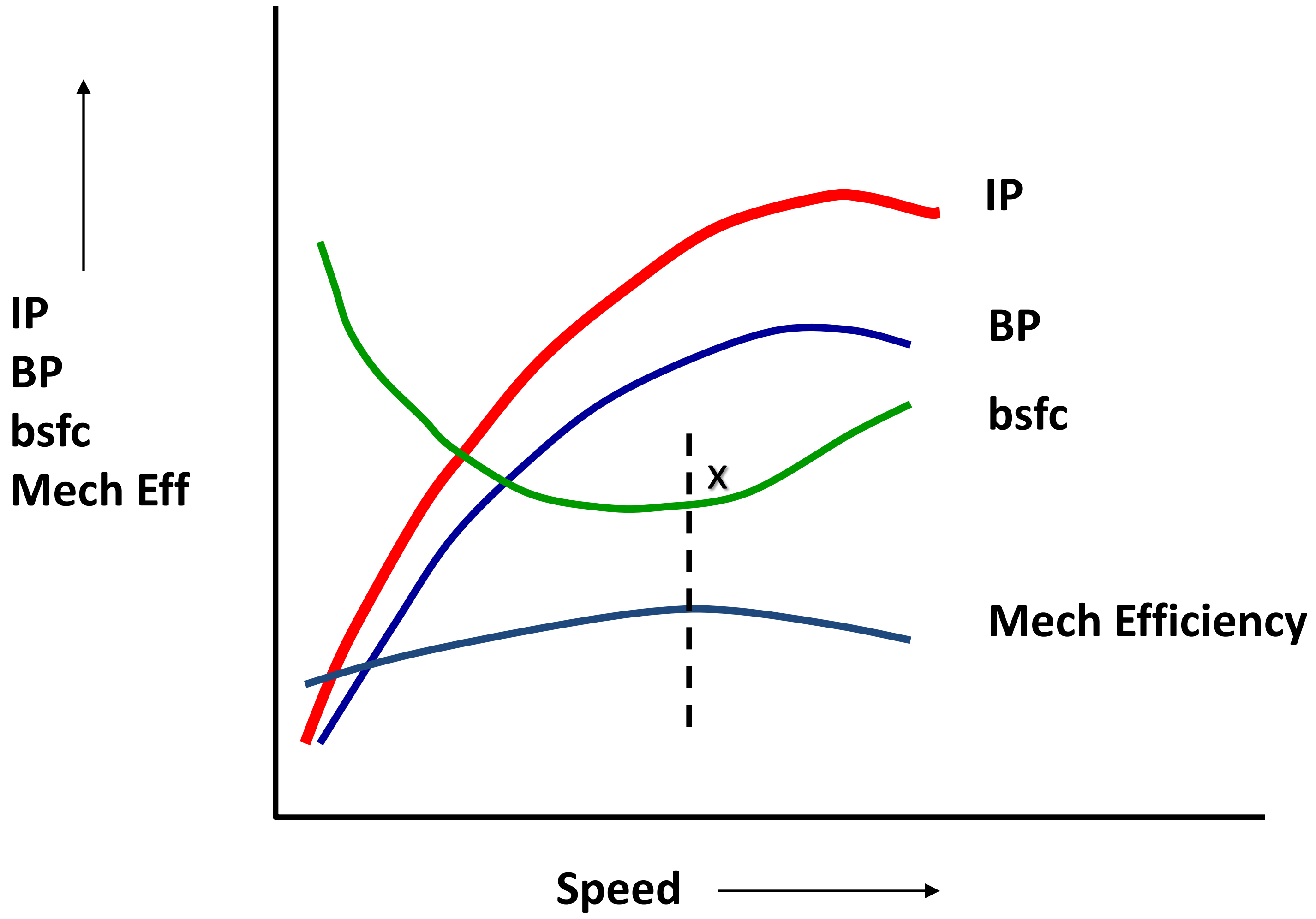
- At rated output, max p-V diagram area, hence max imep; For given torque; power ∞ N

SI Engine Characteristic Curves

- IP increases when imep or speed or both increase
- IP initially increases faster with speed, if inlet conditions are kept constant
- However, after certain limit, rate of increase of IP reduces with speed due to reduction in vol efficiency as air/charge velocity increase results in inlet pr drop
- Mech losses increase with increase in speed(FP∞ N^2) due to which increase in IP is off-set by steep increase in FP

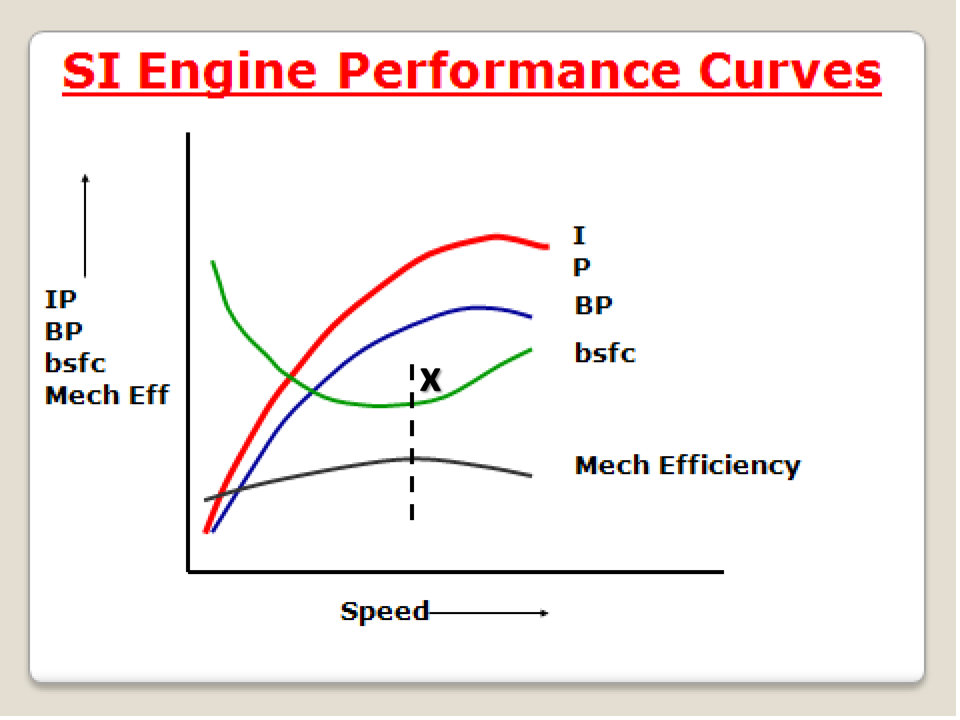


SI Engine Characteristic Curves

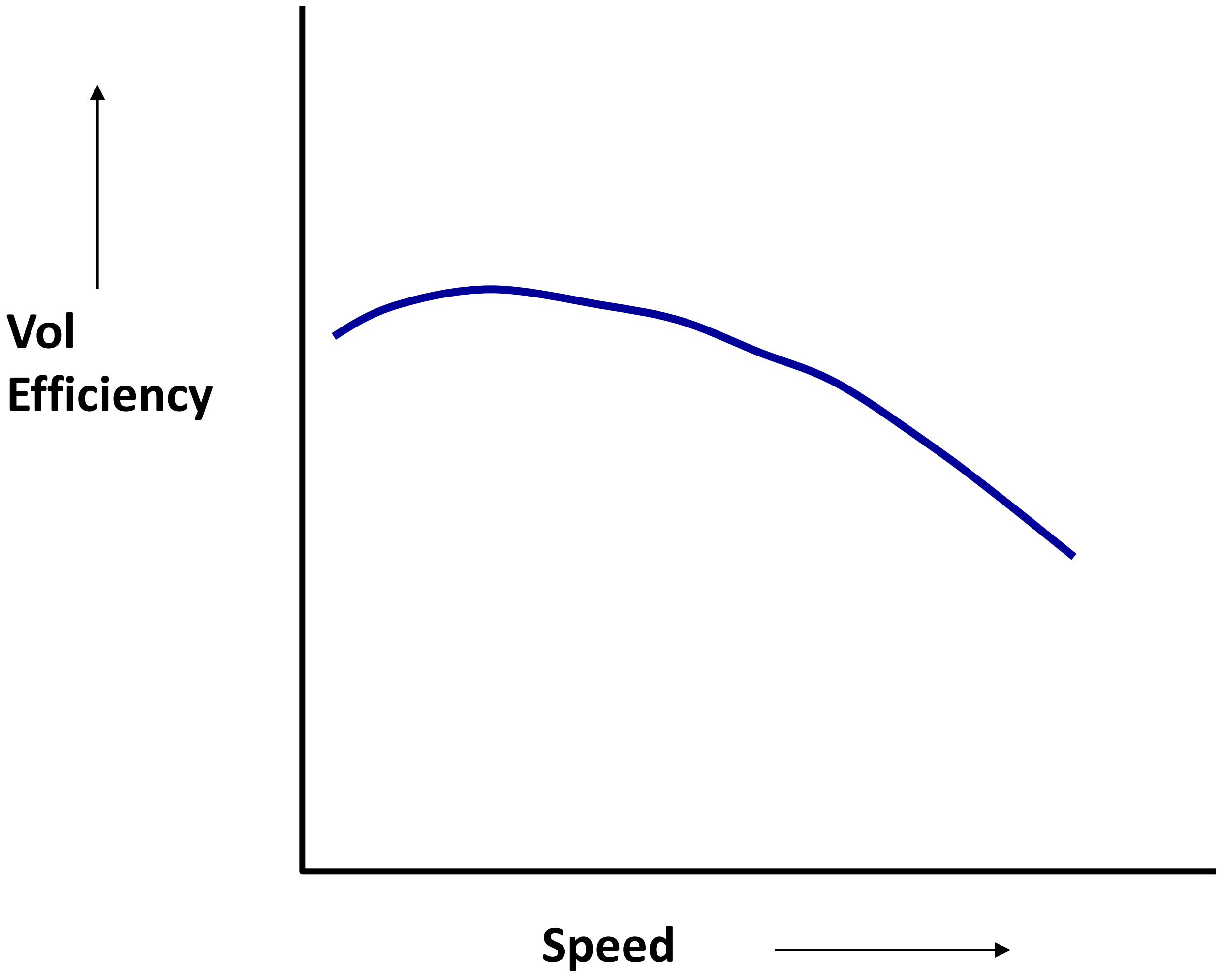


SI Engine Characteristic Curves

- As FP ∞ N^2, mech efficiency reduces due to steep increase in FP
- At lower speeds, due to lower charge velocity because of low piston speed, bsfc reduces since volumetric efficiency increases and mech efficiency also increases
- After certain speed, bsfc increases due to reduction in volumetric efficiency and increase in mech losses
- Point x represents economical speed of eng for min fuel consumption



SI Engine Characteristic Curve

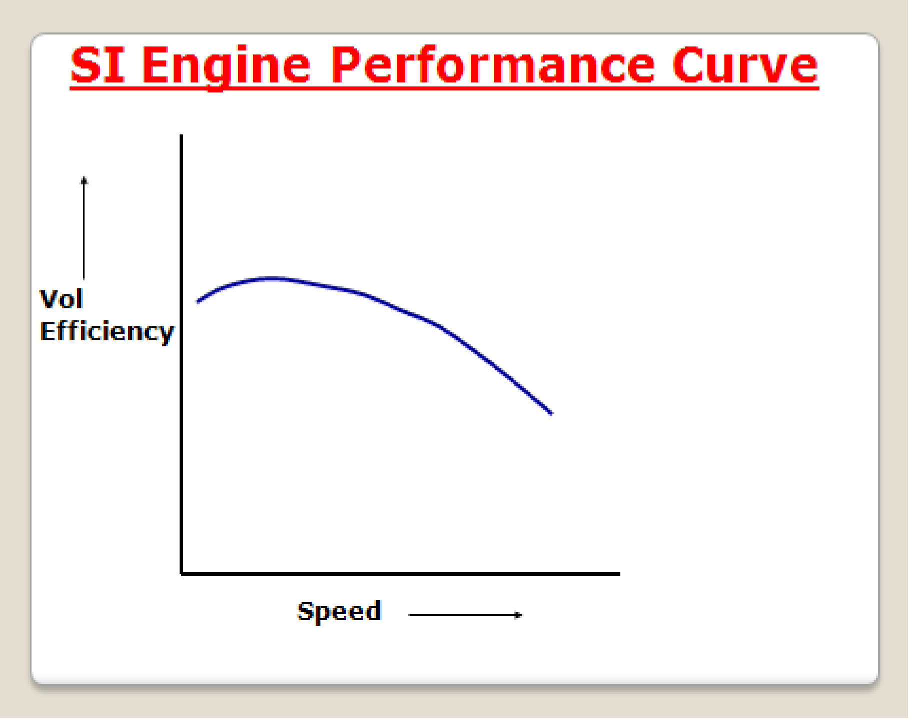


SI Engine Characteristic Curve

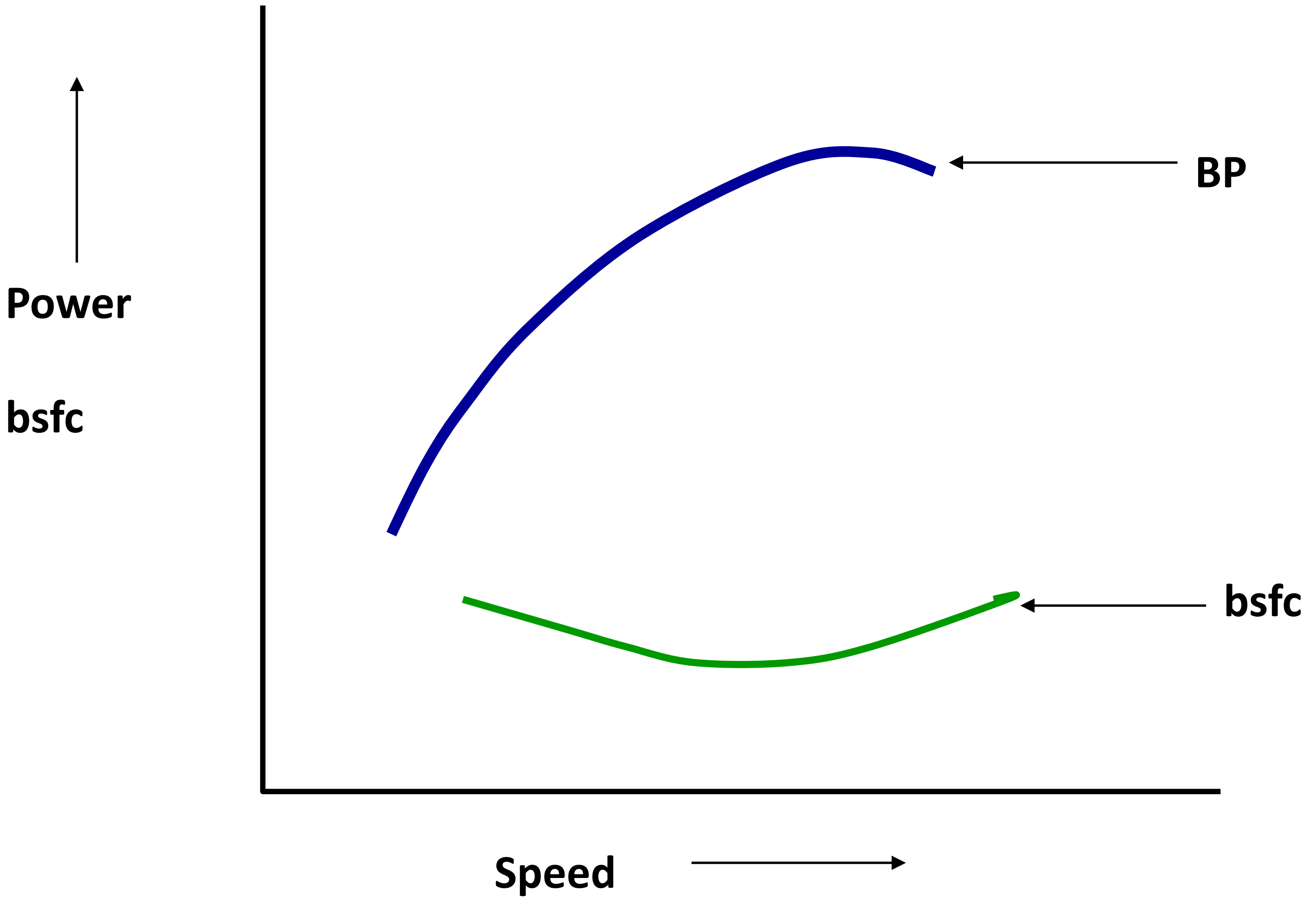
- Volumetric Efficiency reduces with increase in speed due to increase in intake velocity resulting in drop of suction pressure

- Higher the speed, lesser the time available for induction of charge

- Suction valve fully opens only when pressure inside cylinder slightly below the surrounding pressure, thus reducing effective suction stroke



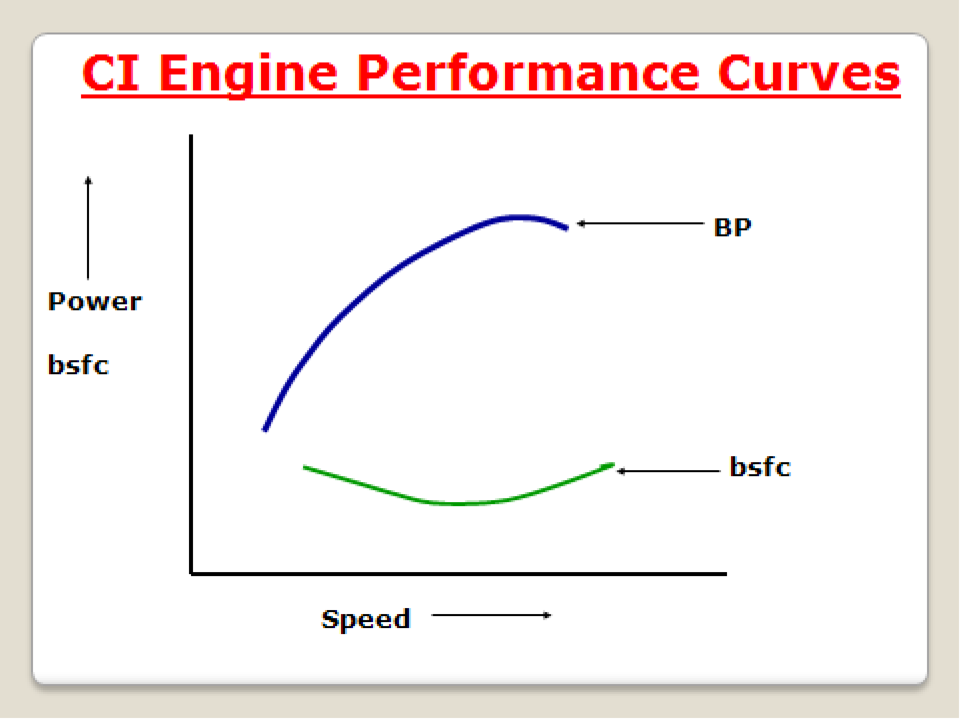
CI Engine Characteristic Curves



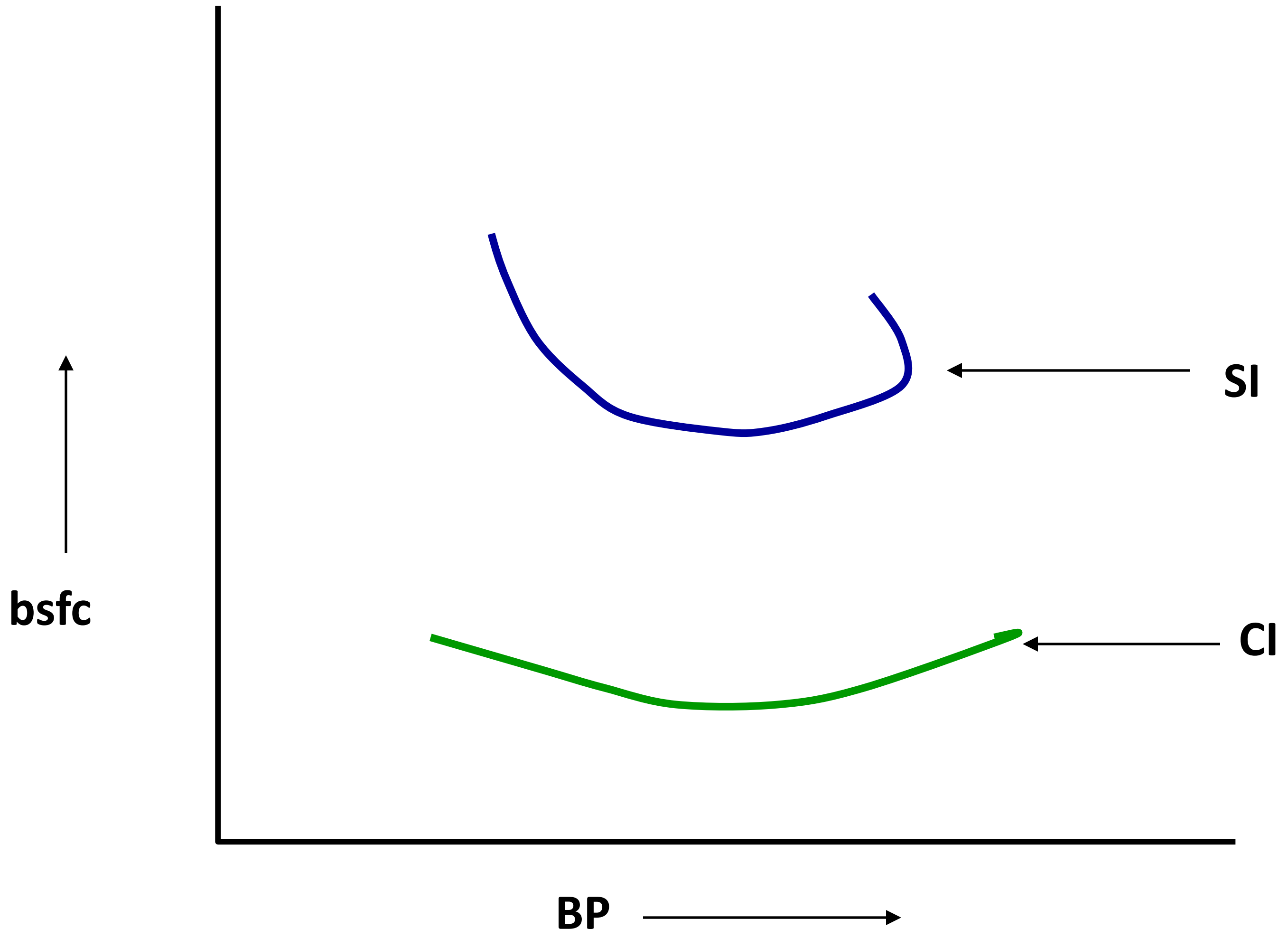
CI Engine Characteristic Curves

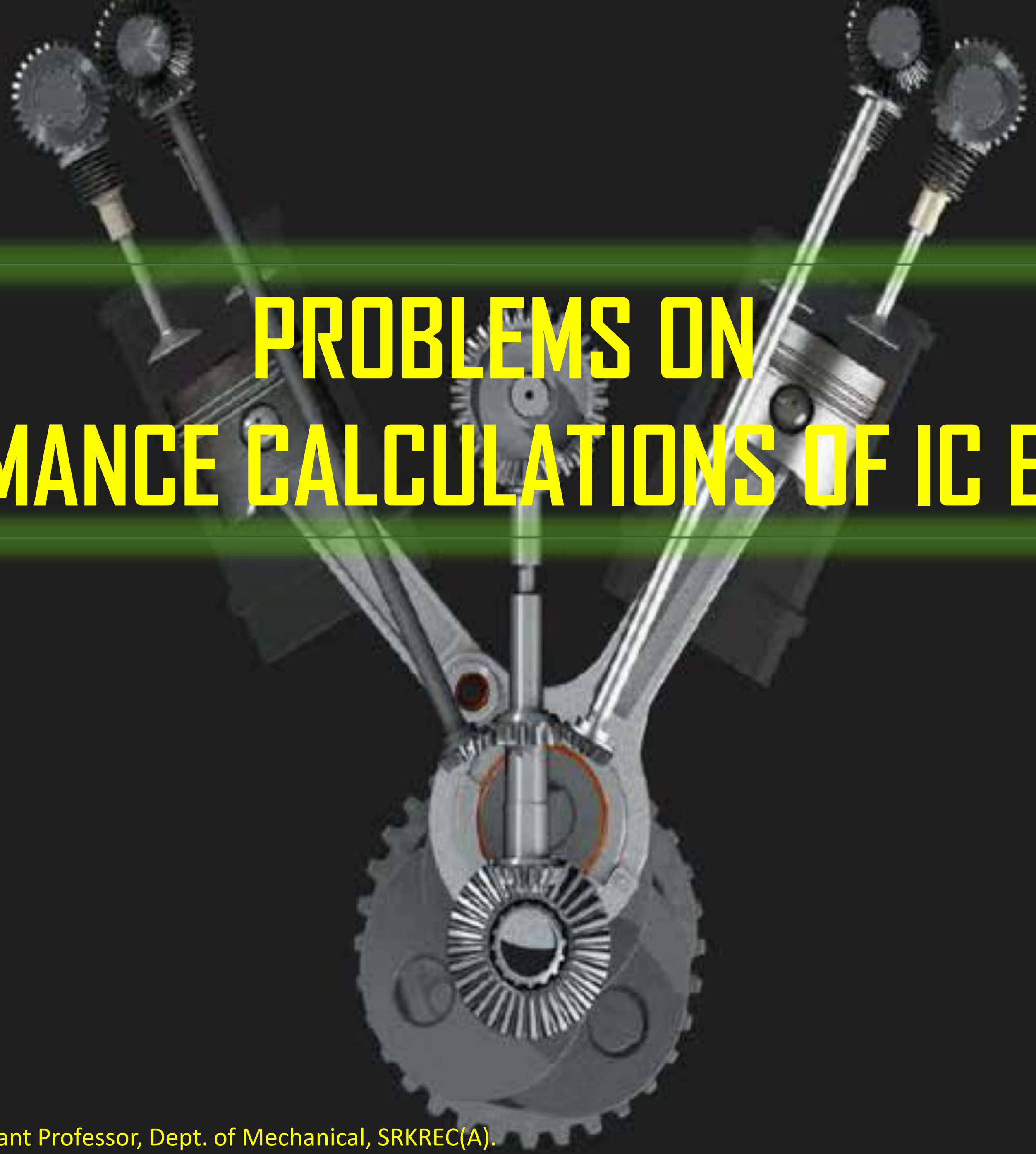
- IP and BP increase with speed but due to steep increase in FP, IP and BP start coming down

- For bsfc curve, same reasons as in SI engine



Engine Characteristic Curves





PROBLEMS ON PERFORMANCE CALCULATIONS OF IC ENGINES

V.Manikanth, Assistant Professor, Dept. of Mechanical, SRKREC(A).

Problem 1: *During test on single cylinder oil engine, working on the four stroke cycle fitted with a rope brake, the following readings are taken.*

Effective diameter of Brake wheel = 600 mm. Dead load on brake = 200 N; spring balance reading = 30 N; speed = 450 r.p.m; Area of indicator diagram = 400 mm^2*; length of indicator diagram = 60 mm; spring scale = 1.1 bar per mm. Bore = 100 mm; stroke = 150 mm; Quantity of oil = 0.815 kg/hr. Calorific value of oil = 42000 kJ/kg. Calculate the brake power, indicated power, mechanical efficiency, brake thermal efficiency and brake specific fuel consumption and Indicated thermal efficiency.*

Steps to solve the problem:

1. Read the problem carefully and understand each and every parameter.

2. Identify whether it is SI/CI engine, 2S/4S, No. of cylinders.

3. Identify the type of Dynamometer and its corresponding values and hints.

4. Identify what to find out & draw a rough flow chart

5. Note down the given data and start solving for the unknown data.

6. Note down all the formulae required to solve the problem

7. Now identify the missing date and solve it

8. Substitute all the values and find out what problem needs.

9. Verify logically all the final results.

Solution:

Given: Effective Radius $R = \dfrac{600}{2} = 300 \text{ mm} = 0.3 \text{ m}$

(Dead load)(W) = 200 N, S = 30 N, N = 450 r.p.m.

$a_d = 400 \text{ mm}^2, l_d = 60 \text{ mm}$, $s = 1.1$ bar/mm,

Bore dia. $D = 100 \text{ mm} = 0.1 \text{ m}, L = 150 \text{ mm} = 0.15 \text{ m},$

Mass of fuel $(\dot{m}_f) = 0.815$ kg/hr, $C.V = 42{,}000$ kJ/kg.

Brake power (BP)

$$\text{B.P} = \frac{2\pi N(W - S) \times R}{60}$$

$$= \frac{2\pi \times 450 \times (200 - 30) \times 0.3}{60}$$

$$B.P = 2403.32 \text{ W} = 2.403 \text{ kW}$$

Indicated Power (IP)

Before that, we have to find mean effective pressure (P_m)

$$P_m = \frac{a_d \times s}{l_d} = \frac{400}{60} \times 1.1 = 7.333 \text{ bar}$$

$$P_m = 7.333 \times 10^2 \text{KPa}$$

$$\text{Area of cylinder } A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4} \times 0.1^2$$

$$A = 7.854 \times 10^{-3} \text{m}^2$$

$$\text{Indicated Power I.P} = \frac{P_m AL(N/2) \times n}{60}$$

[$\because N/2$ for 4 stroke engine]

$$\text{I.P.} = \frac{7.333 \times 10^2 \times 7.854 \times 10^{-3} \times 0.15 \times (450/2) \times 1}{60}$$

[$\because n = 1$ for single cylinder]

$$= 3.2396 \text{ kW}$$

Mechanical Efficiency ($\eta_{\mathbf{mech}}$)

$$\eta_{mech} = \frac{B.P}{I.P} = \frac{2.403}{3.2396}$$

$$= 0.74175$$

$$= 74.175\,\%$$

Brake Thermal Efficiency

$$\eta_{\text{Brake}} = \frac{B.P. \times 3600}{\dot{m}_f \times C.V}$$

$$= \frac{2.403 \times 3600}{0.815 \times 42{,}000} = 0.25273$$

$$= 25.273\,\%$$

Indicated Thermal Efficiency

$$\eta_{\text{indicated}} = \frac{I.P \times 3600}{\dot{m}_f \times C.V}$$

$$= \frac{3.2396 \times 3600}{0.815 \times 42{,}000} = 0.3407$$

$$= 34.07\,\%$$

Brake specific fuel consumption $(SFC)_{\text{Brake}}$

$$(SFC)_{\text{Brake}} = \frac{\dot{m}_f}{\text{B.P.}}$$

$$= \frac{0.815}{2.403}\,\frac{\text{kg}}{\text{kW-hr}}$$

$$= \mathbf{0.3392}\,\frac{\textbf{kg}}{\textbf{kW-hr}}$$

2. During the test on a four stroke four-cylinder gasoline engine has a bore of 60 mm and a stroke of 100 mm develops a torque of 66.5 N-m when running at 3000 rpm. If the clearance volume in each cylinder is 60 cc, the relative efficiency with respect to break thermal efficiency is 0.5 and calorific value of the fuel is 42 MJ/kg, determine the fuel consumption in kg/h and the break mean effective pressure.

Given data: 4 stroke; 4 cylinder; $D = 0.06$ m ;

$L = 0.1$ m ; $T = 66.5$ Nm ; $N = 3000$ rpm

$V_c = 60 \text{ cm}^3 = 60 \times 10^{-6} \text{ m}^3$; $\eta_{\text{relative}} = 0.5$;

$CV = 42 \times 10^3$ kJ/kg

Solution:

$$\boxed{\begin{array}{c}\textbf{Compression ratio} \rightarrow \textbf{Air std } \eta \rightarrow \textbf{Brake thermal } \eta \\ \boldsymbol{BP} \rightarrow \boldsymbol{\dot{m}_f} \rightarrow (\boldsymbol{P_m})_{\textbf{brake}}\end{array}}$$

To find r

$$V_s = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4} \times 0.06^2 \times 0.1 = 2.83 \times 10^{-4} \text{ m}^3$$

$$r = \frac{V_s + V_c}{V_c} = \frac{2.83 \times 10^{-4} + 60 \times 10^{-6}}{60 \times 10^{-6}} = 5.712$$

$$\text{Air standard efficiency } = 1 - \frac{1}{(r)^{\gamma - 1}}$$

$$= 1 - \frac{1}{(5.712)^{0.4}} = 0.5$$

[Since it is gasoline engine, it is considered as petrol engine. So otto cycle η]

$$\text{Relative efficiency } = \frac{\text{Brake thermal efficency}}{\text{Air standard efficiency}}$$

$$\text{Brake thermal efficiency } = 0.5 \times 0.5 = 0.25$$

$$\text{Brake power } BP = \frac{2\,\pi\,NT}{60} = \frac{2\pi \times 3000 \times 66.5}{60} = 20891.6 \text{ watts}$$

$$= 20.892 \text{ kW}$$

To find fuel consumption ($\dot{m}_f$)

We know, brake thermal efficiency $= \dfrac{\text{B.P} \times 3600}{\dot{m}_f \times Cv}$

$$\dot{m}_f = \frac{20.892 \times 3600}{0.25 \times 42 \times 10^3}$$

$$= 7.16 \text{ kg/hr}$$

To find brake mean effective pressure $(P_m)_{\text{brake}}$

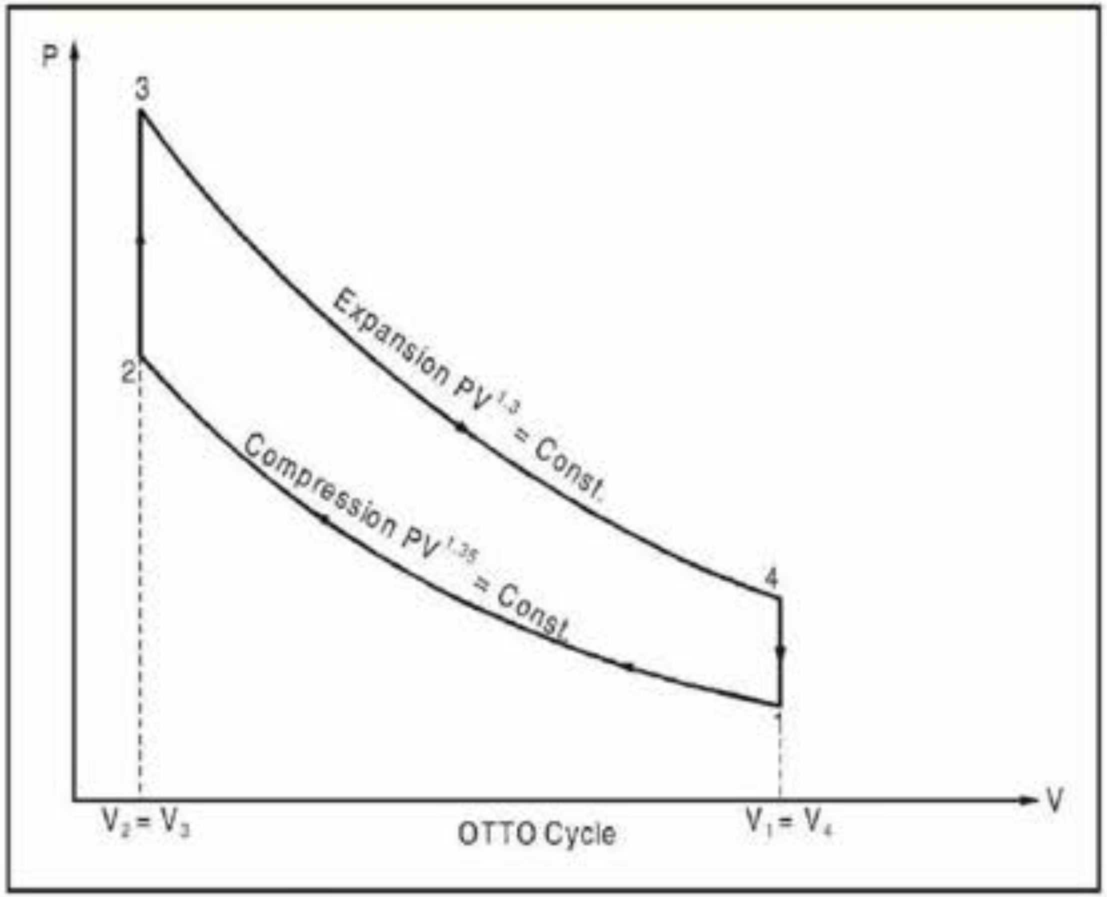
$$(P_m)_{\text{brake}} = \frac{\text{B.P} \times 60}{LA \times \left(\dfrac{N}{2}\right) \times n}$$

[$\dfrac{N}{2}$ for 4 stroke; $n = 4$ for 4 cylinder]

$$= \frac{20.892 \times 60}{0.1 \times \dfrac{\pi}{4} \times (0.06)^2 \times \left(\dfrac{3000}{2}\right) \times 4}$$

$$= 738.9 \text{ kPa} = 7.4 \text{ bar}$$

5. **Consider the following data refer to an oil engine working on Otto four-stroke cycle: Brake power is 14.7 kW, Suction pressure is 0.9 bar, Mechanical efficiency is 80%, Ratio of compression is 5, Index of compression curve is 1.35, Index of expansion curve is 1.3, Maximum explosion pressure is 24 bar, Engine speed is 1000 rpm, Ratio of stroke: bore is 1.5. Find the diameter and stroke of the piston.**



Given:

B.P. = 14.7 kW, $P_1 = 0.9$ bar, $\eta_{mech} = 80\%$,

$r = 5$, $P_3 = 24$ bar

$N = 1000$ r.p.m, $\dfrac{L}{D} = 1.5$; $D = ?, L = ?$

$$\text{Compression ratio } r = \frac{V_1}{V_2} = \frac{V_4}{V_3}$$

To find P_2: (compression process 1-2)

$$P_1 V_1^{1.35} = P_2 V_2^{1.35}$$

or $$P_2 = \left(\frac{V_1}{V_2}\right)^{1.35} \times P_1 = (5)^{1.35} \times 0.9$$

$\therefore$ $$P_2 = P_1 \times 8.78 = 0.9 \times 8.78 = 7.9 \text{ bar}$$

To find P_4: (Expansion process 3-4)

$$P_3 V_3^{1.3} = P_4 V_4^{1.3}$$

$$[P_3/P_4] = \left(\frac{V_4}{V_3}\right)^{1.3} = (5)^{1.3} = 8.1$$

$$P_4 = \frac{P_3}{8.1} = \frac{24}{8.1} = 2.96 \text{ bar}$$

Work done/cycles = Area $1-2-3-4$

= (Area under the curve $3-4$) – (area under the curve $1-2$)

$$= \frac{P_3 V_3 - P_4 V_4}{1.3-1} - \frac{P_2 V_2 - P_1 V_1}{1.35-1}$$

$$= \frac{10^2 (24 V_3 - 2.96 V_4)}{0.3} - \frac{10^2 (7.9 V_3 - 0.9 V_4)}{0.35} \quad [\because V_1 = V_4 \text{ and } V_2 = V_3]$$

$$= [(80\ V_3 - 9.87\ V_4) - (22.57 V_3 - 2.57 V_4)] \times 10^2$$

$$= (57.43\ V_3 - 7.3\ V_4) \times 10^2$$

$$= (57.43\ V_3 - 7.3 \times 5V_3) \times 10^2 \qquad \left[\because \frac{V_4}{V_3} = 5\right]$$

$$= 2093\ V_3 \text{ kN} - \text{m}$$

Mean effective pressure

$$P_m = \frac{\text{Work done/cycle}}{\text{Stroke volume } (V_s)}$$

$$= \frac{2093 V_3}{(V_4 - V_3)} = \frac{2093\ V_3}{5V_3 - V_3} = 523.25 \text{ kPa} = 5.23 \text{ bar}$$

Now, $\eta_{mech} = \dfrac{\text{B.P.}}{\text{I.P.}}$

$$\therefore \quad \text{I.P.} = \frac{\text{B.P.}}{\eta_{mech}} = \frac{14.7}{0.8} = 18.37 \text{ kW}$$

To find D and L:

$$I.P = \frac{P_m\ LA\left(\frac{N}{2}\right) \times n}{60}$$

[$\because \dfrac{N}{2}$ for 4 stroke $n = 1$ for single cylinder]

$$18.37 = \frac{523 \times 1.5D \times \left(\frac{\pi}{4} \times D^2\right) \times \frac{1000}{2} \times 1}{60}$$

$$D^3 = 3.5795 \times 10^{-3}$$

$$D = 0.152 \text{ m}$$

$$L = 1.5D = 1.5 \times 0.152 = 0.229 \text{ m}$$

3. On a four cylinder, four stroke petrol engine test rig, the following results were found: B.P with all cylinders working = 24.25 kW, B.P with cylinder No:1 cut off = 16.53 kW, B.P with cylinder No:2 cut off = 17.2 kW, B.P with cylinder No:3 cut off = 17.34 kW, B.P with cylinder No:4 cut off = 17.8 kW. Estimate the indicated power of the engine and its mechanical efficiency.

