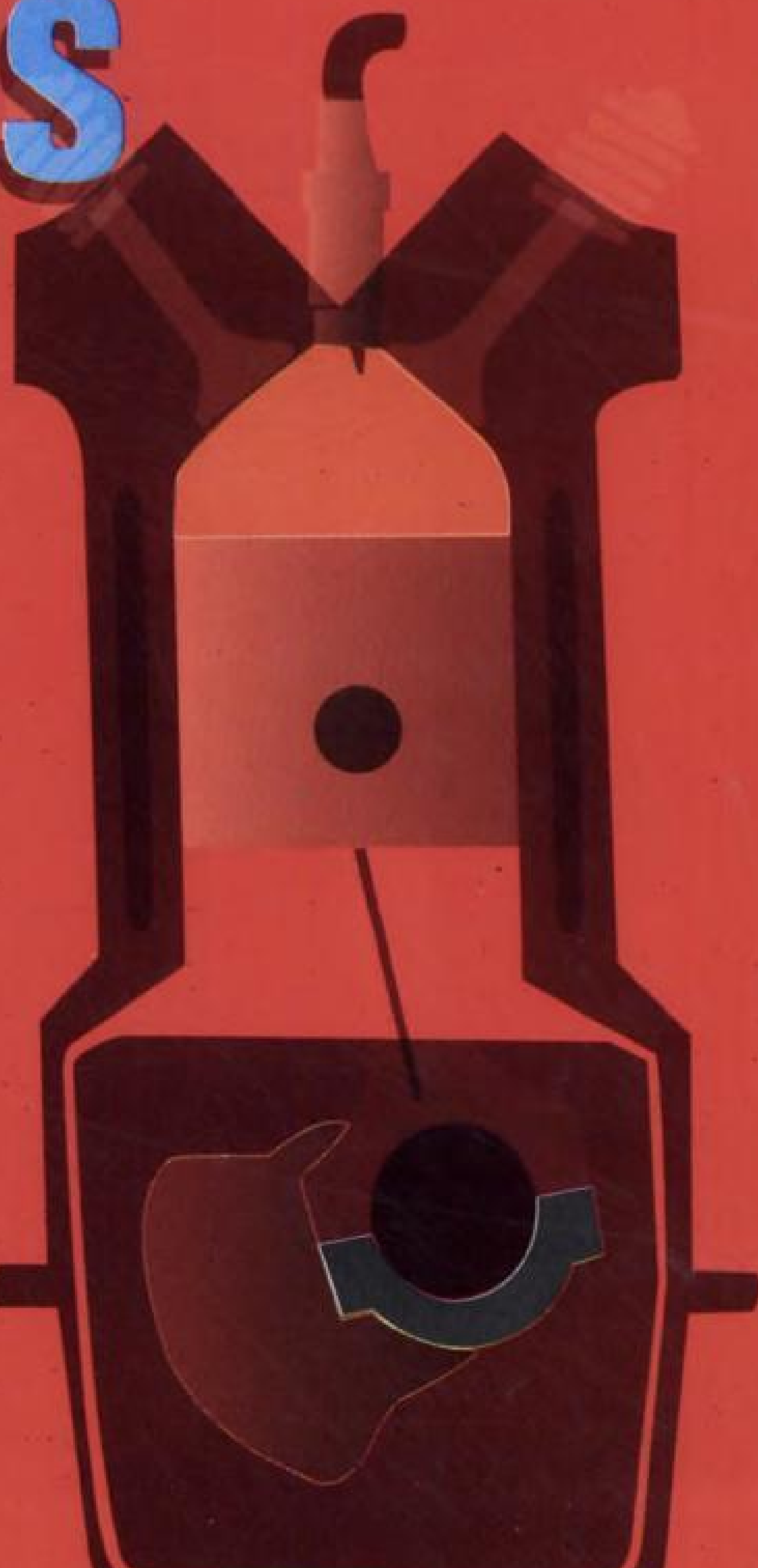
INTERNAL COMBUSTION ENGINES

Second Edition



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CONTENTS

NOMENCLATURE

A

a_1	constant
$amep$	mean effective pressure required to drive the auxiliary components
A	piston area [Chp.1]
A	area of heat transfer [Chp.14]
A	average projected area of each particles [Chp.15]
A	TEL in ml/gal of fuel [Chp.6]
A_1	cross-sectional area at inlet of the carburettor
A_2	cross-sectional area at venturi of the carburettor
A_{act}	actual amount of air in kg for combustion per kg of fuel
A_f	area of cross-section of the fuel nozzle [Chp.8]
A_f	area of fin [Chp.14]
A_e	effective area
A_{th}	theoretical amount of air in kg per kg of fuel
A/F	air-fuel ratio

B

b_1	constant
bp	brake power
bhp	brake horsepower
$bmep$	brake mean effective pressure
$bsfc$	brake specific fuel consumption
BDC	Bottom Dead Centre

C

$cmep$	mean effective pressure required to drive the compressor or scavenging pump
C	velocity [Chp.6]

C_d	coefficient of discharge for the orifice [Chp.9]
C_{da}	coefficient of discharge for the venturi
C_{df}	coefficient of discharge for fuel nozzle
C_f	fuel velocity at the nozzle exit
C_p	specific heat of gas at constant pressure
C_{rel}	relative charge
C_v	specific heat at constant volume
CV	calorific value of the fuel

D

d	cylinder bore diameter [Chp.1]
d	diameter of orifice [Chp.9]
D	brake drum diameter

E

ϵ	expansion ratio
E	stored energy [Chp.2]
E	enrichment [Chp.8]
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening

F

f	coefficient of friction
$fmep$	frictional mean effective pressure
fp	frictional power
F	force
F/A	fuel-air ratio
F_R	relative fuel-air ratio

G

g	acceleration due to gravity
g_c	gravitational constant
gp	gross power

H

h	specific enthalpy
h	pressure difference [Chp.9]
h	convective heat transfer coefficient [Chp.14]
H	enthalpy

I

ip	indicated power
$imep$	indicated mean effective pressure
$isfc$	indicated specific fuel consumption
I	intensity
IDC	Inner Dead Centre
IVC	Inlet Valve Closing
IVO	Inlet Valve Opening

K

k	thermal conductivity of gases
k_1	constant [Chp.4]
K	number of cylinders
K_{ac}	optical absorption coefficient

L

l	characteristic length
l	distance [Chp.16]
L	stroke
L_ℓ	length of the light path

M

m	mass
m	exponent [Chp.14]
mep	mean effective pressure
$mmep$	mechanical mean effective pressure
$\dot{m}_a$	mass flow rate of air
$\dot{m}_{act}$	actual mass flow rate of air
$\dot{m}_{th}$	theoretical mass flow rate of air
M_{del}	mass of fresh air delivered
M_f	molecular weight of the fuel
M	molecular weight
M_{ref}	reference mass

N

n	number of power strokes
n	polytropic index [Chp.2]
n	number of soot particles per unit volume [Chp.16]
N	speed in revolutions per minute
N_i	number of injections per minute [Chp.9]

O

ODC	Outer Dead Centre
ON	Octane Numbers

P

p	pressure
$pmep$	charging mean effective pressure
pp	pumping power
p_{ar}	pure air ratio
p_{bm}	brake mean effective pressure
p_e	exhaust pressure
p_i	inlet pressure
p_{im}	indicated mean effective pressure
p_m	mean effective pressure
P_{cyl}	pressure of charge inside the cylinder
P_{inj}	fuel pressure at the inlet to injector
P_l	pressure loss coefficient
P_s	specific power output
PN	performance number

Q

q	heat transfer
$\dot{q}$	rate of heat transfer
Q_R	heat rejected
Q_S	heat supplied

R

r	compression ratio
rpn	relative performance number
r_c	cut-off ratio
r_p	pressure ratio
R	length of the moment arm
R	delivery ratio [Chp.20]
$\overline{R}$	universal gas constant
R_{del}	delivery ratio

S

$\overline{s}_p$	mean piston speed
sfc	specific fuel consumption
S	spring scale reading

T

t	time
T	absolute temperature
T	torque [Chp.17]
TDC	Top Dead Centre
T_b	black body temperature
T_f	friction torque
T_g	mean gas temperature
T_l	load torque

U

u	specific internal energy
U	internal energy
U_c	chemical energy
U_s	stored energy

V

v	specific volume
V	volume
V_{ch}	volume of cylinder charge
V_{cp}	volume of combustion products
V_{del}	volume of air delivered
V_f	fuel jet velocity
V_{pure}	volume of pure air
V_{ref}	reference volume
V_{res}	volume of residual gas
V_{ret}	volume of retained air or mixture
V_s	displacement volume
V_s	swept volume
V_{short}	short circuiting air
V_{tot}	total volume
V_C	clearance volume
V_T	volume at bottom dead centre

W

w	specific weight
w	work transfer [Chp.8]
W	net work
W	weight [Chp.16]
W	number of quartz windows [Chp.16]
W	load [Chp.13]

WOT	Wide Open Throttle
W_C	compressor work
W_T	turbine work
W_x	external work

Z

z	height of the nozzle exit [Chp.8]
z_2	datum height [Chp.2]
Z	constant

GREEK

α	air coefficient
γ	ratio of specific heats
Δp	pressure difference
ΔT	temperature difference between the gas and the wall
ϵ	heat exchanger efficiency
η	efficiency
$\eta_{air\ std}$	air standard efficiency
η_{bth}	brake thermal efficiency
η_c	compressor efficiency
η_{ch}	charging efficiency
η_{ith}	indicated thermal efficiency
η_m	mechanical efficiency
η_{rel}	relative efficiency
η_{sc}	scavenging efficiency
η_t	turbine efficiency
η_{th}	thermal efficiency
η_{trap}	trapping efficiency
η_v	volumetric efficiency
θ	crank angle [Chp.12]
θ	specific absorbance per particle [Chp.16]
λ	wave length [Chp.16]
λ	excess air factor [Chp.20]
μ	kinematic viscosity of gases
ν	dynamic viscosity
ρ	density
ρ_f	density of fuel
ϕ	equivalence ratio
ψ	magnetic field strength
ω	angular velocity

1

INTRODUCTION

1.1 ENERGY CONVERSION

The distinctive feature of our civilization today, one that makes it different from all others, is the wide use of mechanical power. At one time, the primary source of power for the work of peace or war was chiefly man's muscles. Later, animals were trained to help and afterwards the wind and the running stream were harnessed. But, the great step was taken in this direction when man learned the art of energy conversion from one form to another. The machine which does this job of energy conversion is called an engine.

1.1.1 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'.

1.1.2 Definition of 'Heat Engine'

Heat engine is a device which transforms the chemical energy of a fuel into thermal energy and utilizes this thermal energy to perform useful work. Thus, thermal energy is converted to mechanical energy in a heat engine.

Heat engines can be broadly classified into two categories:

(i) Internal Combustion Engines (IC Engines)

(ii) External Combustion Engines (EC Engines)

1.1.3 Classification and Some Basic Details of Heat Engines

Engines whether Internal Combustion or External Combustion are of two types, viz.,

(i) Rotary engines

(ii) Reciprocating engines

A detailed classification of heat engines is given in Fig.1.1. Of the various types of heat engines, the most widely used ones are the reciprocating internal combustion engine, the gas turbine and the steam turbine. The steam engine is rarely used nowadays. The reciprocating internal combustion engine enjoys some advantages over the steam turbine due to the absence of heat exchangers in the passage of the working fluid (boilers and condensers in steam turbine plant). This results in a considerable mechanical simplicity and improved power plant efficiency of the internal combustion engine.