Solution:

Given B.P. or simply B = 24.25 kW, $B_1 = 16.53$ kW;

$B_2 = 17.2$ kW; $B_3 = 17.34$ kW; $B_4 = 17.8$ kW

$$I_1 = B - B_1 = 24.25 - 16.53 = 7.72 \text{ kW}$$

$$I_2 = B - B_2 = 24.25 - 17.2 = 7.05 \text{ kW}$$

$$I_3 = B - B_3 = 24.25 - 17.34 = 6.91 \text{ kW}$$

$$I_4 = B - B_4 = 24.25 - 17.8 = 6.45 \text{ kW}$$

Total Indicated Power $IP = I_1 + I_2 + I_3 + I_4$

$$= 7.72 + 7.05 + 6.91 + 6.45$$

$$= 28.13 \text{ kW}$$

Mechanical efficiency $\eta_{mech} = \dfrac{B.P}{I.P}$

$$= \frac{24.25}{28.13} = 0.8621$$

$$= 86.21\ \%$$

4. **The following readings were obtained during a brake on a four cylinder, four stroke engine coupled to a hydraulic dynamometer at constant speed. B.P. with all cylinders working = 14.7 kW, B.P. with cylinder No.1. Cut off = 10.14 kW, B.P. with cylinder No.2. Cut off = 10.3 kW, B.P. with cylinder No.3. Cut off = 10.36 kW, B.P. with cylinder No.4. Cut off = 10.21 kW. Petrol consumption = 5.5 kg/hr, Calorific value of petrol = 44,000 kJ/Kg, Dia. of cylinder is 8cm, Stroke of piston is 10 cm, Clearance volume is 0.1 liter. Calculate (i) Mechanical efficiency (ii) Relative efficiency on the basis of IP.**

Given: B.P (or) simply B = 14.7 kW; $B_1 = 10.14$; $B_2 = 10.3$;

$B_3 = 10.36$; $B_4 = 10.21$; $\dot{m}_f = 5.5$ kg/hr

C.V = 42,000 kJ/Kg; D = 0.08 m; L = 0.1 m

$$V_c = \text{Clearance volume} = 0.1 \text{ litre} = 0.1 \times 10^{-3} \text{ m}^3$$

$$[\because 1000 \text{ lit} = 1 \text{ m}^3; \text{So } 1 \text{ lit} = \frac{1}{1000} \text{ m}^3]$$

$$I_1 = B - B_1 = 14.7 - 10.14 = 4.56 \text{ kW}$$

$$I_2 = B - B_2 = 14.7 - 10.3 = 4.4 \text{ kW}$$

$$I_3 = B - B_3 = 14.7 - 10.36 = 4.34 \text{ kW}$$

$$I_4 = B - B_4 = 14.7 - 10.21 = 4.49 \text{ kW}$$

$$\text{Total I.P.} = I_1 + I_2 + I_3 + I_4 = 4.56 + 4.4 + 4.34 + 4.49$$

$$= 17.79 \text{ kW}$$

1. **Mechanical efficiency:** $\eta_{\text{mech}} = \dfrac{\text{B.P.}}{\text{I.P}} = \dfrac{14.7}{17.79}$

$$= 0.8263 = 82.63\%$$

Indicated thermal efficiency: $\eta_{\textbf{indicated}}$

$$\eta_{\text{indicated}} = \frac{\text{I.P.} \times 3600}{\dot{m}_f \times \text{C.V}}$$

$$= \frac{17.79 \times 3600}{5.5. \times 42{,}000} = 0.27724$$

$$= 27.725\ \%$$

Air Standard efficiency: $\eta_{\textbf{Air standard}}$

$$\eta_{\text{air standard}} = 1 - \frac{1}{(r)^{\gamma - 1}}$$

[Since it is a petrol engine the otto cycle efficiency is used] where r = compression ratio $\frac{V_s + V_c}{V_c}$

where V_s = Swept volume and V_c = Clearance volume

$$V_s = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4} \times (0.08)^2 \times 0.1 = 5.027 \times 10^{-4} \text{ m}^3$$

$$V_c = 0.1 \times 10^{-3} \text{ m}^3 \text{ (given)}$$

$$\text{So, } r = \frac{V_s + V_c}{V_c} = \frac{5.027 \times 10^{-4} + 0.1 \times 10^{-3}}{0.1 \times 10^{-3}}$$

$$= 6.027$$

$$\eta_{\text{air standard}} = 1 - \frac{1}{(6.027)^{1.4-1}} = 0.51252$$

$$= 51.252\,\%$$

$$\left.\begin{array}{c}\text{Relative efficiency} \\ \text{(or)} \\ \text{Efficiency ratio} \\ \text{on the basis of IP}\end{array}\right\} = \frac{\eta_{\text{i}}\text{ndicated}}{\eta_{\text{air standard}}} = = \frac{0.27724}{0.51252} = 0.541$$

$$\eta_{\textbf{Relative}} = \mathbf{54.1\%}$$

7.During the trial of a single cylinder 4-stroke oil engine, the following results were obtained. Cylinder dia. is 20 cm; stroke=40 cm, MEP=6 bar, torque=407 N-m, speed=250 rpm, Oil consumption is 4 kg/h, Calorific value is 43 MJ/kg, cooling water flow rate is 45 kg/min, air used per kg of fuel 30 kg, rise in cooling water temperature 450C, temperature of exhaust gases 4200C, room temperature 200C, mean specific heat of exhaust gas is 1 kJ/kg-K. Find the indicated power, brake power, and draw the heat balance sheet for the test in kJ/hr.

To find Indicated Power

$$\text{I.P} = \frac{P_m \times L \times A \times \frac{N}{2}}{60} \qquad \left[\because \frac{N}{2} \text{ for 4 stroke} \right]$$

$$= \frac{600 \times 0.4 \times \left[\frac{\pi}{4} \times 0.2^2 \right] \times \frac{250}{2}}{60} = 15.71 \text{ kW}$$

To find Brake Power

$$BP = \frac{2\,\pi\, NT}{60} = \frac{2\,\pi \times 250 \times 0.407}{60} = 10.65 \text{ kW}$$

To draw Heat Balance Sheet [in hour basis]

Heat supplied by fuel $Q_s = \dot{m}_f \times C.V$

$$= 4 \times 43 \times 10^3 = 172 \times 10^3 \text{ kJ/hr}$$

$\boldsymbol{Q_{ip}}$

Heat utilized for I.P $= 15.71 \times 3600$

$$= 56.556 \times 10^3 \text{ kJ/hr}$$

$\boldsymbol{Q_w}$

Heat carried out by cooling water

$$Q_w = \dot{m}_w \, C_{pw} \, (t_2 - t_1)$$

$$= 4.5 \times 60 \times 4.187 \ (45) = 50.872 \times 10^3 \text{ kJ/hr}$$

Heat lost through exhaust gases 'Q_g'

$$Q_g = \dot{m}_g \times C_{pg} \times (\Delta t)_g$$

$$A : F = 30 : 1$$

$$\dot{m}_g = (\text{Air + Oil}) \text{ consumption/hr}$$

$$= (30 \times 4 + 4) = 124 \text{ kg/hr}$$

$$Q_g = 124 \times 1 \times (420 - 20) = 49.6 \times 10^3 \text{ kJ/hr}$$

To find Unaccounted loss 'Q_U'

$$Q_u = Q_s - [Q_{IP} + Q_w + Q_g]$$

$$= 172 \times 10^3 - [56.56 \times 10^3 + 50.872 \times 10^3 + 49.6 \times 10^3]$$

$$Q = 14.972 \times 10^3 \text{ kJ/hr}$$

Now we can draw the heat balance sheet.

Heat Balance Sheet

CREDIT			DEBIT		
Heat supplied per hour	**kJ/hr**	**%**	**Heat expenditure per hour**	$\frac{kJ}{hr}$	**%**
Heat supplied by the combustion of fuel	172×10^3		1. Heat utilized for I.P. ($Q_{I.P.}$)	56.56×10^3	32.9.%
			2. Heat carried out by cooling water (Q_w)	50.87×10^3	29.6 %
			3. Heat lost through exhaust gases (Q_g)	49.6×10^3	28.8%
			4. Unaccounted heat loss (Q_u)	14.97×10^3	8.7%
Total	172×10^3	100%		172×10^3	100%

Problem *Calculate the relative efficiency based on indicated power and A:F ratio for a four stroke gas engine working on otto cycle from the following data:*

Brake power = 5 kW; Speed = 180 r.p.m; Volumetric efficiency=85%; Clearance volume $= 1500\ cm^3$; *Swept volume* $= 6500\ cm^3$; $\eta_{mech} = 80\%$; *Fuel consumption 4* m^3/hr $C.V = 17{,}000\ kJ/m^3$.

Solution:

To Find $\eta_{\mathbf{relative}}$

$$\eta_{relative} = \frac{\text{Indicated thermal } \eta}{\text{Air standard } \eta}$$

$$\text{I.P} = \frac{\text{B.P}}{\eta_{\text{mech}}} = \frac{5}{0.8} = 6.25 \text{ kW}$$

$$\eta_{\text{Indicated thermal}} = \frac{I.P \times 3600}{V_f \times C.V}$$

$$= \frac{6.25 \times 3600}{4 \times 17{,}000} = 0.331 = 33.1\%$$

$$\eta_{\text{Air standard}} = 1 - \frac{1}{(r)^{\gamma - 1}}$$

where $r = \text{Compression ratio} = \dfrac{V_c + V_s}{V_c}$

$$= \frac{1500 + 6500}{1500} = 5.333$$

$$\eta_{\text{Air standard}} = 1 - \frac{1}{(5.333)^{0.4}} = 0.4881 = 48.81\%$$

$$\eta_{\text{relative}} = \frac{\eta_{\text{indicated thermal}}}{\eta_{\text{air std}}} = \frac{0.331}{0.4881} = 0.6782$$

$$= 67.82\%$$

V_{cycle} in $m^3 = V_s \times \eta_{\text{vol}}$ [If η_{vol} is not given, then $\eta_{\text{vol}} = 1$]

$$\dot{V} \text{ in } m^3/\text{sec} = V_{\text{cycle}} \times \frac{(N \text{ or } N/2)}{60} \times \text{No. of cylinders}$$

[N for 2 stroke and $N/2$ for 4 stroke]

Volume of mixture admitted into the cylinder per cycle

$$V_{\text{cycle}} \text{ in } m^3 = V_s \times \eta_{\text{vol}} = 6500 \times 0.85 = 5525 \text{ cm}^3$$

$$= 5525 \times 10^{-6} m^3$$

Volume of fuel consumed per cycle $= \dfrac{4}{60} \times \dfrac{1}{N/2}$

$$= \frac{4}{60} \times \frac{1}{90}$$

$$= 7.4074 \times 10^{-4} \text{m}^3$$

So volume of air per cycle

$$= 5525 \times 10^{-6} - 7.4074 \times 10^{-4}$$

$$= 4.7842 \times 10^{-3} \text{m}^3$$

A:F ratio by volume $= \dfrac{4.7842 \times 10^{-3}}{7.4074 \times 10^{-4}}$

$$= 6.4588 : 1$$

Problem : *A 2 cylinder 4 stroke engine runs at 240 rpm developing a torque of 5 kN-m. The bore and stroke of cylinder are 30 cm and 60 cm respectively. Engine runs with gaseous fuel having calorific value of* $16.8\ \mathrm{MJ/m^3}$. *The gas and air mixture is supplied in proposition of 1:7 by volume. The volumetric efficiency is 0.85. Determine, (i) Brake Power (ii) The piston speed in m/s. (iii) The brake mean effective pressure (iv) The brake thermal efficiency.*

Solution:

(i) Brake Power

$$B.P = \frac{2\,\pi\,NT}{60} = \frac{2\,\pi \times 240 \times 5}{60} = 125.67 \text{ kW}$$

(ii) Mean piston speed $\left(2\,\frac{LN}{60}\right)$

$$v_p = \frac{2 \times 0.6 \times 240}{60} = 4.8 \text{ m/s}$$

(iii) P_{mb} Brake mean effective pressure

$$B.P = \frac{P_{mb} \times LA \times \frac{N}{2} \times n}{60}$$

$$\left[\because \frac{N}{2} \text{ for 4 stroke } n = 2 \text{ cylinder}\right]$$

$$P_{mb} = \frac{125.67 \times 60}{0.6 \times \left(\frac{\pi}{4} \times 0.3^2\right) \times \left(\frac{240}{2}\right) \times 2}$$

$$= 740.95 \text{ kPa} = 7.41 \text{ bar}$$

(iv) Brake thermal efficiency η_{brake}

$$\eta_{\text{brake}} = \frac{\text{B.P} \times 3600}{\dot{V}_f \times CV}$$

To find $\dot{V}_f$

$$V_s = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4} \times 0.3^2 \times 0.6 = 0.0424 \text{ m}^3$$

Actual volume $V = 0.0424 \times 0.85 = 0.036 \text{ m}^3$ in 1 cycle

Actual volume of mixture in m^3/s $= \dot{V} = V \times \dfrac{N}{2 \times 60} \times n$

$$\left[\because \frac{N}{2} \text{ for 4 stroke } n = 2 \text{ for 2 cylinder}\right]$$

$$\dot{V} = 0.036 \times \frac{240}{2 \times 60} \times 2$$

$$= 0.144 \text{ m}^3/s$$

$$\dot{V} = \text{ Volume of mixture } = \dot{V}_a + \dot{V}_f = 0.1442$$

$$[\dot{V}_f = \text{ Volume of fuel in m}^3\text{/s}]$$

$$= 7\,\dot{V}_f + \dot{V}_f = 8\,\dot{V}_f = 0.1442$$

$$\dot{V}_f = \frac{0.1442}{8} = 0.018 \text{ m}^3\text{/s}$$

$$\dot{V}_f \text{ in m}^3\text{/hr} = 64.8 \text{ m}^3\text{/hr}$$

$$\text{Brake thermal } \eta = \frac{\text{B.P} \times 3600}{\dot{V}_f \times CV}$$

$$= \frac{125.67 \times 3600}{64.8 \times 16.8 \times 10^3} = 0.4156$$

$$= 41.56\%$$

Example 23.45. *From the data given below, calculate indicated power, brake power and draw a heat balance sheet for a two-stroke diesel engine run for 20 minutes at full load :*

r.p.m.	= 350
m.e.p.	= 3.1 bar
Net brake load	= 640 N
Fuel consumption	= 1.52 kg
Cooling water	= 162 kg
Water inlet temperature	= 30°C
Water outlet temperature	= 55°C
Air used/kg of fuel	= 32 kg
Room temperature	= 25°C
Exhaust temperature	= 305°C
Cylinder bore	= 200 mm
Cylinder stroke	= 280 mm
Brake diameter	= 1 metre
Calorific value of fuel	= 43900 kJ/kg
Steam formed per kg of fuel in the exhaust	= 1.4 kg
Specific heat of steam in exhaust	= 2.09 kJ/kg K
Specific heat of dry exhaust gases	= 1.0 kJ/kg K.

Solution. *Given* : $N = 350$ r.p.m., $p_{mi} = 3.1$ bar, $(W\text{-}S) = 640$ N, $m_f = 1.52$ kg, $m_w = 162$ kg, $t_{w_1} = 30°C$, $t_{w_2} = 55°C$, $m_a = 32$ kg/kg of fuel, $t_r = 25°C$, $t_g = 305°C$, $D = 0.2$ m, $L = 0.28$ m, $D_b = 1$ m, $C = 43900$ kJ/kg, $c_{ps} = 2.09$, $c_{pq} = 1.0$ and $k = 1$ ..... for two-stroke cycle engine.

(*i*) **Indicated power, I.P. :**

$$\text{I.P.} = \frac{np_{mi}LANk \times 10}{6}$$

$$= \frac{1 \times 3.1 \times 0.28 \times \pi / 4 \times 0.2^2 \times 350 \times 1 \times 10}{6}$$

$= \mathbf{15.9\ kW.}$ **(Ans.)**

(*ii*) **Brake power, B.P. :**

$$\text{B.P.} = \frac{(W - S)\,\pi\, D_b N}{60 \times 1000} = \frac{640 \times \pi \times 1 \times 350}{60 \times 1000}$$

$= \mathbf{11.73\ kW.}$ **(Ans.)**

—Heat supplied in *20 minutes*

$= 1.52 \times 43900 = 66728$ kJ

(*i*) Heat equivalent of I.P. in 20 minutes

$= \text{I.P.} \times 60 \times 20 = 15.9 \times 60 \times 20 = 19080$ kJ

(ii) Heat carried away by cooling water

$$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

$$= 162 \times 4.18 \times (55 - 30) = 16929 \text{ kJ}$$

Total mass of air $= 32 \times 1.52 = 48.64$ kg

Total mass of exhaust gases

$$= \text{Mass of fuel} + \text{mass of air}$$

$$= 1.52 + 48.64 = 50.16 \text{ kg}$$

Mass of steam formed $= 1.4 \times 1.52 = 2.13$ kg

$\therefore$ Mass of dry exhaust gases

$$= 50.16 - 2.13 = 48.03 \text{ kg}$$

(*iii*) Heat carried away by dry exhaust gases

$$= m_g \times c_{pg} \times (t_g - t_r)$$

$$= 48.03 \times 1.0 \times (305 - 25) = 13448 \text{ kJ}$$

(*iv*) Heat carried away by steam

$$= 2.13[h_f + h_{fg} + c_{ps}(t_{sup} - t_s)]$$

At 1.013 bar pressure (atmospheric pressure assumed) :

$$h_f = 417.5 \text{ kJ/kg}, h_{fg} = 2257.9 \text{ kJ/kg}$$

$$= 2.13 [417.5 + 2257.9 + 2.09 (305 - 99.6)]$$

$= 6613$ kJ/kg ..... neglecting sensible heat of water at room temperature.

Heat balance sheet (20 minute basis) :

Item	kJ	Per cent
Heat supplied by fuel	**66728**	**100**
(*i*) Heat equivalent of I.P.	19080	28.60
(*ii*) Heat carried away by cooling water	16929	25.40
(*iii*) Heat carried away by dry exhaust gases	13448	20.10
(*iv*) Heat carried away by steam in exhaust gases	6613	9.90
(*v*) Heat unaccounted for (by difference)	10658	16.00
Total	66728	100.00

A Lecture on

COMBUSTION IN SI ENGINES

By

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Syllabus: Combustion in SI Engines

Normal combustion and abnormal combustion Importance of flame speed and effect of engine variables-types of abnormal combustion pre-ignition and knock, Fuel requirements and fuel rating, anti-knock additions, Combustion chamber requirements and Types of combustion chamber, Design principles of combustion chambers.

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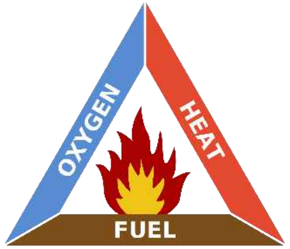
What is Combustion?



Combustion is a chemical reaction in which certain elements of the fuel like hydrogen and carbon combine with oxygen liberating heat energy and causing an increase in temperature of the gases. The conditions necessary for combustion are the presence of combustible mixture and some means of initiating the process. The theory of combustion is a very complex subject and has been a topic of intensive research for many years. In spite of this, not much knowledge is available concerning the phenomenon of combustion.

The process of combustion in engines generally takes place either in a homogeneous or a heterogeneous fuel vapour-air mixture depending on the type of engine.

HOMOGENEOUS MIXTURE

In spark-ignition engines a nearly homogeneous mixture of air and fuel is formed in the carburettor. Homogeneous mixture is thus formed outside the engine cylinder and the combustion is initiated inside the cylinder at a particular instant towards the end of the compression stroke. The flame front spreads over a combustible mixture with a certain velocity. In a homogeneous gas mixture the fuel and oxygen molecules are more or less, uniformly distributed.

Once the fuel vapour-air mixture is ignited, a flame front appears and rapidly spreads through the mixture. The flame propagation is caused by heat transfer and diffusion of burning fuel molecules from the combustion zone to the adjacent layers of unburnt mixture. The flame front is a narrow zone separating the fresh mixture from the combustion products. The velocity with which the flame front moves, with respect to the unburned mixture in a direction normal to its surface is called the normal flame velocity.

In a homogeneous mixture with an equivalence ratio, ϕ, *(the ratio of the actual fuel-air ratio to the stoichiometric fuel-air ratio)* close to 1.0, the flame speed is normally of the order of 40 cm/s. However, in a spark-ignition engine the maximum flame speed is obtained when ϕ is between 1.1 and 1.2, i.e., when the mixture is slightly richer than stoichiometric.

If the equivalence ratio is outside this range the flame speed drops rapidly to a low value. When the flame speed drops to a very low value, the heat loss from the combustion zone becomes equal to the amount of heat-release due to combustion and the flame gets extinguished. Therefore, it is quite preferable to operate the engine within an equivalence ratio of 1.1 to 1.2 for proper combustion. However, by introducing turbulence and incorporating proper air movement, the flame speed can be increased in mixtures outside the above range.

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HETEROGENEOUS MIXTURE

In a heterogeneous gas mixture, the rate of combustion is determined by the velocity of mutual diffusion of fuel vapours and air and the rate of chemical reaction is of minor importance. Self-ignition or spontaneous ignition of fuel-air mixture, at the high temperature developed due to higher compression ratios, is of primary importance in determining the combustion characteristics.

When the mixture is heterogeneous the combustion can take place in an overall lean mixture since, there are always local zones where ϕ varies between 1.0 and 1.2 corresponding to maximum rate of chemical reaction. Ignition starts in this zone and the flame produced helps to burn the fuel in the adjoining zones where the mixture is leaner. Similarly, in the zones where the mixture is rich the combustion occurs because of the high temperature produced due to combustion initiated in the zones where ϕ is 1.0 to 1.2.

A comprehensive study of combustion in both spark-ignition and compression-ignition engines is given in the following sections.

Combustion phenomenon in SI engines:

Normal Combustion. In a spark-ignition engine, a single intensely high temperature spark passes across the electrodes, leaving behind a thin thread of flame. From this thin thread, combustion spreads to the envelope of mixture immediately surrounding it at a rate which depends primarily upon the temperature of the flame front itself and to a secondary degree, upon both the temperature and the density of the surrounding envelope. In the actual engine cylinder, the mixture is not at rest but is in highly turbulent condition. The turbulence breaks the filament of a flame into a ragged front, thus presenting a far greater area of surface from which heat is being radiated ; hence its advance is speeded up enormously.

According to Ricardo, the combustion process can be imagined as if developing in two stages, *one the growth and development of a self-propagating nucleus of flame (ignition lag), and the other the spread of that flame throughout the combustion chamber*. The former is a chemical process depending upon the nature of the fuel, temperature and pressure, the proportion of the exhaust gas and also upon the temperature co-efficient of the fuel, *i.e.*, the relationship between temperature and rate of acceleration of oxidation or burning.

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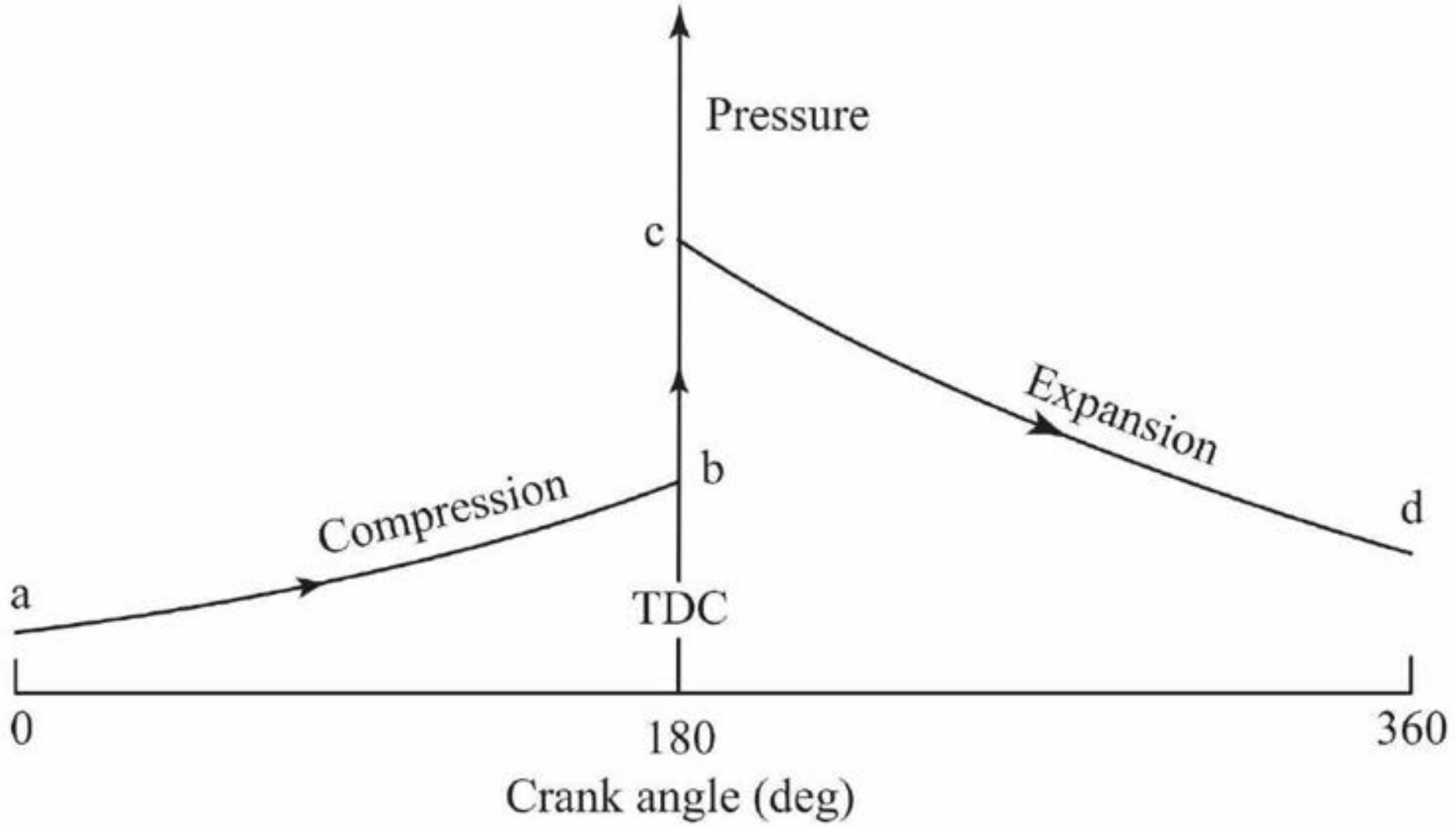

Fig. Theoretical p-θ diagram

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Combustion can be broadly classified into two types

1. **Normal combustion**
2. **Abnormal combustion**

Stages of combustion in SI Engines:

1. **Ignition lag**
2. **Flame propagation**
3. **After burning**

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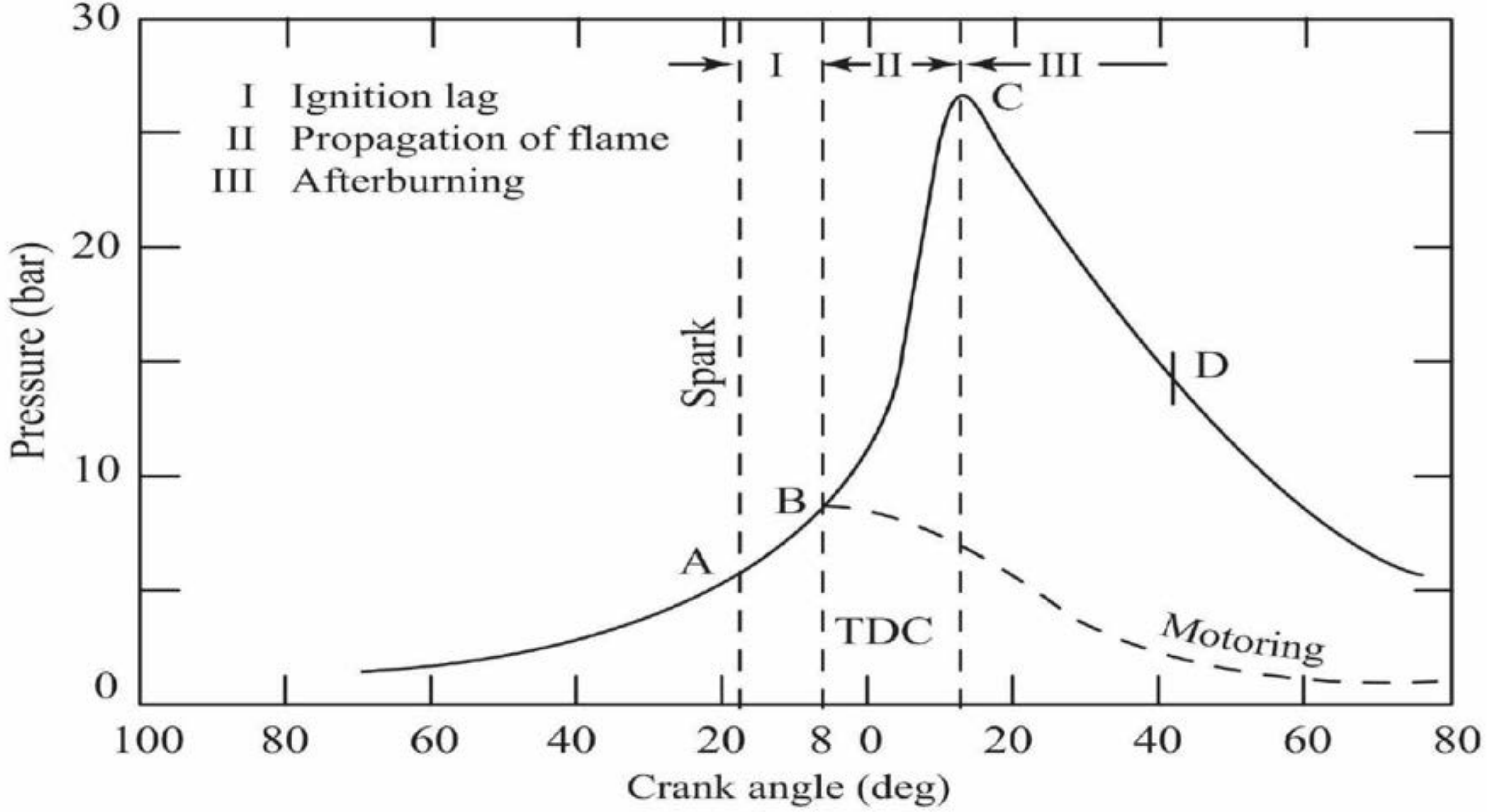

Fig. Stages of combustion in an SI engine

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The *first stage* (A→B) is referred to as the ignition lag or preparation phase in which growth and development of a self propagating nucleus of flame takes place. This is a chemical process depending upon both temperature and pressure, the nature of the fuel and the proportion of the exhaust residual gas. Further, it also depends upon the relationship between the temperature and the rate of reaction.

The *second stage* (B→C) is a physical one and it is concerned with the spread of the flame throughout the combustion chamber. The starting point of the second stage is where the first measurable rise of pressure is seen on the indicator diagram i.e., the point where the line of combustion departs from the compression line (point B). This can be seen from the deviation from the motoring curve.

During the second stage the flame propagates practically at a constant velocity. Heat transfer to the cylinder wall is low, because only a small part of the burning mixture comes in contact with the cylinder wall during this period. The rate of heat-release depends largely on the turbulence intensity and also on the reaction rate which is dependent on the mixture composition. The rate of pressure rise is proportional to the rate of heat-release because during this stage, the combustion chamber volume remains practically constant (since piston is near the top dead centre).

The starting point of the *third stage* is usually taken as the instant at which the maximum pressure is reached on the indicator diagram (point C). The flame velocity decreases during this stage. The rate of combustion becomes low due to lower flame velocity and reduced flame front surface. Since the expansion stroke starts before this stage of combustion, with the piston moving away from the top dead centre, there can be no pressure rise during this stage.

FLAME FRONT PROPAGATION:

For efficient combustion the rate of propagation of the flame front within the cylinder is quite critical. The two important factors which determine the rate of movement of the flame front across the combustion chamber are the reaction rate and the transposition rate. The reaction rate is the result of a purely chemical combination process in which the flame eats its way into the unburned charge. The transposition rate is due to the physical movement of the flame front relative to the cylinder wall and is also the result of the pressure differential between the burning gases and the unburnt gases in the combustion chamber.

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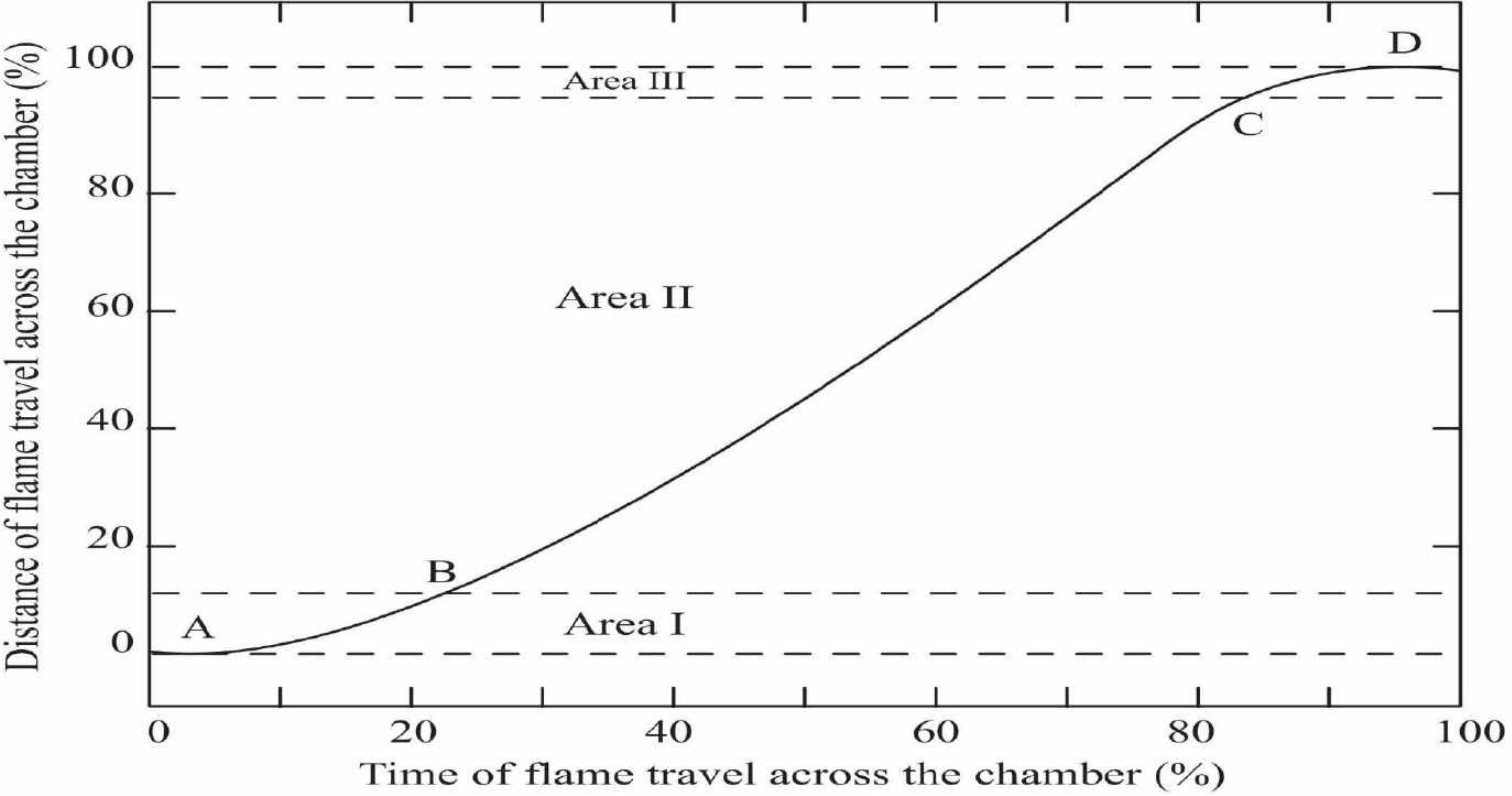

Fig. Details of flame travel

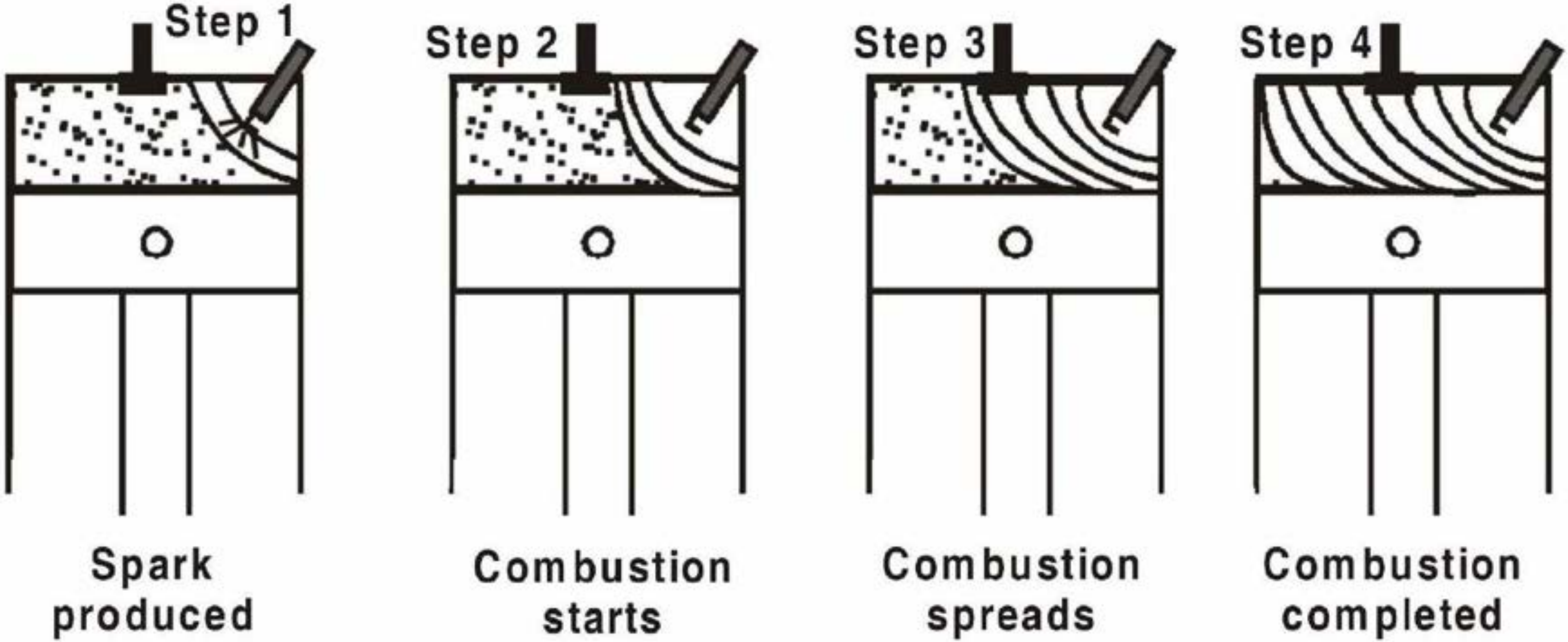

Fig. different steps of normal combustion

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Factors influencing normal combustion in S.I Engines:

1. Induction pressure

2. Engine speed

3. Ignition timing

4. Fuel choice

5. Combustion chamber

6. Compression ratio

7. Mixture strength

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1. Induction pressure

As the pressure falls, delay period increases, and the ignition must be earlier at low pressures.

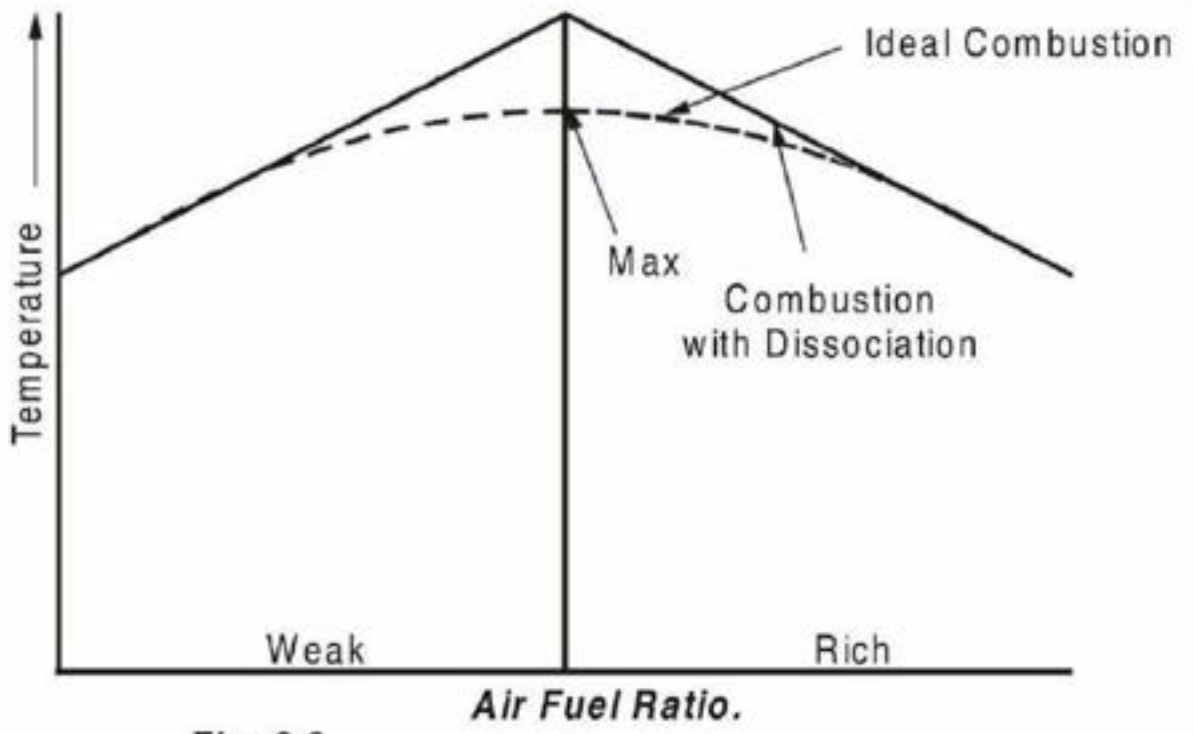

Fig. 2.3.