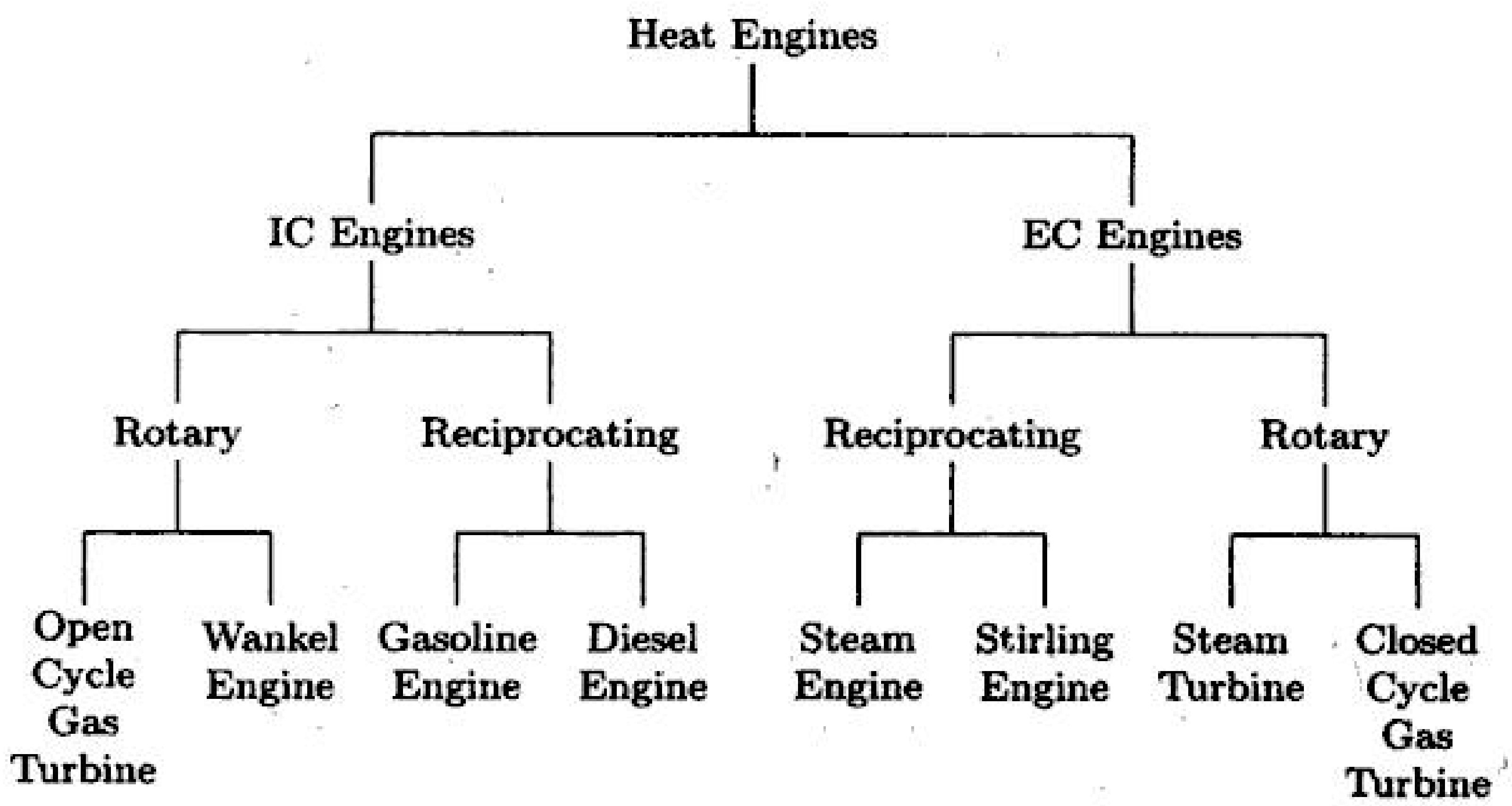


Fig. 1.1 Classification of Heat Engines

Another advantage of the reciprocating internal combustion engine over the other two types is that all its components work at an average temperature which is much below the maximum temperature of the working fluid in the cycle. This is because the high temperature of the working fluid in the cycle persists only for a very small fraction of the cycle time. Therefore, very high working fluid temperatures can be employed resulting in higher thermal efficiency.

Further, in internal combustion engines, higher thermal efficiency can be obtained with moderate maximum working pressure of the fluid in the cycle, and therefore, the weight to power ratio is less than that of the steam turbine plant. Also, it has been possible to develop reciprocating internal combustion engines of very small power output (power output of even a fraction of a kilowatt) with reasonable thermal efficiency and cost.

The main disadvantage of this type of engine is the problem of vibration caused by the reciprocating components. Also, it is not possible to use a variety of fuels in these engines. Only liquid or gaseous fuels of given specification can be efficiently used. These fuels are relatively more expensive.

Considering all the above factors the reciprocating internal combustion engines have been found suitable for use in automobiles, motor-cycles and scooters, power boats, ships, slow speed aircraft, locomotives and power units of relatively small output.

1.1.4 External Combustion and Internal Combustion Engines

External combustion engines are those in which combustion takes place outside the engine whereas in internal combustion engines combustion takes place within the engine. For example, in a steam engine or a steam turbine, the heat generated due to the combustion of fuel is employed to generate high pressure steam which is used as the working fluid in a reciprocating engine or a turbine.

In case of gasoline or diesel engines, the products of combustion generated by the combustion of fuel and air within the cylinder form the working fluid.

1.2 BASIC ENGINE COMPONENTS AND NOMENCLATURE

Even though reciprocating internal combustion engines look quite simple, they are highly complex machines. There are hundreds of components which have to perform their functions satisfactorily to produce output power. There are two types of engines, viz., spark-ignition (SI) and compression-ignition (CI) engine . Let us now go through the important engine components and the nomenclature associated with an engine.

1.2.1 Engine Components

A cross section of a single cylinder spark-ignition engine with overhead valves is shown in Fig.1.2. The major components of the engine and their functions are briefly described below.

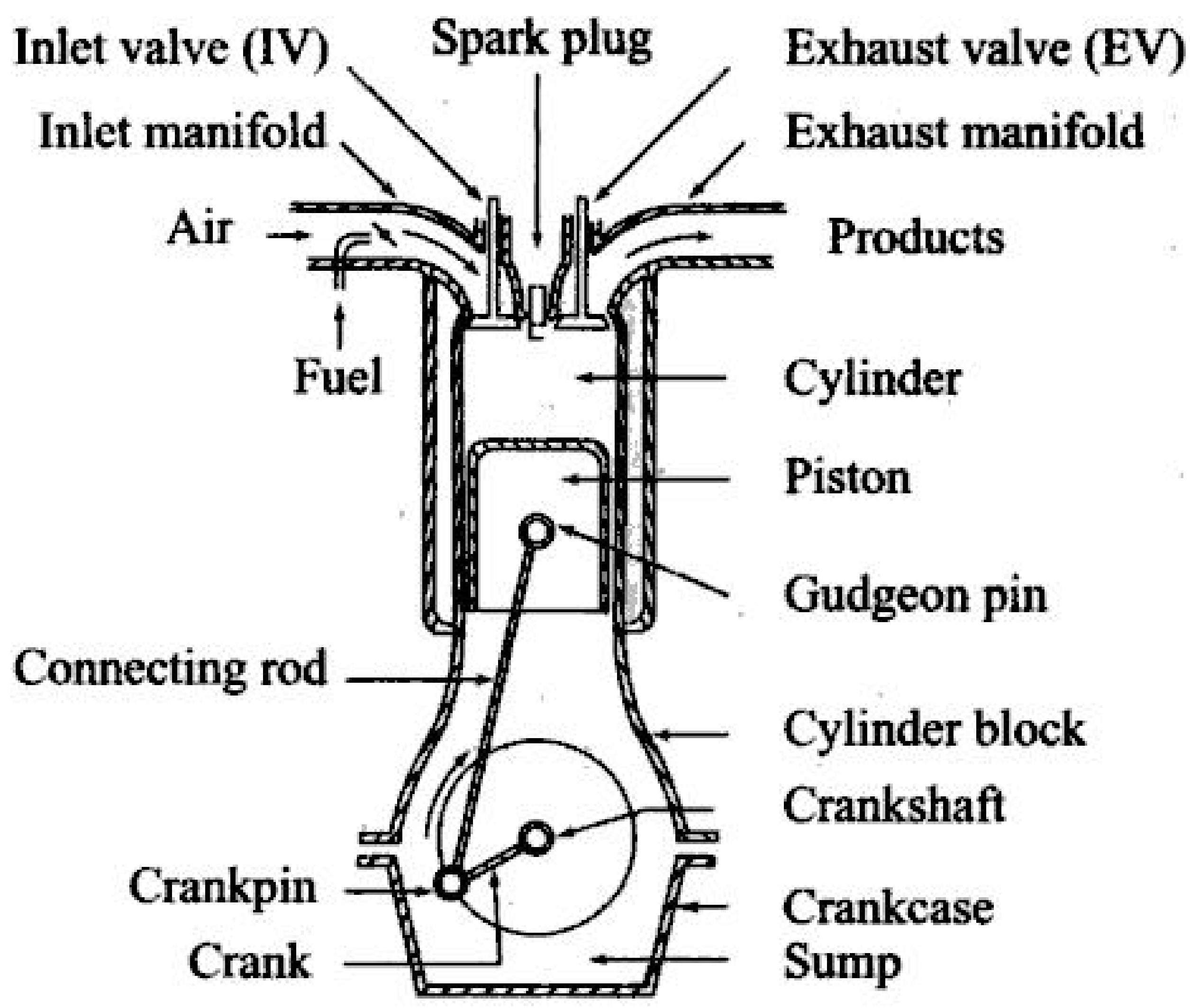


Fig. 1.2 Cross-section of a Spark-Ignition Engine

Cylinder Block : The cylinder block is the main supporting structure for the various components. The cylinder of a multicylinder engine are cast as a single unit, called cylinder block. The cylinder head is mounted on the cylinder block. The cylinder head and cylinder block are provided with water jackets in the case of water cooling or with cooling fins in the case of air cooling. Cylinder head gasket is incorporated between the cylinder block and cylinder head. The cylinder head is held tight to the cylinder block by number of bolts or studs. The bottom portion of the cylinder block is called crankcase. A cover called crankcase which becomes a sump for lubricating oil is fastened to the bottom of the crankcase. The inner surface of the cylinder block which is machined and finished accurately to cylindrical shape is called bore or face.

Cylinder : As the name implies it is a cylindrical vessel or space in which the piston makes a reciprocating motion. The varying volume created in the cylinder during the operation of the engine is filled with the working fluid and subjected to different thermodynamic processes. The cylinder is supported in the cylinder block.

Piston : It is a cylindrical component fitted into the cylinder forming the moving boundary of the combustion system. It fits perfectly (snugly) into the cylinder providing a gas-tight space with the piston rings and the lubricant. It forms the first link in transmitting the gas forces to the output shaft.

Combustion Chamber : The space enclosed in the upper part of the cylinder, by the cylinder head and the piston top during the combustion process, is called the combustion chamber. The combustion of fuel and the consequent release of thermal energy results in the building up of pressure in this part of the cylinder.

Inlet Manifold : The pipe which connects the intake system to the inlet valve of the engine and through which air or air-fuel mixture is drawn into the cylinder is called the inlet manifold.

Exhaust Manifold : The pipe which connects the exhaust system to the exhaust valve of the engine and through which the products of combustion escape into the atmosphere is called the exhaust manifold.

Inlet and Exhaust Valves : Valves are commonly mushroom shaped poppet type. They are provided either on the cylinder head or on the side of the cylinder for regulating the charge coming into the cylinder (inlet valve) and for discharging the products of combustion (exhaust valve) from the cylinder.

Spark Plug : It is a component to initiate the combustion process in Spark-Ignition (SI) engines and is usually located on the cylinder head.

Connecting Rod : It interconnects the piston and the crankshaft and transmits the gas forces from the piston to the crankshaft. The two ends of the connecting rod are called as small end and the big end (Fig.1.3). Small end is connected to the piston by gudgeon pin and the big end is connected to the crankshaft by crankpin.

Crankshaft : It converts the reciprocating motion of the piston into useful rotary motion of the output shaft. In the crankshaft of a single cylinder engine there are a pair of crank arms and balance weights. The balance weights are provided for static and dynamic balancing of the rotating system. The crankshaft is enclosed in a crankcase.