2. Engine speed

When the engine speed increases, the delay period time needs more crank angle and ignition should take place earlier.

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3. Ignition timing

If the ignition takes place too early, then the peak pressure will occur early and work transfer falls. If the ignition takes place too late, then peak pressure will be low and the work transfer falls.

4. Fuel choice

The calorific value and enthalpy of vaporisation will affect the temperature achieved. The induction period of the fuel will affect the delay period.

5. Combustion chamber

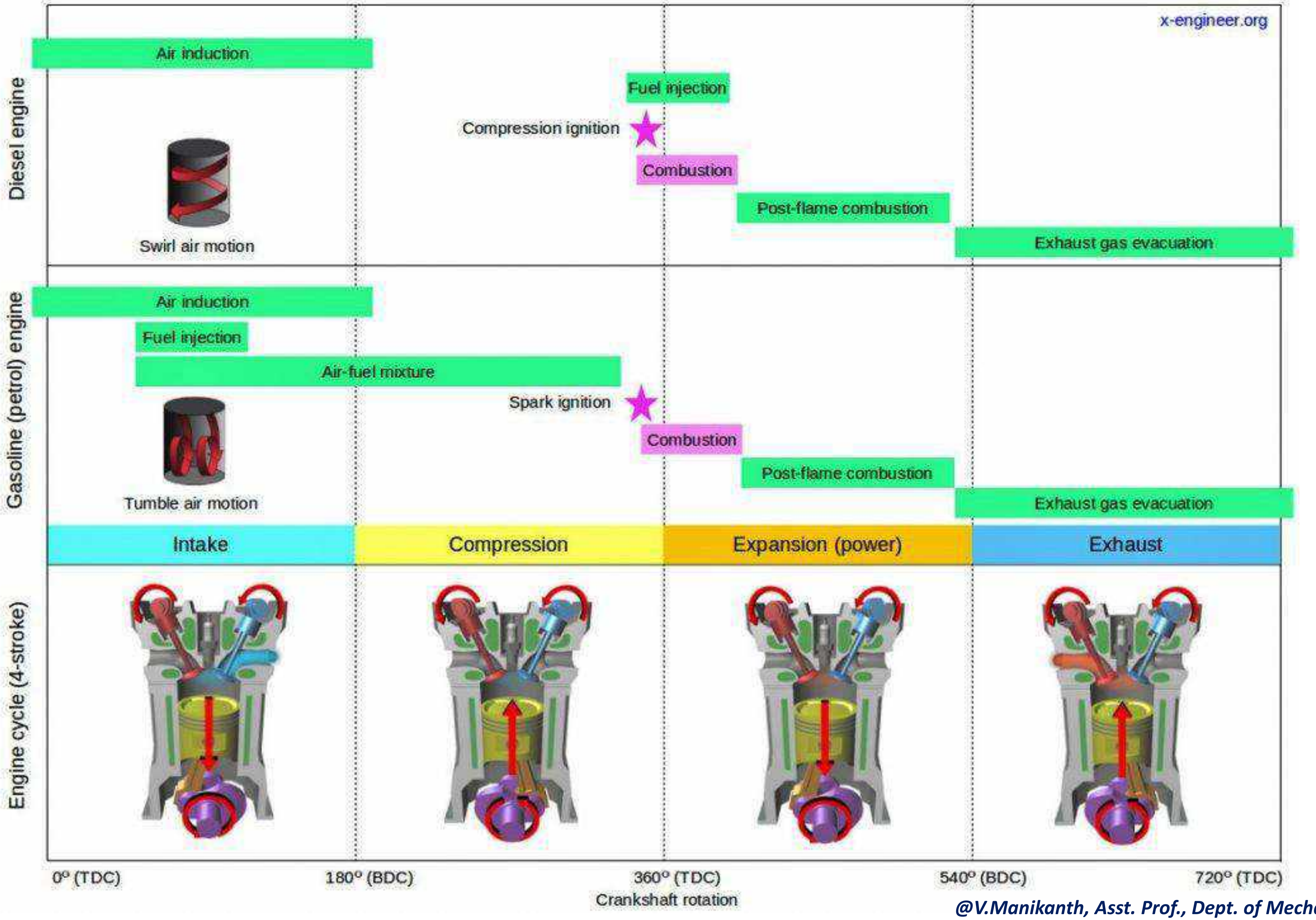
The combustion chamber should be designed to give shorter flame path to avoid knocking and it should give proper turbulence.

6. Compression ratio

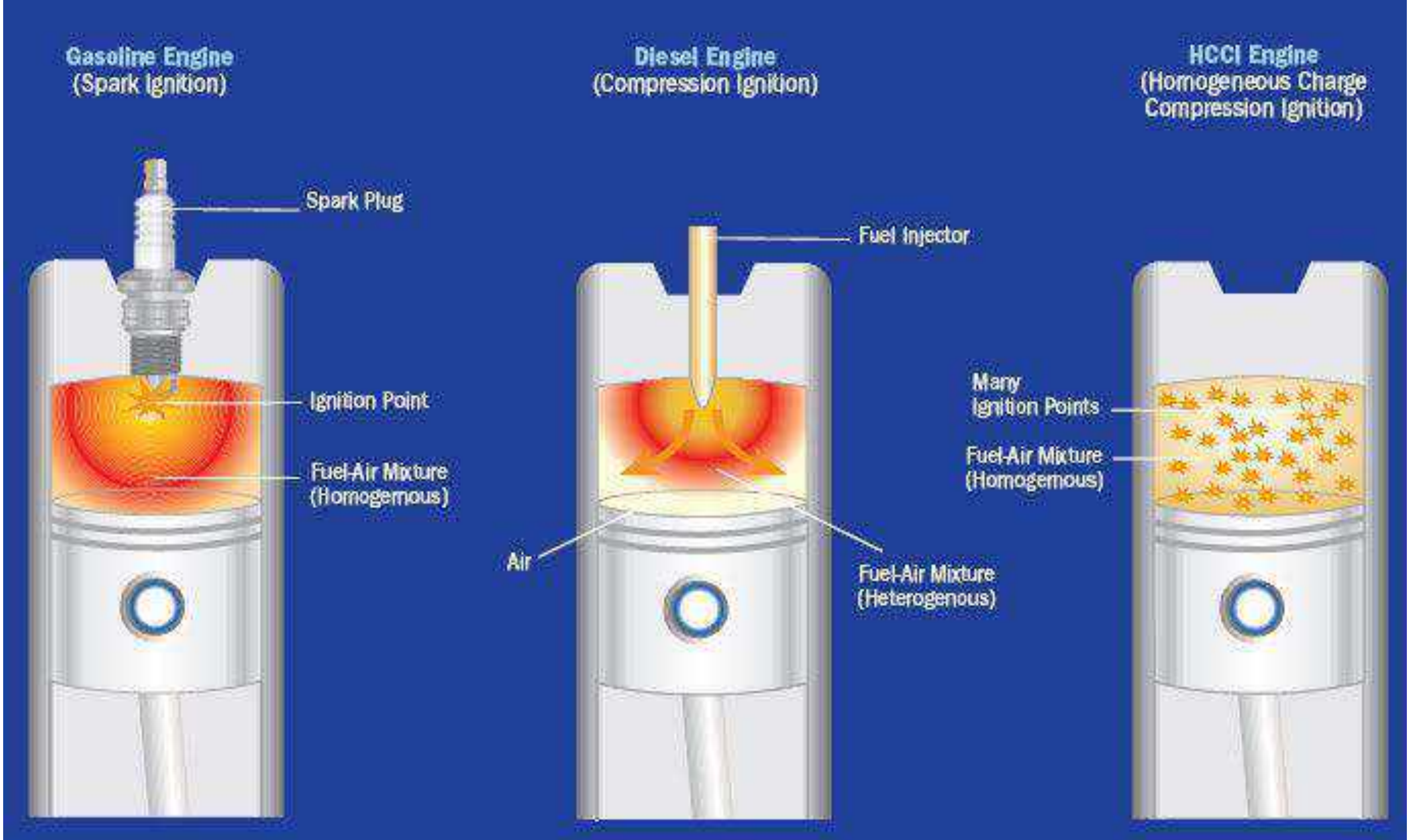
When the compression ratio increases, it increases the maximum pressure and the work transfer.

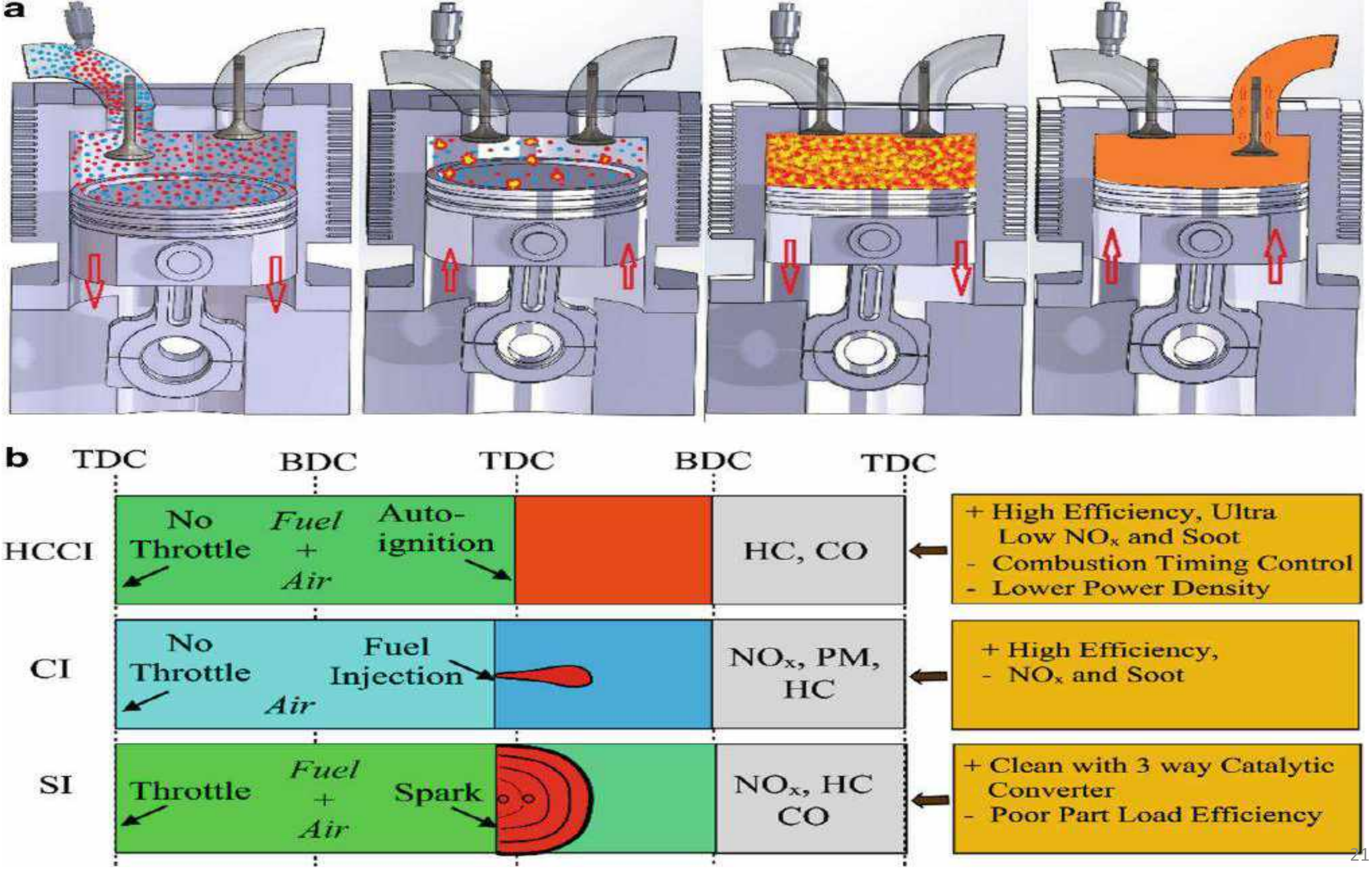
7. Mixture strength

The rich mixture is necessary for producing the maximum work transfer.



x-engineer.org
Diesel engine
Air induction
Fuel injection
Compression ignition
Combustion
Post-flame combustion
Exhaust gas evacuation
Swirl air motion
Gasoline (petrol) engine
Air induction
Fuel injection
Air-fuel mixture
Spark ignition
Combustion
Post-flame combustion
Exhaust gas evacuation
Tumble air motion
Intake
Compression
Expansion (power)
Exhaust
Engine cycle (4-stroke)
0° (TDC)
180° (BDC)
360° (TDC)
540° (BDC)
720° (TDC)
Crankshaft rotation


Gasoline Engine
(Spark Ignition)
Diesel Engine
(Compression Ignition)
HCCI Engine
(Homogeneous Charge
Compression Ignition)
Spark Plug
Fuel Injector
Ignition Point
Fuel-Air Mixture
(Homogemous)
Many
Ignition Points
Fuel-Air Mixture
(Homogemous)
Air
Fuel-Air Mixture
(Heterogenous)


a
b
TDC
BDC
TDC
BDC
TDC
HCCI
No Throttle
Fuel + Air
Auto-ignition
HC, CO
+ High Efficiency, Ultra Low NO_x and Soot
- Combustion Timing Control
- Lower Power Density
CI
No Throttle
Air
Fuel Injection
NO_x, PM, HC
+ High Efficiency,
- NO_x and Soot
SI
Throttle
Fuel + Air
Spark
NO_x, HC CO
+ Clean with 3 way Catalytic Converter
- Poor Part Load Efficiency

FACTORS INFLUENCING THE FLAME SPEED:

1. Turbulence
2. Fuel-Air Ratio
3. Temperature and Pressure
4. Compression Ratio
5. Engine Output
6. Engine Speed
7. Engine Size

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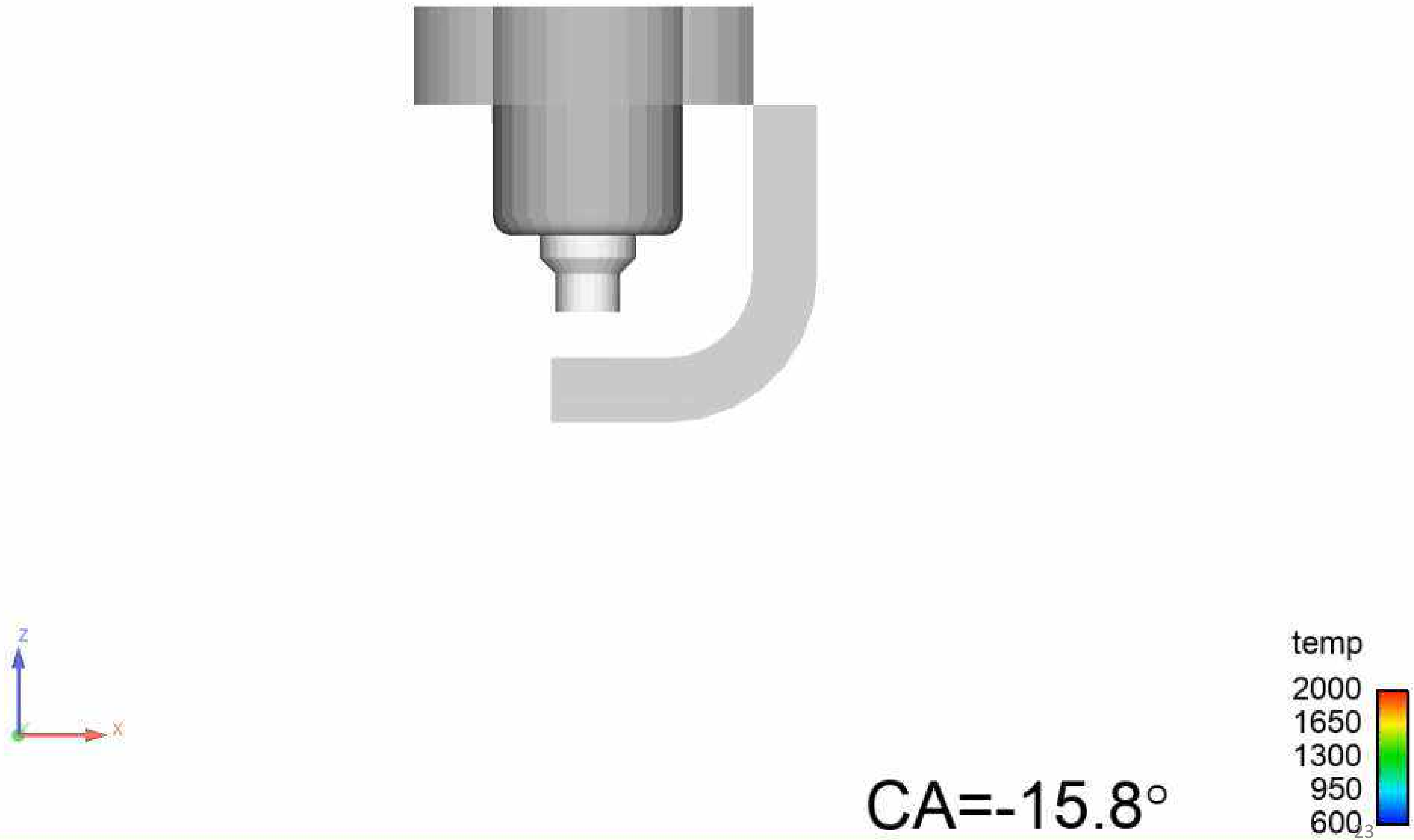
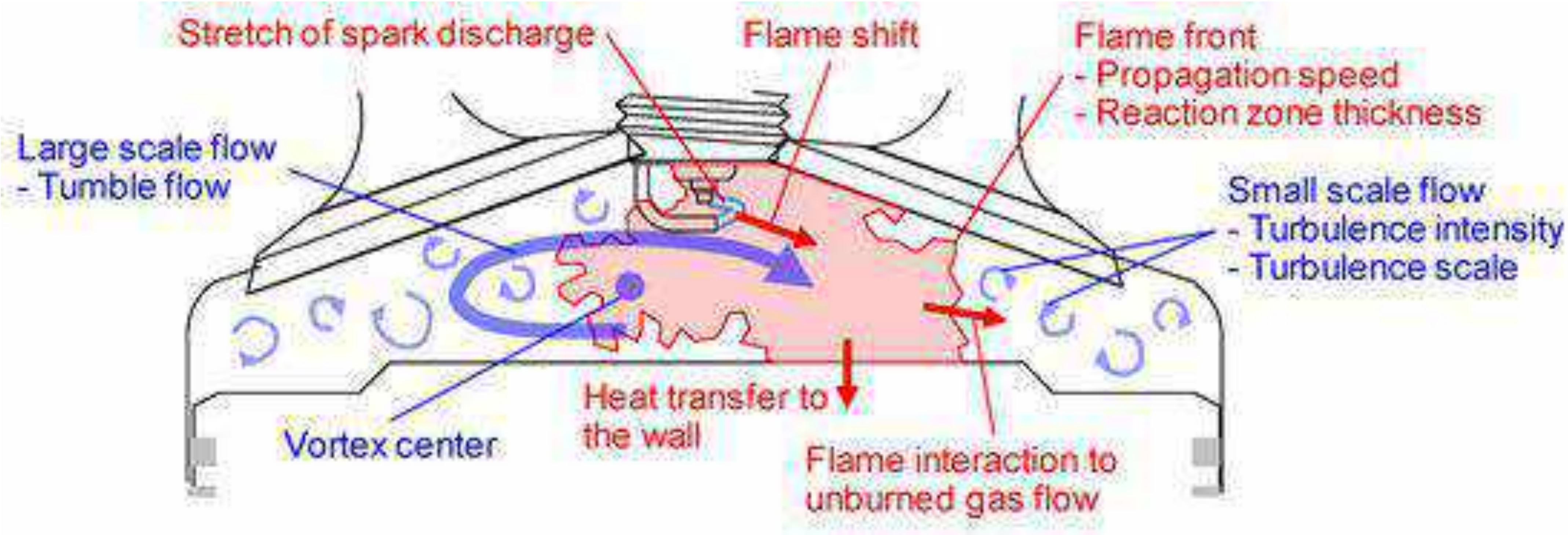

z
x
CA=-15.8°
temp
2000
1650
1300
950
600

The study of factors which affect the velocity of flame propagation is important since the flame velocity influences the rate of pressure rise in the cylinder and is related to certain types of abnormal combustion that occur t in spark ignition engines. There are several factors which affect the flame speed, to a varying degree, the most important being the turbulence and the fuel-air ratio.



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1. Turbulence: The flame speed is quite low in non-turbulent mixtures and increases with increasing turbulence. This is mainly due to the additional physical intermingling of the burning and unburned particles at the flame front which expedites reaction by increasing the rate of contact.

The turbulence in the incoming mixture is generated during the admission of fuel-air mixture through comparatively narrow sections of the intake pipe, valve openings etc., in the suction stroke.

Turbulence which is supposed to consist of many minute swirls appears to increase the rate of reaction and produce a higher flame speed than that made up of larger and fewer swirls.

A suitable design of the combustion chamber which involves the geometry of cylinder head and piston crown increases the turbulence during the compression stroke.

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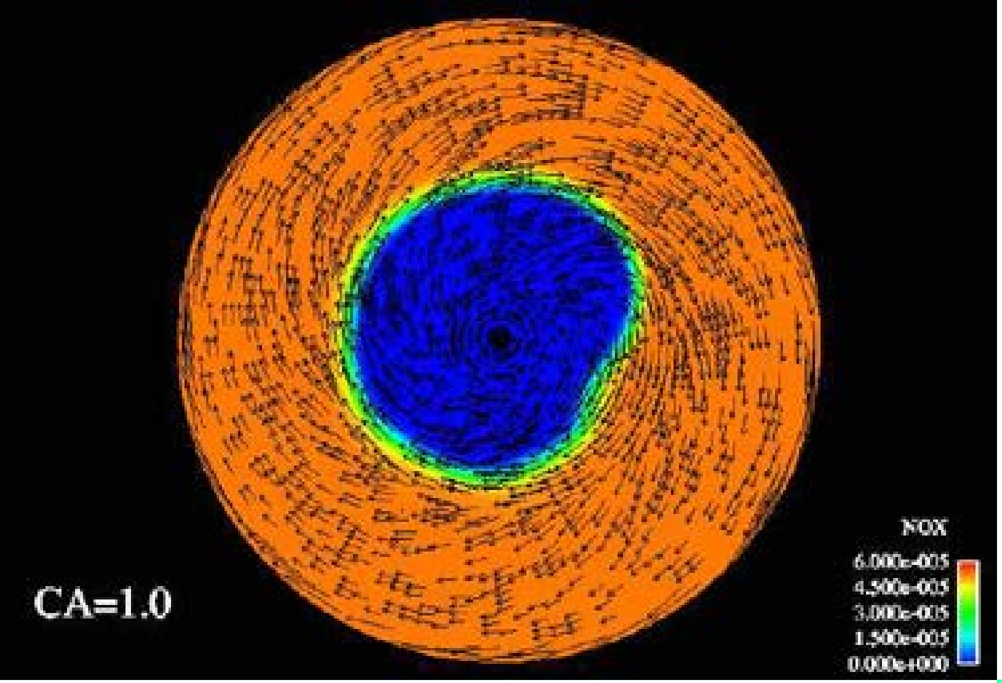
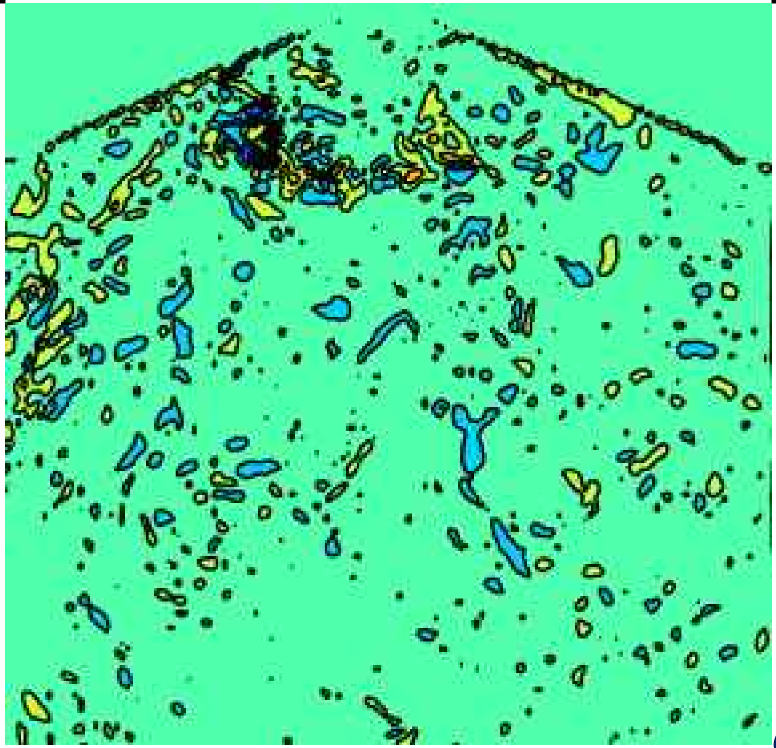
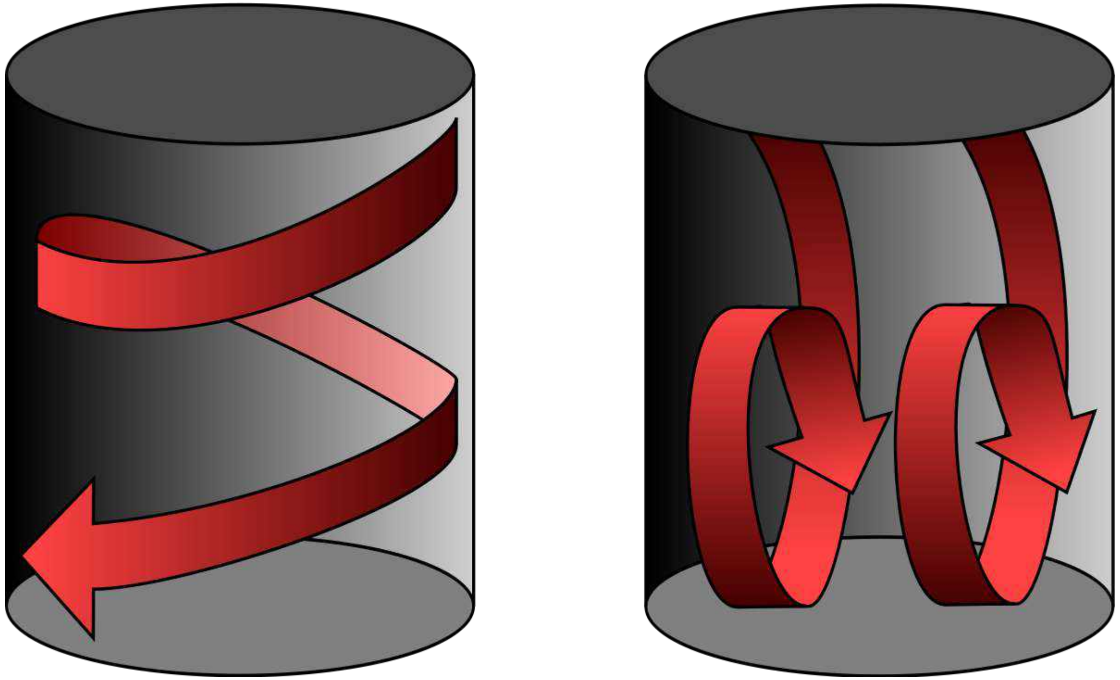
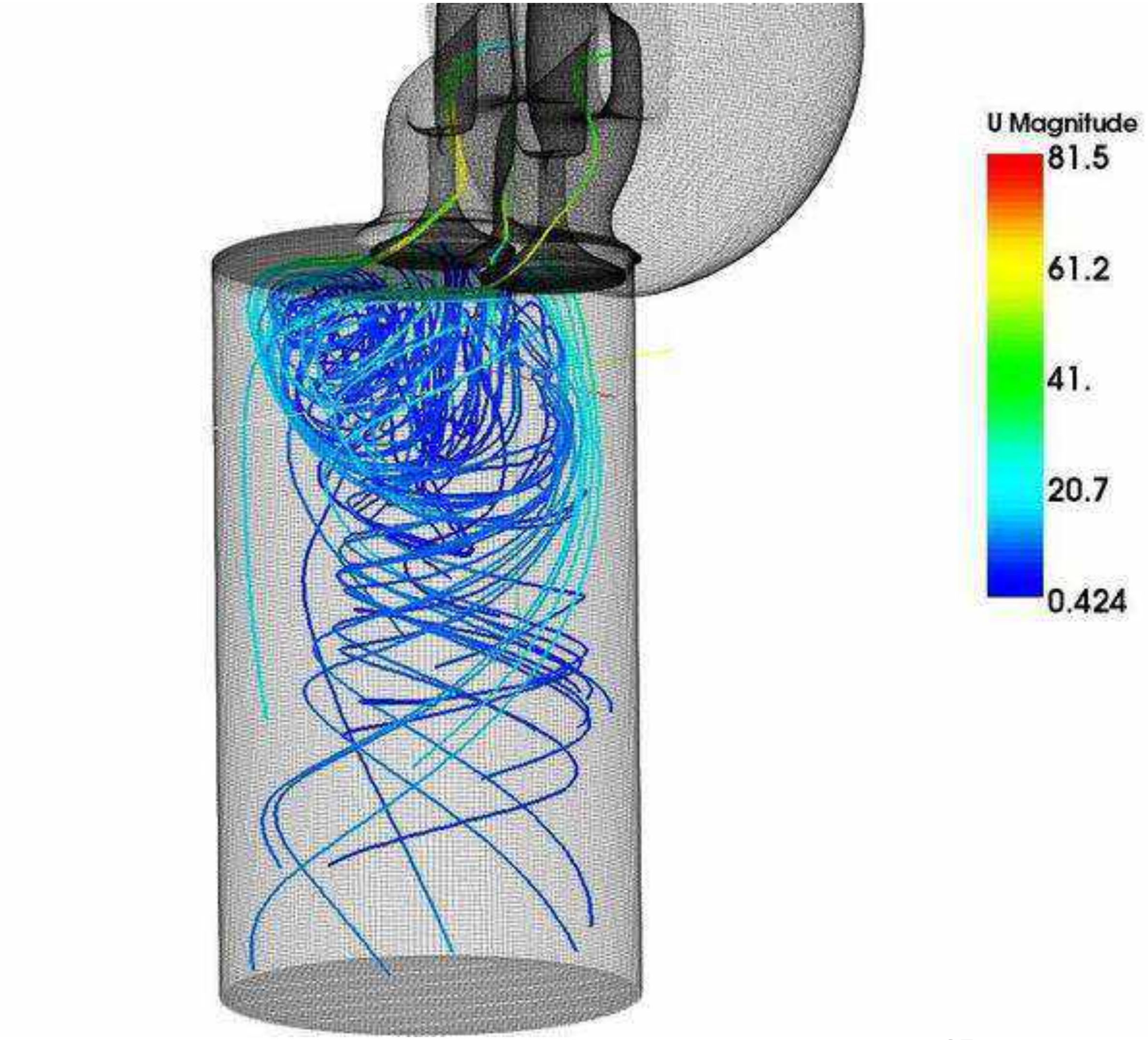

NOX
6.000e-005
4.500e-005
3.000e-005
1.500e-005
0.000e+000
CA=1.0



Swirl Tumble





It also accelerates the chemical reaction by intimate mixing of fuel and oxygen so that spark advance may be reduced. This helps in burning lean mixtures also. The increase of flame speed due to turbulence reduces the combustion duration and hence minimizes the tendency of abnormal combustion

Disadvantages of Turbulence:

1. Generally, turbulence increases the heat flow to the cylinder wall.
2. Excessive turbulence may extinguish the flame resulting in rough and noisy operation of the engine.

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2.Fuel-Air Ratio: The fuel-air ratio has a very significant influence on the flame speed. The highest flame velocities (minimum time for complete combustion) are obtained with somewhat richer mixture (point A) as shown in Fig. which shows the effect of mixture strength on the rate of burning as indicated by the time taken for complete burning in a given engine. When

When the mixture is made leaner or richer (see point A in Fig.) the flame speed decreases. Less thermal energy is released in the case of lean mixtures resulting in lower flame temperature. Very rich mixtures lead to incomplete combustion which results again in the release of less thermal energy.

3.Temperature and Pressure: Flame speed increases with an increase in intake temperature and pressure. A higher initial pressure and temperature may help to form a better homogeneous air-vapour mixture which helps in increasing the flame speed. This is possible because of an overall increase in the density of the charge.

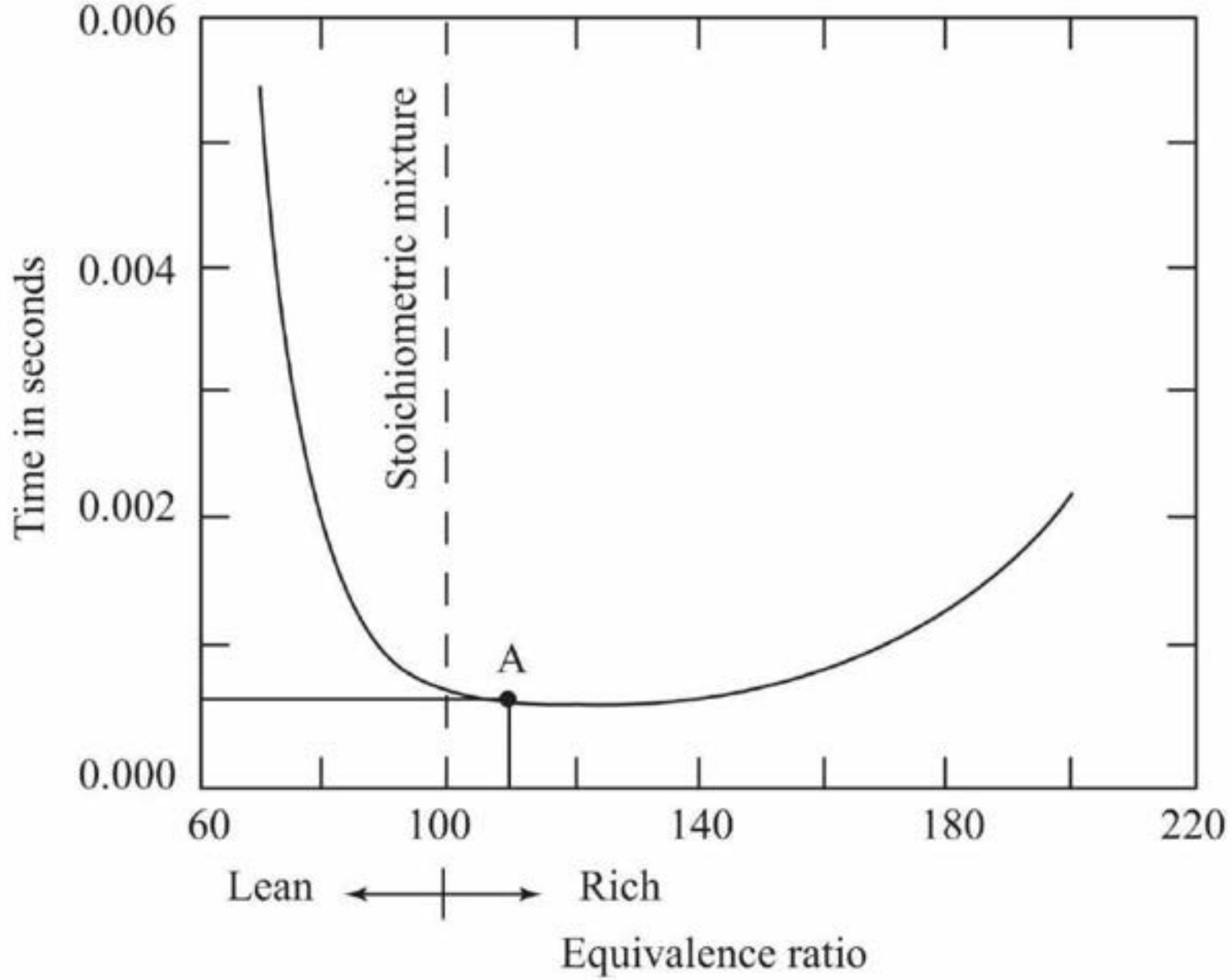

Fig. Effect of mixture strength on the rate of burning

4.Compression Ratio:

A higher compression ratio increases the pressure and temperature of the working mixture which reduce the initial preparation phase of combustion and hence less ignition advance is needed. High pressures and temperatures of the compressed mixture also speed up the second phase of combustion. Increased compression ratio reduces the clearance volume and therefore increases the density of the cylinder gases during burning. This increases the peak pressure and temperature and the total combustion duration is reduced. Thus engines having higher compression ratios have higher flame speeds.

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5.Engine Output:

The cycle pressure increases when the engine output is increased. With the increased throttle opening the cylinder gets filled to a higher density. This results in increased flame speed. When the output is decreased by throttling, the initial and final compression pressures decrease and the dilution of the working mixture increases. The smooth development of self-propagating nucleus of flame becomes unsteady and difficult. The main disadvantages of SI engines are the poor combustion at low loads and the necessity of mixture enrichment (ϕ between 1.2 to 1.3) which causes wastage of fuel and discharge of unburnt hydrocarbon and the products of incomplete combustion like carbon monoxide etc. in the atmosphere.

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6.Engine Speed:

The flame speed increases almost linearly with engine speed since the increase in engine speed increases the turbulence inside the cylinder. The time required for the flame to traverse the combustion space would be halved, if the engine speed is doubled. Double the engine speed and hence half the original time would give the same number of crank degrees for flame propagation. The crank angle required for the flame propagation during the entire phase of combustion, will remain nearly constant at all speeds.

7.Engine Size:

The size of the engine does not have much effect on the rate of flame propagation. In large engines the time required for complete combustion is more because the flame has to travel a longer distance. This requires increased crank angle duration during the combustion. This is one of the reasons why large sized engines are designed to operate at low speeds.

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RATE OF PRESSURE RISE:

The rate of pressure rise in an engine combustion chamber exerts a considerable influence on the peak pressure developed, the power produced and the smoothness with which the forces are transmitted to the piston. The rate of pressure rise is mainly dependent upon the mass rate of combustion of mixture in the cylinder.

it may be noted that higher rate of combustion results in higher rate of pressure rise producing higher peak pressures at a point closer to TDC. This generally is a desirable feature because higher peak pressures closer to T DC produce a greater force acting through a large part of the power stroke and hence, increase the power output of the engine. The higher rate of pressure rise causes rough running of the engine because of vibrations produced in the crankshaft rotation. It also tends to promote an undesirable occurrence known as knocking.

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A compromise between these opposing factors is accomplished by designing and operating the engine in such a manner that approximately one-half of the maximum pressure is reached by the time the piston reaches TDC. This results in the peak pressure being reasonably close to the beginning of the power stroke, yet maintaining smooth engine operation.