Piston Rings : Piston rings, fitted into the slots around the piston, provide a tight seal between the piston and the cylinder wall thus preventing leakage of combustion gases (Fig.1.3).

Gudgeon Pin : It forms the link between the small end of the connecting rod and the piston.

Camshaft : The camshaft and its associated parts control the opening and closing of the two valves. The associated parts are push rods, rocker arms, valve springs and tappets. This shaft also provides the drive to the ignition system. The camshaft is driven by the crankshaft through timing gears.

Cams : These are made as integral parts of the camshaft and are designed in such a way to open the valves at the correct timing and to keep them open for the necessary duration.

Fly Wheel : The net torque imparted to the crankshaft during one complete cycle of operation of the engine fluctuates causing a change in the angular velocity of the shaft. In order to achieve a uniform torque an inertia mass in the form of a wheel is attached to the output shaft and this wheel is called the flywheel.

1.2.2 Nomenclature

Cylinder Bore (d) **:** The nominal inner diameter of the working cylinder is called the cylinder bore and is designated by the letter d and is usually expressed in millimeter (mm).

Piston Area (A) **:** The area of a circle of diameter equal to the cylinder bore is called the piston area and is designated by the letter A and is usually expressed in square centimeter (cm^2).

Stroke (L) **:** The nominal distance through which a working piston moves between two successive reversals of its direction of motion is called the stroke and is designated by the letter L and is expressed usually in millimeter (mm).

Stroke to Bore Ratio : L/d ratio is an important parameter in classifying the size of the engine.

If $d < L$, it is called under-square engine. If $d = L$, it is called square engine. If $d > L$, it is called over-square engine.

An over-square engine can operate at higher speeds because of larger bore and shorter stroke.

Dead Centre : The position of the working piston and the moving parts which are mechanically connected to it, at the moment when the direction of the piston motion is reversed at either end of the stroke is called the dead centre. There are two dead centres in the engine as indicated in Fig.1.3. They are:

(i) Top Dead Centre (ii) Bottom Dead Centre

(i) *Top Dead Centre* (TDC) *:* It is the dead centre when the piston is farthest from the crankshaft. It is designated as TDC for vertical engines and Inner Dead Centre (IDC) for horizontal engines.

(ii) *Bottom Dead Centre* (BDC) *:* It is the dead centre when the piston is nearest to the crankshaft. It is designated as BDC for vertical engines and Outer Dead Centre (ODC) for horizontal engines.

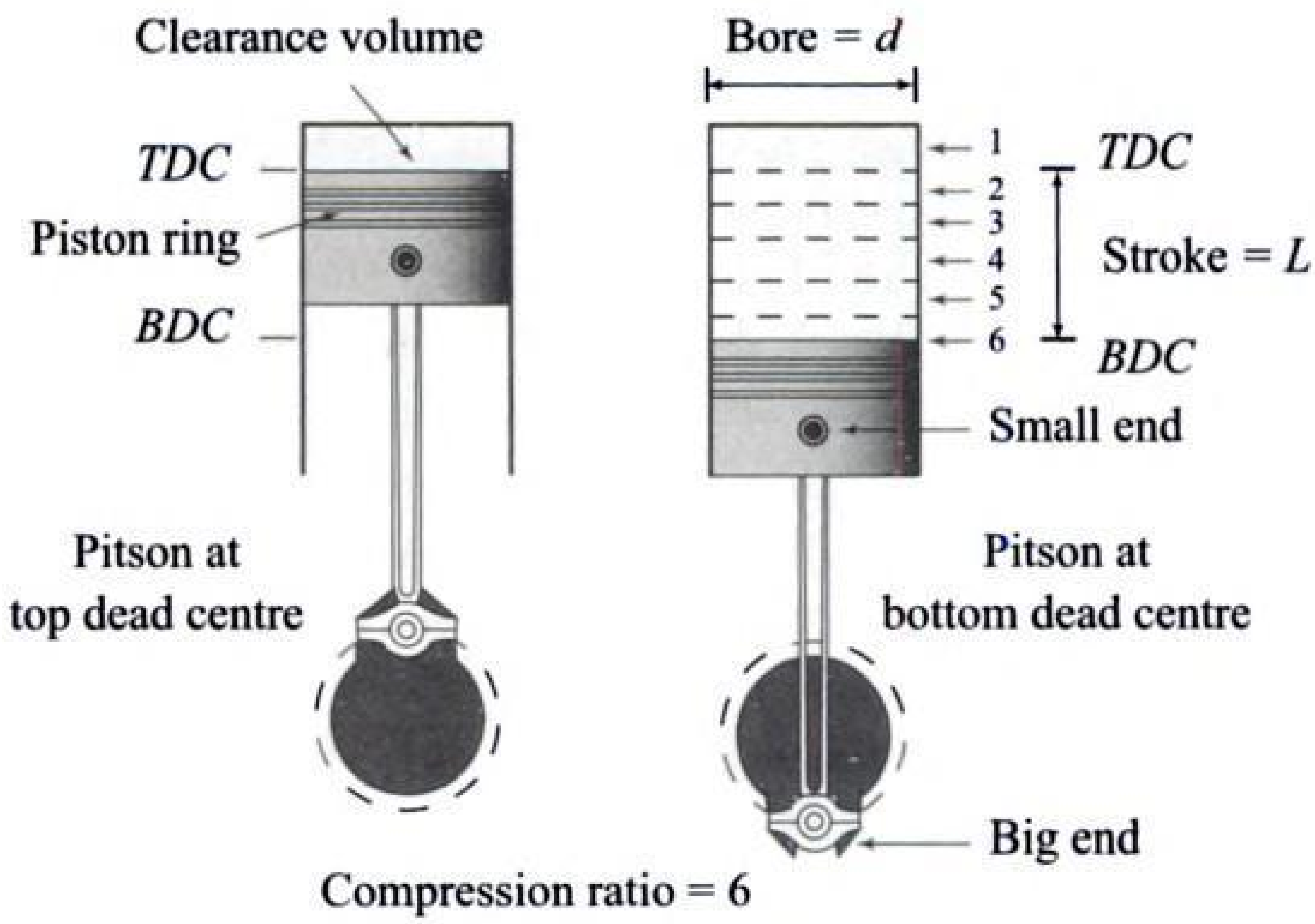


Fig. 1.3 Top and Bottom Dead Centres

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

$$V_s = A \times L \quad = \quad \frac{\pi}{4} d^2 L \tag{1.1}$$

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.

$$r = \frac{V_T}{V_C} = \frac{V_C + V_s}{V_C} = 1 + \frac{V_s}{V_C} \tag{1.2}$$

1.3 THE WORKING PRINCIPLE OF ENGINES

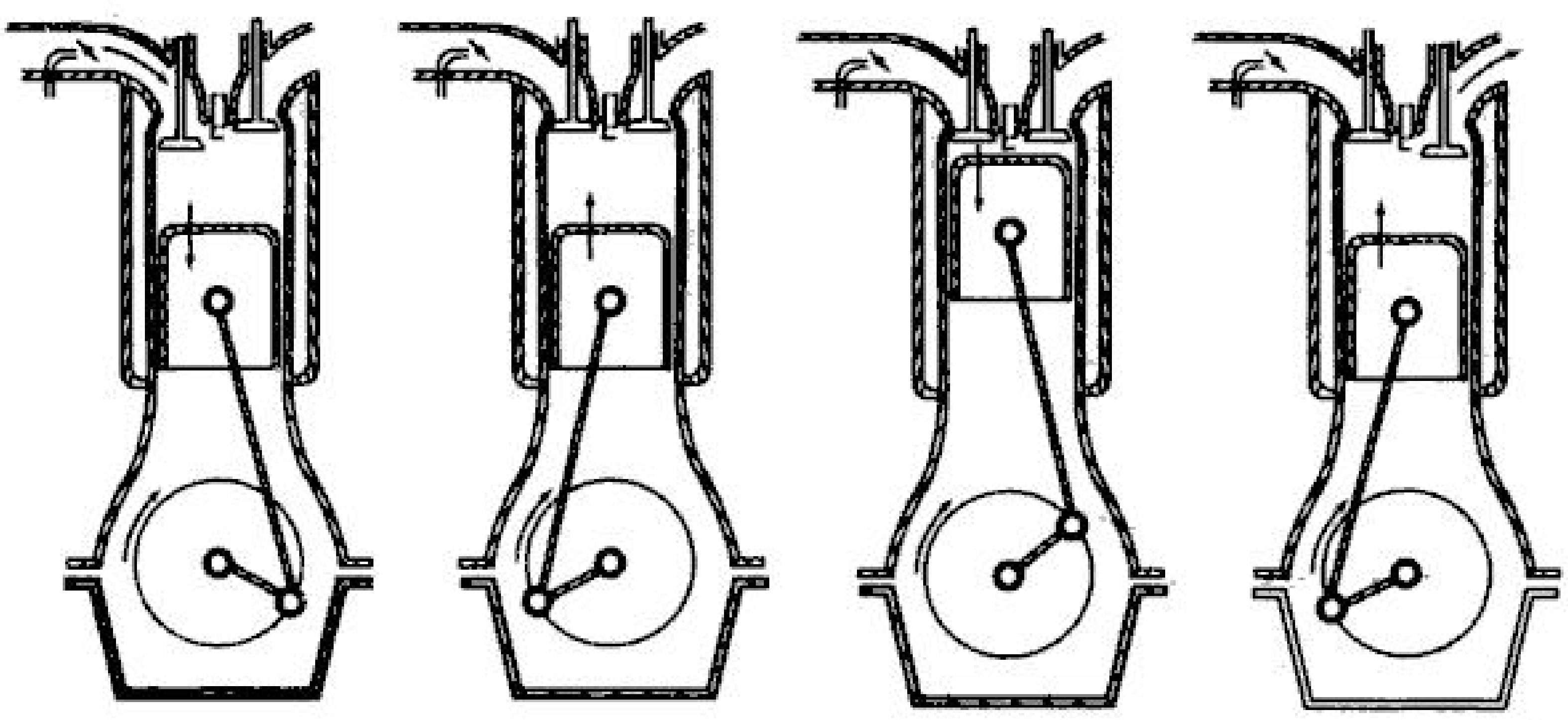
If an engine is to work successfully then it has to follow a cycle of operations in a sequential manner. The sequence is quite rigid and cannot be changed. In the following sections the working principle of both SI and CI engines is described. Even though both engines have much in common there are certain fundamental differences.

The credit of inventing the spark-ignition engine goes to Nicolaus A. Otto (1876) whereas compression-ignition engine was invented by Rudolf Diesel (1892). Therefore, they are often referred to as Otto engine and Diesel engine.

1.3.1 Four-Stroke Spark-Ignition Engine

In a four-stroke engine, the cycle of operations is completed in four strokes of the piston or two revolutions of the crankshaft. During the four strokes, there are five events to be completed, viz., suction, compression, combustion, expansion and exhaust. Each stroke consists of 180° of crankshaft rotation and hence a four-stroke cycle is completed through 720° of crank rotation. The cycle of operation for an ideal four-stroke SI engine consists of the following four strokes : (i) suction or intake stroke; (ii) compression stroke; (iii) expansion or power stroke and (iv) exhaust stroke.

The details of various processes of a four-stroke spark-ignition engine with overhead valves are shown in Fig.1.4 (a-d). When the engine completes all the five events under ideal cycle mode, the p-V diagram will be as shown in Fig.1.5.



(a) Intake (b) Compression (c) Expansion (d) Exhaust

Fig. 1.4 Working Principle of a Four-Stroke SI Engine

(i) *Suction or Intake Stroke :* Suction stroke 0→1 (Fig.1.5) starts when the piston is at the top dead centre and about to move downwards. The inlet valve is open at this time and the exhaust valve is closed, Fig.1.4(a). Due to the suction created by the motion of the piston towards the bottom dead centre, the charge consisting of fuel-air mixture is drawn into the cylinder. When the piston reaches the bottom dead centre the suction stroke ends and the inlet valve closes.