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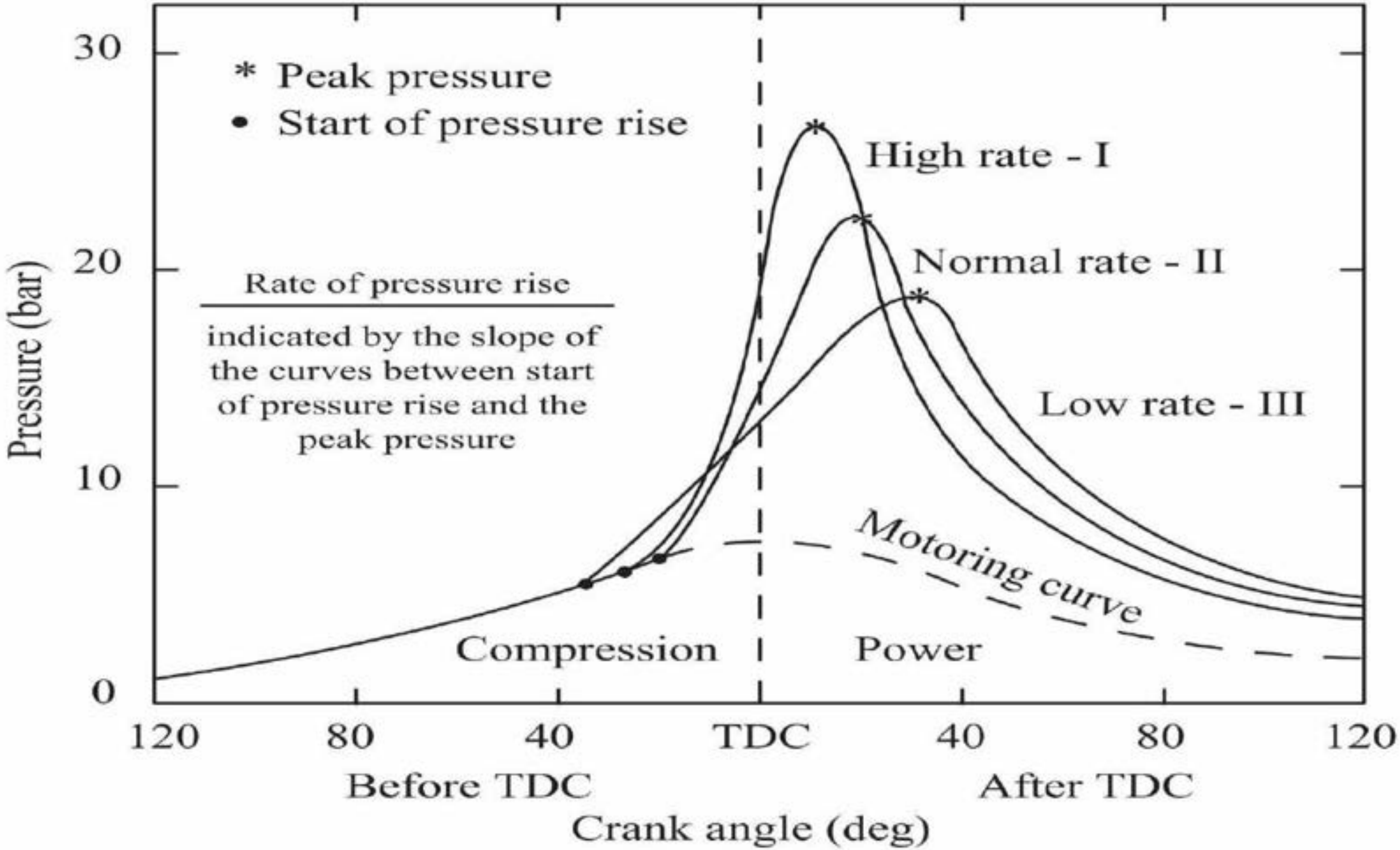

Fig. Illustrations of various combustion rates

ABNORMAL COMBUSTION IN SI ENGINES:

In normal combustion, the flame initiated by the spark travels across the combustion chamber in a fairly uniform manner. Under certain operating conditions the combustion deviates from its normal course leading to loss of performance and possible damage to the engine. This type of combustion may be termed as an abnormal combustion or knocking combustion. The consequences of this abnormal combustion process are the loss of power, recurring pre--ignition and mechanical damage to the engine.

The abnormal combustion deviates from the normal behavior resulting in loss of performance and physical damage to the engine.

There are two types of Abnormal combustion.

1. Pre-ignition

2. Knocking (or) Detonation (or) Pinking

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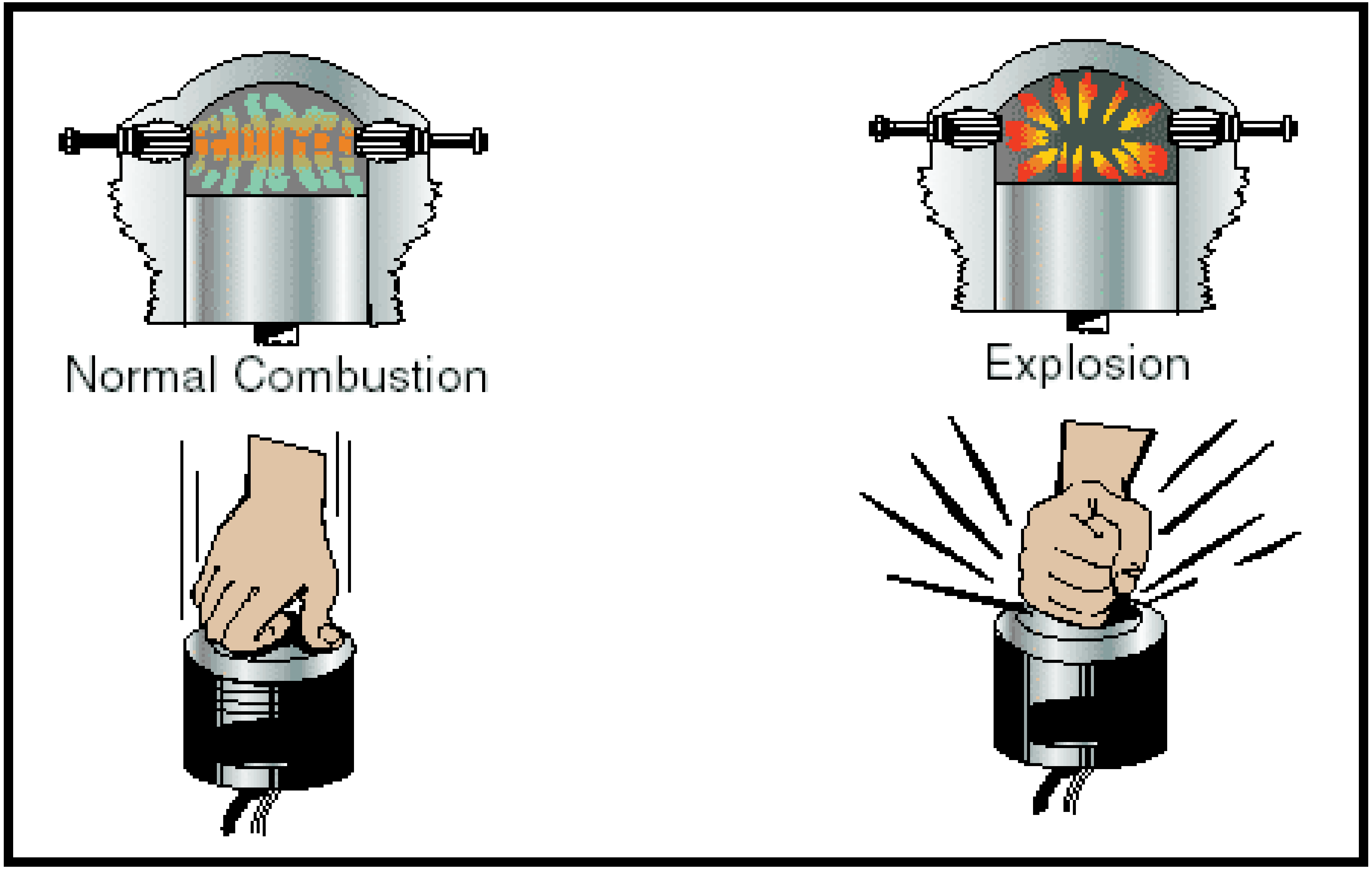
Normal Combustion
Explosion

1. Pre-ignition:

High temperature carbon deposits formed inside the combustion chamber ignite the air-fuel mixture before normal ignition occurs by spark plug. This ignition due to hot carbon deposits is called pre-ignition. After some time of Pre-ignition, the normal ignition starts and both the flames get collided.

If Pre-ignition occurs much early in the compression stroke, the work to compress the charge will be increased. So the net power output will be reduced. Also this may cause crank failure due to high load to compress charge. Pre-ignition causes very high pressure and temperature. It causes the detonation. Pre-ignition is considered as abnormal combustion.

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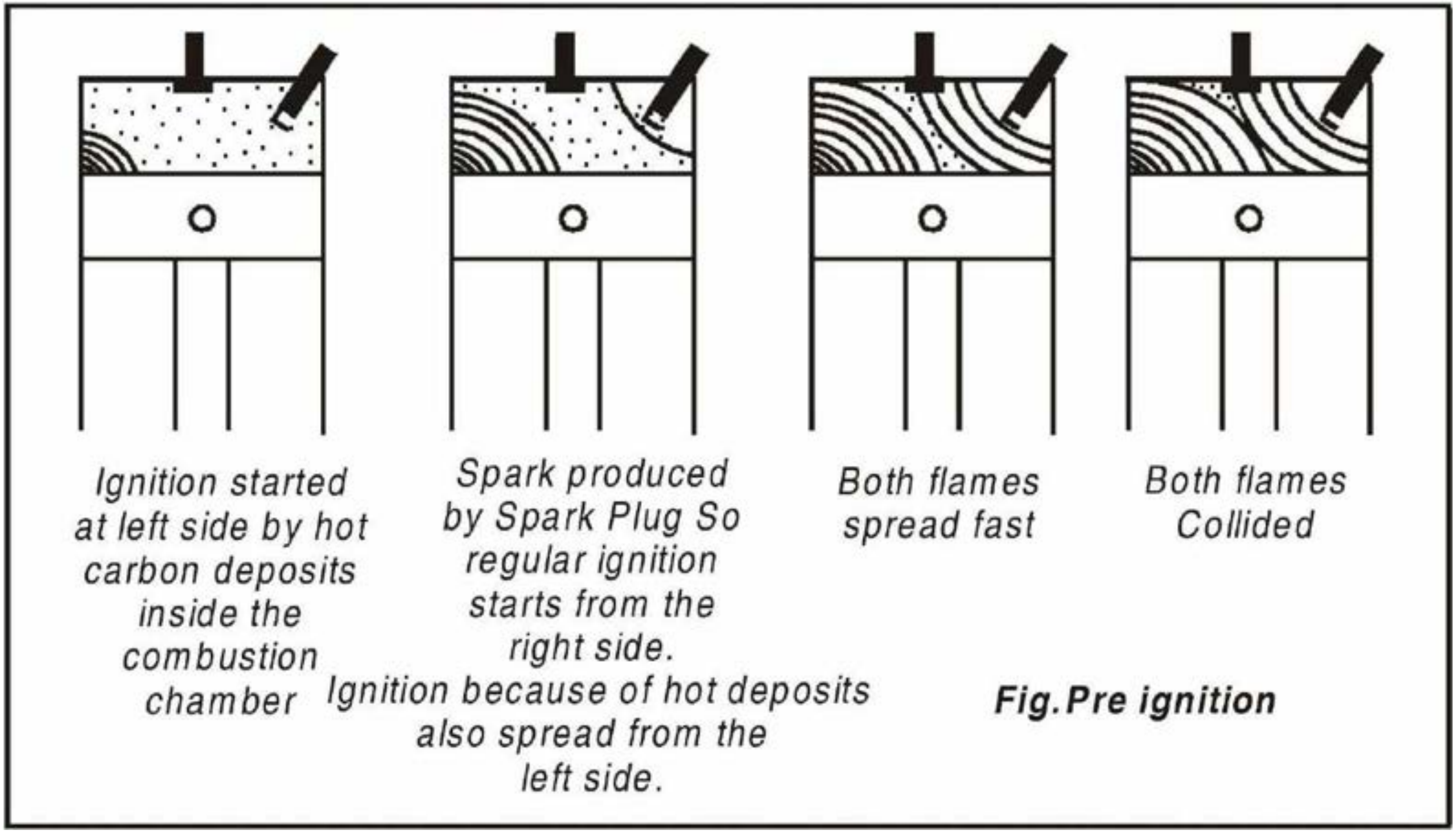

Fig.Pre ignition

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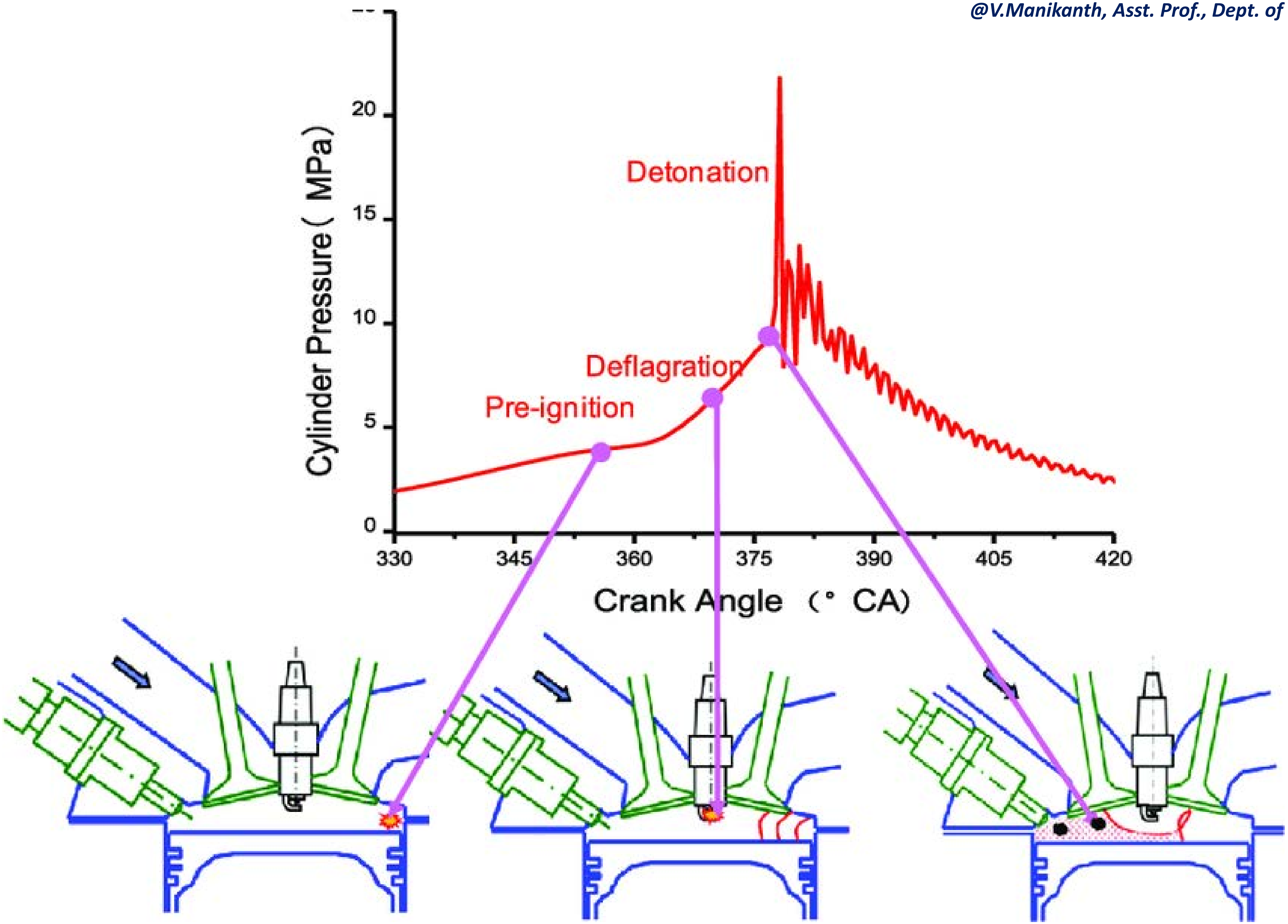
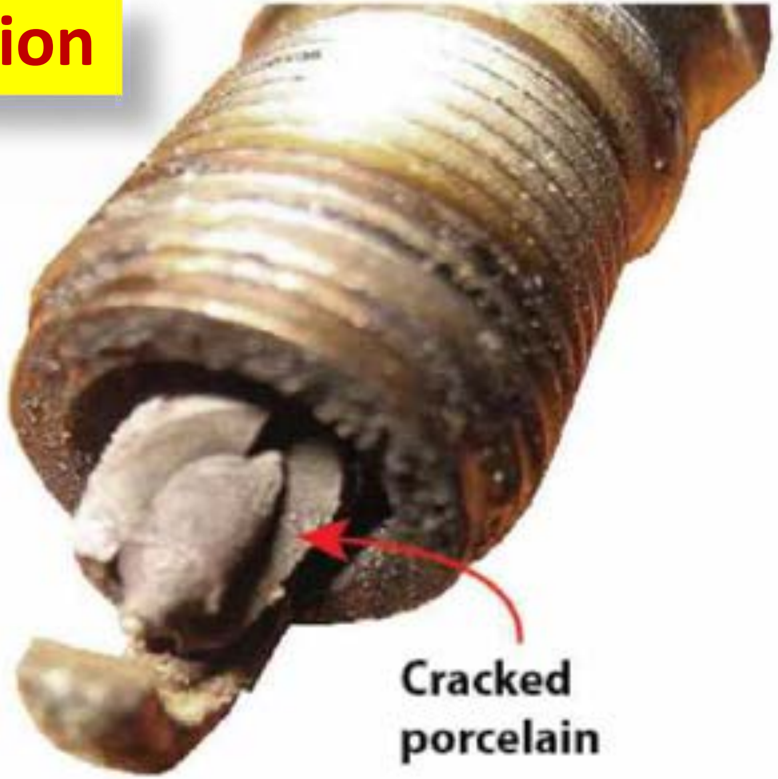

Detonation
Deflagration
Pre-ignition
Cylinder Pressure（ MPa）
Crank Angle （° CA）
0
5
10
15
20
330
345
360
375
390
405
420

Causes of pre ignition



Tests for pre-ignition:

The standard test for pre-ignition is to shut off the ignition. If the engine still fires, it is assumed that pre-ignition was taking place when the ignition was on. Experience shows that this assumption is not always valid. Sudden loss of power with no evidence of mechanical malfunctioning is fairly good evidence of pre-ignition. The best proof of pre-ignition is the appearance of an indicator card taken with a high speed indicator of the balanced-pressure type.

2. Knocking (or) Detonation (or) Pinking :

In a spark-ignition engine combustion which is initiated between the spark plug electrodes spreads across the mixture. A definite flame front which separates the fresh mixture from the products of combustion travels from the spark plug to the other end of the combustion chamber. Heat-release due to combustion increases the temperature and consequently the pressure, of the burned part of the mixture above those of the unburned mixture. In order to effect pressure equalization the burned part of the mixture will expand, and compress the unburned mixture adiabatically thereby increasing its pressure and temperature. This process continues as the flame front advances through the mixture and the temperature and pressure of the unburned mixture are increased further.

If the temperature of the unburnt mixture exceeds the self-ignition temperature of the fuel and remains at or above this temperature during the period of preflame reactions (ignition lag), spontaneous ignition or auto ignition occurs at various pin-point locations. This phenomenon is called '*knocking*'. The process of auto ignition leads towards engine knock.

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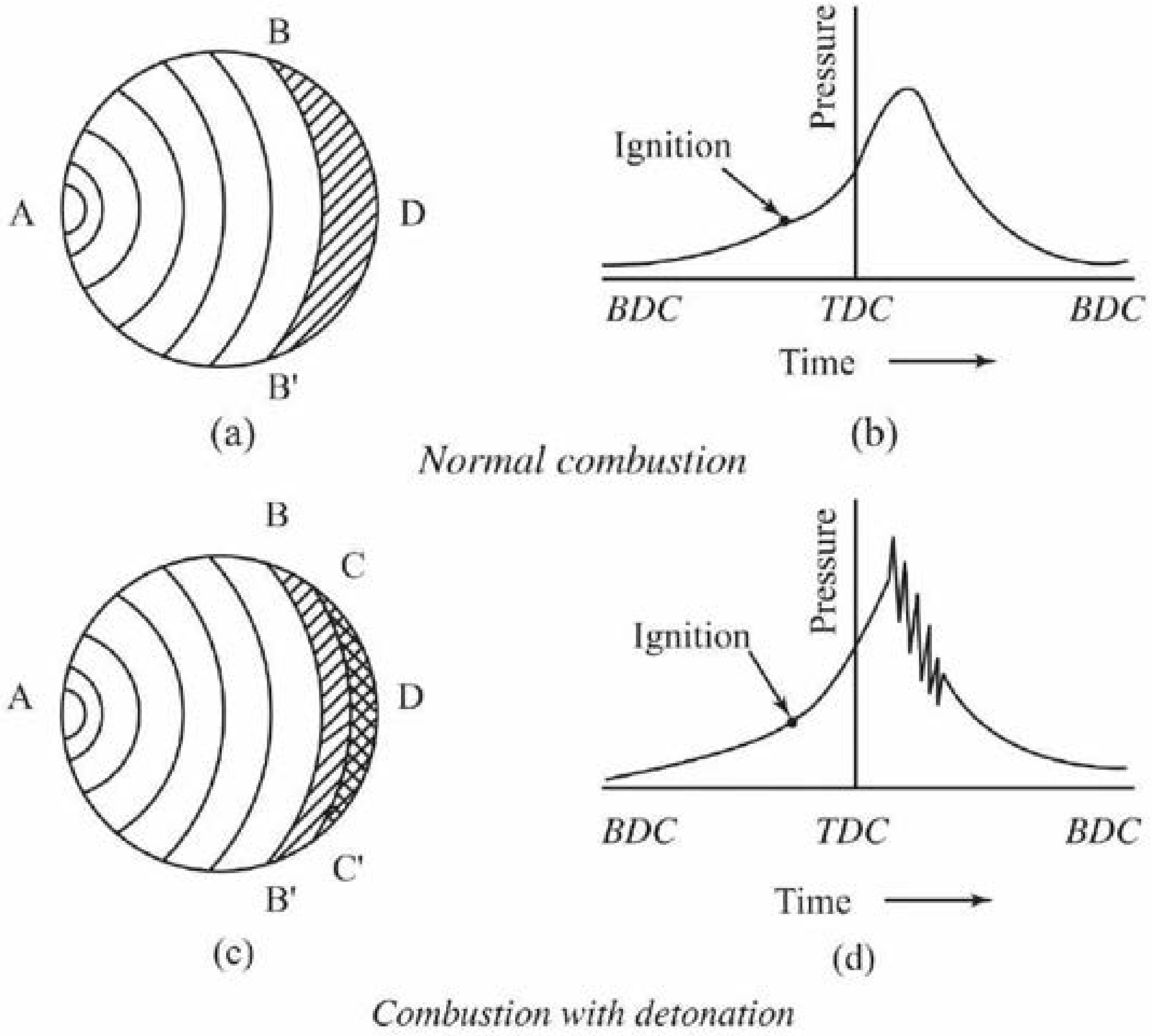

Fig. Normal and abnormal combustion

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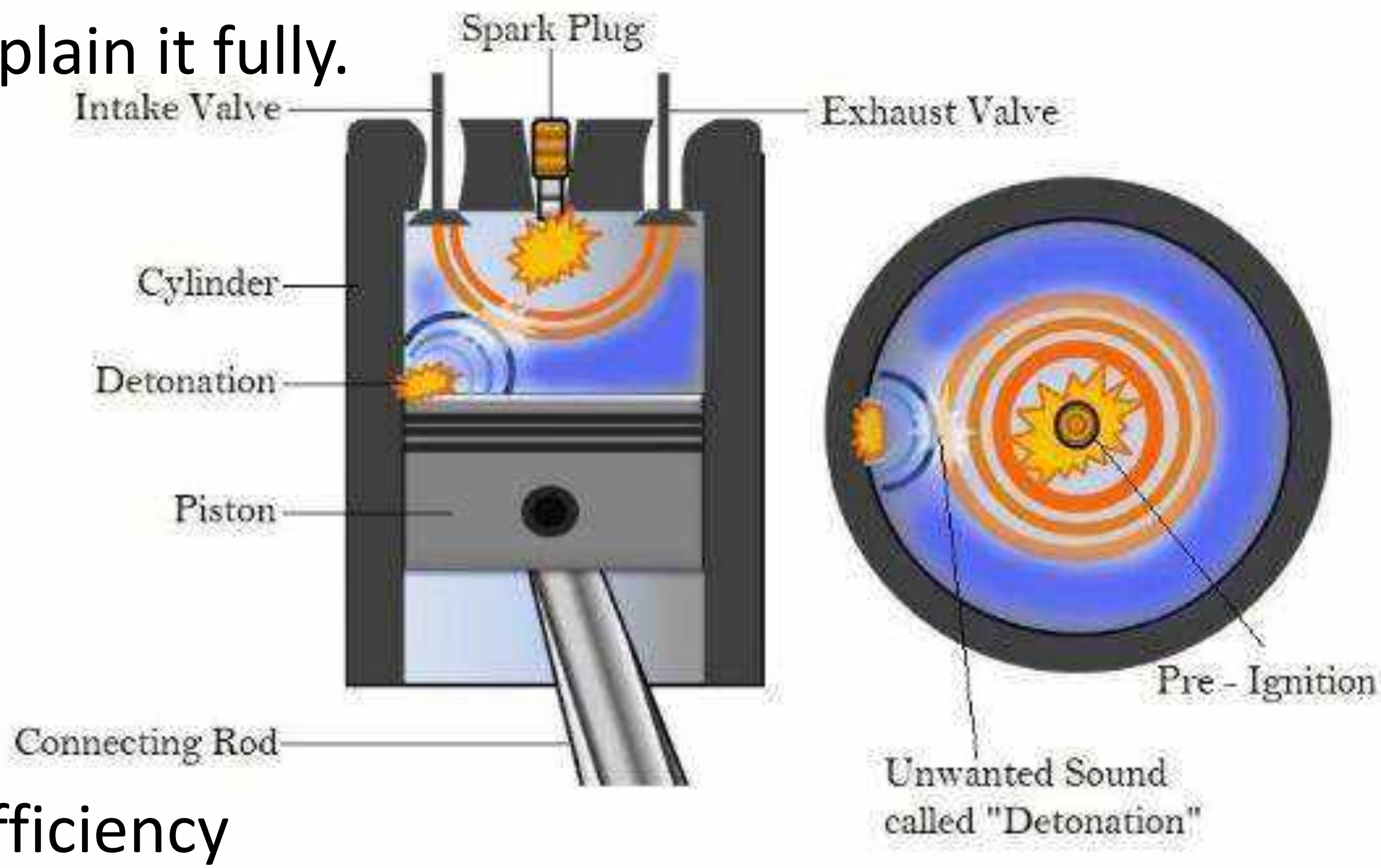
In fact knocking or detonation is a complex phenomenon and no single explanation may be sufficient to explain it fully.



Effects of detonation :

1. Noise and roughness
2. Mechanical damage
3. Carbon deposits
4. Increase in heat transfer
5. Decrease in power output and efficiency
6. Pre-ignition.

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Control of detonation :

The detonation can be controlled or even stopped by the following methods :

1. Increasing engine r.p.m.
2. Retarding spark.
3. Reducing pressure in the inlet manifold by throttling.
4. Making the ratio too lean or too rich, preferably latter.
5. Water injection. Water injection increases the delay period as well as reduces the flame temperature.
6. Use of high octane fuel can eliminate detonation. High octane fuels are obtained by adding additives known as dopes (such as tetra-ethyl of lead, benzol, xylene etc.), to petrol.

FACTORS AFFECTING KNOCK

The *likelihood of knock is increased by any reduction in the induction period of combustion and any reduction in the progressive explosion flame velocity*. Particular factors are listed below :

1. **Fuel choice :**	A low self ignition temperature promotes knock.
2. **Induction pressure :**	Increase of pressure decreases the self ignition temperature and the induction period. Knock will tend to occur *at full throttle*.
3. **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.
4. **Ignition timing :**	*Advanced ignition timing* increases peak pressures and *promotes* knock.
5. **Mixture strength :**	*Optimum mixture strength* gives high pressures and *promotes* knock.
6. **Compression ratio :**	*High compression ratios* increase the cylinder pressures and *promote* knock.
7. **Combustion chamber design :**	Poor design gives long flame paths, poor turbulence and insufficient cooling all of which *promote* knock.
8. **Cylinder cooling :**	*Poor cooling* raises the mixture temperature and *promotes* knock.

Summary of variables affecting knock in an SI engine

Increase in variable	Major effect on unburned reduce charge	Action to be taken to knocking	Can operator usually control?
Compression ratio	Increases temperature & pressure	Reduce	No
Mass of charge inducted	Increases pressure	Reduce	Yes
Inlet temperature	Increases temperature	Reduce	In some cases
Chamber wall temperature	Increases temperature	Reduce	Not ordinarily
Spark advance	Increases temperature & pressure	Retard	In some cases
A/F ratio	Increases temperature & pressure	Make very rich	In some cases
Turbulence	Decreases time factor	Increase	Somewhat (through engine speed)
Engine speed	Decreases time factor	Increase	Yes
Distance of flame travel	Increases time factor	Reduce	No

COMBUSTION CHAMBER DESIGN IN S.I. ENGINES

From the point of view of attaining highest resistance to detonation under a given set of service conditions, and also promote high power output ; high thermal efficiency, and smooth operation, the following would appear to be desirable characteristics for S.I. engines :

1. Short combustion time.
2. Short ratio of flame path to bore.
3. Absence of hot surfaces in the end gas region.
4. Use of squish areas particularly in the end gas region.
5. High velocity through the inlet valve.
6. Large surface to volume ratio for the end gas (to have adequate cooling in the detonation zone).
7. Cooling of hot spots.

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Combustion chambers are usually designed with every possible attempt made to meet the following objectives :

1. To regulate the rate of pressure rise such that the greatest force is applied to the piston as closely after T.D.C. on the 'power stroke' as possible, with a gradual decrease in the force on the piston during the power stroke. The forces must be applied to the piston smoothly, however, thus placing a limit on the rate of the pressure rise, as well as the position of the peak pressure with respect to T.D.C.

2. To prevent the possibility of detonation at all times. To obtain these objectives attempt is made to design the combustion chambers with the

following factors in mind :

(i) To achieve the highest possible flame front velocity through the creation of high turbulence of the minute "*swirl*" type.

(ii) To burn the largest mass of charge as soon as possible after ignition (consistent with a smooth application of force), with progressive reduction in the mass of the charge burnt toward the end of the combustion.

(iii) To reduce the possibility of detonation by :

— Reducing the temperature of the last portion of the charge to burn, through the application of a high surface to volume ratio in that part of the combustion chamber where this portion burns. Such a ratio increases the heat transfer to the combustion chamber walls and thereby tends to reduce the temperature of the final unburned charge.

— Reducing the distance for the flame to travel by centrally locating the spark plug, or in some engines, by using dual spark plugs.

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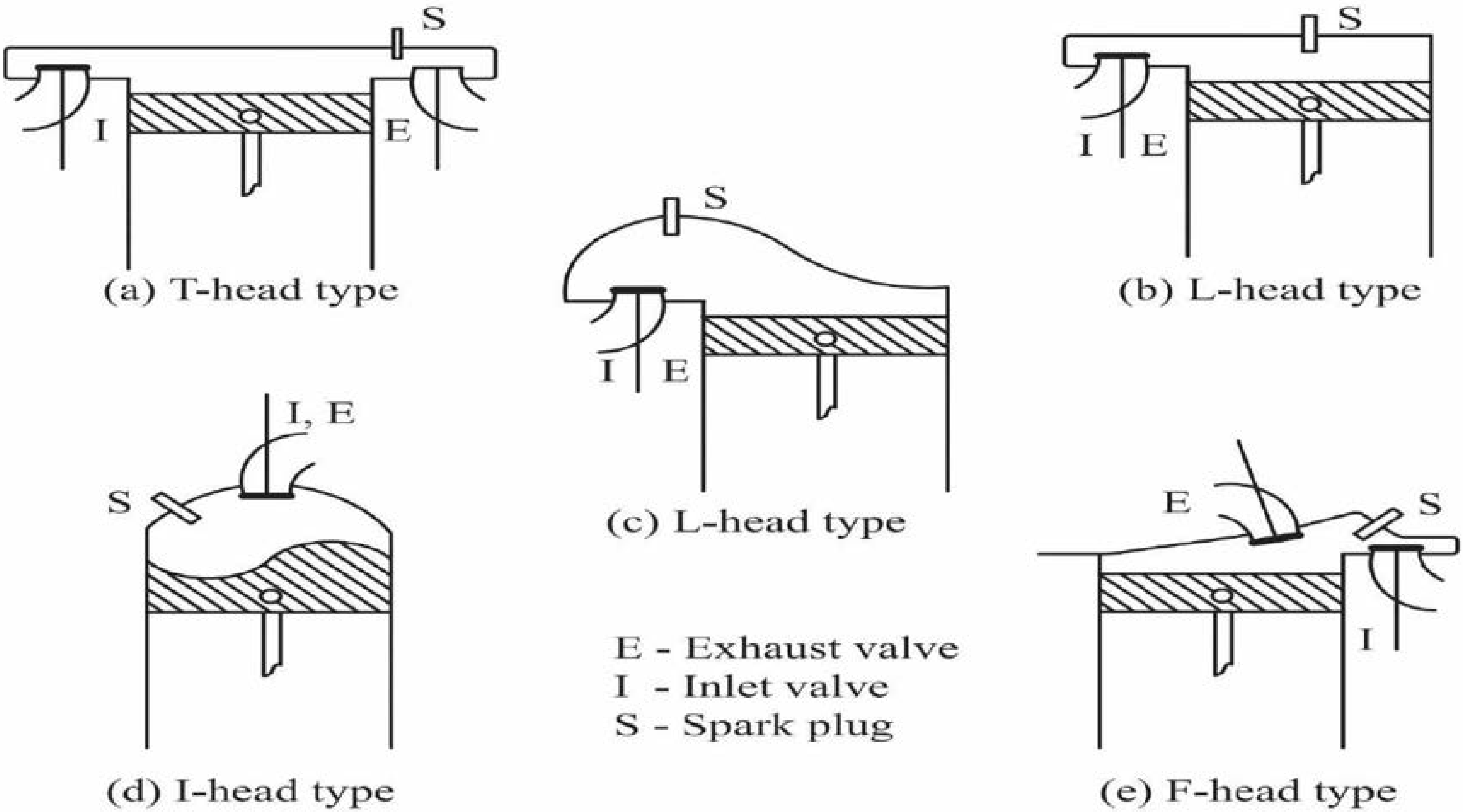

Fig. Examples of typical combustion chamber

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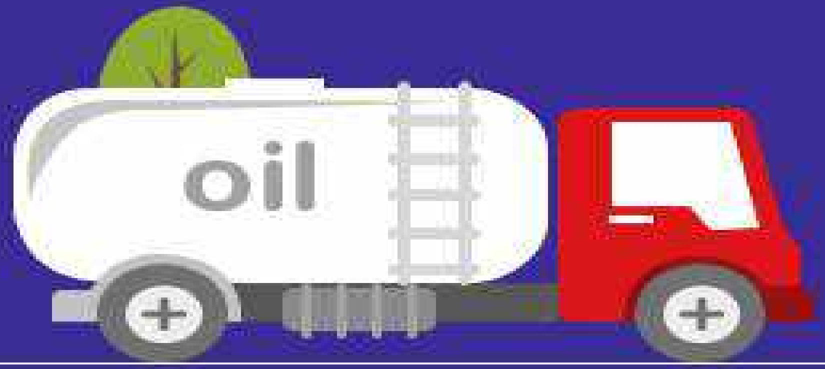
FUEL REQUIREMENTS &
FUEL RATING



The study of fuels for IC engines has been carried out ever since these engines came into existence. The engine converts heat energy which is obtained from the chemical combination of the fuel with the oxygen, into mechanical energy. Since the heat energy is derived from the fuel, a fundamental knowledge of fuels and their characteristics is essential in order to understand the combustion phenomenon. The characteristics of the fuel used have considerable influence on the design, efficiency, output and particularly, the reliability and durability of the engine. Further, the fuel characteristics play a vital role in the atmospheric pollution caused by the engines used in automobiles.

The above families of hydrocarbons exhibit some general characteristics due to their molecular structure which are summarized below:

(i) Normal paraffins exhibit the poorest antiknock quality when used in an SI engine. But the antiknock quality improves with the increasing number of carbon atoms and the compactness of the molecular structure. The aromatics offer the best resistance to knocking in SI Engines.

(ii) For CI engines, the order is reversed i.e., the normal paraffins are the best fuels and aromatics are the least desirable.

(iii) As the number of atoms in the molecular structure increases, the boiling temperature increases. Thus fuels with fewer atoms in the molecule tend to be more volatile.

(iv) The heating value generally increases as the proportion of hydrogen atoms to carbon atoms in the molecule increases due to the higher heating value of hydrogen than carbon. Thus, paraffins have the highest heating value and the aromatics the least.

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IMPORTANT QUALITIES OF ENGINE FUELS

Fuels used in IC engines should possess certain basic qualities which are important for the smooth running of the engines. In this section, the important qualities of fuels for both SI and CI engines are reviewed.

SI Engine Fuels :

Gasoline which is mostly used in the present day SI engines is usually a blend of several low boiling paraffins, naphthenes and aromatics in varying proportions. Some of the important qualities of gasoline are discussed below.

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Important properties of fuel in SI Engine:

The fuel characteristics that are important for the performances of internal combustion engines are

- ❖ Volatility of the fuel.
- ❖ Detonation characteristics.
- ❖ Good thermal properties like heat of combustion and heat of evaporation.
- ❖ Sulphur content.
- ❖ Aromatic content.
- ❖ Cleanliness of fuel.

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Important characteristics of SI Engine fuel:

Every engine is designed for a particular fuel according to its desired qualities. For good performance of SI engine, the fuel used must have the proper characteristics like,

- ❖ It should readily mix with air to make an uniform mixture at inlet.
- ❖ It must be knock resistant.
- ❖ It should not pre-ignite easily.
- ❖ It should not tend to decrease the volumetric efficiency of the engine.
- ❖ Its sulphur content should be low.
- ❖ It must have adequate calorific value.
- ❖ It must have proper viscosity.

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RATING OF FUELS:

Normally fuels are rated for their antiknock qualities. The rating of fuels is done by defining two parameters called Octane number and Cetane number for gasoline and diesel oil respectively.

Rating of SI Engine Fuels:

Resistance to knocking is an extremely important characteristic of fuel for spark-ignition engines. These fuels differ widely in their ability to resist knock depending on their chemical composition. In addition to the chemical characteristics of hydrocarbons in the fuel, other operating parameters such as fuel-air ratio, ignition timing, dilution, engine speed, shape of the combustion chamber, ambient conditions, compression ratio etc. affect the tendency to knock in the engine cylinder. Therefore, in order to determine the knock resistance characteristic of the fuel, the engine and its operating variables must be fixed at standard values.

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Octane Number (ON)

Octane Number (gaseous fuel) indicates the anti-knock properties of a fuel, based on the comparison of mixtures of Iso octane and normal heptane. This is a number to rate the petrol fuel according to its detonating tendency. If the fuel has the tendency to detonate less, then it has high octane number and vice versa.

- ✓ Iso-octane (2,2,4-Trimethylpentane) is a high rating fuel (i.e. detonation is less).
- ✓ Normal heptane(***n*-heptane**) is a low rating fuel (i.e. detonation is more).

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Iso-octane and normal heptane are mixed together and this sample mixture is used for running a test engine. The octane number of the fuel is the percentage of octane in this sample mixture which detonates in similar way as the fuel under the same condition. High octane fuel's number is 100. This type of fuel will not have tendency to detonate. We can make given fuel into octane number 90 to 100 by adding tetra ethyl lead. But this addition will reduce the engine life. Fuels with a higher octane ratings are used in high performance gasoline engines that require higher compression ratio. Fuels with lower octane number are ideal for diesel engines, because diesel engines do not compress the fuel but rather compress only air and then inject the fuel.

Two methods that are employed for measuring octane number are Research Octane Number (RON) and Motor Octane Number (MON). The octane numbers measured under two different engine conditions in a standard "Cooperative Fuels Research (CFR)" engine has a variable compression ratio

Research Octane Number (RON)

The most common type of octane rating is Research Octane Number (RON). RON is determined by using the fuel in a test engine running at 600 rpm with the variable compression ratio under controlled condition, and comparing the results with the mixture of iso-octane and *n*-heptane.

Motor Octane Number (MON)

Motor Octane Number is determined at 900 rpm engine speed instead of 600 rpm used in RON. MON testing uses a similar test engine used in RON testing but with a preheated fuel mixture, higher engine speed and variable ignition timing.

$$\boxed{\textbf{Anti-knock}\ \text{Index} = \frac{(RON + MON)}{2}}$$

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Advantages of High-Octane Fuel:

1. We can increase the compression ratio without detonation.

2. Engine efficiency can be increased without detonation.

3. Super charging can be done without detonation. So totally, the unwanted detonation can be reduced.

The widely used antiknock agents are:

- Tetraethyl lead [TEL] $(CH_3CH_2)_4Pb$
- Methylcyclopentadienyl manganese tricarbonyl (MMT) $(CH_3C_5H_4)Mn(CO)_3$
- Ferrocene $Fe(C_5H_5)_2$
- Iron pentacarbonyl
- Toluene
- Iso octane

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Anti-knock Agents

Anti-knock agents are classified into high-percentage additives like alcohol and low-percentage additives based on heavy elements.

Internal combustion engine discharges various substances to the atmosphere. Some of these emissions are harmful to the environment such as Carbon monoxide, Nitrogen oxides, unburnt hydrocarbons and certain compounds of lead.

The catalytic converter is used to oxidize the unburnt hydrocarbons and carbon monoxide to carbon dioxide and to decompose nitrogen oxides into nitrogen and oxygen.

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High percentage additives are those organic compounds that do not contain metals, but require high blending ratios, such as 20-30% for benzene and ethanol. Ethanol is inexpensive, and widely available but being corrosive in nature, it is not used.

Tetra ethyl lead (TEL) $(CH_3CH_2)_4$ Pb is a main additive and it is a common anti knock agent.

Adding a small amount of Tetra ethyl load (TEL) improves the anti-knock quality of fuel.

Effects of Anti knock additives

- The main problem in using Tetra ehtyl lead is the lead content in it since lead is extremely toxic and poisonous.

- A manganese - carrying additive like methylcyclopentadienyl manganese tricarbonyl (MMT) directly affects the humans.