(ii) *Compression Stroke :* The charge taken into the cylinder during the suction stroke is compressed by the return stroke of the piston 1→2, (Fig.1.5). During this stroke both inlet and exhaust valves are in closed position, Fig.1.4(b). The mixture which fills the entire cylinder volume is now compressed into the clearance volume. At the end of the compression stroke the mixture is ignited with the help of a spark plug located on the cylinder head. In ideal engines it is assumed that burning takes place instantaneously when the piston is at the top dead centre and hence the burning process can be approximated as heat addition at constant volume. During the burning process the chemical energy of the fuel is converted into heat energy producing a temperature rise of about 2000 °C (process 2→3), Fig.1.5. The pressure at the end of the combustion process is considerably increased due to the heat release from the fuel.

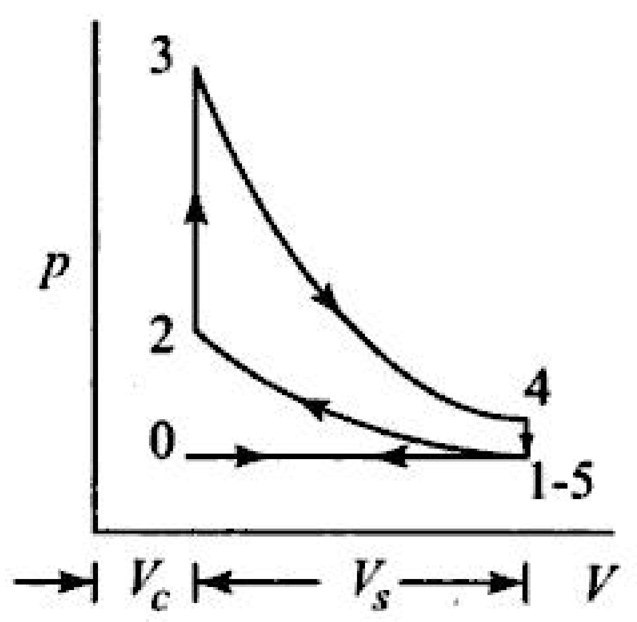


Fig. 1.5 Ideal p-V Diagram of a Four-Stroke SI Engine

(iii) *Expansion or Power Stroke :* The high pressure of the burnt gases forces the piston towards the *BDC*, (stroke 3→4) Fig.1.5. Both the valves are in closed position, Fig.1.4(c). Of the four-strokes only during this stroke power is produced. Both pressure and temperature decrease during expansion.

(iv) *Exhaust Stroke :* At the end of the expansion stroke the exhaust valve opens and the inlet valve remains closed, Fig.1.4(d). The pressure falls to atmospheric level a part of the burnt gases escape. The piston starts moving from the bottom dead centre to top dead centre (stroke 5→0), Fig.1.5 and sweeps the burnt gases out from the cylinder almost at atmospheric pressure. The exhaust valve closes when the piston reaches TDC. at the end of the exhaust stroke and some residual gases trapped in the clearance volume remain in the cylinder.

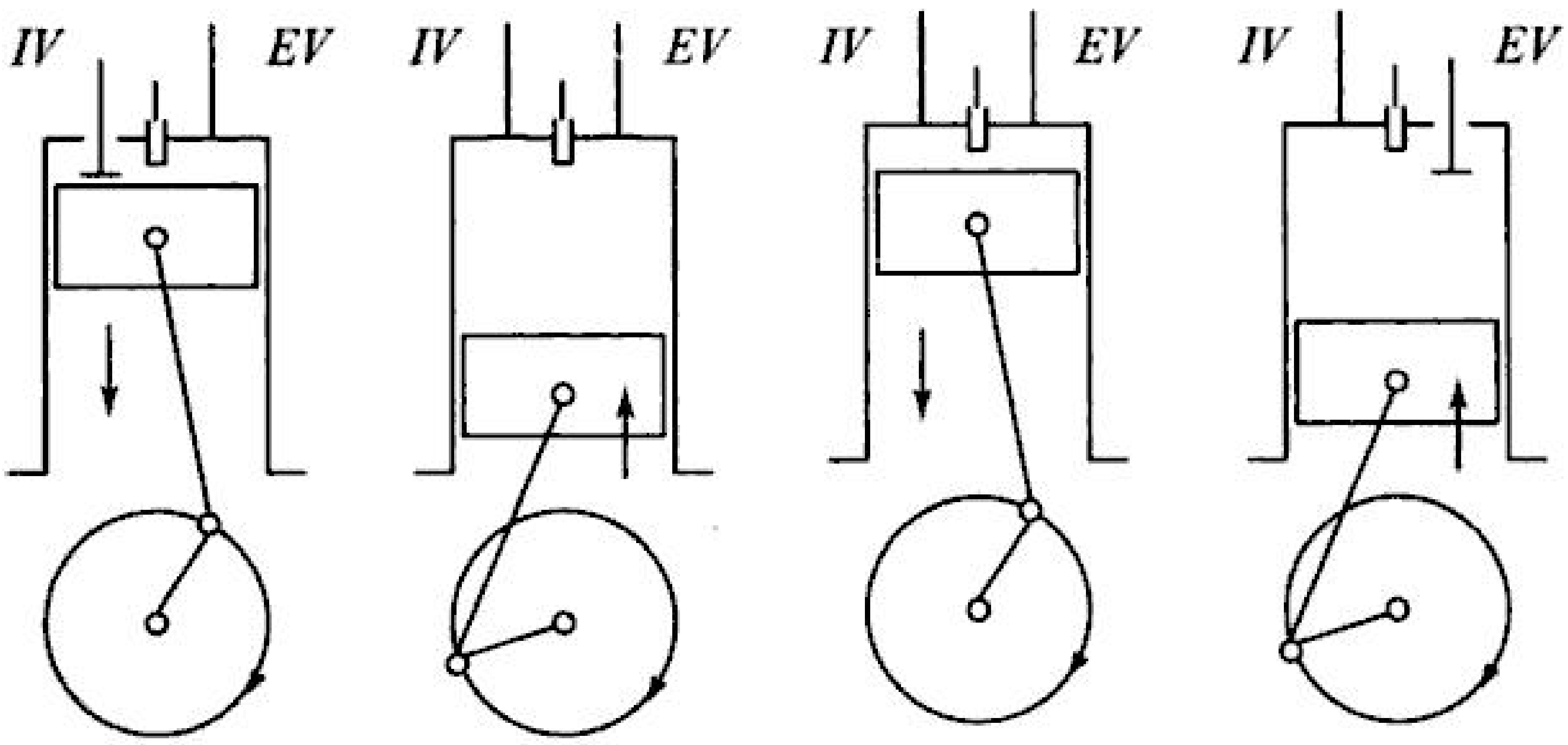
These residual gases mix with the fresh charge coming in during the following cycle, forming its working fluid. Each cylinder of a four-stroke engine completes the above four operations in two engine revolutions, one revolution of the crankshaft occurs during the suction and compression strokes and the second revolution during the power and exhaust strokes. Thus for one complete cycle there is only one power stroke while the crankshaft turns by two revolutions. For getting higher output from the engine the heat release (process 2→3) should be as high as possible and the heat rejection (process 3→4) should be as small as possible. So one should be careful in drawing the ideal p-V diagram (Fig.1.5).

1.3.2 Four-Stroke Compression-Ignition Engine

The four-stroke CI engine is similar to the four-stroke SI engine but it operates at a much higher compression ratio. The compression ratio of an SI engine is between 6 and 10 while for a CI engine it is from 16 to 20. In the CI engine during suction stroke, air, instead of a fuel-air mixture, is inducted. Due to the high compression ratio employed, the temperature at the end of the compression stroke is sufficiently high to self ignite the fuel which is injected into the combustion chamber. In CI engines, a high pressure fuel pump and an injector are provided to inject the fuel into the combustion chamber. The carburettor and ignition system necessary in the SI engine are not required in the CI engine.

The ideal sequence of operations for the four-stroke CI engine is as follows:

(i) *Suction Stroke :* Air alone is inducted during the suction stroke. During this stroke intake valve is open and exhaust valve is closed, Fig.1.6(a).



(a) Suction (b) Compression (c) Expansion (d) Exhaust

Fig. 1.6 Cycle of Operation of a CI Engine

(ii) *Compression Stroke :* Air inducted during the suction stroke is compressed into the clearance volume. Both valves remain closed during this stroke, Fig.1.6(b).

(iii) *Expansion Stroke :* Fuel injection starts nearly at the end of the compression stroke. The rate of injection is such that combustion maintains the pressure constant in spite of the piston movement on its expansion stroke increasing the volume. Heat is assumed to have been added at constant pressure. After the injection of fuel is completed (i.e. after cut-off) the products of combustion expand. Both the valves remain closed during the expansion stroke, Fig.1.6(c).

(iv) *Exhaust Stroke :* The piston travelling from *BDC* to *TDC* pushes out the products of combustion. The exhaust valve is open and the intake valve is closed during this stroke, Fig.1.6(d). The ideal *p-V* diagram is shown in Fig.1.7.

Due to higher pressures in the cycle of operations the CI engine has to be more sturdy than a SI engine for the same output. This results in a CI engine being heavier than the SI engine. However, it has a higher thermal efficiency on account of the high compression ratio (of about 18 as against about 8 in SI engines) used.

1.3.3 Comparison of SI and CI Engines

In four-stroke engines, there is one power stroke for every two revolutions of the crankshaft. There are two non-productive strokes of

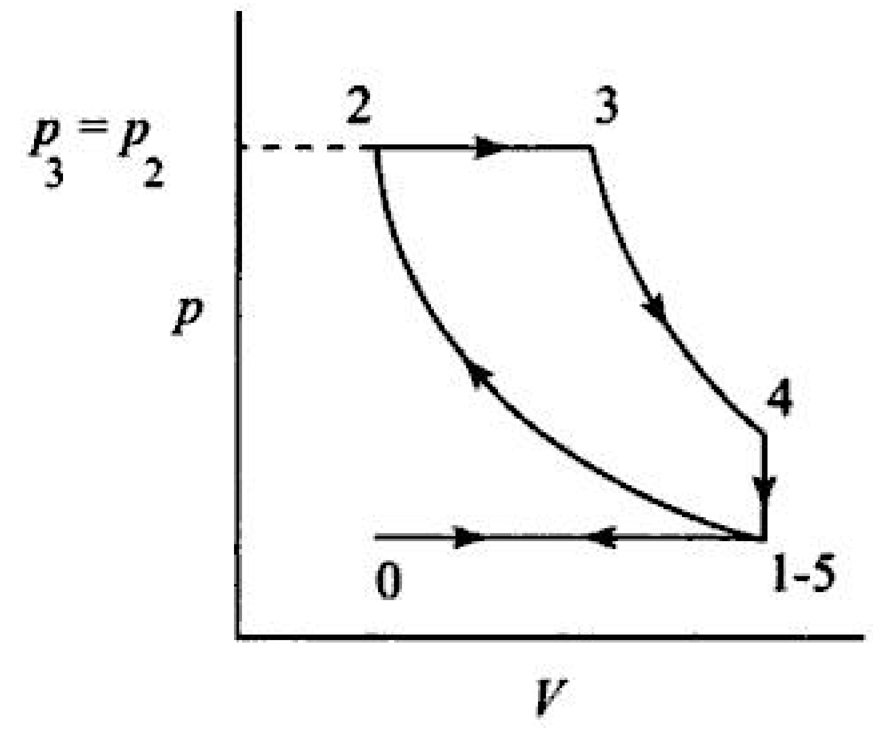


Fig. 1.7 Ideal p-V Diagram for a Four-Stroke CI Engine

exhaust and suction which are necessary for flushing the products of combustion from the cylinder and filling it with the fresh charge. If this purpose could be served by an alternative arrangement, without the movement of the piston, it is possible to obtain a power stroke for every revolution of the crankshaft increasing the output of the engine. However, in both SI and CI engines operating on four-stroke cycle, power can be obtained only in every two revolution of the crankshaft.