The exposure of MMT results in eye irritation, giddiness, headache and it causes difficulties in breathing.

- Ferrocene ($Fe(C_5H_2)_2$) is an organometallic compound of iron. The iron contents in ferrocene forms a conductive coating on the spark plug.

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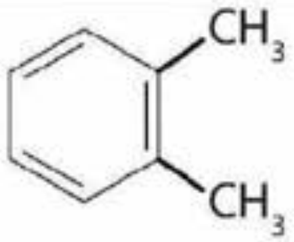
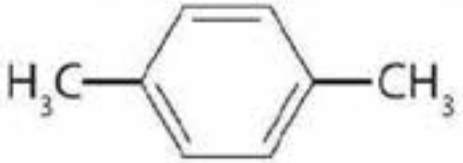
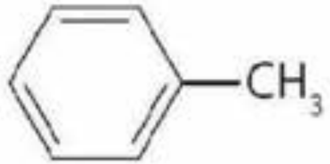
Name	Condensed Structural Formula	Octane Rating	Name	Condensed Structural Formula	Octane Rating
n-heptane	$CH_3CH_2CH_2CH_2CH_2CH_2CH_3$	0	*o*-xylene		107
n-hexane	$CH_3CH_2CH_2CH_2CH_2CH_3$	25	ethanol	CH_3CH_2OH	108
n-pentane	$CH_3CH_2CH_2CH_2CH_3$	62	*t*-butyl alcohol	$(CH_3)_3COH$	113
isooctane	$(CH_3)_3CCH_2CH(CH_3)_2$	100	*p*-xylene		116
benzene		106	methyl *t*-butyl ether	$H_3COC(CH_3)_3$	116
methanol	CH_3OH	107	toluene		118

COMBUSTION IN CI ENGINES

A lecture by

V.Manikanth

Assistant Professor

Dept. Of Mechanical Engineering

SRKR Engineering College (A)

UNIT-III

Combustion in C.I. Engines

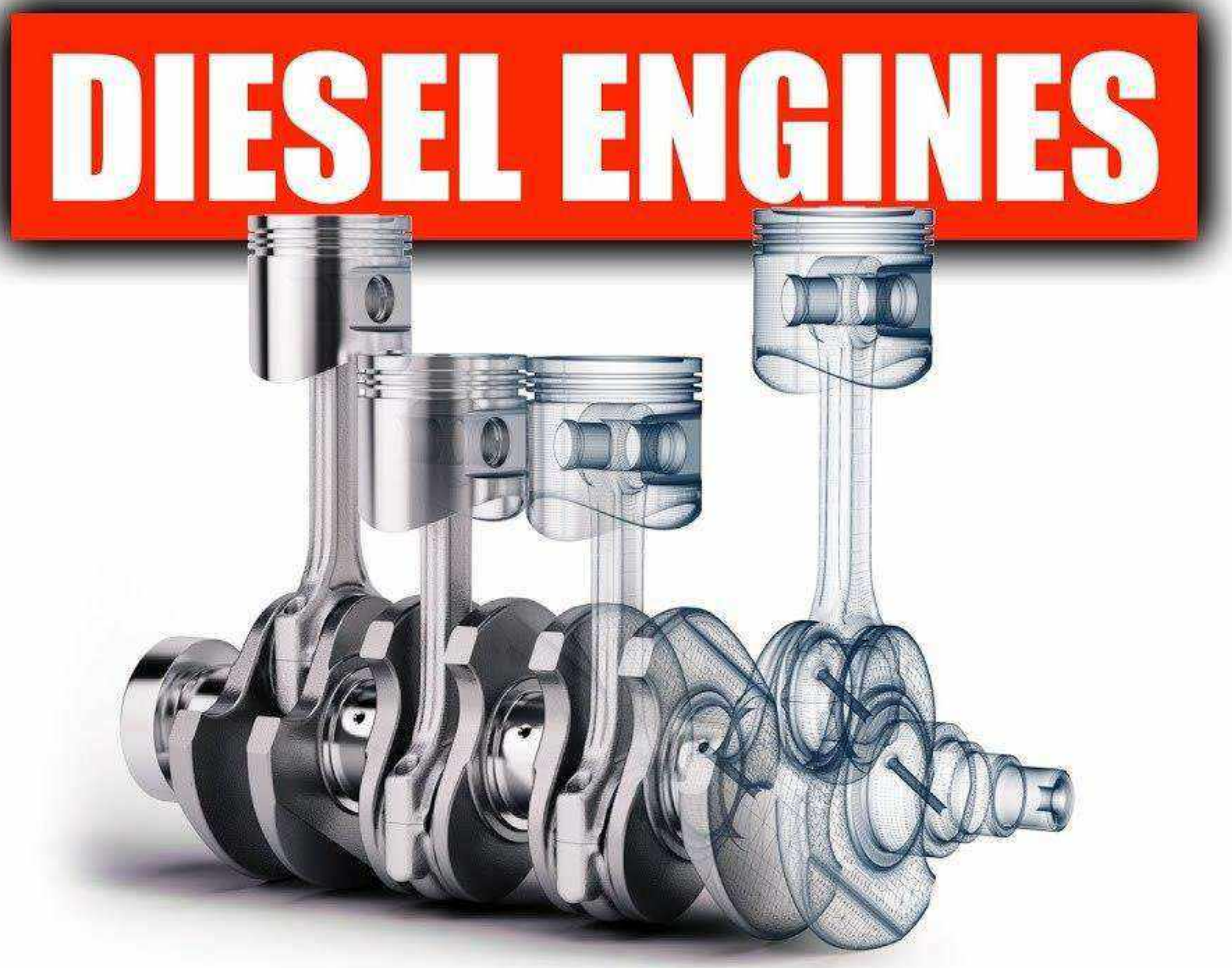
Stages of combustion, Delay period and its importance, effect of engine variables, diesel knock, suction compression and combustion induced turbulence, open and divided combustion chambers.

Textbooks:

1. A Treatise on Heat Engineering by Vasandhani and Kumar.
2. Applied Thermodynamics-II by R. Yada

Reference Books:

1. I.C. Engines by V. Ganesan.
2. Thermal Engineering, by R.K.Rajput.
3. I.C. Engines, by Mathur and Nehata.


DIESEL ENGINES

COMBUSTION IN CI ENGINES

In CI engine, combustion occurs because of the high temperature of the compressed air. Since the fuel is ignited with the high temperature of compressed air, it is called auto-ignition. For the auto ignition, compression ratio should be maximum (about 12). It requires heavier construction. So CI engines are heavier and bigger.

The air is compressed and the fuel is injected with high pressure in the form of fine spray near the end of the compression. This leads to **delay period.** This is also called **ignition lag**.

The fuel which is in atomized form is slightly colder than the hot compressed air in the cylinder. An appreciable time elapses before the actual combustion starts. This elapsed time is called **delay period** or **ignition lag.**

Four stages of combustion in CI Engine.

1. Ignition delay period.
2. Period of rapid or uncontrolled combustion.
3. Period of controlled combustion.
4. After burning.

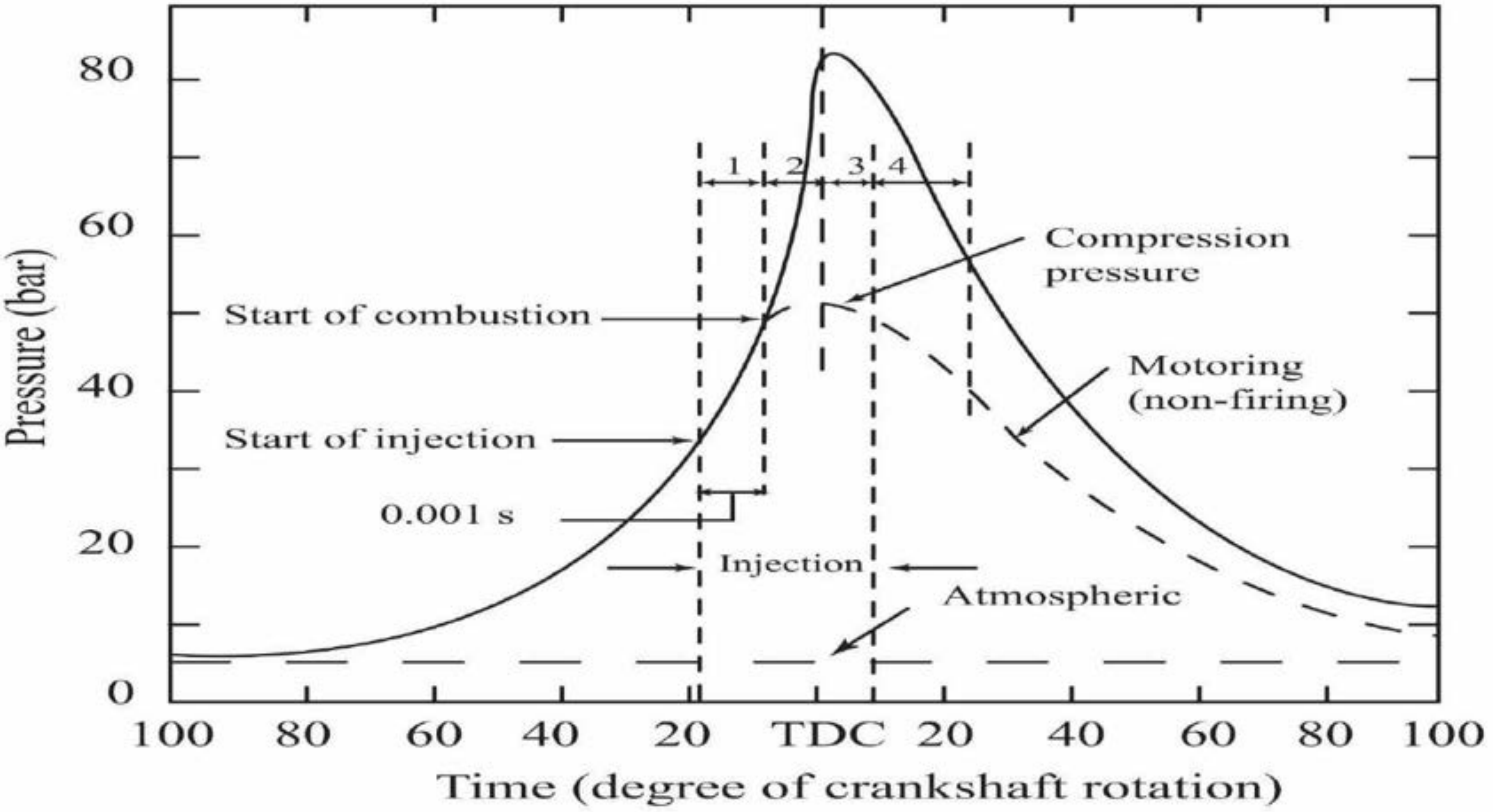
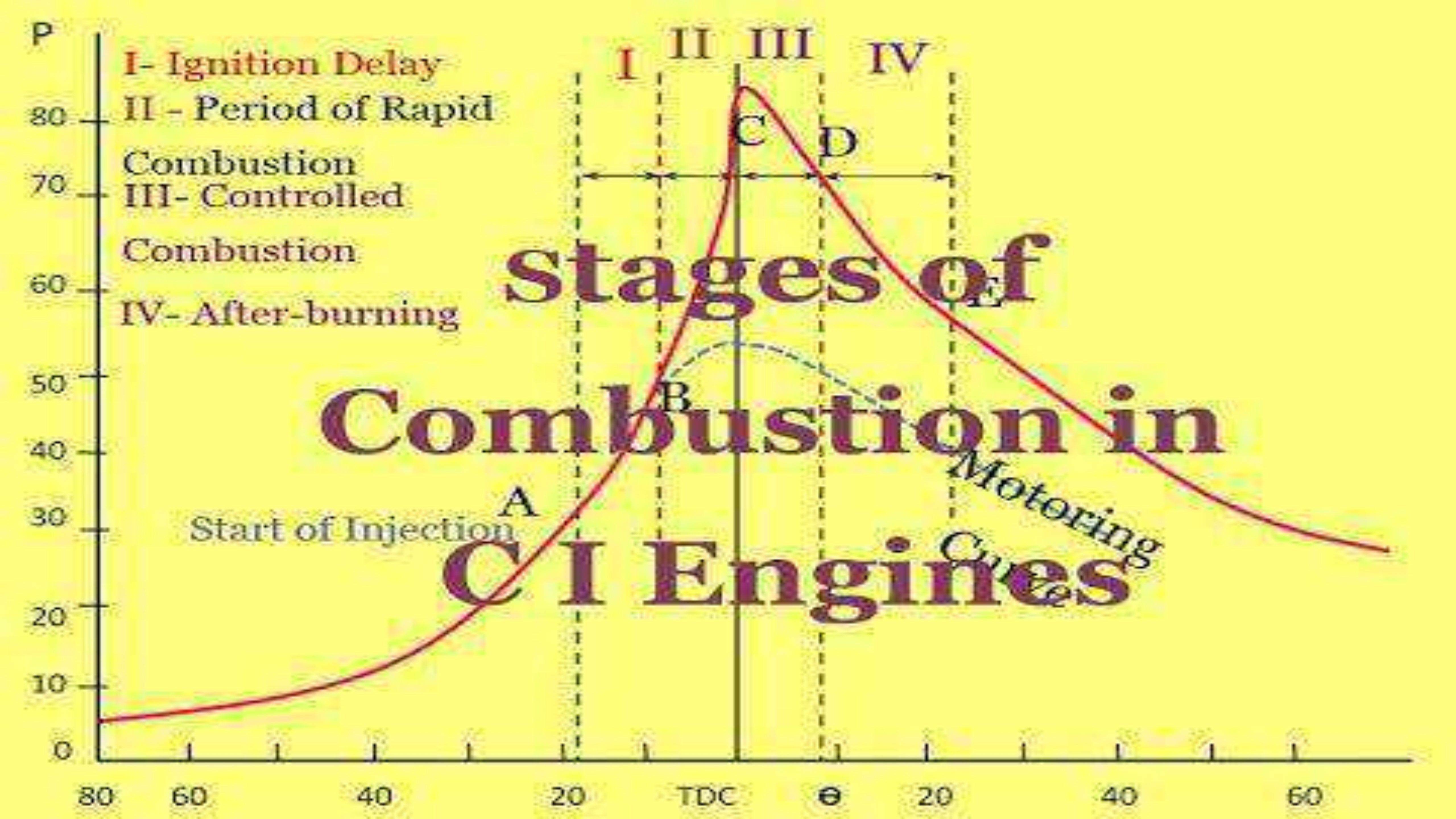


Fig. Stages of combustion in a CI engine


Stages of Combustion in C I Engines
I- Ignition Delay
II - Period of Rapid Combustion
III- Controlled Combustion
IV- After-burning
I
II
III
IV
A
B
C
D
E
Start of Injection
Motoring Curve
P
0
10
20
30
40
50
60
70
80
80
60
40
20
TDC
θ
20
40
60

1.Ignition Delay Period:

There is a definite period of inactivity between the time when the first droplet of fuel hits the hot air in the combustion chamber and the time it starts through the actual burning phase. This period is known as the ignition delay period. In Fig. the delay period is shown on pressure crank angle (or time) diagram between points a and b. Point a represents the time of injection and point b represents the time at which the pressure curve (caused by combustion) first separates from the motoring curve.

The ignition delay period can be divided into two parts,

i. Physical delay

ii. Chemical delay.

The delay period in the CI engine is a very great incluencing factor on both engine design and performance. It influences the following:

(i) The combustion rate

(ii) Knocking

(iii) Starting ability

(iv) The presence of smoke in the exhaust

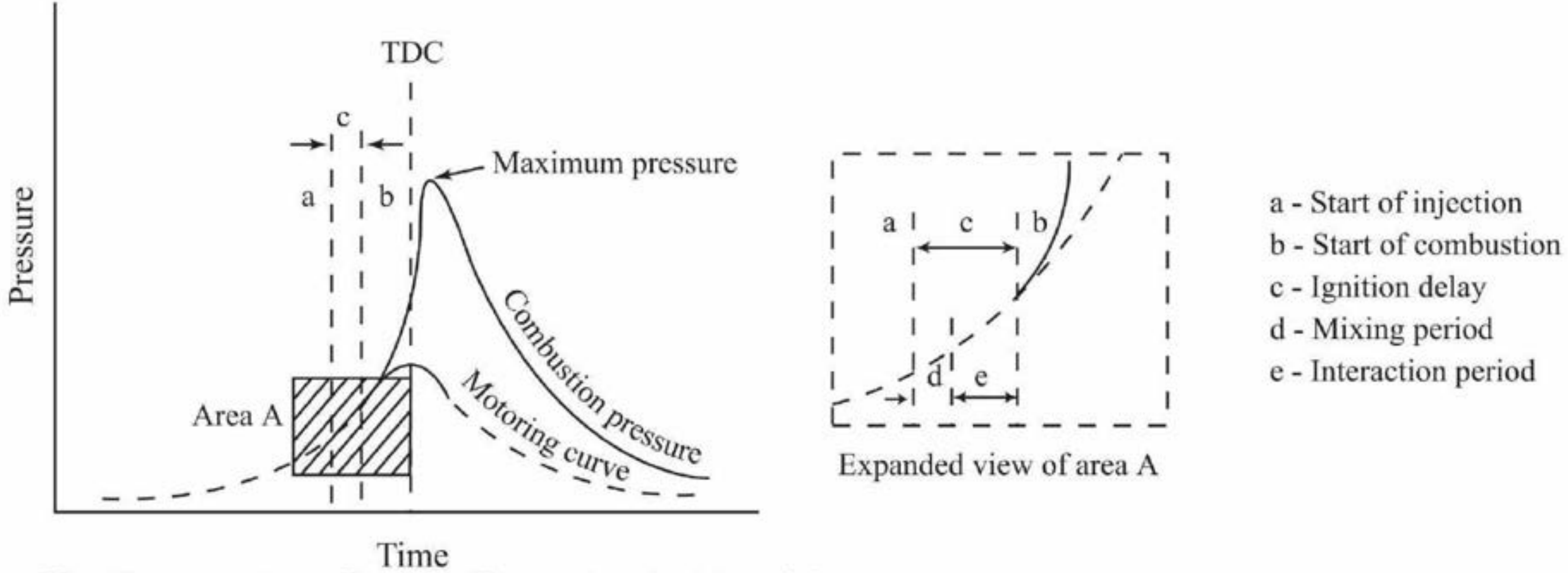


Fig. Pressure-time diagram illustrating ignition delay

i. Physical Delay:

The physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions. During this period, the fuel is atomized, vaporized, mixed with air and raised to its self-ignition temperature. This physical delay depends on the type of fuel, i.e., for light fuel the physical delay is small while for heavy viscous fuels the physical delay is high. The physical delay is greatly reduced by using high injection pressures, higher combustion chamber temperatures and high turbulence to facilitate breakup of the jet and improving evaporation.

ii. Chemical Delay:

During the chemical delay, reactions start slowly and then accelerate until inflammation or ignition takes place. Generally, the chemical delay is larger than the physical delay. However, it depends on the temperature of the surroundings and at high temperatures, the chemical reactions are faster and the physical delay becomes longer than the chemical delay. It is clear that, the ignition lag in the SI engine is essentially equivalent to the chemical delay for the CI engine. In most CI engines the ignition lag is shorter than the duration of injection.

Factors Affecting Delay Period

1. Temperature and pressure in the combustion chamber at the time of injection.
2. Air-fuel ratio.
3. Turbulence of air.
4. Presence of residual gases.
5. Rate of fuel injection.
6. The extent of atomization and vaporization and fineness of fuel spray.

2. Period of Rapid Combustion (or) Uncontrolled Combustion

After delay period, this period starts. This period is counted from the end of delay period to the point of maximum pressure on the indicator diagram. In this stage, the pressure rise is rapid. About one-third of heat is released at this stage.

The rate of pressure raised in this stage depends on

1. the amount of fuel sprayed in the delay period.
2. the degree of turbulence.
3. fineness of fuel-spray.

3. Period of Controlled Combustion: This is the third stage starting after rapid combustion period. The fuel injected in the stage is burnt immediately and any further pressure rise can be controlled by injection rate. The period of controlled combustion is coming to an end at maximum cycle temperature.

4. After Burning: The unburnt fuel particles will get inflamed even after fuel injection is over. This is called after burning. This after burning may continue in the expansion stroke upto 70° to 80° of crank angle from TDC.

Table. Effect of Variables on the Delay Period

Increases in variable	Effect on Delay Period	Reason
Cetane number of fuel	Reduces	Reduces the self-ignition temperature
Injection pressure	Reduces	Reduces physical delay due to greater surface-volume ratio
Injection timing advance	Reduces	Reduced pressures and temperatures when the injection begins
Compression ratio	Reduces	Increases air temperature and pressure and reduces autoignition temperature
Intake temperature	Reduces	Increases air temperature
Jacket water temperature	Reduces	Increases wall and hence air temperature

Increases in variable	Effect on Delay Period	Reason
Fuel temperature	Reduces	Increases chemical reaction due to better vaporization
Intake pressure (supercharging)	Reduces	Increases density and also reduces autoignition temperature
Speed	Increases in terms of crank angle. Reduces in terms of milliseconds	Reduces loss of heat
Load (fuel-air ratio)	Decreases	Increases the operating temperature
Engine size	Decreases in terms of crank angle. Little effect in terms of milliseconds	Larger engines operate normally at low speeds
Type of combustion chamber	Lower for engines with precombustion chamber	Due to compactness of the chamber

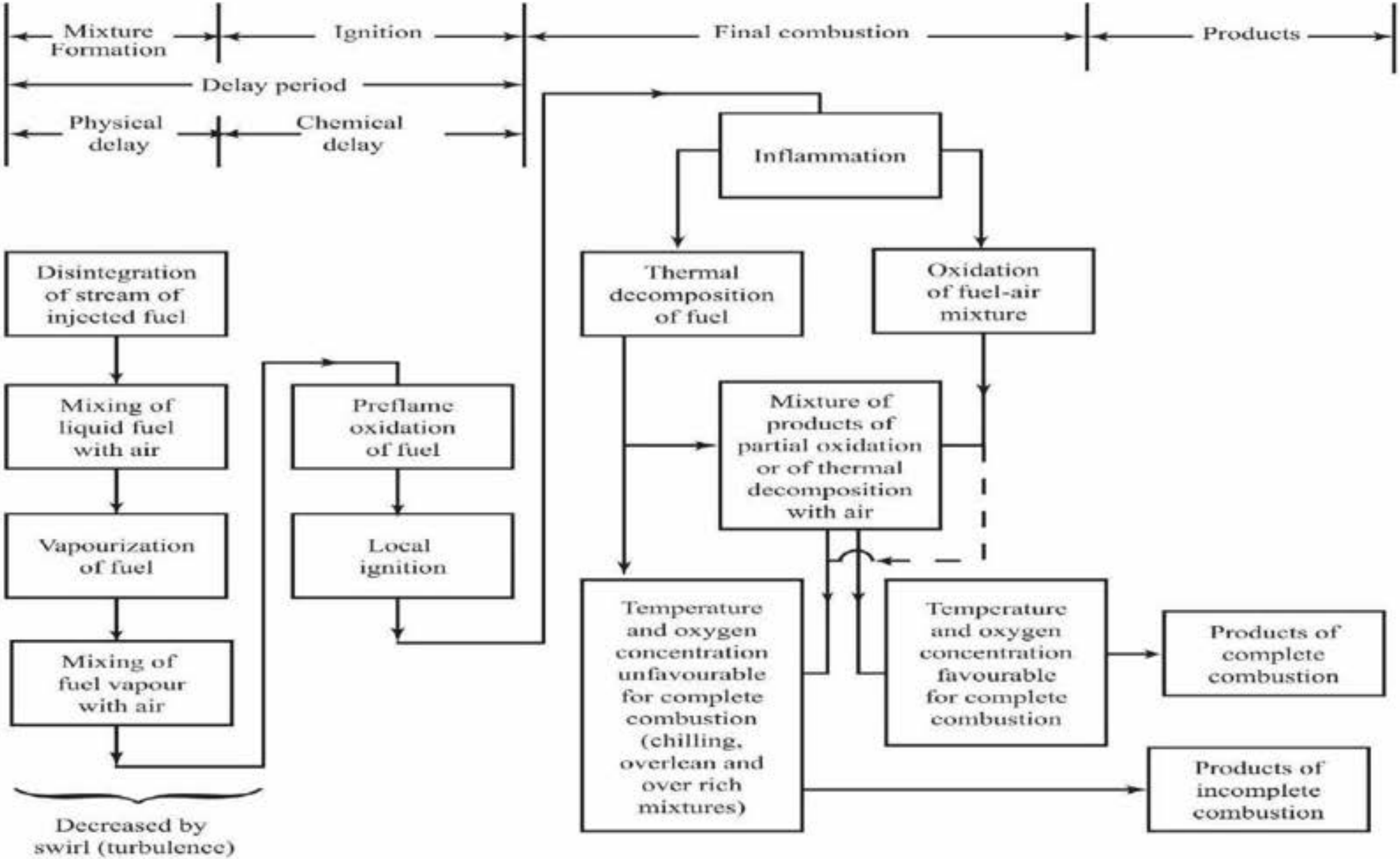


Fig. Block diagram illustrating the combustion process in a CI engine

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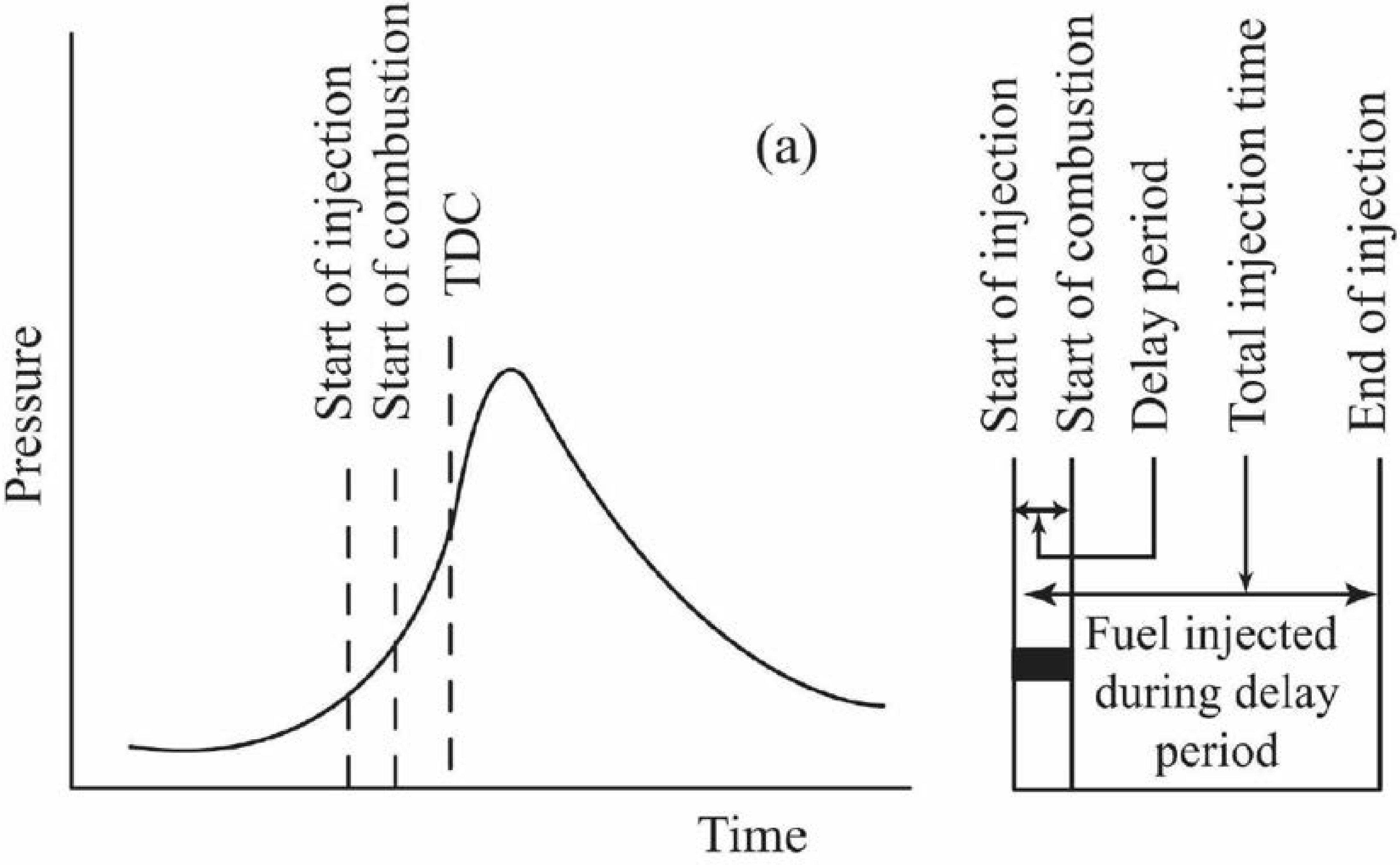
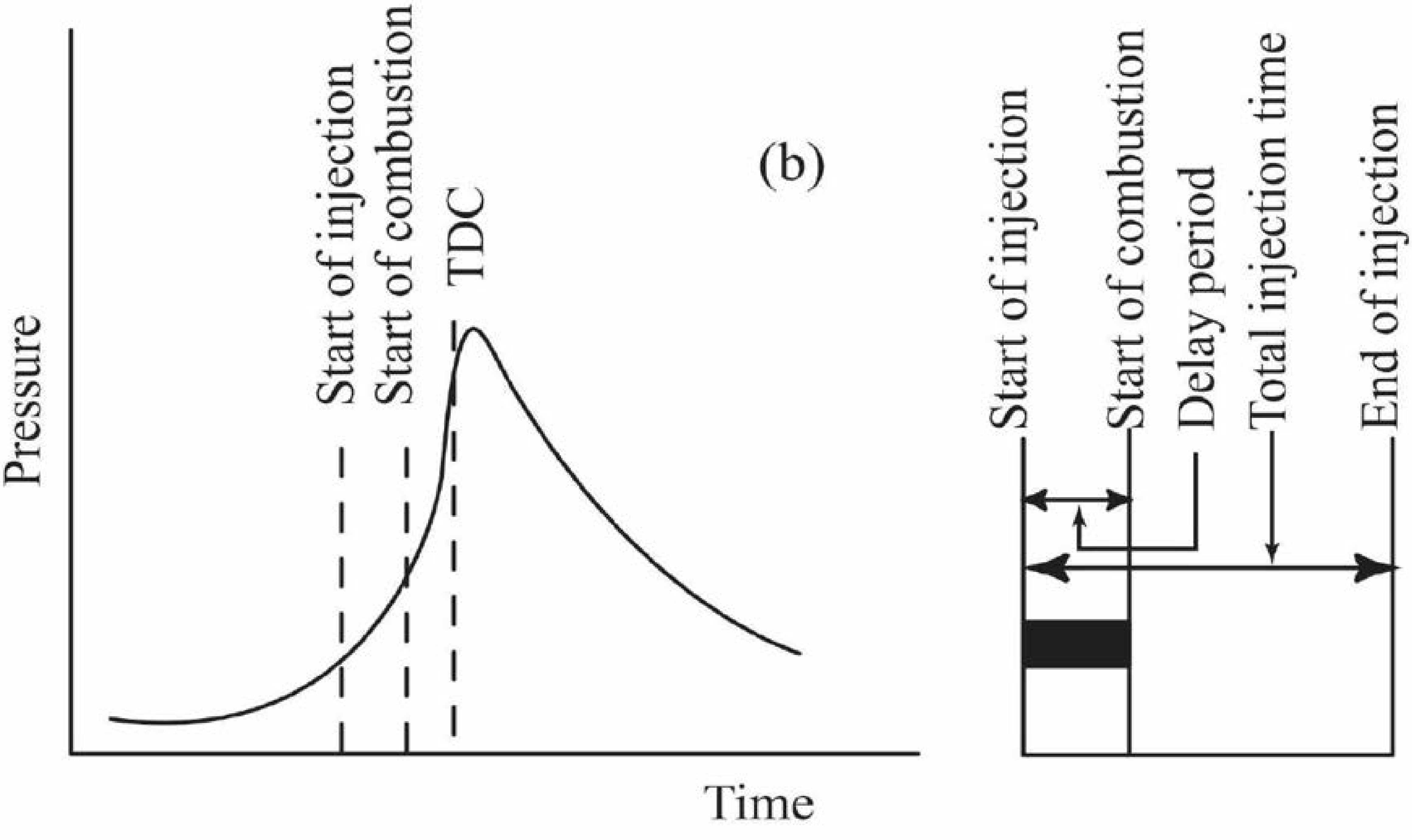
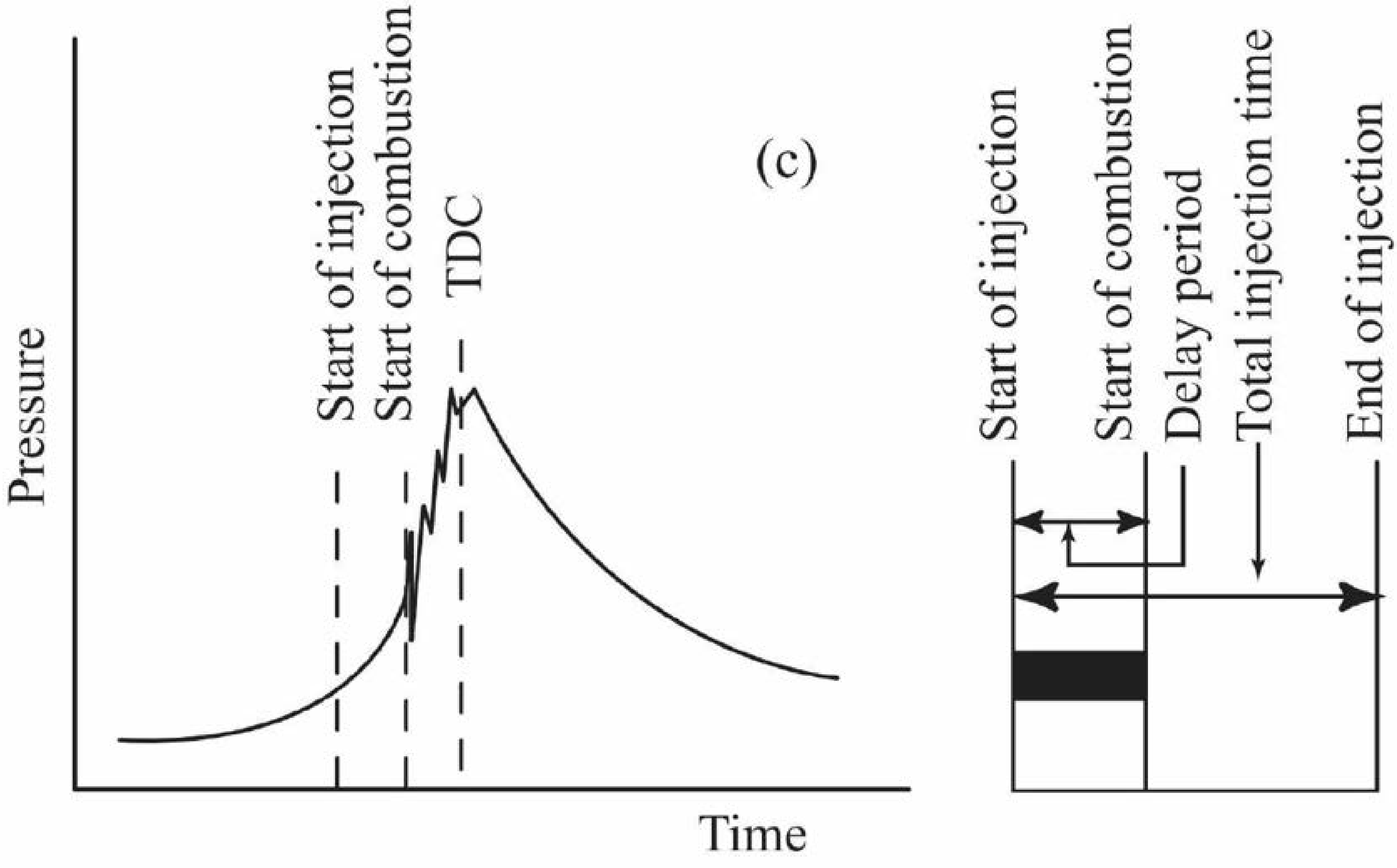
KNOCKING (OR) DIESEL KNOCK

If the ignition delay is longer, the actual burning of the first few droplets is delayed and a greater quantity of fuel droplets gets accumulated in the chamber. When the actual burning commences, the additional fuel can cause too rapid a rate of pressure rise as resulting in a jamming of forces against the piston and rough engine operation. If the ignition delay is quite long, so much fuel can accumulate that the rate of pressure rise is almost instantaneous. Such a @V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A). situation produces the extreme pressure differentials and violent gas vibrations known as knocking and is evidenced by audible knock.

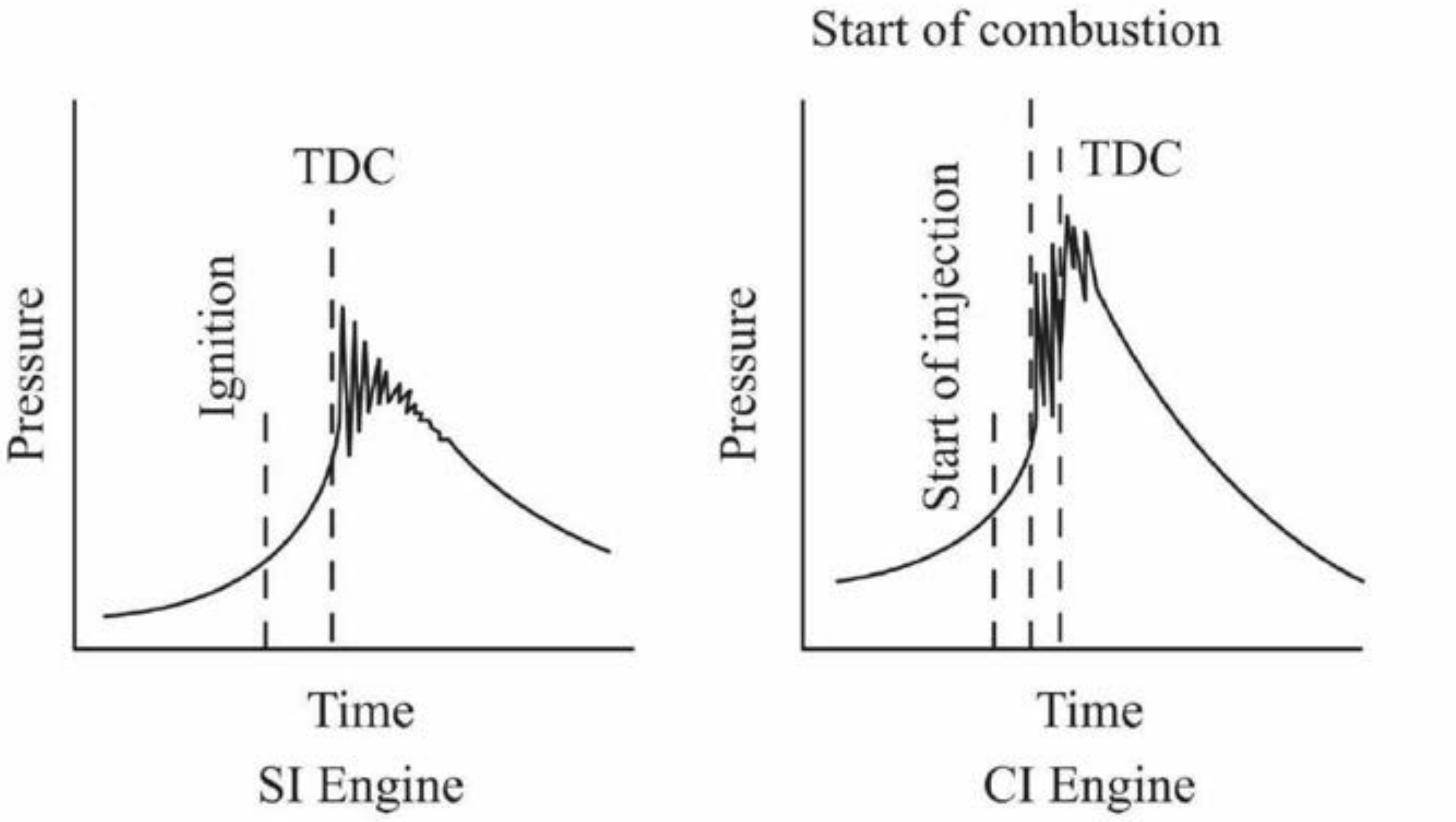
In order to decrease the tendency of knock it is necessary to start the actual burning as early as possible after the injection begins. In other words, it is necessary to decrease the ignition delay and thus decrease the amount of fuel present when the actual burning of the first few droplets start.


Pressure
Start of injection
Start of combustion
TDC
(a)
Time
Start of injection
Start of combustion
Delay period
Total injection time
End of injection
Fuel injected
during delay
period


(b)
Pressure
Time
Start of injection
Start of combustion
TDC
Start of injection
Start of combustion
Delay period
Total injection time
End of injection


Pressure
Start of injection
Start of combustion
TDC
(c)
Time
Start of injection
Start of combustion
Delay period
Total injection time
End of injection

In the SI engine, knocking occurs near the end of combustion whereas in the CI engine, knocking occurs near the beginning of combustion.



Diagrams illustrating knocking combustion in SI and CI engines

The methods to prevent knocking

1. By reducing the delay period by doping. Note: Doping is the process of adding 1% ethyl nitrate to accelerate the combustion and as a result, we can reduce knocking.

2. By raising the compression ratio, we can raise the temperature of air much higher than that required for auto ignition of the fuel. By doing so, we can reduce knocking.

Note: In petrol engine, detonation occurs if we increase compression ratio. Here in CI engine, we can prevent knocking by increasing compression ratio.

3. By increasing the turbulence of the compressed air, we can prevent knocking.

4. By adjusting the fuel injector so as to inject only a small quantity of fuel in beginning.

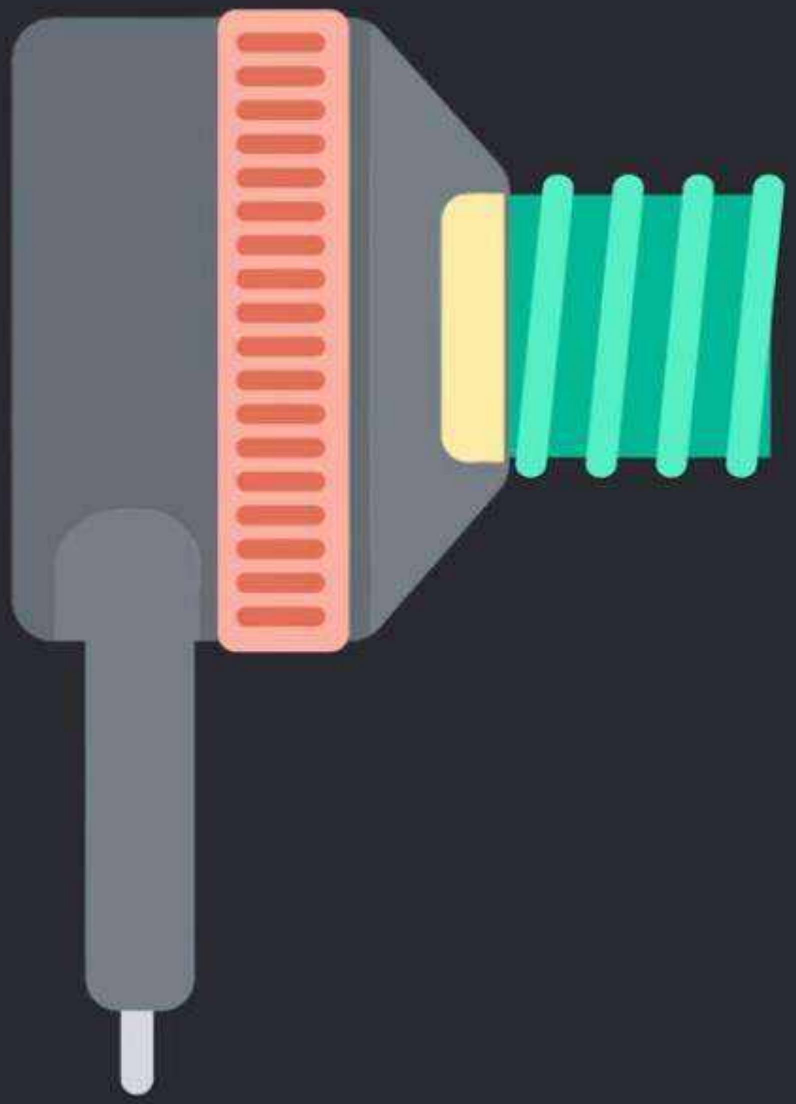
5. By super charging, we can reduce knocking.

Note: Super charging is the process of increasing the inlet pressure of air. But super charging will increase the tendency of detonation in SI engine.

6. By increasing the injector pressure, we can atomize the fuel efficiently to avoid knocking.

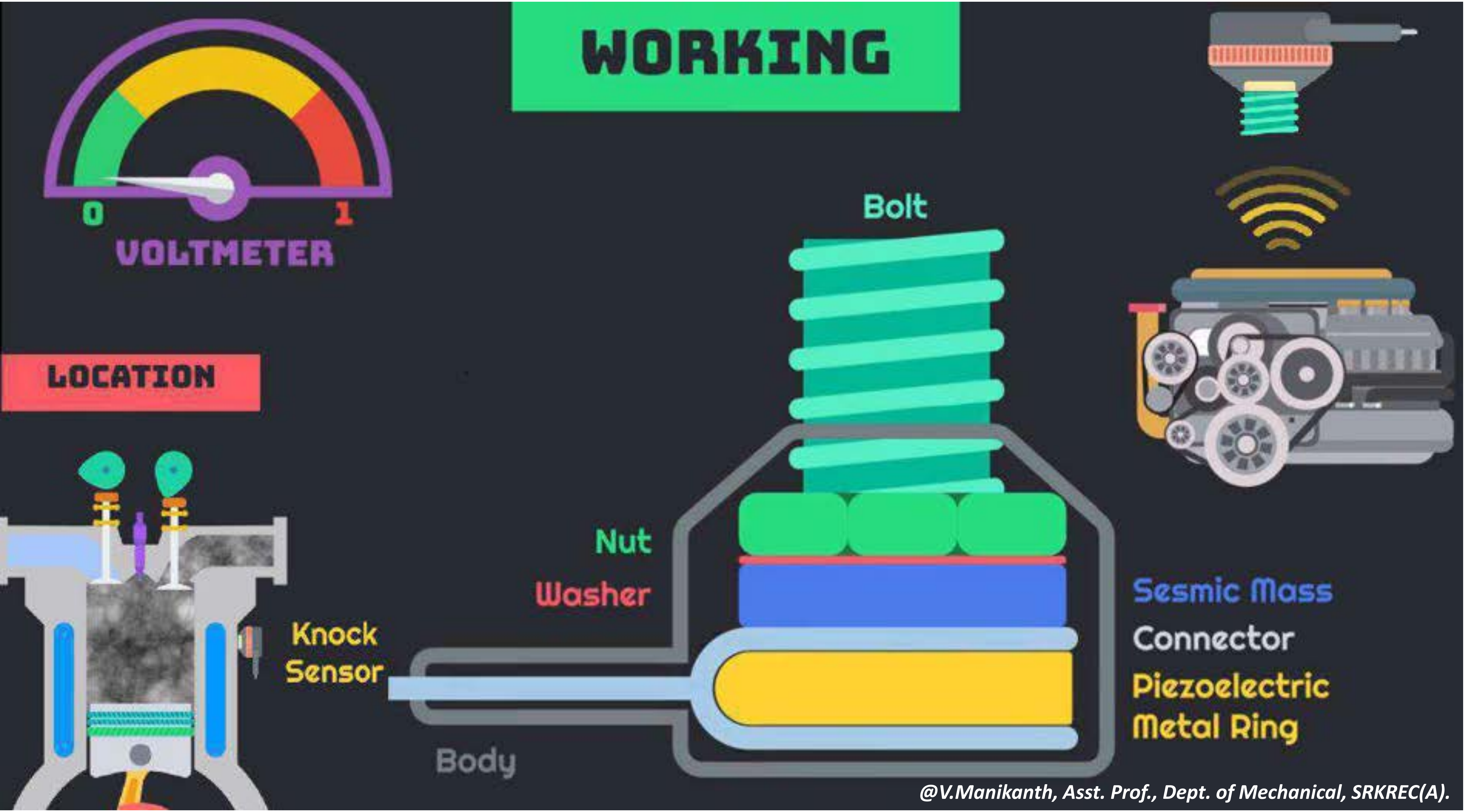
Table. Characteristics tending to reduce detonation or knock

S.No.	Characteristics	SI Engines	CI Engines
1.	Ignition temperature of fuel	High	Low
2.	Ignition delay	Long	Short
3.	Compression ratio	Low	High
4.	Inlet temperature	Low	High
5.	Inlet pressure	Low	High
6.	Combustion wall temperature	Low	High
7.	Speed, rpm	High	Low
8.	Cylinder size	Small	Large



HOW KNOCK SENSOR WORKS?

WORKING


0
1
VOLTMETER
LOCATION
Bolt
Nut
Washer
Knock
Sensor
Sesmic Mass
Connector
Piezoelectric
Metal Ring
Body

COMBUSTION CHAMBERS FOR CI ENGINES

The combustion chamber characteristics must be such as

1. To provide proper mixing of fuel and air in a short time.

2. To avoid maximum cylinder pressure.

3. To avoid excessive pressure rise.

4. It is to be designed so that the fuel should be burnt fully in the expansion stroke.

CI engine combustion chambers are classified into two categories:

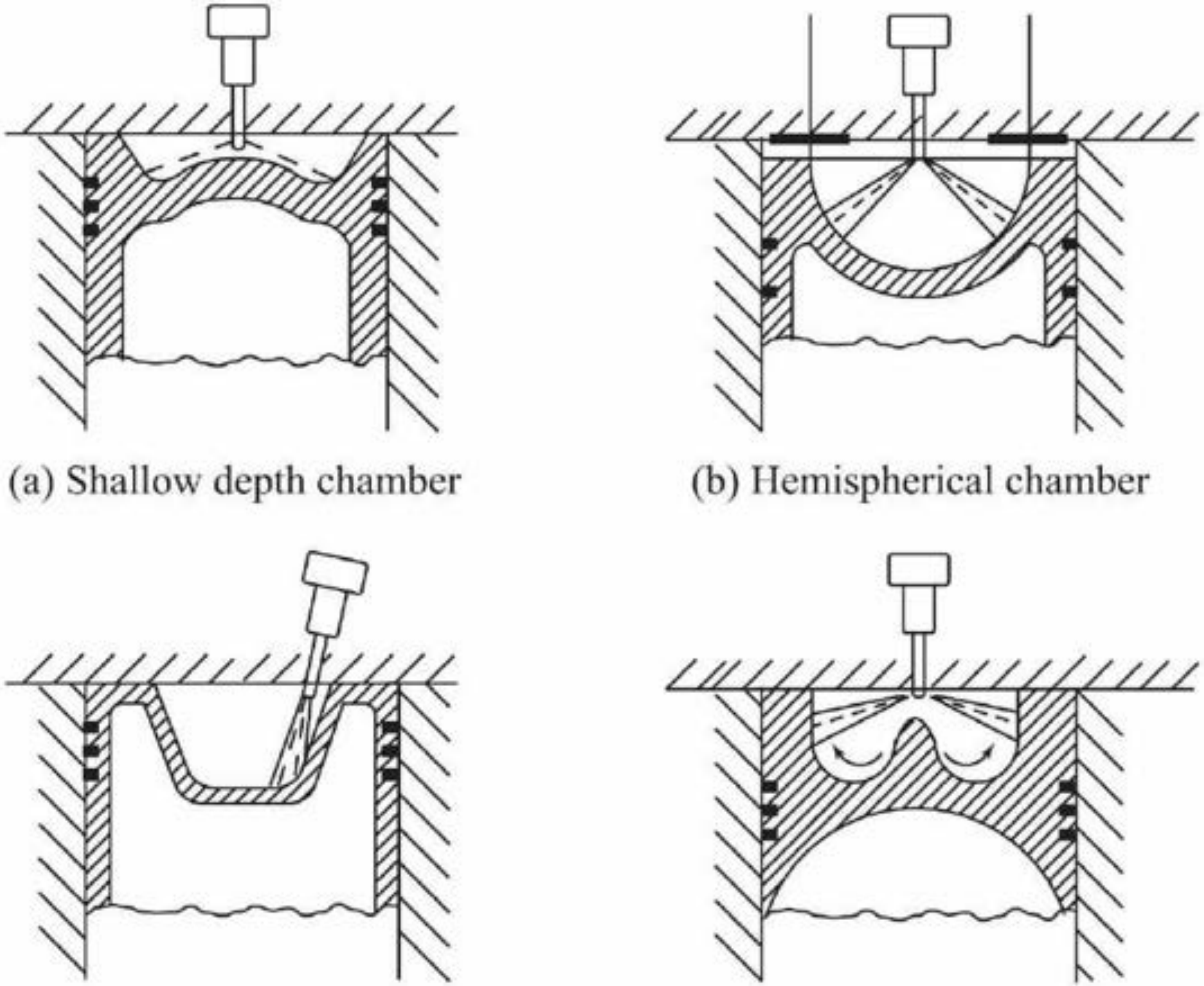
(i) Direct-Injection (DI) Type: This type of combustion chamber is also called an open combustion chamber. In this type the entire volume of the combustion chamber is in the main cylinder and the fuel is injected into this volume.

(ii) Indirect-Injection (IDI) Type: In this type of combustion chambers, the combustion space is divided into two parts, one part in the main cylinder and the other part in the cylinder head. The fuel-injection is affected usually into that part of the chamber located in the cylinder head. These chambers are classified further into:

(a) Swirl chamber in which compression swirl is generated.

(b) Pre combustion chamber in which combustion swirl is induced.

(c) Air cell chamber in which both compression and combustion swirl are induced.



(a) Shallow depth chamber

(b) Hemispherical chamber

(c) Cylindrical chamber

(d) Toroidal chamber

Fig. Open combustion chambers

The main advantages of this type of chambers are:

(i) Minimum heat loss during compression because of lower surface area to volume ratio and hence, better efficiency.

(ii) No cold starting problems.

(iii) Fine atomization because of multi hole nozzle.

The drawbacks of these combustion chambers are:

(i) High fuel-injection pressure required and hence complex design of fuel injection pump.

(ii) Necessity of accurate metering of fuel by the injection system, particularly for small engines.

Indirect–Injection Chambers

A divided combustion chamber is defined as one in which the combustion space is divided into two or more distinct compartments connected by restricted passages. This creates considerable pressure differences between them during the combustion process.

These chambers are further classified into

- Turbulent combustion chamber
- Pre combustion chamber
- Energy cell

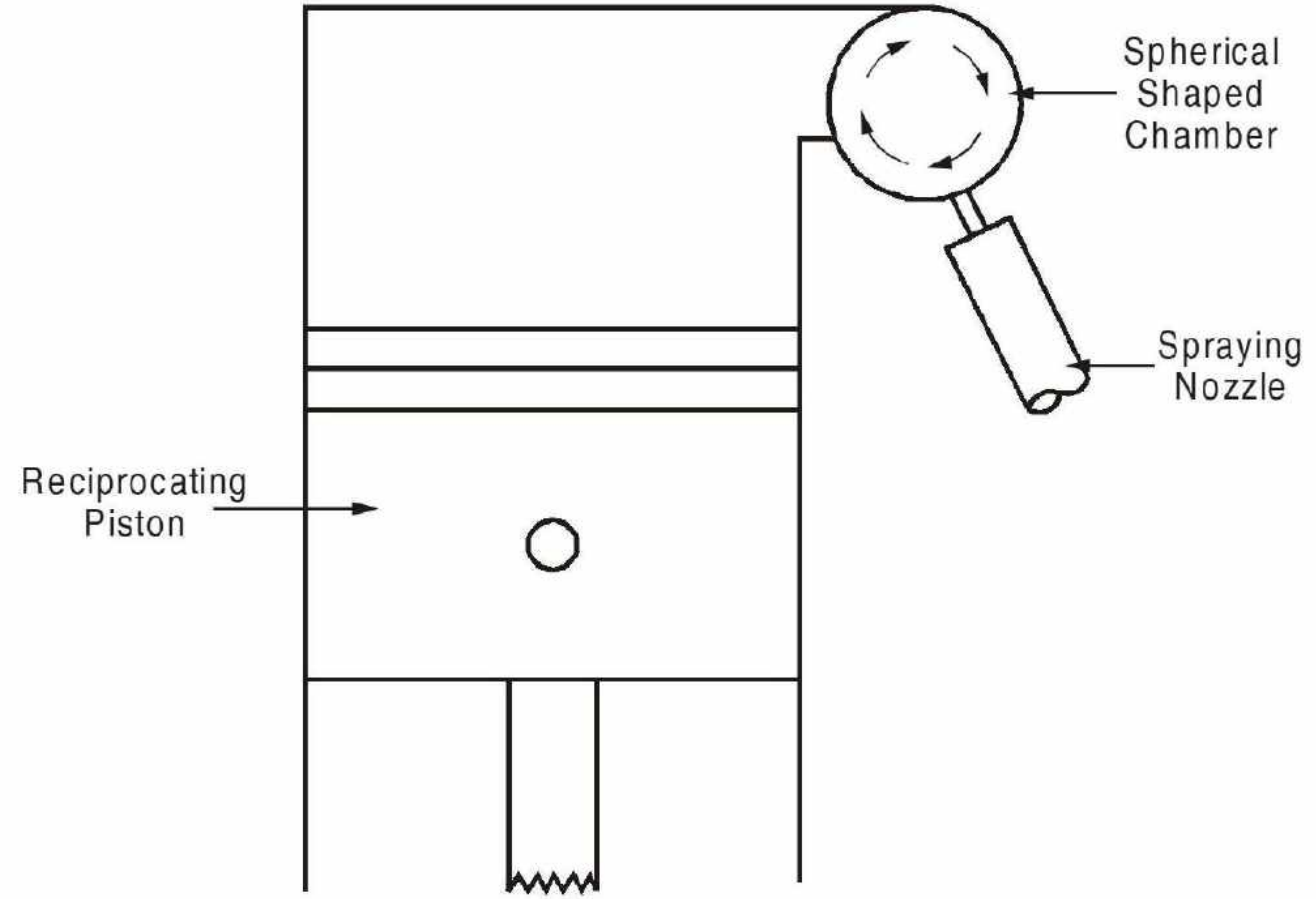


Fig. Turbulent Combustion Chamber

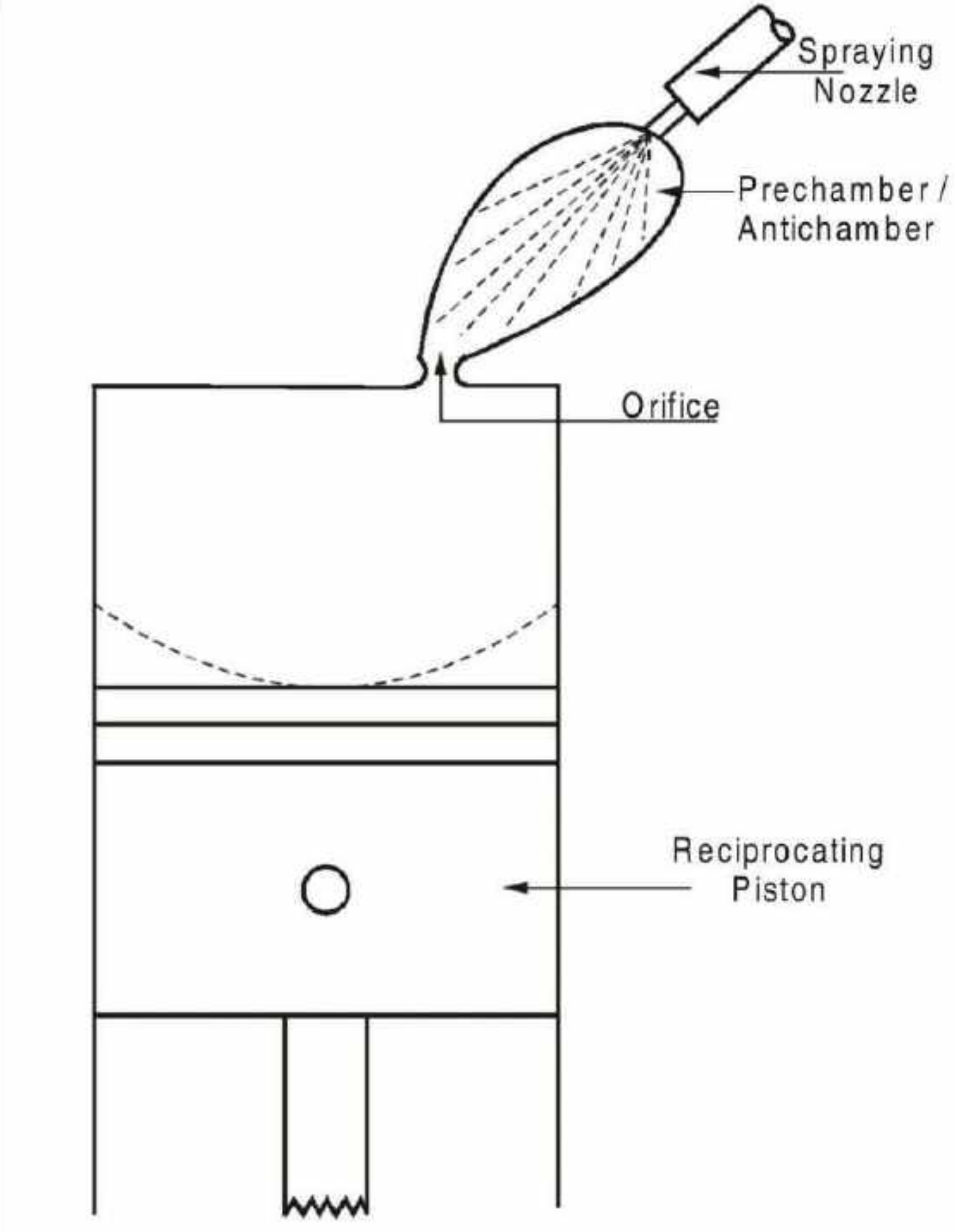


Fig. Precombustion Chamber

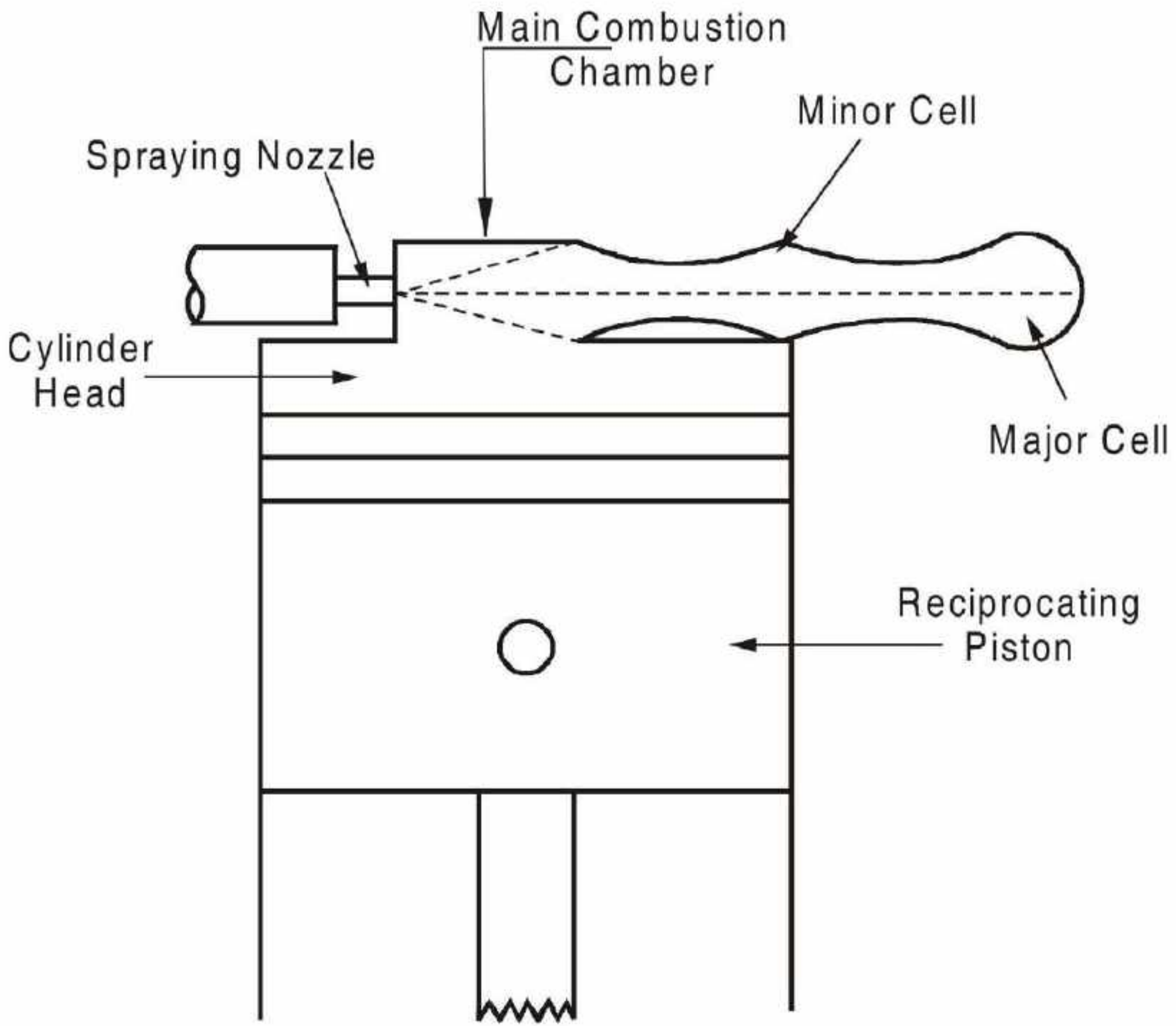
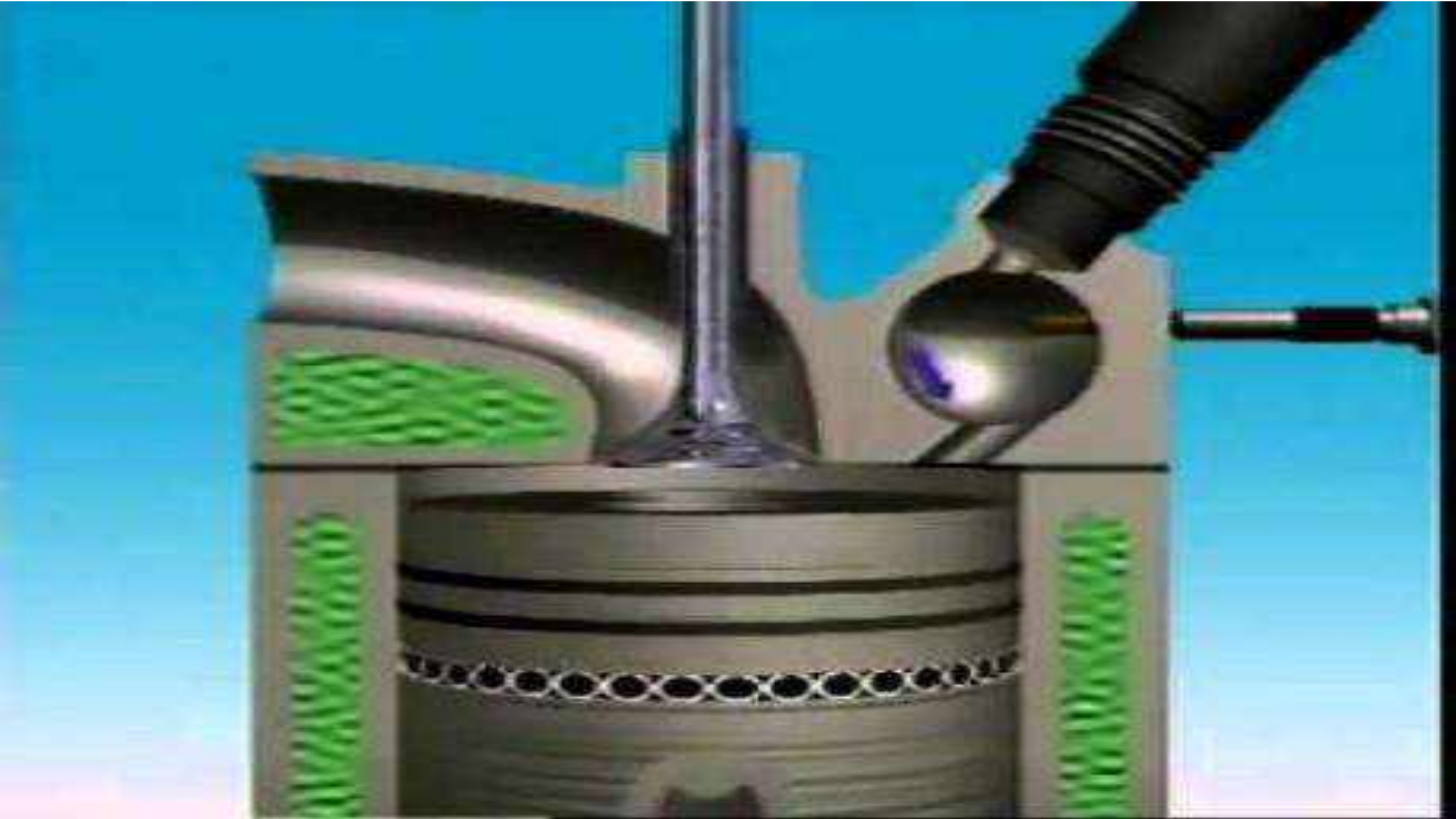


Fig. Energy Cell



Advantages of Indirect-injection combustion chamber

- No complex design is needed for fuel-injection pump since the injection pressure required is very low.
- Direction of spraying is not very important.
- Higher engine speeds can be achieved since burning is continuous in pre chamber.

Disadvantages

- Heater plugs are required, because of poor cold starting performance.
- Specific fuel consumption is high as there is a loss of pressure due to air motion through the duct and heat loss due to large heat transfer area.
- The increase in temperature and pressure on the part of the piston leads to cracking and distortion.

Open combustion chamber	Divided combustion chamber
1. It requires multiple hole injection nozzles.	1. It requires single hole injection nozzle.
2. It can consume good ignition quality fuels.	2. It consumes poor ignition quality fuels.
3. Open combustion chamber is more efficient.	3. Divided combustion chamber leads to pressure loss and heat losses during compression and expansion. So this type is not efficient.
4. It requires high injection pressure.	4. It requires moderate injection pressure.
5. Cylinder construction is simple and less cost.	5. Cylinder construction is hard and more expensive.

Diesel Fuel Requirements & Ratings

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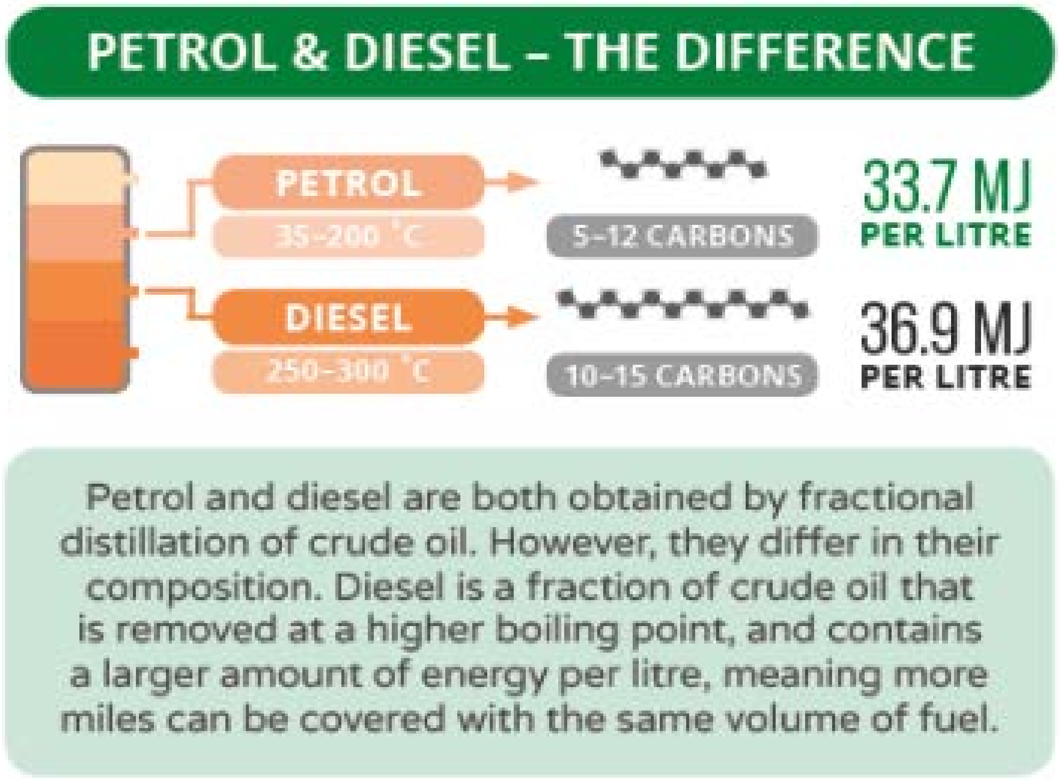
DIESEL FUEL

The fuel used in Compression Ignition (CI) engine is diesel, which is a type of hydrocarbon. The most common type of diesel fuel is a specific fractional distillate of petroleum fuel oil, but alternatives that are not derived from petroleum, such as biodiesel, biomass to liquid (BTL) or gas to liquid (GTL) diesel, are increasingly being developed and adopted..

Chemical Composition:

petroleum-derived diesel is composed of about 75% saturated hydrocarbons (primarily paraffins including n, iso, and cycloparaffins), and 25% aromatic hydrocarbons (including naphthalenes and alkylbenzenes). The average chemical formula for common diesel fuel is $C_{12}H_{23}$, ranging approximately from $C_{10}H_{20}$ to $C_{15}H_{28}$.

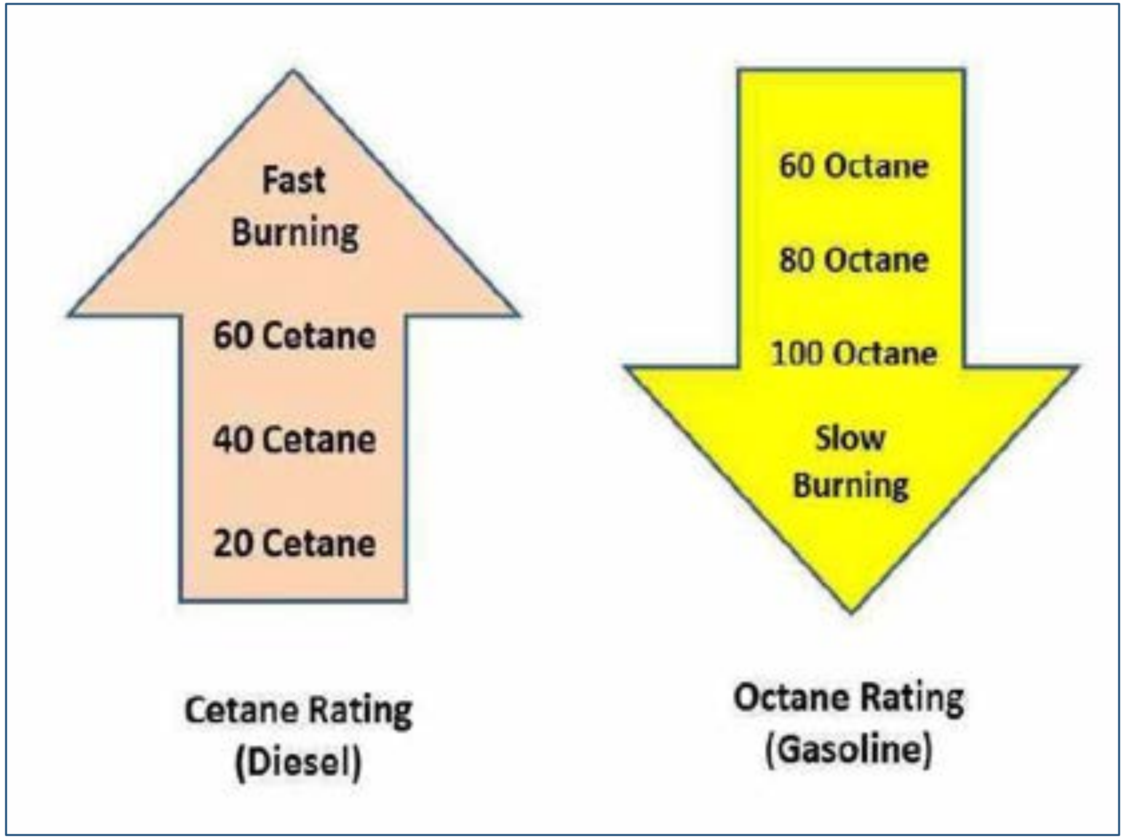


Some of the desired characteristics of Diesel:

1. Knocking characteristics
2. Volatility of the fuel
3. Starting characteristics of the fuel
4. Smoke produced by fuel and its odour
5. Viscosity of fuel
6. Corrosion and wear
7. Easy to handle

Cetane Number (CN)

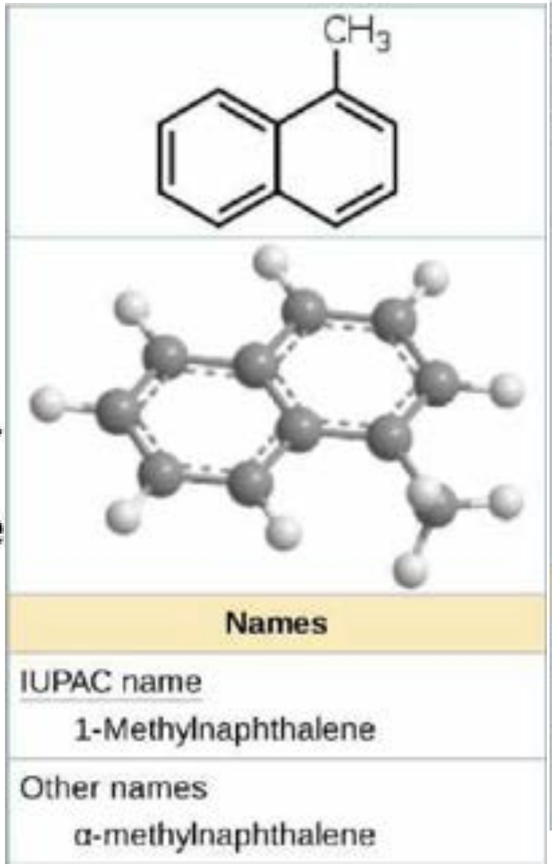
Cetane number (or CN) is an inverse function of a fuel's ignition delay, the time period between the start of ignition and the first identifiable pressure increase during combustion of the fuel. In a particular diesel engine, higher cetane fuels will have shorter ignition delay periods than lower Cetane fuels. Cetane numbers are only used for the relatively light distillate diesel oils.

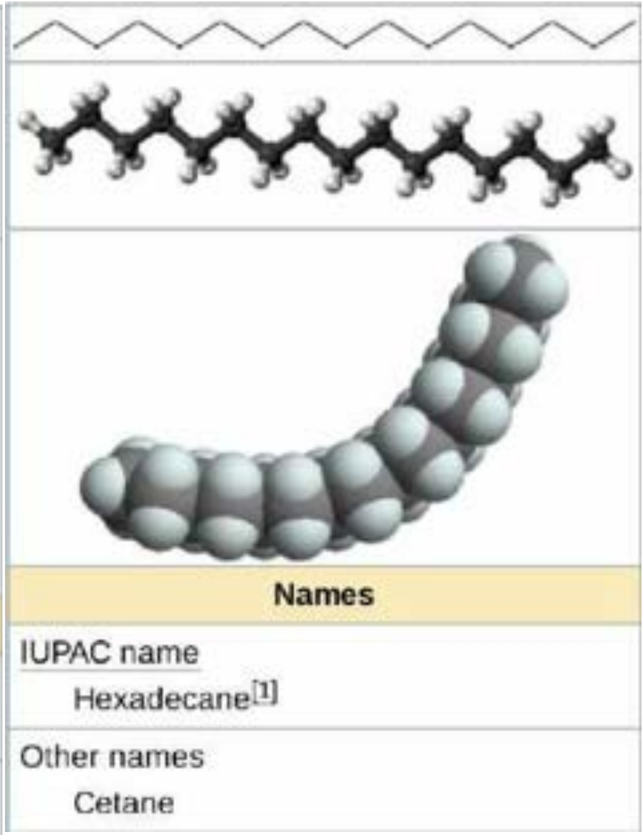


Diesel Index

The Diesel index indicates the ignition quality of the fuel. It is found to correlate, approximately to the cetane number of commercial fuels. Diesel index and cetane number are usually about 50. Lower value will result in smoky exhaust.

The cetane number is a number to rate diesel fuel's ability to auto ignite quickly when it is injected into the high pressure, high temperature air in the cylinder. Higher the cetane number, lesser is the **'Diesel knocking'** tendency.





Procedure for finding Cetane Number

- Cetane ($C_{10}H_{34}$) has high ignitabilty.
- α–methyl-napthalene ($C_{11}H_{10}$) has low ignitability.
- Both are mixed together and this sample mixture is used for running a test engine.