Since both SI and CI engines have much in common, it is worthwhile to compare them based on important parameters like basic cycle of operation, fuel induction, compression ratio etc. The detailed comparison is given in Table 1.1.

1.3.4 Two-Stroke Engine

As already mentioned, if the two unproductive strokes, viz., the suction and exhaust could be served by an alternative arrangement, especially without the movement of the piston then there will be a power stroke for each revolution of the crankshaft. In such an arrangement, theoretically the power output of the engine can be doubled for the same speed compared to a four-stroke engine. Based on this concept, Dugald Clark (1878) invented the two-stroke engine.

In two-stroke engines the cycle is completed in one revolution of the crankshaft. The main difference between two-stroke and four-stroke engines is in the method of filling the fresh charge and removing the burnt gases from the cylinder. In the four-stroke engine these operations are performed by the engine piston during the suction and exhaust strokes respectively. In a two-stroke engine, the filling pro-

Table 1.1 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 introduced.	The quantity of fuel is regulated. 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 lower peak pressures.	Heavier due to higher peak pressures.

cess is accomplished by the charge compressed in crankcase or by a blower. The induction of the compressed charge moves out the product of combustion through exhaust ports. Therefore, no piston strokes are required for these two operations. Two strokes are sufficient to complete the cycle, one for compressing the fresh charge and the other for expansion or power stroke.

Figure 1.8 shows one of the simplest two-stroke engines, viz., the crankcase scavenged engine. Figure 1.9 shows the ideal indicator diagram of such an engine. The air or charge is inducted into the crankcase through the spring loaded inlet valve when the pressure in the crankcase is reduced due to upward motion of the piston during compression stroke. After the compression and ignition, expansion takes place in the usual way.

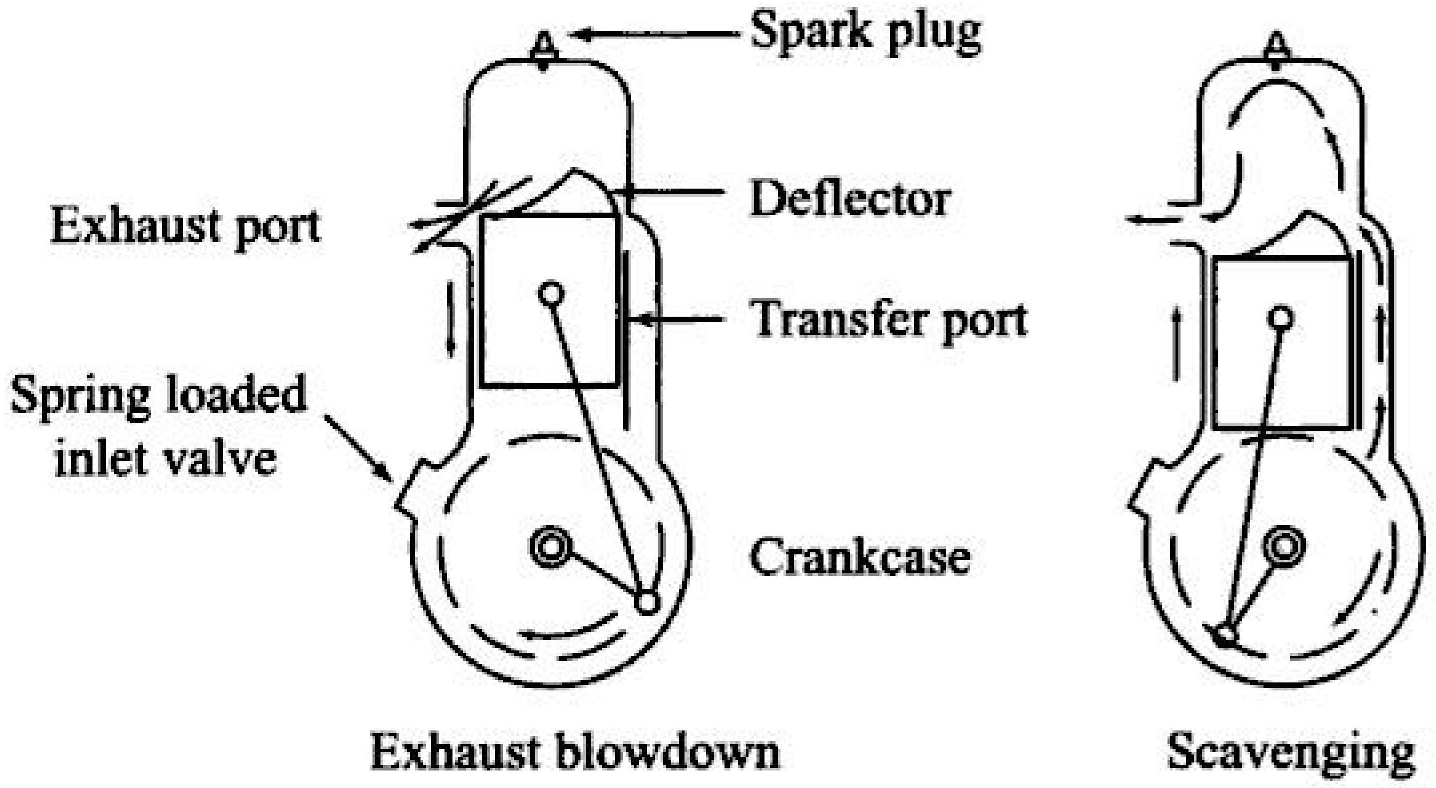


Fig. 1.8 Crankcase Scavenged Two-Stroke Engine

During the expansion stroke the charge in the crankcase is compressed. Near the end of the expansion stroke, the piston uncovers the exhaust ports and the cylinder pressure drops to atmospheric pressure as the combustion products leave the cylinder. Further movement of the piston uncovers the transfer ports, permitting the slightly compressed charge in the crankcase to enter the engine cylinder. The top of the piston has usually a projection to deflect the fresh charge towards the top of the cylinder before flowing to the exhaust ports. This serves the double purpose of scavenging the upper part of the cylinder of the combustion products and preventing the fresh charge from flowing directly to the exhaust ports.

The same objective can be achieved without piston deflector by proper shaping of the transfer port. During the upward motion of the

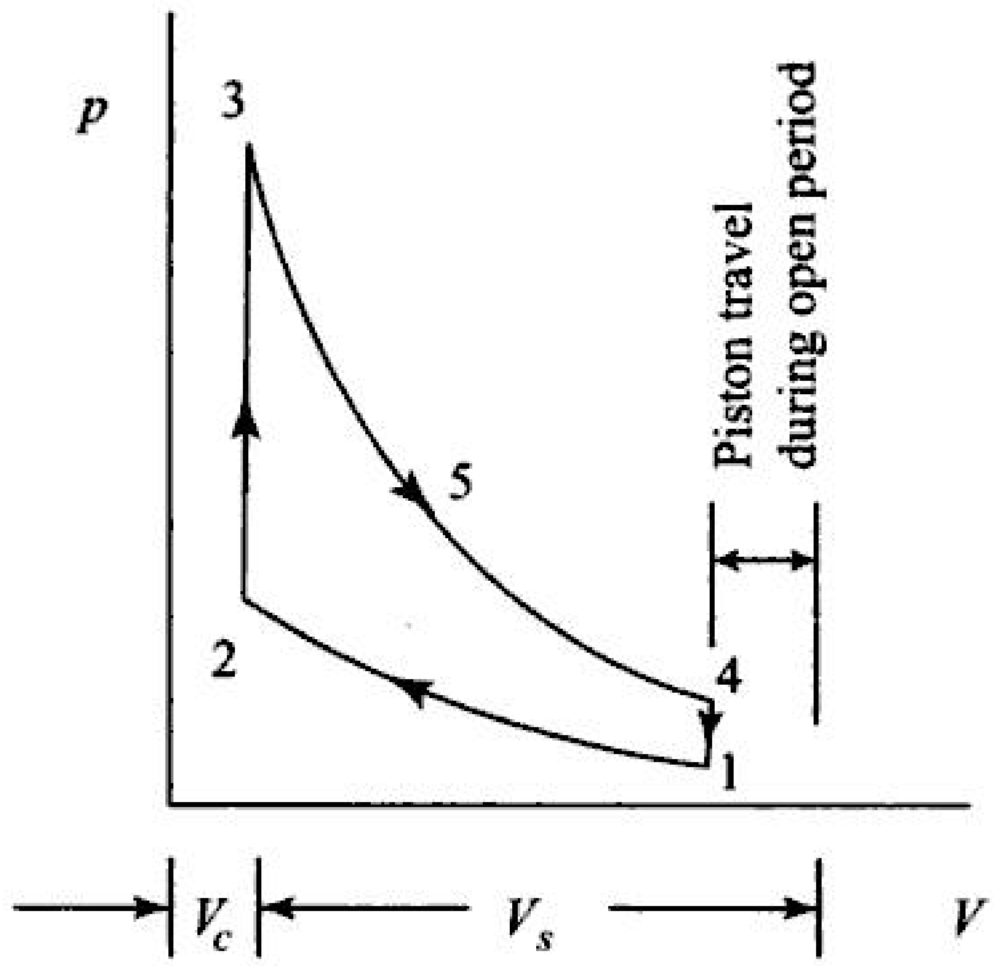


Fig. 1.9 Ideal Indicator Diagram of a Two-Stroke SI Engine

piston from *BDC* the transfer ports close first and then the exhaust ports close when compression of the charge begins and the cycle is repeated.

1.3.5 Comparison of Four-Stroke and Two-Stroke Engines

The two-stroke engine was developed to obtain a greater output from the same size of the engine. The engine mechanism also eliminates the valve arrangement making it mechanically simpler. Almost all two-stroke engines have no conventional valves but only ports (some have an exhaust valve). This simplicity of the two-stroke engine makes it cheaper to produce and easy to maintain. Theoretically a two-stroke engine develops twice the power of a comparable four-stroke engine because of one power stroke every revolution (compared to one power stroke every two revolutions of a four-stroke engine). This makes the two-stroke engine more compact than a comparable four-stroke engine. In actual practice power output is not exactly doubled but increased by only about 30% because of

(i) reduced effective expansion stroke and

(ii) increased heating caused by increased number of power strokes which limits the maximum speed.

The other advantages of the two-stroke engine are more uniform torque on crankshaft and comparatively less exhaust gas dilution.

However, when applied to the spark-ignition engine the two-stroke cycle has certain disadvantages which have restricted its application to only small engines suitable for motor cycles, scooters, lawn mowers, outboard engines etc. In the SI engine, the incoming charge consists of fuel and air. During scavenging, as both inlet and exhaust ports are open simultaneously for some time, there is a possibility that some of the fresh charge containing fuel escapes with the exhaust. This results in high fuel consumption and lower thermal efficiency. The other drawback of two-stroke engine is the lack of flexibility, viz., the capacity to operate with the same efficiency at all speeds. At part throttle operating condition, the amount of fresh mixture entering the cylinder is not enough to clear all the exhaust gases and a part of it remains in the cylinder to contaminate the charge. This results in irregular operation of the engine.

The two-stroke diesel engine does not suffer from these defects. There is no loss of fuel with exhaust gases as the intake charge in diesel engine is only air. The two-stroke diesel engine is used quite widely. Many of the high output diesel engines work on this cycle. A disadvantage common to all two-stroke engines, gasoline as well as diesel, is the greater cooling and lubricating oil requirements due to one power stroke in each revolution of the crankshaft. Consumption of lubricating oil is high in two-stroke engines due to higher temperature. A detailed comparison of two-stroke and four-stroke engines is given in Table 1.2.

1.4 ACTUAL ENGINES

Actual engines differ from the ideal engines because of various constraints in their operation. The indicator diagram also differs considerably from the ideal indicator diagrams. Actual indicator diagrams of a two-stroke and a four-stroke SI engines are shown in Figs.1.10(a) and 1.10(b) respectively. The various processes are indicated in the respective figures.