Additives

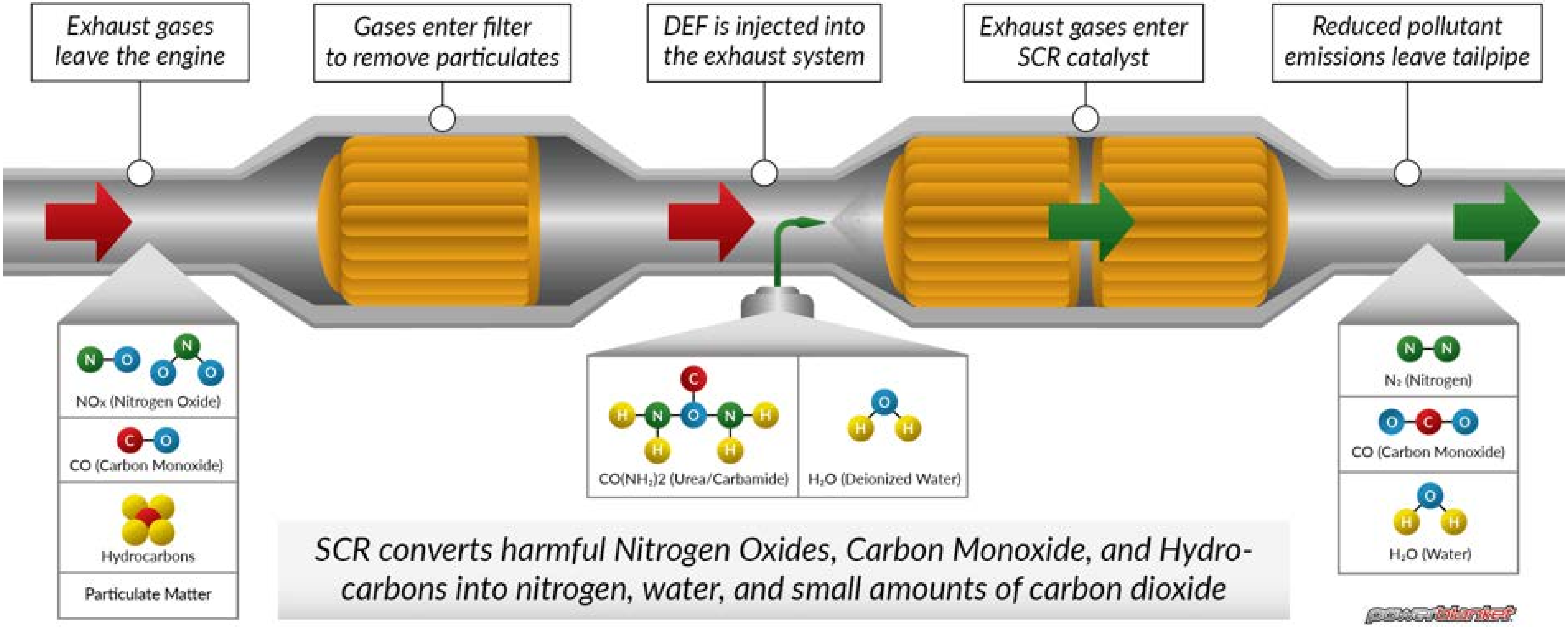
Alkyl nitrates (2-ethylhexyl nitrate) and di-tert-butyl peroxide are used as additives to raise the cetane number.

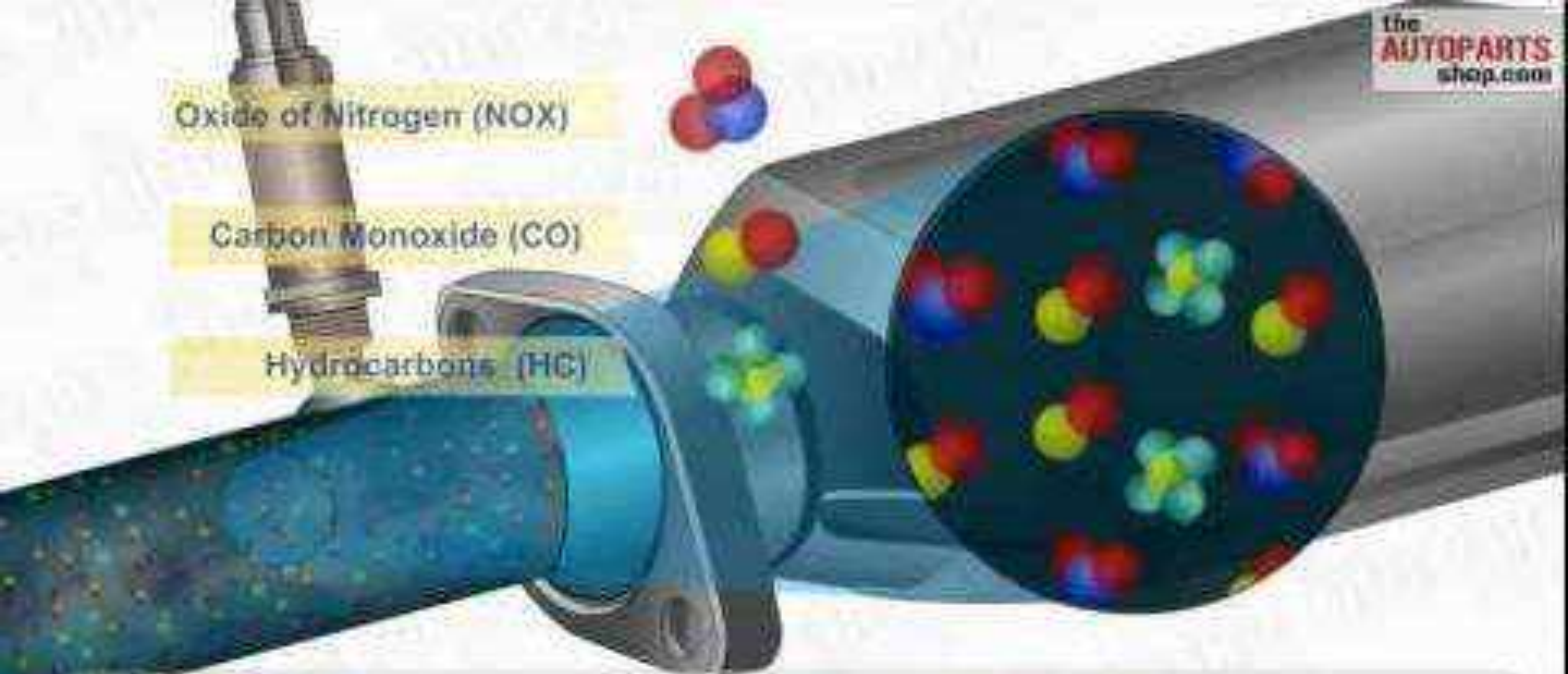
The cetane number of the diesel is the percentage of cetane in this sample mixture which knocks in a similar way as the diesel under the same condition.

- Lower cetane number, higher are the hydrocarbon emission in the exhaust gases.
- Lower cetane number, higher are the noise level.
- Lower cetane number, increases the ignition delay.

WHAT IS SELECTIVE CATALYTIC REDUCTION?

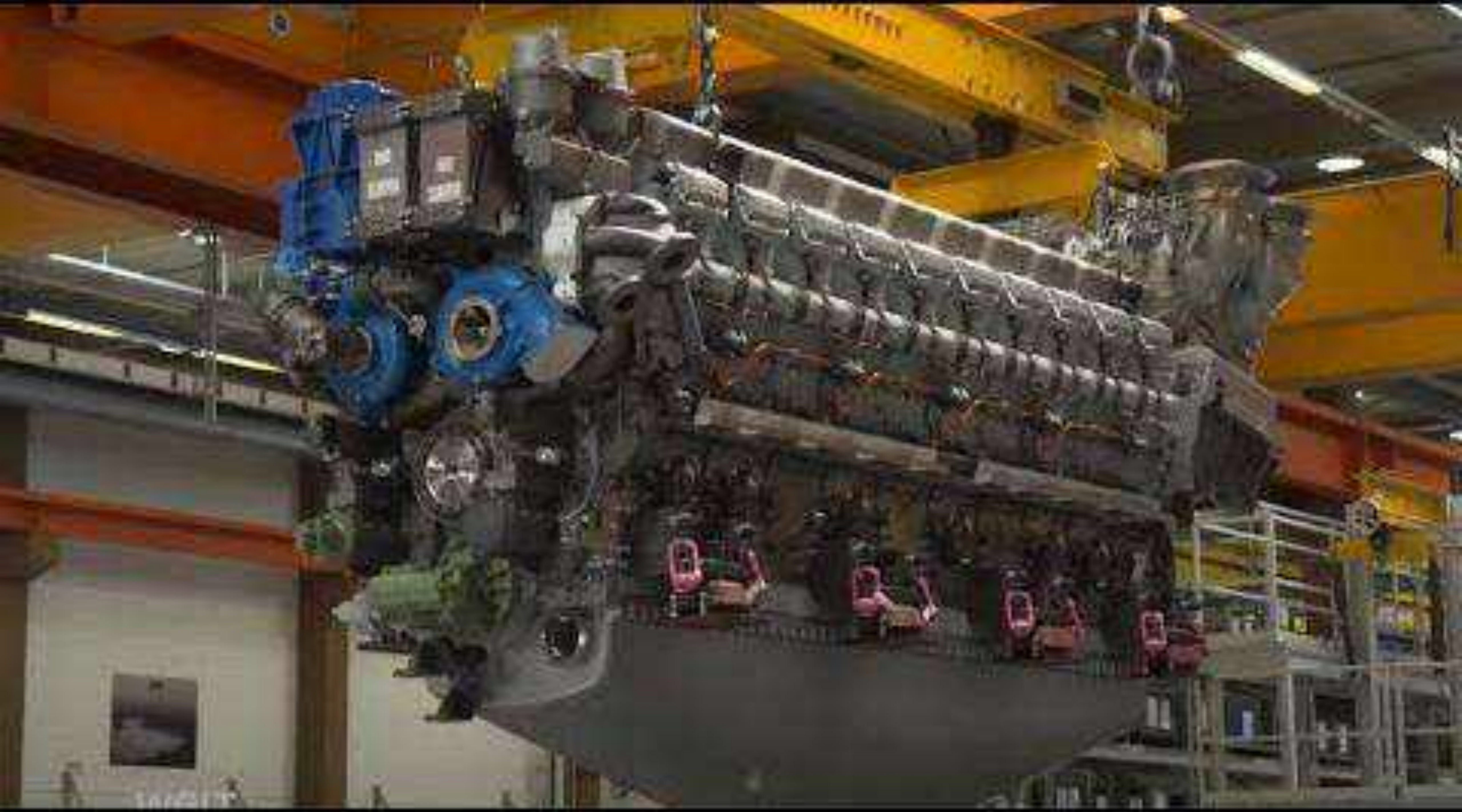
SCR is a pollution reduction process used in diesel engines.
DEF is injected to convert dangerous emissions into cleaner exhaust.





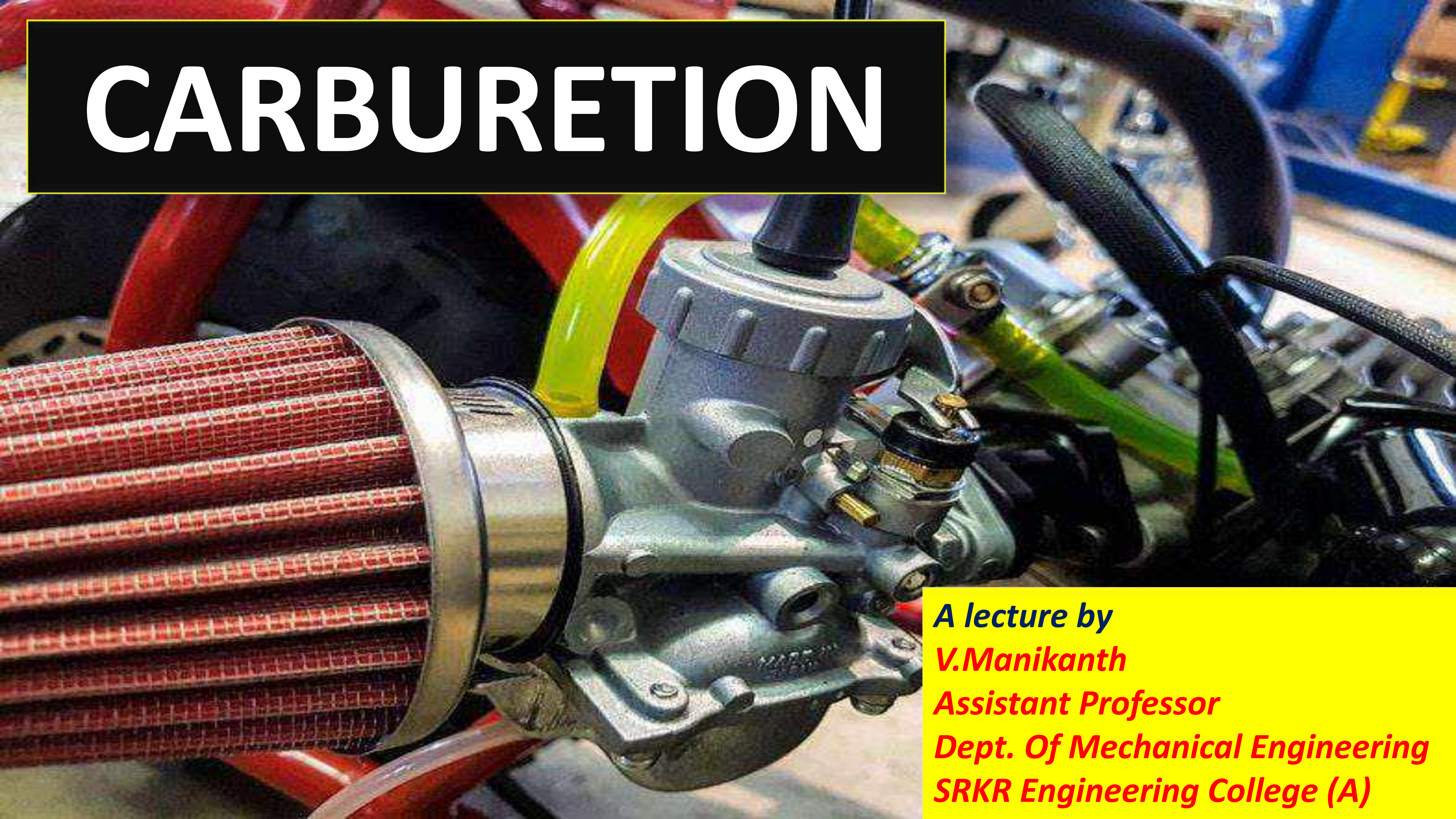
Due to partial combustion, the gases entering inside the catalytic converter consists of a mixture of Carbon Monoxide (CO), Unburned Hydrocarbons (HC) and Oxides of Nitrogen (Nox), which are harmful to the environment.

SOME USEFUL DOCUMENTARIES




CNBC
THE END
OF DIESEL
CARS

CARBURETION



A lecture by
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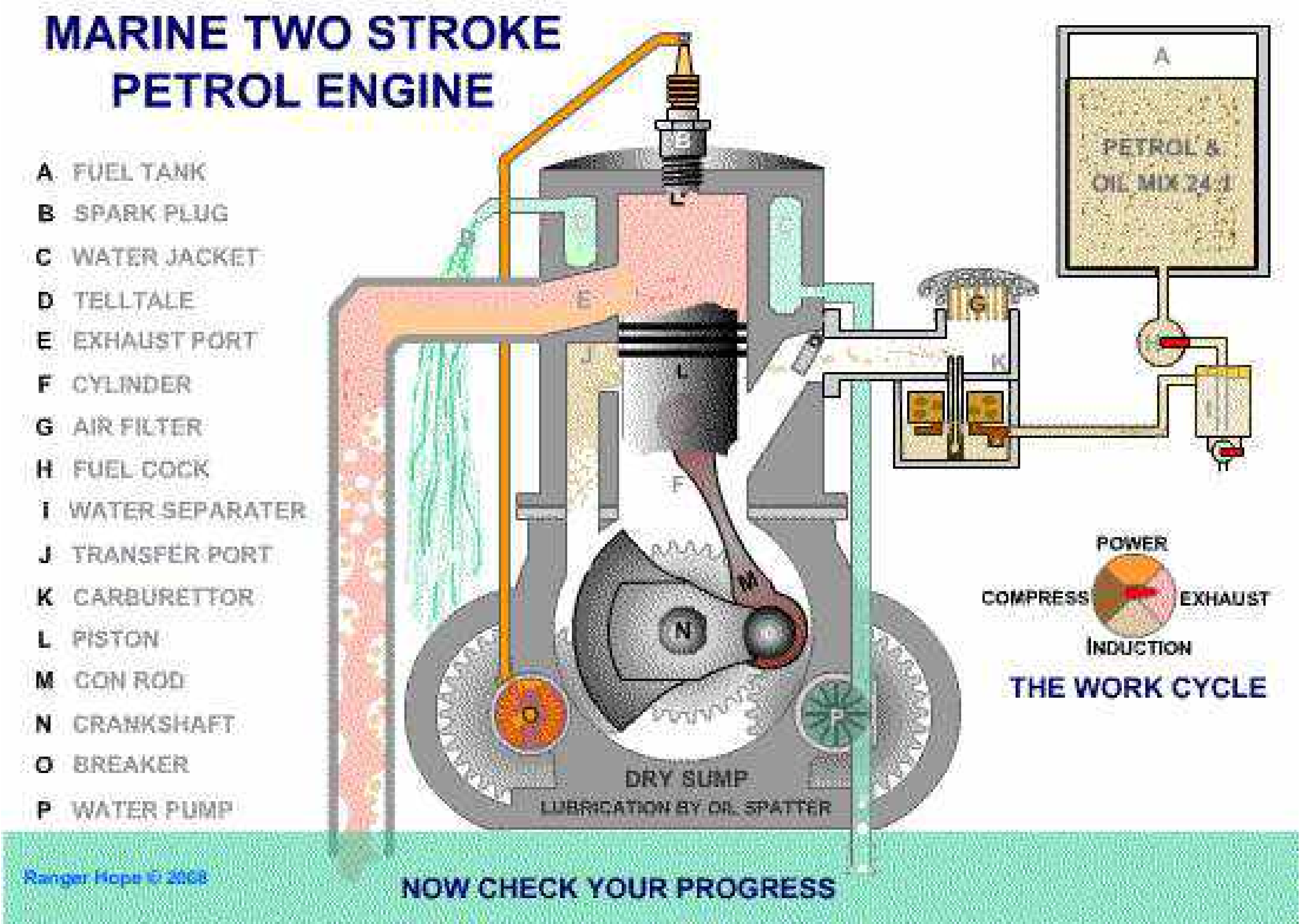

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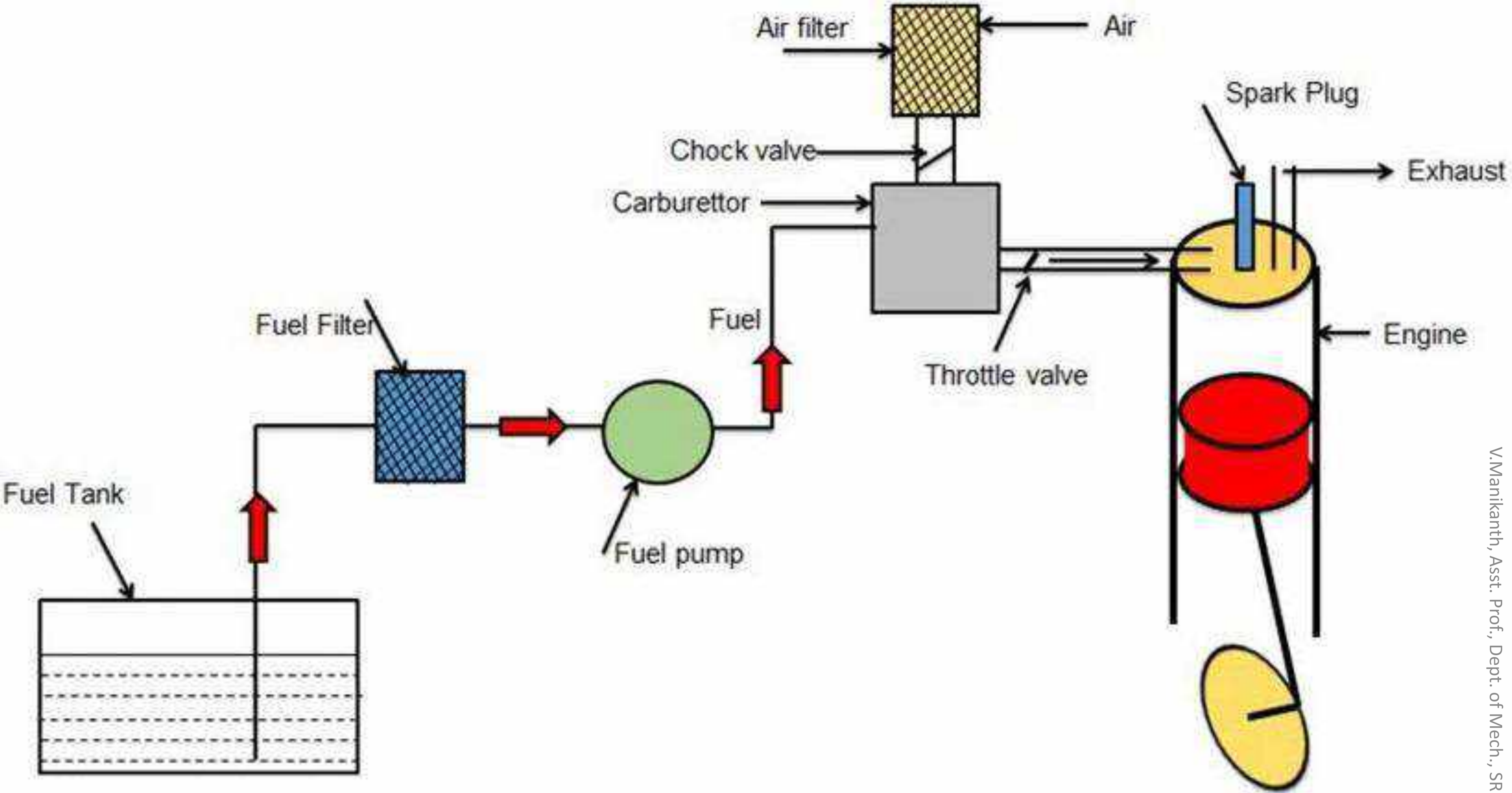

FIG: Fuel supply system in Automobiles

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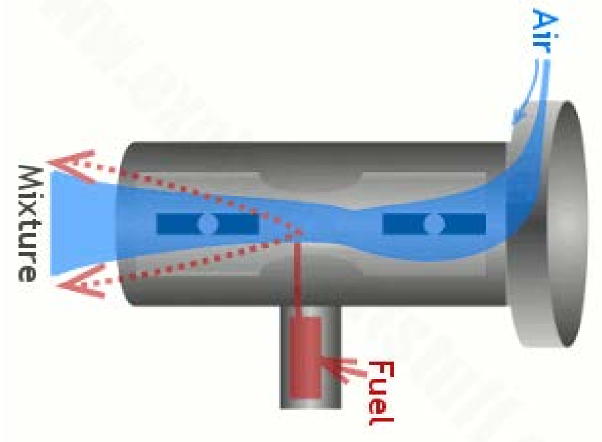
The process of mixture preparation is extremely important for spark-ignition engines. The purpose of carburetion is to provide a combustible mixture of fuel and air in the required quantity and quality for efficient operation of the engine under all conditions

A good carburetor must produce the desired air-fuel mixture ratio and supply the mixture to the engine at all speeds and loads and must do the same automatically

DEFINITION OF CARBURETION:

"The process of formation of a combustible fuel-air mixture by mixing the proper amount of fuel with air before admission to engine cylinder is called carburetion and the device which does this job is called a carburetor."



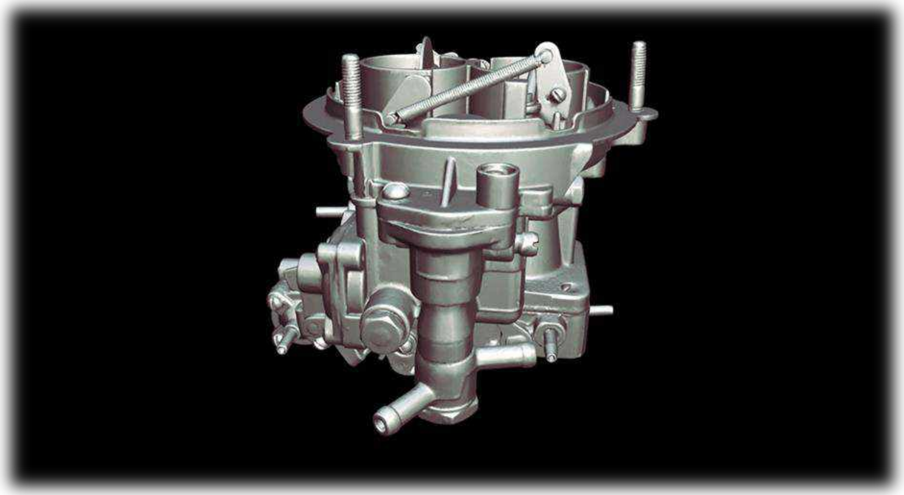
The important requirements of an automobile carburetors.

(i) Ease of starting the engine, particularly under low ambient conditions.

(ii) Ability to give full power quickly after starting the engine.

(iii) Equally good and smooth engine operation at various loads.

(iv) Good and quick acceleration of the engine.



(v) Developing sufficient power at high engine speeds.

(vi) Simple and compact in construction.

(vii) Good fuel economy.

(viii) Absence of racing of the engine under idling conditions.

(ix) Ensuring full torque at low speeds.

FACTORS AFFECTING CARBURETION

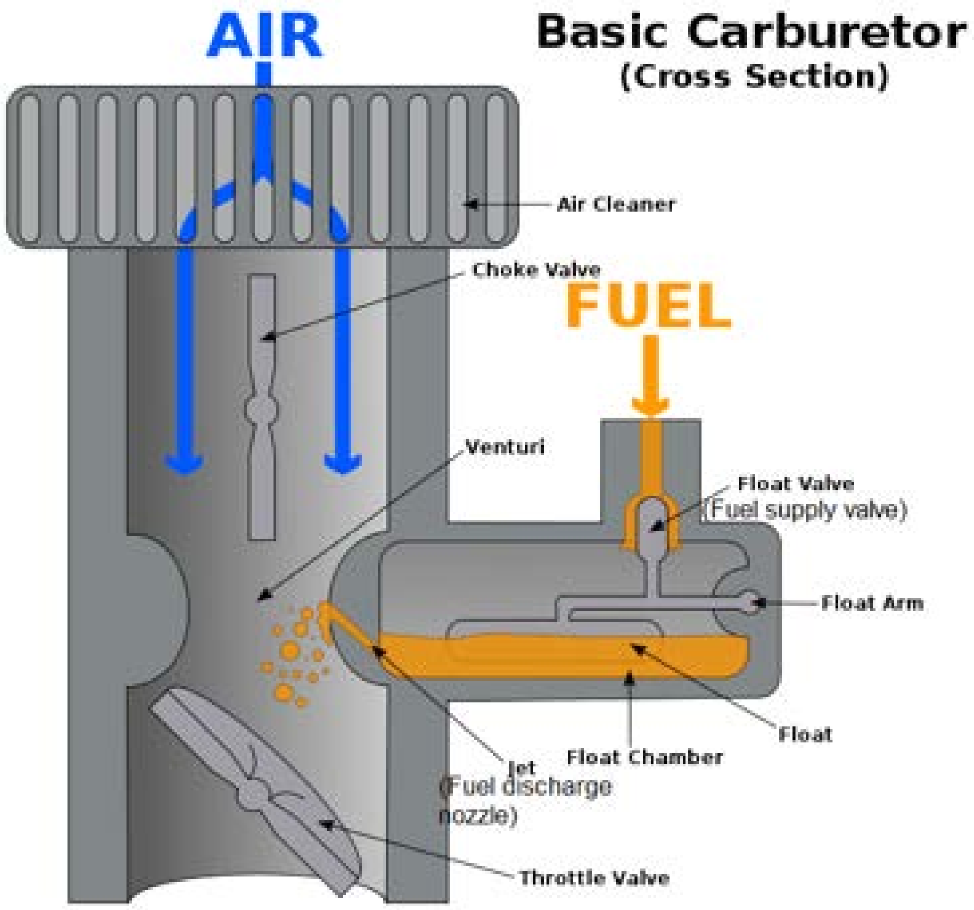
Of the various factors, the process of carburetion is influenced by

(i) the engine speed

(ii) the vaporization characteristics of the fuel

(iii) the temperature of the incoming air and

(iv) the design of the carburetor



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Since modern engines are of high speed type, the time available for mixture formation is very limited. For example, an engine running at 3000 rpm has only about 10 milliseconds (ms) for mixture induction during intake stroke. When the speed becomes 6000 rpm the time available is only 5 ms. Therefore, in order to have high quality carburetion (that is mixture with high vapour content) the velocity of the air stream at the point where the fuel is injected has to be increased.

Suitable evaporation characteristics of the fuel, indicated by its distillation curve, are necessary for efficient carburetion especially at high engine speeds. An increase in atmospheric temperature, however, leads to a decrease in power output of the engine when the air-fuel ratio is constant due to reduced mass flow into the cylinder or, in other words, reduced volumetric efficiency.

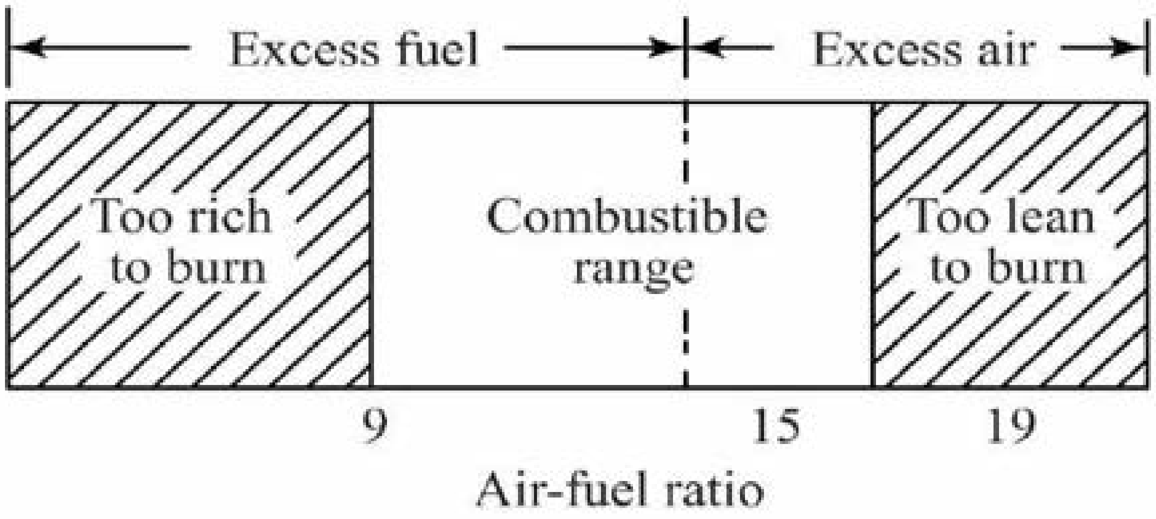
AIR–FUEL MIXTURES:

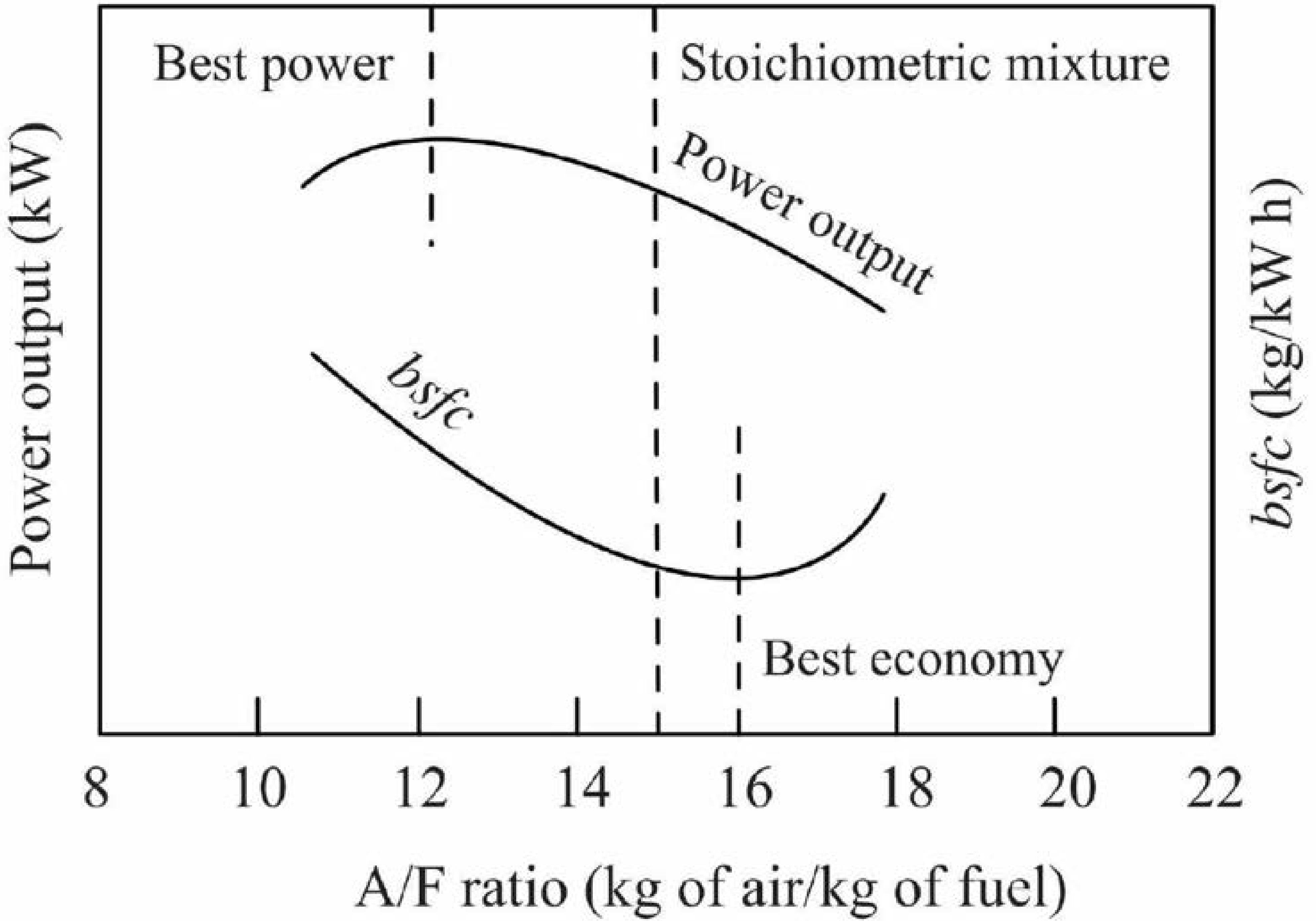
An engine is generally operated at different loads and speeds. For this, proper air-fuel mixture should be supplied to the engine cylinder. Fuel and air are mixed to form three different types of mixtures.

(i) chemically correct mixture (A/F ratio for C_8H_{18} is 15.12:1; usually approximated to 15:1)

(ii) rich mixture (A/F ratio of 12:1, 10:1 etc.)

(iii) lean mixture (A/F ratio of 17:1, 20:1 etc.)




Best power
Stoichiometric mixture
Power output
bsfc
Best economy
Power output (kW)
bsfc (kg/kW h)
8
10
12
14
16
18
20
22
A/F ratio (kg of air/kg of fuel)

AUTOMOTIVE ENGINE AIR–FUEL MIXTURE REQUIREMENTS:

The carburetor must be able to supply the required air-fuel ratio to satisfy these demands.

These ranges are:

(i) Idling (mixture must be enriched)

(ii) Cruising (mixture must be leaned)

(iii) High Power (mixture must be enricheda)

How the Air/Fuel Ratio Changes

Engine Operation	Air/Fuel Ratio
Cold Start	Extra Rich
Cold Idle	Rich
Warm Idle	Balanced
Light Acceleration	Slightly Rich
Medium Acceleration	Rich
Hard Acceleration	Extra Rich
Cruise Light Load	Balanced to Lean
Cruise Moderate Load	Balanced to Rich
Cruise Heavy Load	Rich
Coasting	Lean to Very Lean
Decelerating	Very Lean to Fuel Off

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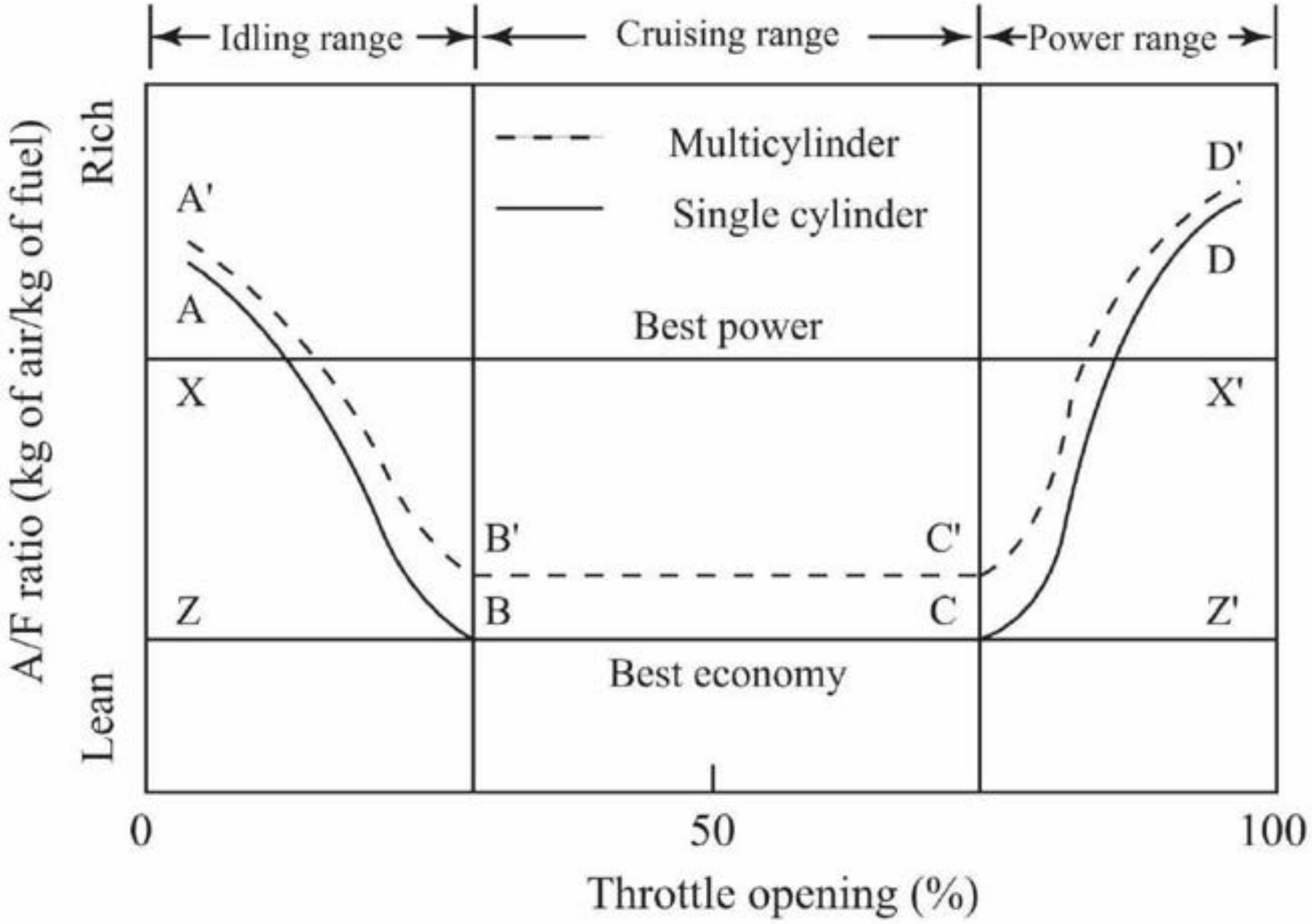


Anticipated carburetor performance to fulfill engine requirements

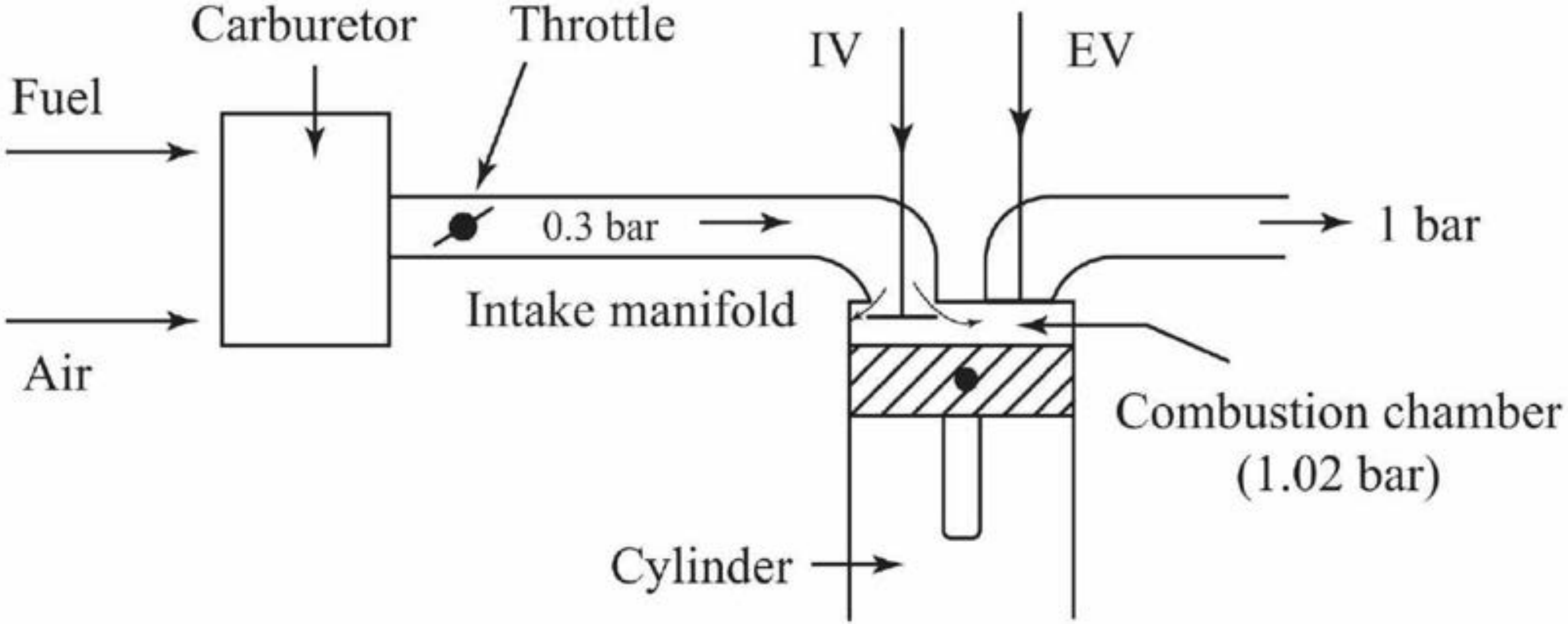


Fig. Schematic diagram of combustion chamber and induction system at the start of intake stroke

Idling Range:

An idling engine is one which operates at no load and with nearly closed throttle. Under idling conditions, the engine requires a rich mixture, as indicated by point A in Fig. This is due to the existing pressure conditions within the combustion chamber and the intake manifold which cause exhaust gas dilution of the fresh charge.

with nearly closed throttle the pressure in the intake manifold is considerably below atmospheric due to restriction to the air flow. When the intake valve opens, the pressure differential between the combustion chamber and the intake manifold results in initial backward flow of exhaust gases into the intake manifold.