1.5 CLASSIFICATION OF IC ENGINES

Internal combustion engines are usually classified on the basis of the thermodynamic cycle of operation, type of fuel used, method of charging the cylinder, type of ignition, type of cooling and the cylinder arrangement etc. Details are given in Fig.1.11.

Table 1.2 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 one power stroke is obtained in each 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.
Volumetric efficiency is more due to more time for induction.	Volumetric efficiency is low due to lesser time for induction.
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, aeroplanes, power generation etc.	Used where low cost, compactness and light weight are important, viz., in mopeds, scooters, motorcycles, hand sprayers etc.

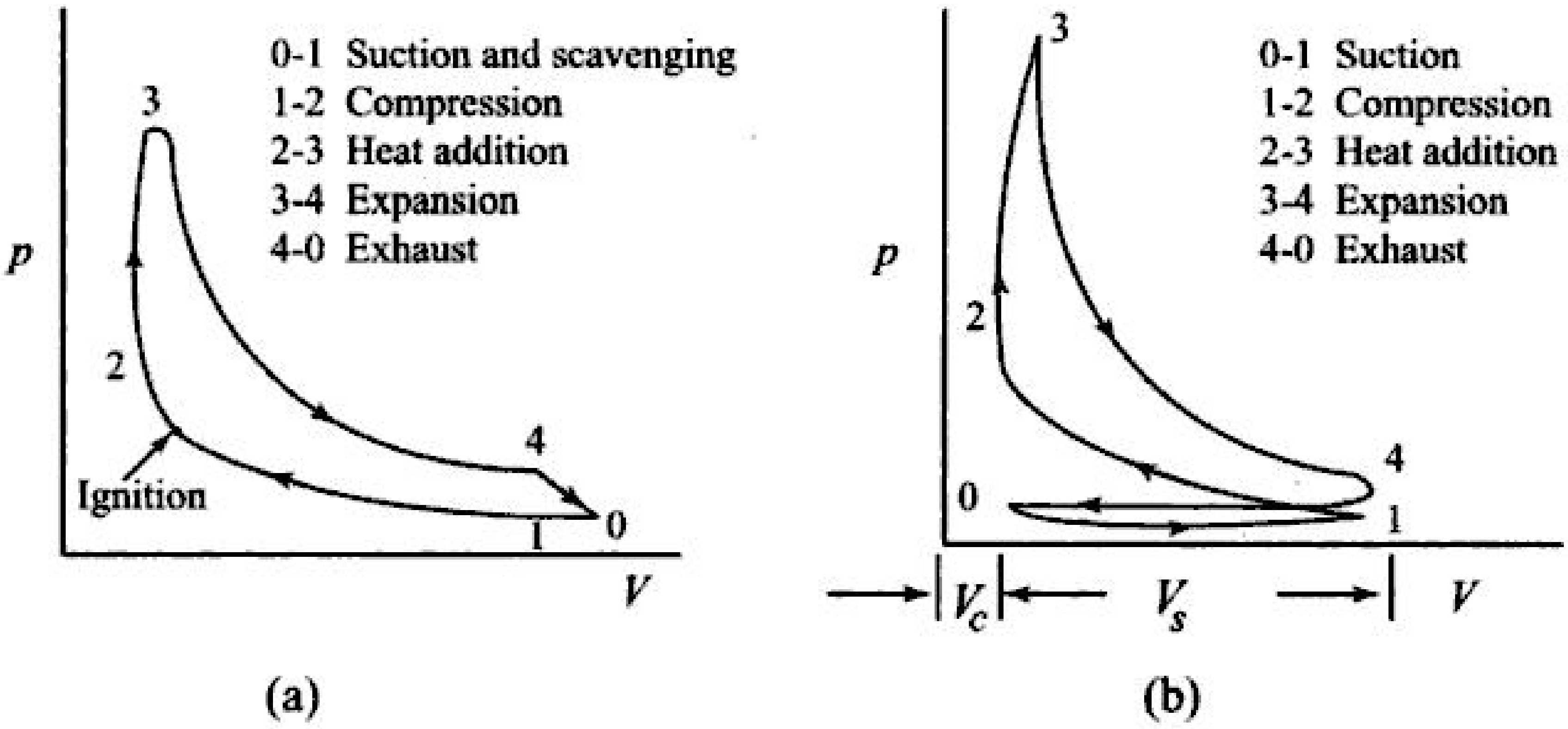


Fig. 1.10 Actual Indicator Diagrams of a Two-Stroke and Four-Stroke SI Engine

1.5.1 Cycle of Operation

According to the cycle of operation, IC engines are basically classified into two categories

(i) Constant volume heat addition cycle engine or Otto cycle engine. It is also called a Spark-Ignition engine, SI engine or Gasoline engine .

(ii) Constant-pressure heat addition cycle engine or Diesel cycle engine. It is also called a compression-ignition engine, CI engine or Diesel engine .

1.5.2 Type of Fuel Used

Based on the type of fuel used engines are classified as

(i) Engines using volatile liquid fuels like gasoline, alcohol, kerosene, benzene etc.
The fuel is generally mixed with air to form a homogeneous charge in a carburettor outside the cylinder and drawn into the cylinder in its suction stroke. The charge is ignited near the end of the compression stroke by an externally applied spark and therefore these engines are called spark-ignition engines.

(ii) Engines using gaseous fuels like natural gas, Liquified Petroleum Gas (LPG), blast furnace gas and biogas.

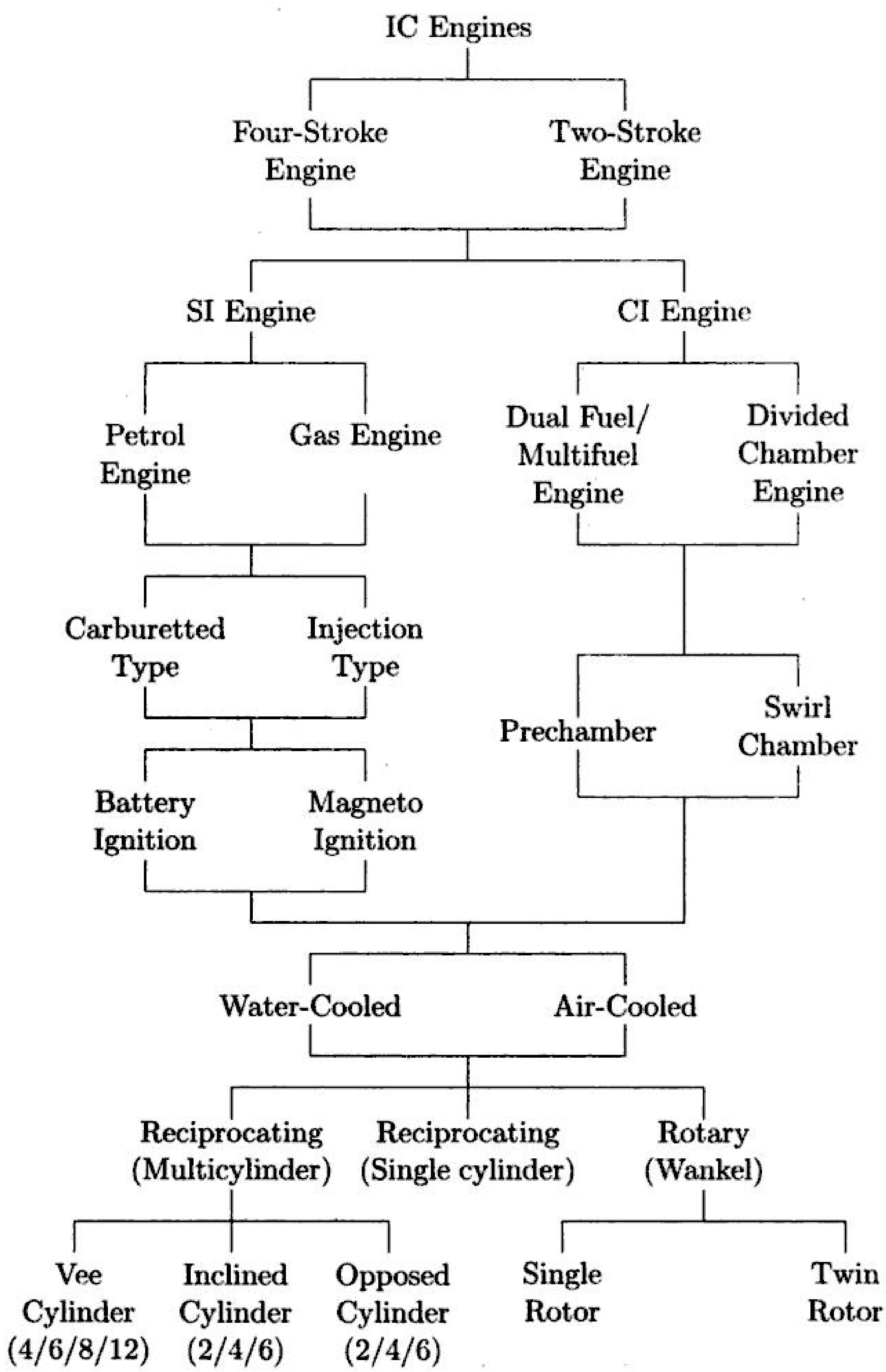


Fig. 1.11 Classification of Internal Combustion Engines

The gas is mixed with air and the mixture is introduced into the cylinder during the suction process. Working of this type of engine is similar to that of the engines using volatile liquid fuels (SI gas engine).

(iii) Engine using solid fuels like charcoal, powdered coal etc. Solid fuels are generally converted into gaseous fuels outside the engine in a separate gas producer and the engine works as a gas engine.

(iv) Engines using viscous (low volatility at normal atmospheric temperatures) liquid fuels like heavy and light diesel oils. The fuel is generally introduced into the cylinder in the form of minute droplets by a fuel injection system near the end of the compression process. Combustion of the fuel takes place due to its coming into contact with the high temperature compressed air in the cylinder. Therefore, these engines are called compression-ignition engines.

(v) Engines using two fuels (dual-fuel engines) A gaseous fuel or a highly volatile liquid fuel is supplied along with air during the suction stroke or during the initial part of compression through a gas valve in the cylinder head and the other fuel (a viscous liquid fuel) is injected into the combustion space near the end of the compression stroke (dual-fuel engines).

1.5.3 Method of Charging

According to the method of charging, the engines are classified as

(i) Naturally aspirated engines : Admission of air or fuel-air mixture at near atmospheric pressure.

(ii) Supercharged Engines : Admission of air or fuel-air mixture under pressure, i.e., above atmospheric pressure.

1.5.4 Type of Ignition

Spark-ignition engines require an external source of energy for the initiation of spark and thereby the combustion process. A high voltage spark is made to jump across the spark plug electrodes. In order to produce the required high voltage there are two types of ignition systems which are normally used. They are :
(i) battery ignition system (ii) magneto ignition system.

They derive their name based on whether a battery or a magneto is used as the primary source of energy for producing the spark.

In the case of CI engines there is no need for an external means to produce the ignition. Because of high compression ratio employed, the resulting temperature at the end of the compression process is high enough to self-ignite the fuel when injected. However, the fuel should be atomized into very fine particles. For this purpose a fuel injection system is normally used.

1.5.5 Type of Cooling

Cooling is very essential for the satisfactory running of an engine. There are two types of cooling systems in use and accordingly, the engines is classified as

(i) air-cooled engine (ii) water-cooled engine

1.5.6 Cylinder Arrangements

Another common method of classifying reciprocating engines is by the cylinder arrangement. The cylinder arrangement is only applicable to multicylinder engines. Two terms used in connection with cylinder arrangements must be defined first.

(i) *Cylinder Row :* An arrangement of cylinders in which the centre-line of the crankshaft journals is perpendicular to the plane containing the centrelines of the engine cylinders.

(ii) *Cylinder Bank :* An arrangement of cylinders in which the centre-line of the crankshaft journals is parallel to the plane containing the centrelines of the engine cylinders.

A number of cylinder arrangements popular with designers are described below. The details of various cylinder arrangements are shown in Fig.1.12.

In-line Engine : The in-line engine is an engine with one cylinder bank, i.e. all cylinders are arranged linearly, and transmit power to a single crankshaft. This type is quite common with automobile engines. Four and six cylinder in-line engines are popular in automotive applications.

'V' Engine : In this engine there are two banks of cylinders (i.e., two in line engines) inclined at an angle to each other and with one crankshaft. Most of the high powered automobiles use the 8 cylinder 'V' engine, four in-line on each side of the 'V'. Engines with more than six cylinders generally employ this configuration.

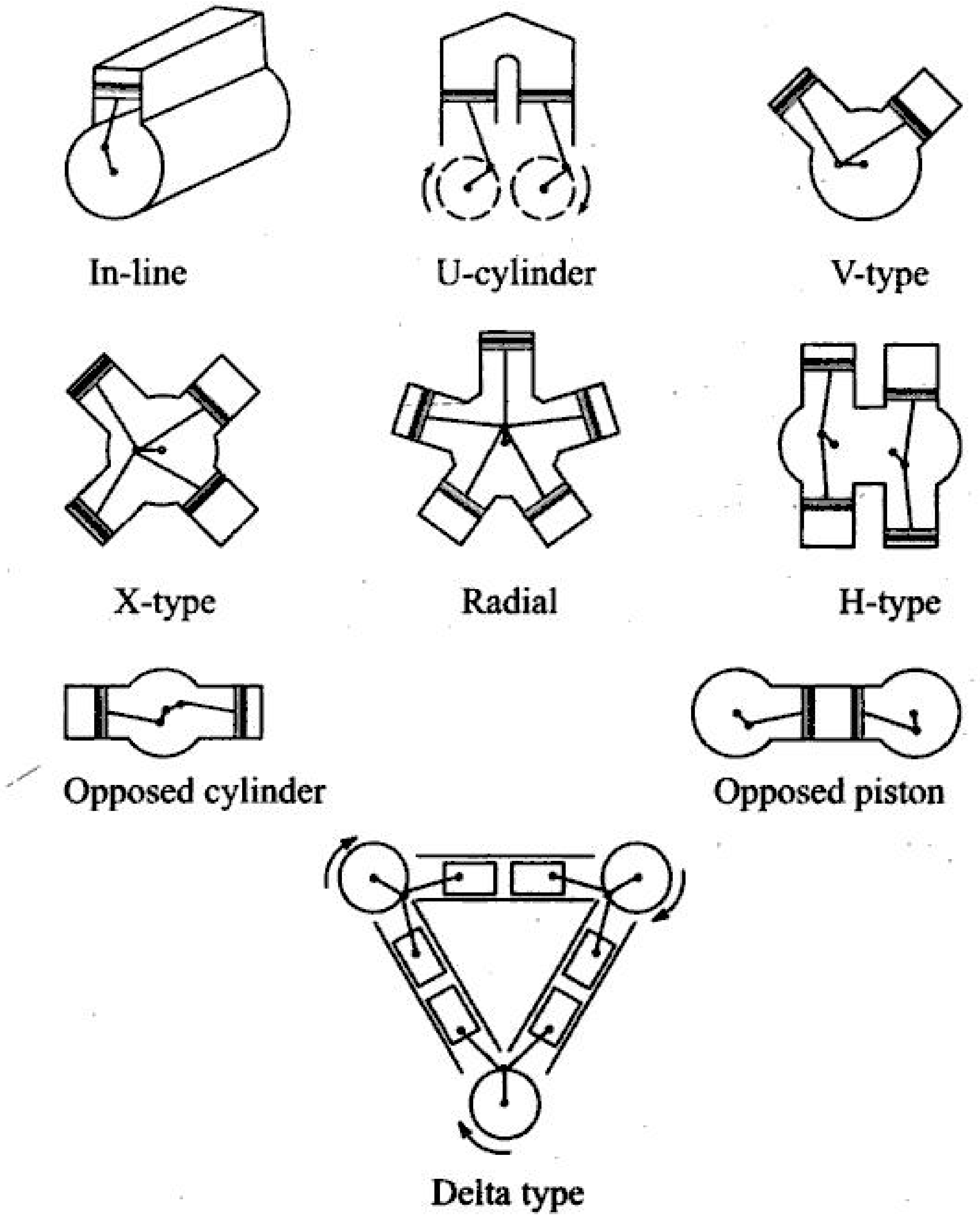


Fig. 1.12 Engine Classification by Cylinder Arrangements

Opposed Cylinder Engine : This engine has two cylinder banks located in the same plane on opposite sides of the crankshaft. It can be visualized as two 'in-line' arrangements 180 degrees apart. It is inherently a well balanced engine and has the advantages of a single crankshaft. This design is used in small aircrafts.

Opposed Piston Engine : When a single cylinder houses two pistons, each of which driving a separate crankshaft, it is called an opposed piston engine. The movement of the pistons is synchronized by coupling the two crankshafts. Opposed piston arrangement, like opposed cylinder arrangement, is inherently well balanced. Further, it has the advantage of requiring no cylinder head. By its inherent features, this engine usually functions on the principle of two-stroke engines.

Radial Engine : Radial engine is one where more than two cylinders in each row are equally spaced around the crankshaft. The radial arrangement of cylinders is most commonly used in conventional air-cooled aircraft engines where 3, 5, 7 or 9 cylinders may be used in one bank and two to four banks of cylinders may be used. The odd number of cylinders is employed from the point of view of balancing. Pistons of all the cylinders are coupled to the same crankshaft.

'X' Type Engine : This design is a variation of 'V' type. It has four banks of cylinders attached to a single crankshaft.

'H' Type Engine : The 'H' type is essentially two 'Opposed cylinder' type utilizing two separate but interconnected crankshafts.

'U' Type Engine : The 'U' type is a variation of opposed piston arrangement.

Delta Type Engine : The delta type is essentially a combination of three opposed piston engine with three crankshafts interlinked to one another.

In general, automobile engines and general purpose engines utilize the 'in-line' and 'V' type configuration or arrangement. The 'radial' engine was used widely in medium and large aircrafts till it was replaced by the gas turbine. Small aircrafts continue to use either the 'opposed cylinder' type or 'in-line' or 'V' type engines. The 'opposed piston' type engine is widely used in large diesel installations. The 'H' and 'X' types do not presently find wide application, except in some diesel installations. A variation of the 'X' type is referred to as the 'pancake' engine.

1.6 APPLICATION OF IC ENGINES

The most important application of IC engines is in transport on land, sea and air. Other applications include industrial power plants and as prime movers for electric generators. Table 1.3 gives, in a nutshell, the applications of both IC and EC engines.

1.6.1 Two-Stroke Gasoline Engines

Small two-stroke gasoline engines are used where simplicity and low cost of the prime mover are the main considerations. In such applications a little higher fuel consumption is acceptable. The smallest engines are used in mopeds (50 cc engine) and lawn mowers. Scooters and motor cycles, the commonly used two wheeler transport, have generally 100-150 cc, two-stroke gasoline engines developing a maximum brake power of about 5 kW at 5500 rpm. High powered motor

Table 1.3 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, Locomotive, Power, Marine	Steam Turbines	Power, Large Marine
Gas turbines	Power, Aircraft, Industrial, Marine	Close Cycle Gas Turbine	Power, Marine

cycles have generally 250 cc two-stroke gasoline engines developing a maximum brake power of about 10 kW at 5000 rpm. Two-stroke gasoline engines may also be used in very small electric generating sets, pumping sets, and outboard motor boats. However, their specific fuel consumption is higher due to the loss of fuel-air charge in the process of scavenging and because of high speed of operation for which such small engines are designed.

1.6.2 Two-Stroke Diesel Engines

Very high power diesel engines used for ship propulsion are commonly two-stroke diesel engines. In fact, all engines between 400 to 900 mm bore are loop scavenged or uniflow type with exhaust valves (see Figs.20.8 and 20.9). The brake power on a single crankshaft can be upto 37000 kW. Nordberg, 12 cylinder 800 mm bore and 1550 mm stroke, two-stroke diesel engine develops 20000 kW at 120 rpm. This speed allows the engine to be directly coupled to the propeller of a ship without the necessity of gear reducers.

1.6.3 Four-Stroke Gasoline Engines

The most important application of small four-stroke gasoline engines is in automobiles. A typical automobile is powered by a four-stroke four cylinder engine developing an output in the range of 30-60 kW at a speed of about 4500 rpm. American automobile engines are much bigger and have 6 or 8 cylinder engines with a power output upto

185 kW. However, the oil crisis and air pollution from automobile engines have reversed this trend towards smaller capacity cars.

Four-stroke gasoline engines were also used for buses and trucks. They were generally 4000 cc, 6 cylinder engines with maximum brake power of about 90 kW. However, in this application gasoline engines have been practically replaced by diesel engines. The four-stroke gasoline engines have also been used in big motor cycles with side cars. Another application of four-stroke gasoline engine is in small pumping sets and mobile electric generating sets.

Small aircraft generally use radial four-stroke gasoline engines. Engines having maximum power output from 400 kW to 4000 kW have been used in aircraft. An example is the Bristol Contours 57, 18 cylinder two row, sleeve valve, air-cooled radial engine developing, a maximum brake power of about 2100 kW.

1.6.4 Four-Stroke Diesel Engines

The four-stroke diesel engine is one of the most efficient and versatile prime movers. It is manufactured in sizes from 50 mm to more than 1000 mm of cylinder diameter and with engine speeds ranging from 100 to 4500 rpm while delivering outputs from 1 to 35000 kW.

Small diesel engines are used in pump sets, construction machinery, air compressors, drilling rigs and many miscellaneous applications. Tractors for agricultural application use about 30 kW diesel engines whereas jeeps, buses and trucks use 40 to 100 kW diesel engines. Generally, the diesel engines with higher outputs than about 100 kW are supercharged. Earth moving machines use supercharged diesel engines in the output range of 200 to 400 kW. Locomotive applications require outputs of 600 to 4000 kW. Marine applications, from fishing vessels to ocean going ships use diesel engines from 100 to 35000 kW. Diesel engines are used both for mobile and stationary electric generating plants of varying capacities. Compared to gasoline engines, diesel engines are more efficient and therefore manufacturers have come out with diesel engines in personal transportation. However, the vibrations from the engine and the unpleasant odour in the exhaust are the main drawbacks.

1.7 THE FIRST LAW ANALYSIS OF ENGINE CYCLE

Before a detailed thermodynamic analysis of the engine cycle is done, it is desirable to have a general picture of the energy flow or energy balance of the system so that one becomes familiar with the various performance parameters. Figure 1.13 shows the energy flow through the reciprocating engine and Fig.1.14 shows its block diagram as an open system.