Cruising Range:

In the cruising range from B to C (*see Fig*.), the exhaust gas dilution problem is relatively insignificant. The primary interest lies in obtaining the maximum fuel economy. Consequently, in this range, it is desirable that the carburetor provides the engine with the best economy mixture.

Power Range:

During peak power operation the engine requires a richer mixture, as indicated by the line CD (*see Fig*.), for the following reasons

(i) To provide best power

(ii) To prevent overheating of exhaust valve and the area near it.

THE SIMPLE CARBURETOR

ESSENTIAL PARTS OF A CARBURETOR

A carburetor consists essentially of the following parts, viz.

(i) fuel strainer

(ii) float chamber

(iii) main fuel metering and idling nozzles

(iv) choke and throttle

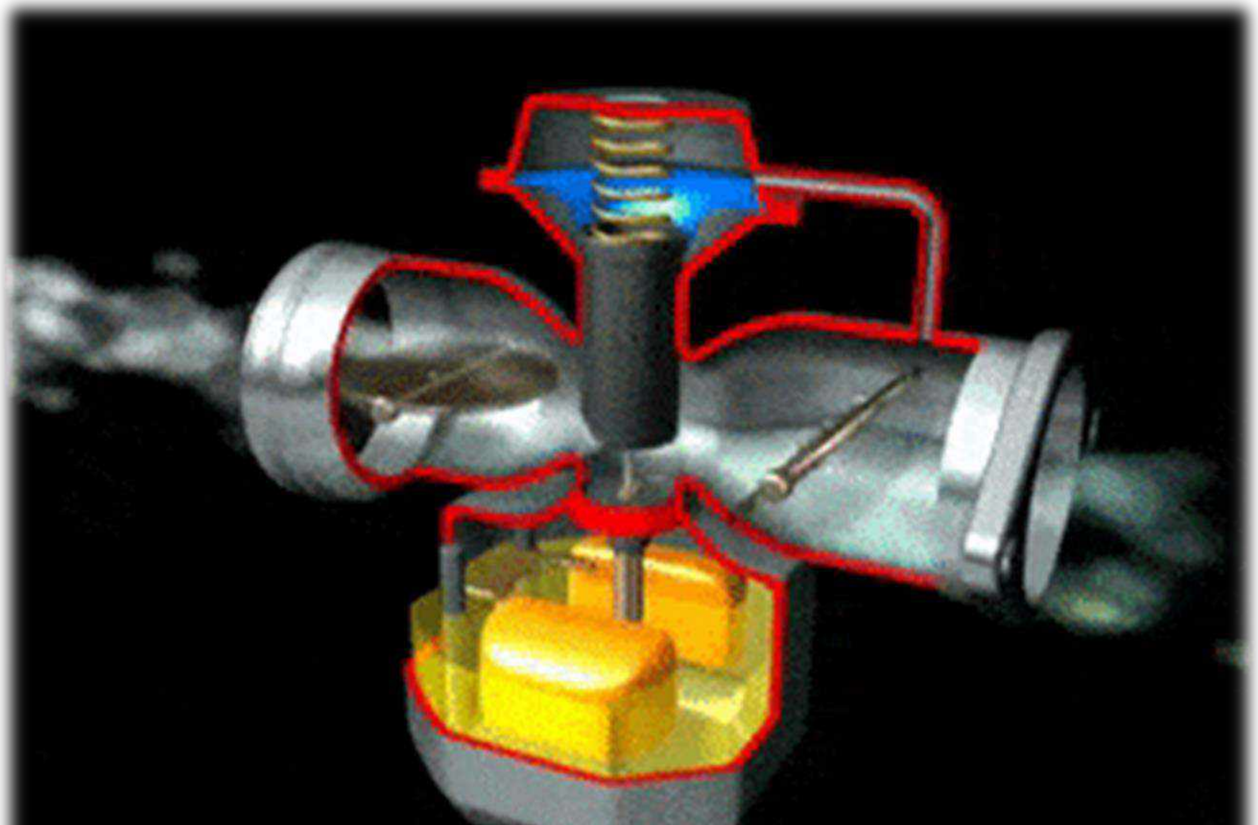


Fig: Simple Carburetor

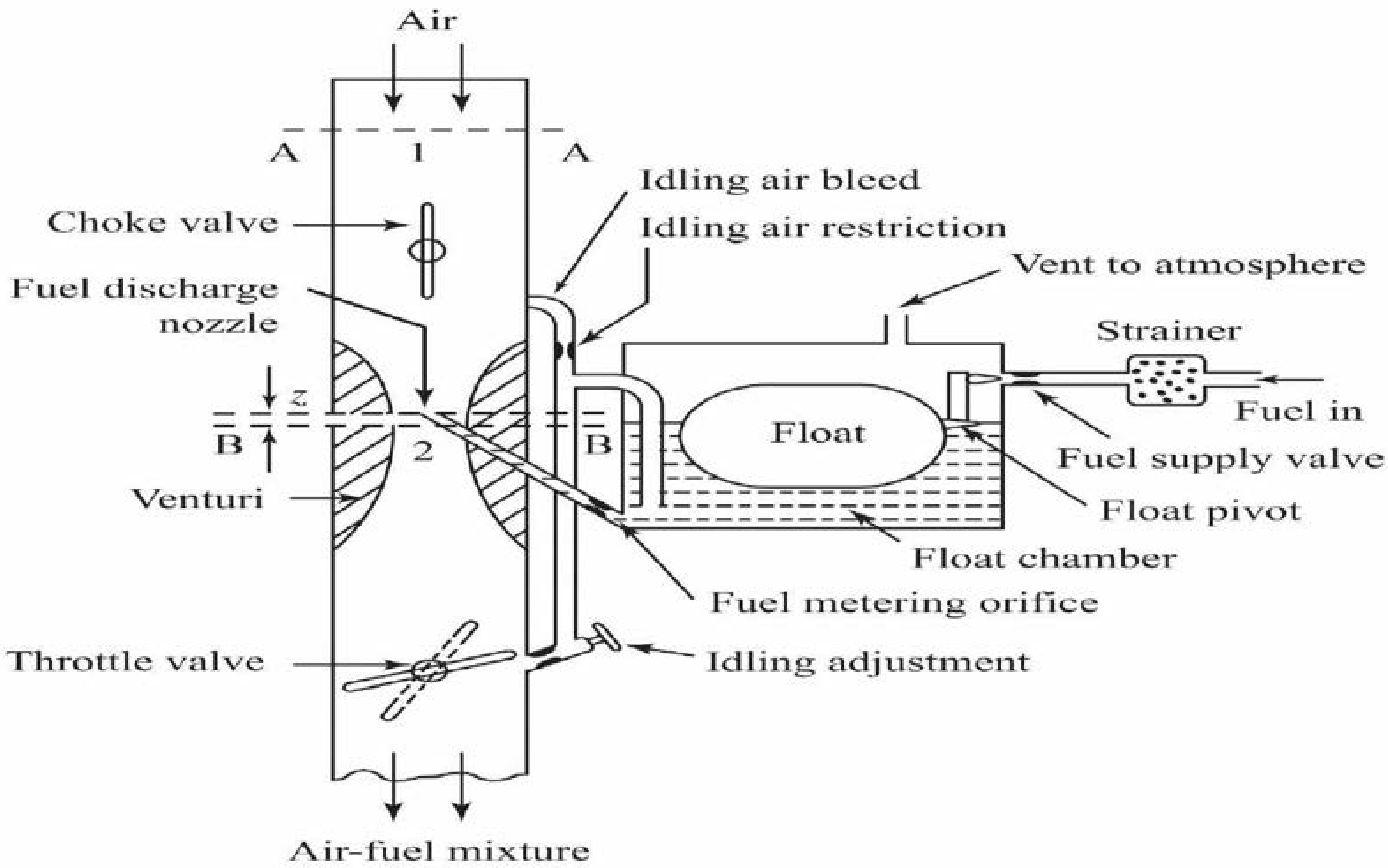
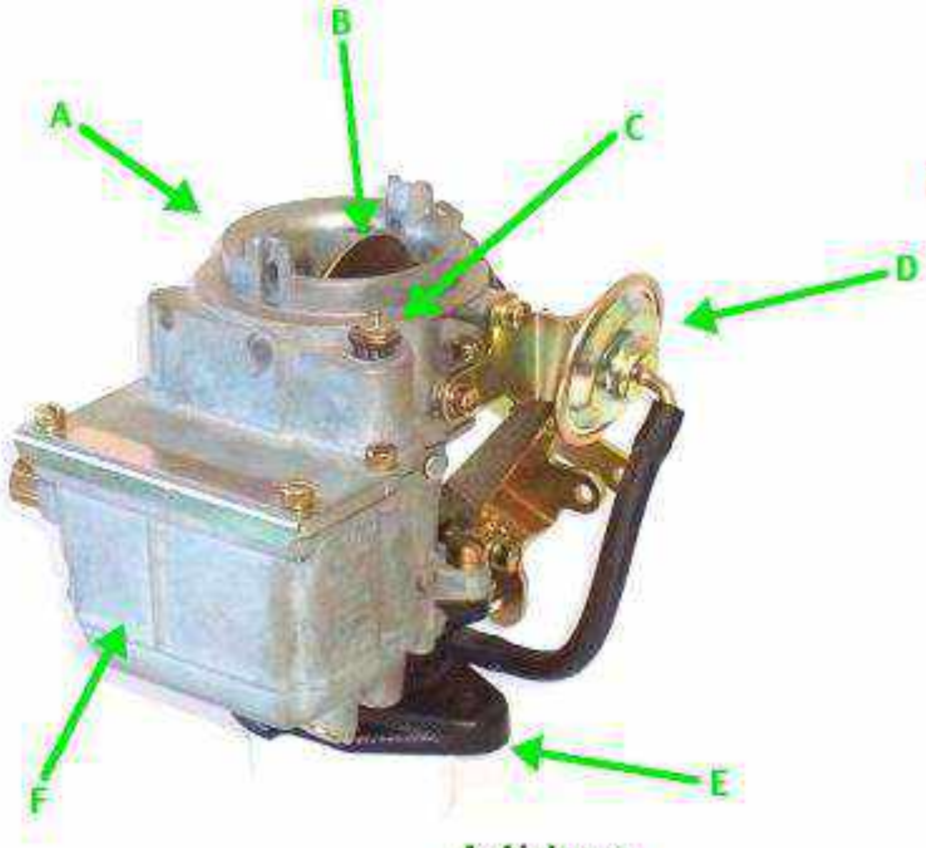
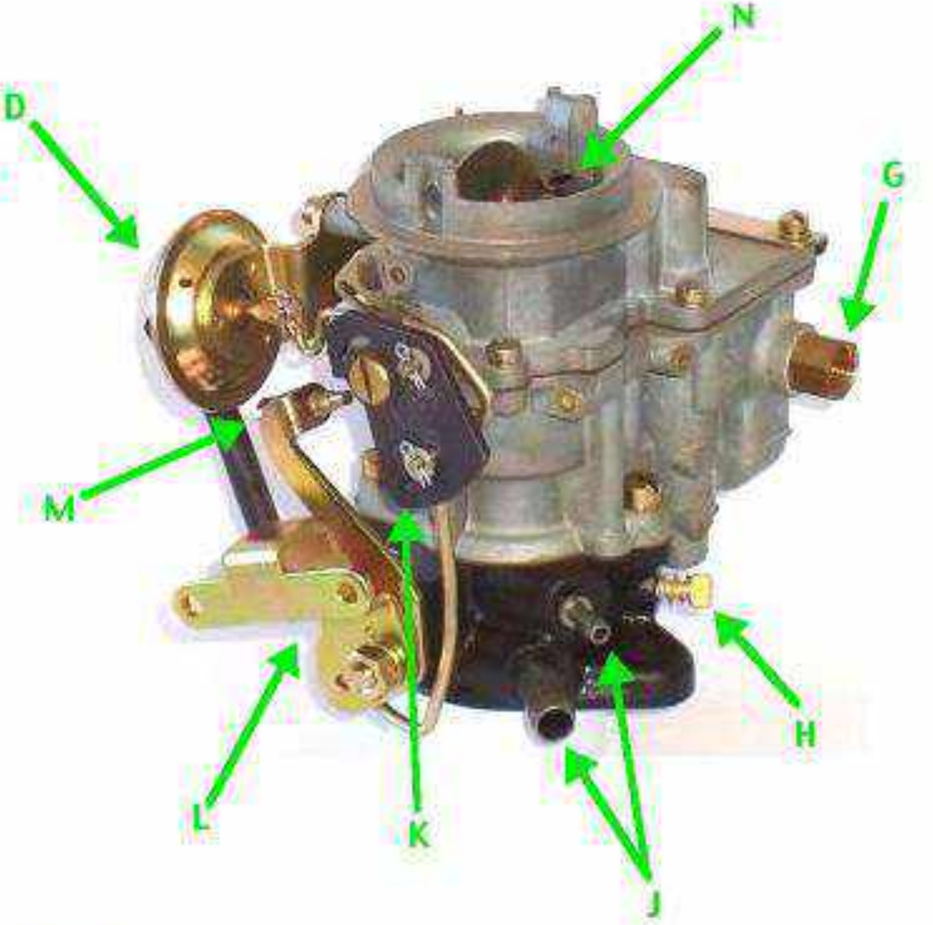


Fig. Simple carburetor


WORKING OF CARBURETOR
Animation
Simplest way





A: Airhorn
B: Choke plate
C: External bowl vent
D: Choke pull-off
E: Throttle body & mounting base
F: Float bowl
G: Fuel inlet
H: Idle mixture adjusting screw
J: Vacuum nipples
K: Fast-idle cam
L: Throttle lever
M: Idle speed adjusting crackscrew
N: Internal bowl vent

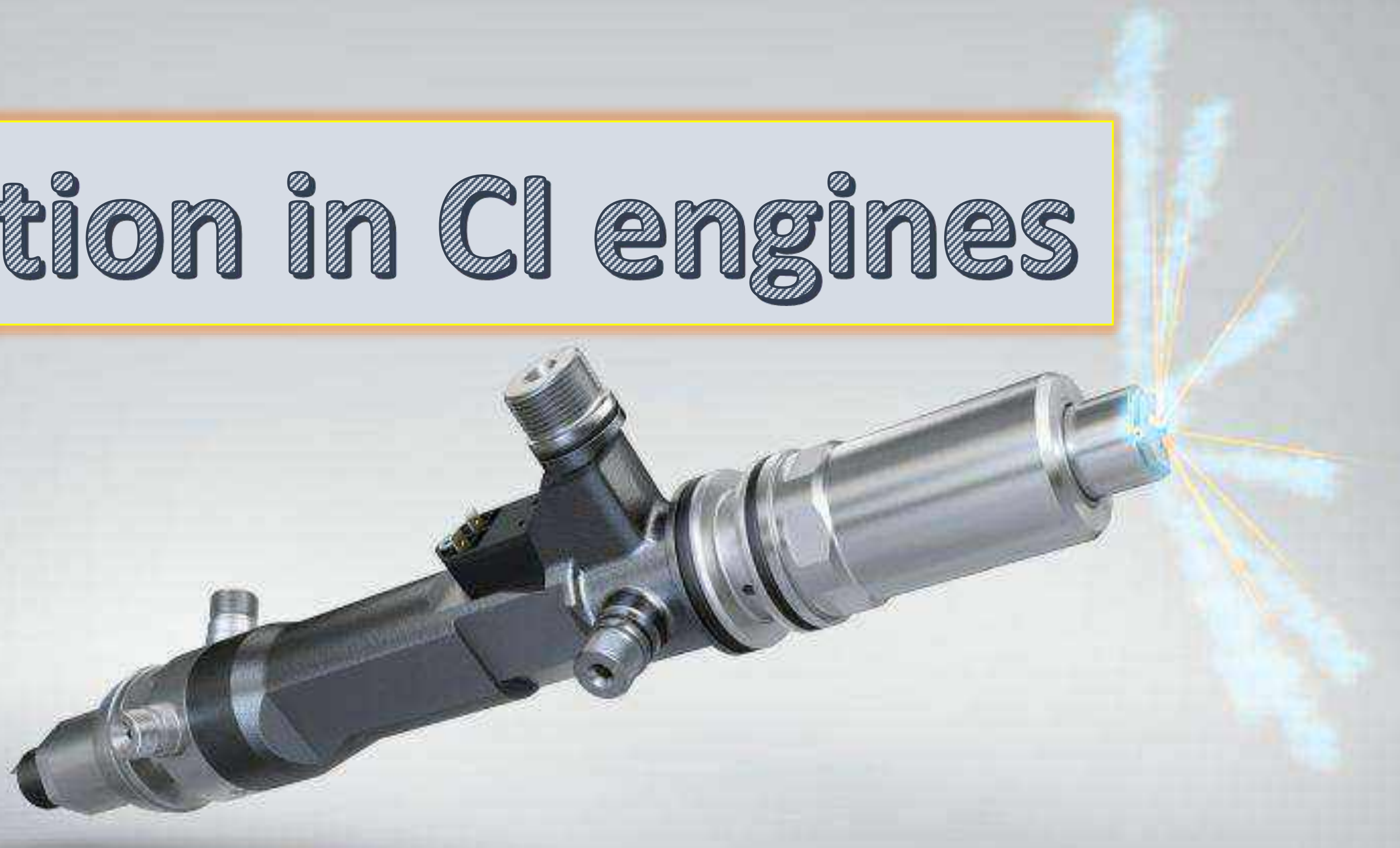
Advantages of Carburetor Feeding Engine

- It’s a simple and low cost fuel feeding system both for two stroke & four stroke motorcycle engine.
- Simplicity and being mechanical its maintenance and repair is possible and quite easy.
- It can be tweaked easily according to user need and environment condition.
- Being mechanical device it responds uniquely every possible position & action of throttle.
- Frequent respond against revving and over raving is very common character and advantage of a carburetor feeding system. That’s why it very widely suitable for off road and dirt bikes.
- Fuel contamination issue can be over looked in carbureted engine though it hampers the performance.
- Very much suitable fuel feeding system for low cost & low capacity motorcycle engines.

Disadvantages of Carburetor Feeding Engine

- Its fuel feeding quantity is not précised as it allows feeding flow according to the suction speed & quantity of the air by the combustion chamber.
- Fuel economy is considerably very low in carbureted engine.
- In carburetor fuel feeding system the engine cold start is a big issue.
- Lean & rich mixture often becomes hassle in carbureted engine.
- Due to inefficient combustion the emission is significantly high in carbureted engine.
- In some case engine gets vibration and spark plug fouling issue is very common.

Fuel Injection in CI engines



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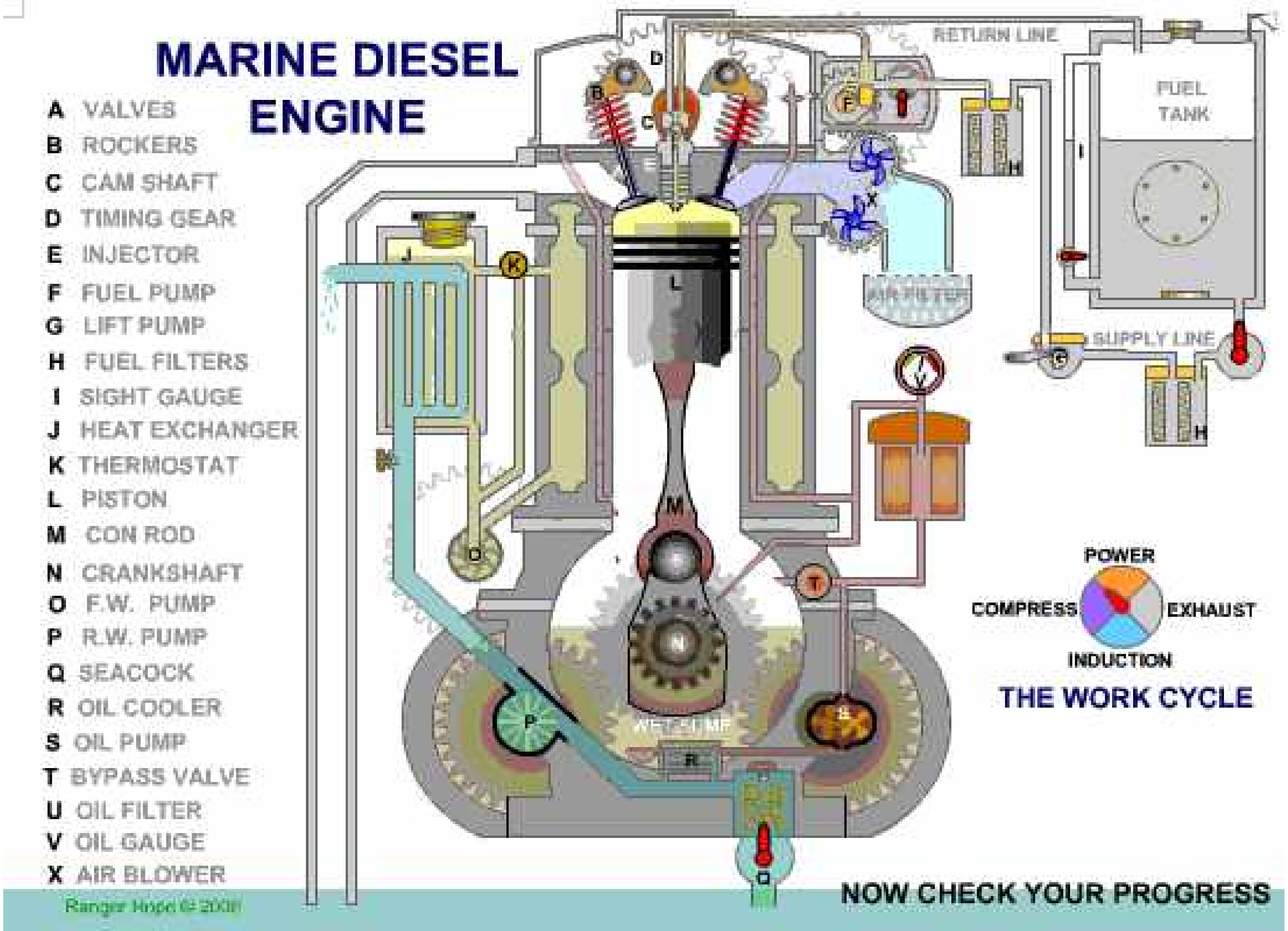

MARINE DIESEL ENGINE
A VALVES
B ROCKERS
C CAM SHAFT
D TIMING GEAR
E INJECTOR
F FUEL PUMP
G LIFT PUMP
H FUEL FILTERS
I SIGHT GAUGE
J HEAT EXCHANGER
K THERMOSTAT
L PISTON
M CON ROD
N CRANKSHAFT
O F.W. PUMP
P R.W. PUMP
Q SEACOCK
R OIL COOLER
S OIL PUMP
T BYPASS VALVE
U OIL FILTER
V OIL GAUGE
X AIR BLOWER
RETURN LINE
FUEL TANK
SUPPLY LINE
AIR FILTER
POWER
COMPRESS
EXHAUST
INDUCTION
THE WORK CYCLE
NOW CHECK YOUR PROGRESS

The performance of diesel engines is heavily influenced by their injection system design. In fact, the most notable advances achieved in diesel engines resulted directly from superior fuel injection system designs. While the main purpose of the system is to deliver fuel to the cylinders of a diesel engine, it is how that fuel is delivered that makes the difference in engine performance, emissions, and noise characteristics.

Unlike its spark-ignited engine counterpart, the diesel fuel injection system delivers fuel under extremely high injection pressures. This implies that the system component designs and materials should be selected to withstand higher stresses in order to perform for extended durations that match the engine's durability targets.

The main purpose of the fuel injection system is to deliver fuel into the cylinders of an engine. In order for the engine to effectively make use of this fuel:

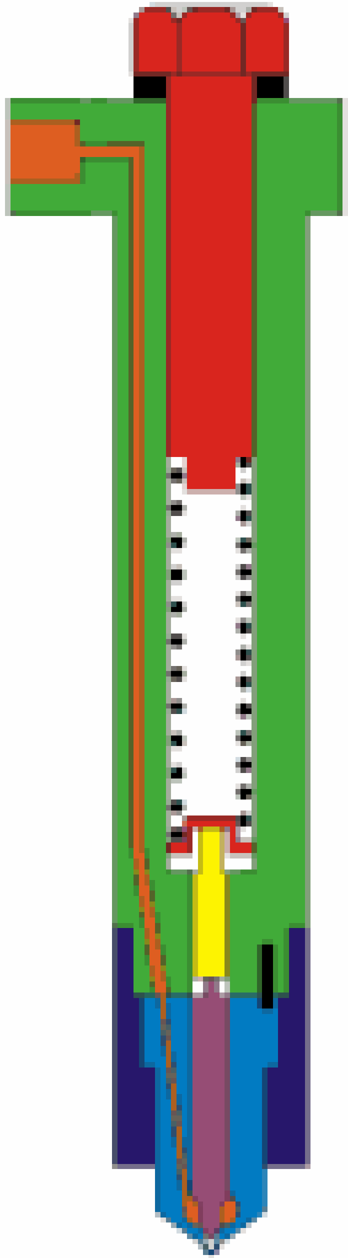
- Fuel must be injected at the proper time, that is, the injection timing must be controlled and
- The correct amount of fuel must be delivered to meet power requirement, that is, injection metering must be controlled.

Figure 1. Main Functions of Diesel Fuel Injection System

The fuel supply system of a diesel engine (CI Engine) consists of ***1. fuel tank, 2. fuel filter, 3. injection pump, 4. injector, 5. fuel lines for necessary connections, and 6. fuel gauge.*** In petrol engines, carburetor and spark plug are used. In Diesel engine, fuel injector is used. Remaining elements are same for both types of engines. The fuel from the fuel tank flows to pump. It then passes out to the inlet side of the main fuel filter. Then filtered fuel proceeds to the inlet side of the fuel injection pump. From the injection pump it flows under pressure in the feed pipes leading to the fuel injectors.

Principle of Working:

It consists of a needle valve fitted on its seat in the nozzle body by a spindle. The spring controls the pressure on the spindle by which the needle valve opens. The nozzle is attached to the body by means of a cap nut. The fuel enters the nozzle through holes in the injector body. When the needle valve is raised from its seat by the pressure of the fuel, the injection of the fuel into the combustion chamber takes place. When the injection pressure falls below the spring pressure, the valve closes.



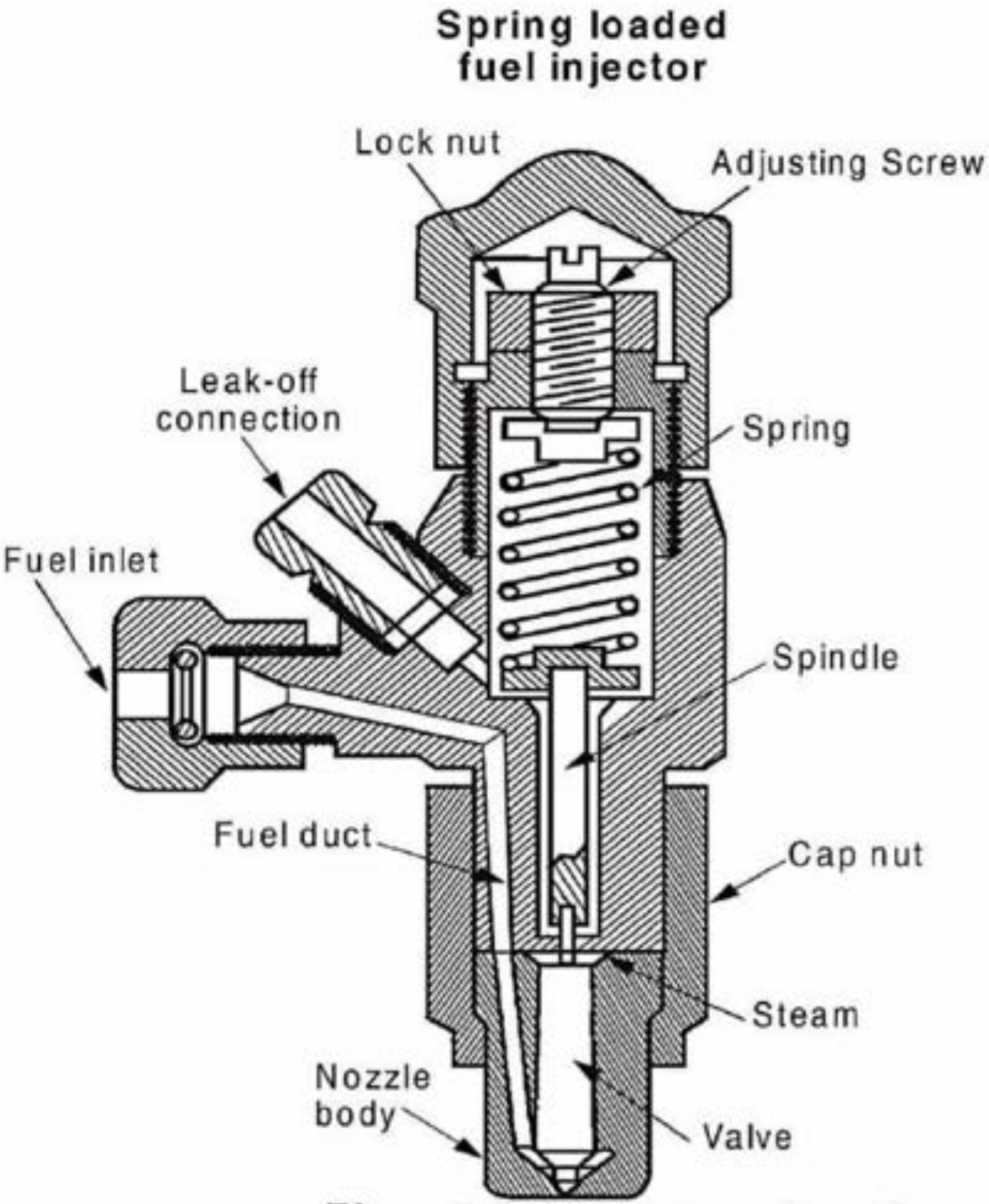


Fig. *Fuel Injecter or Atomiser*

Advantages and Disadvantages of Fuel Injection System

Below are the benefits of fuel injection system:

- Precise fuel mixture of fuel and air ensures maximum possible fuel efficiency and power production.
- Combustion process is significantly much more efficient in fuel injected engine.
- Fuel injection engines are more economical and it maximize and minimize the emission level.
- Cold starting is eliminated in fuel injected engine, making no need for manual chocking.
- It’s also used on modern performance motorcycles.
- Fuel injection system automatically balance the air-fuel mixture considering environment situation.
- Engine vibration is reduced and spark plug fouling issue is minimized.

Disadvantages:

Despite all the benefits of injection system some limitation still occurs. Below are the disadvantages of the system:

- It is a complex electronic controlled device that work with few electronic sensors.
- Maintenance and repairing of the system is very limited. That is, not all workshop can do its working.
- Fuel injection system is quite expensive.
- Good quality material and fuel is highly recommended.
- There is no solution for low cost and low capacity.

Carburetor Vs. Fuel Injection Engine in Bike: Which is Better

You might be wondering which is better when it comes to comparing carburetor vs. fuel injection in bikes. Below are some of the differences between the two:

Versatility: Carburetors are phased out since they are comparatively more pollutant compared to FI systems.

Performance: The ECU is constantly working on complex calculations to offer the best performance of the engine. The carburetors struggle when it comes to ever-changing fuel temperature and air pressure.

Mileage: FI system provides an accurate measurement of fuel and air resulting in higher performance of the engine which leads to better fuel management and mileage. The ECU can be mapped for a higher power ratio if you require.

Maintenance: This is the only category where the carburetor is better compared to the fuel injection system. Carburetors can easily be repaired or replaced while FI systems require professional help which can lead to higher costs.

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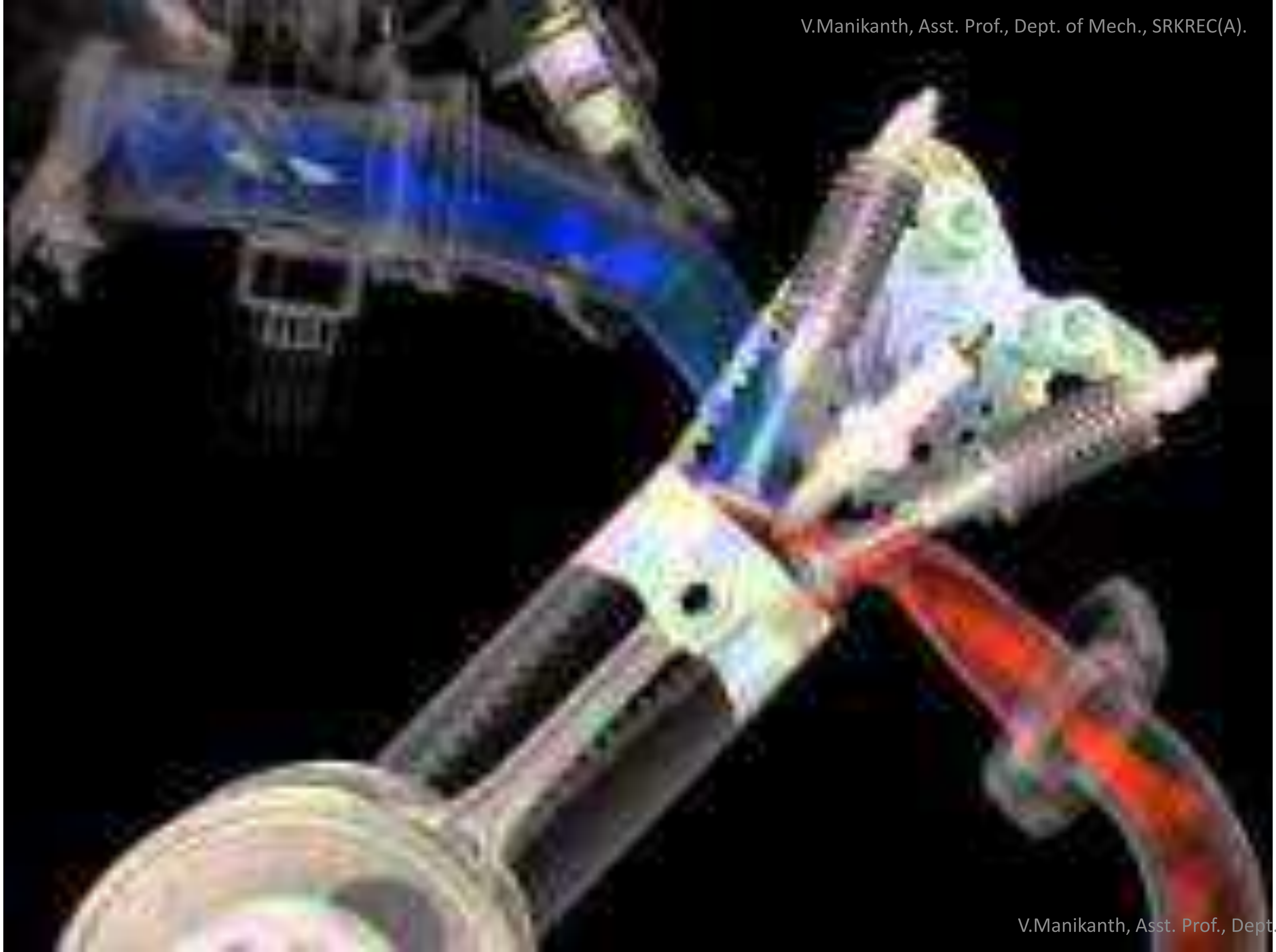

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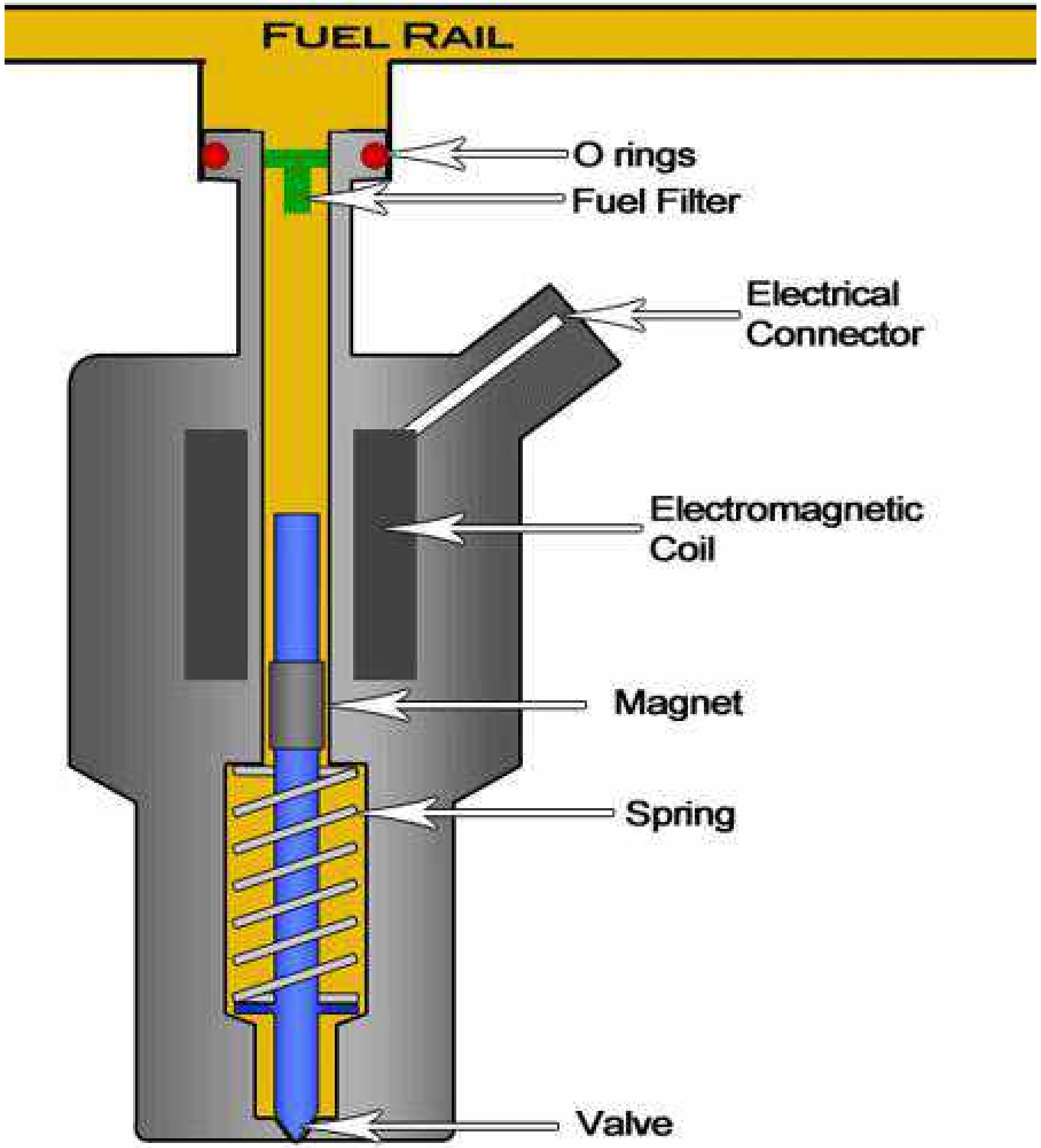

Fig: Working of electronic fuel injector