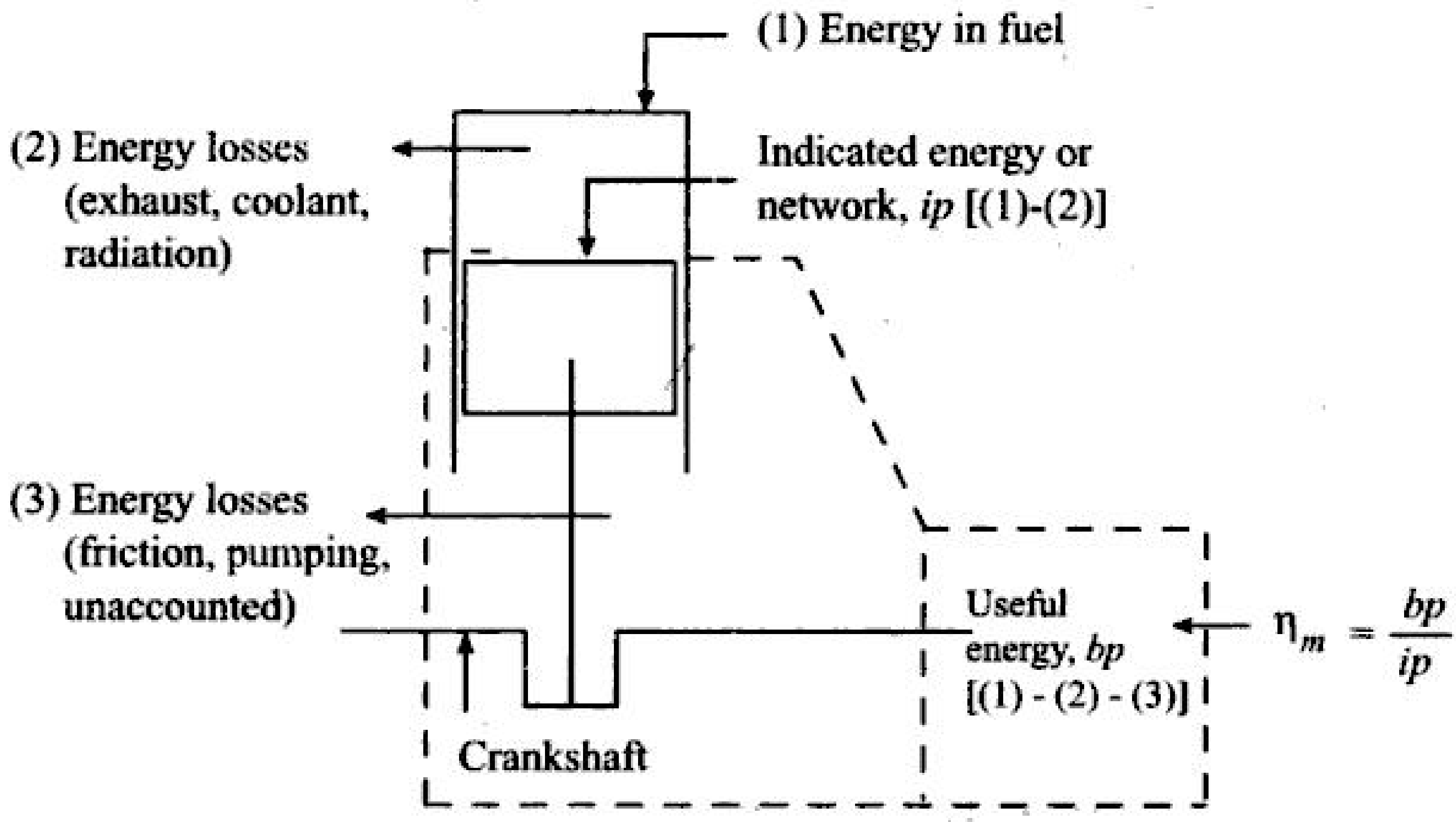


Fig. 1.13 Energy Flow through the Reciprocating Engine

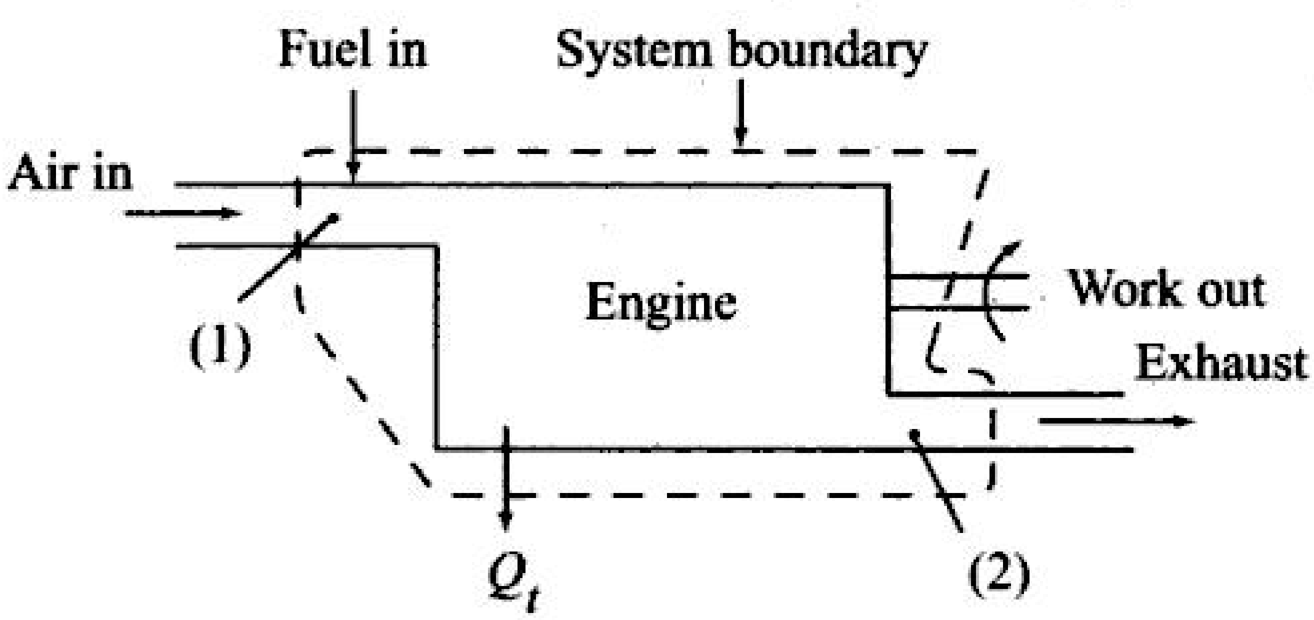


Fig. 1.14 Reciprocating Engine as an Open System

According to the first law of thermodynamics, energy can neither be created nor destroyed. It can only be converted from one form to another. Therefore, there must be an energy balance of input and output to a system. In the reciprocating internal combustion engine the fuel is fed into the combustion chamber where it burns in air converting chemical energy of the fuel into heat. The liberated heat energy cannot be totally utilized for driving the piston as there are losses through the engine exhaust, to the coolant and due to radiation. The heat energy which is converted to power at this stage is called the indicated power, ip and it is utilized to drive the piston. The energy represented by the gas forces on the piston passes through the connecting rod to the crankshaft. In this transmission there are energy losses due to bearing friction, pumping losses etc. In addition, a part of the energy available is utilized in driving the auxiliary devices like feed pump, valve mechanisms, ignition systems etc. The sum of all these losses, expressed in units of power is termed as frictional power, fp . The remaining energy is the useful mechanical energy and is termed as the brake power, bp . In energy balance, generally, frictional power is not shown separately because ultimately this energy is accounted in exhaust, cooling water, radiation, etc.

1.8 ENGINE PERFORMANCE PARAMETERS

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

(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	$(\bar{s}_{p})$
(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 1.15 shows the diagrammatic representation of energy distribution in an IC engine.

1.8.1 Indicated Thermal Efficiency (η_{ith})

Indicated thermal efficiency is the ratio of energy in the indicated power, ip, to the input fuel energy in appropriate units.

$$[ht]\eta_{ith} = \frac{ip\ [\mathrm{kJ/s}]}{\text{energy in fuel per second [kJ/s]}} \tag{1.3}$$

$$= \frac{ip}{\text{mass of fuel/s} \times \text{calorific value of fuel}} \tag{1.4}$$

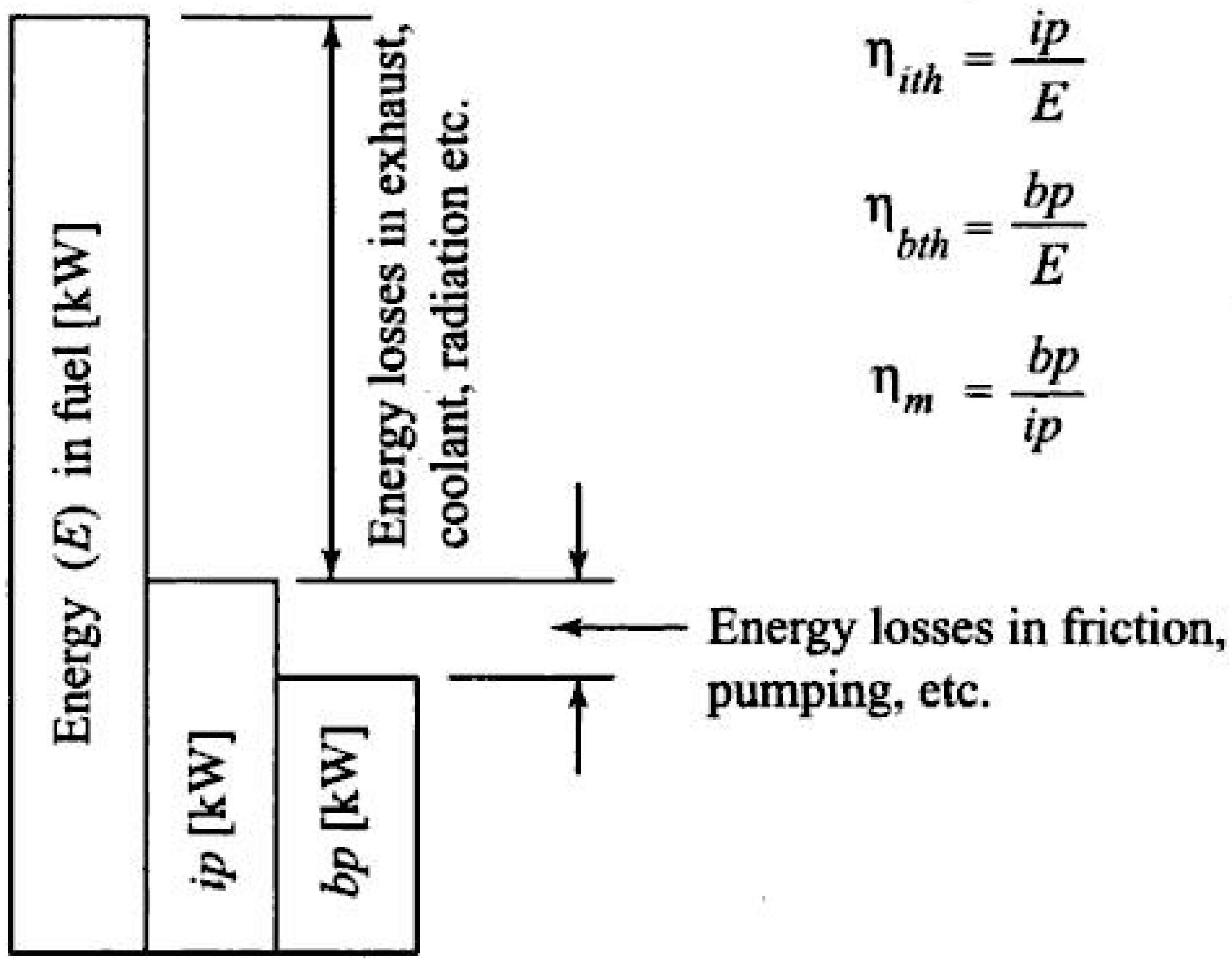


Fig. 1.15 Energy Distribution

1.8.2 Brake Thermal Efficiency (η_{bth})

Brake thermal efficiency is the ratio of energy in the brake power, bp, to the input fuel energy in appropriate units.

$$\eta_{bth} = \frac{bp}{\text{Mass of fuel/s} \times \text{ calorific value of fuel}} \tag{1.5}$$

1.8.3 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).

$$\eta_m = \frac{bp}{ip} = \frac{bp}{bp + fp} \tag{1.6}$$

$$fp = ip - bp \tag{1.7}$$

It can also be defined as the ratio of the brake thermal efficiency to the indicated thermal efficiency.

1.8.4 Volumetric Efficiency (η_v)

This is one of the very important parameters which decides the performance of four-stroke engines. Four-stroke engines have distinct suction stroke and therefore the volumetric efficiency indicates the breathing ability of the engine. It is to be noted that the utilization of the air is what going to determine the power output of the engine. Hence, an engine must be able to take in as much air as possible.

Volumetric efficiency is defined as the volume flow rate of **air** into the intake system divided by the rate at which the volume is displaced by the system.

$$\eta_v = \frac{\dot{m}_a}{\rho_a V_{disp} N/2} \tag{1.8}$$

where ρ_a is the inlet density

An alternative equivalent definition for volumetric efficiency is

$$\eta_v = \frac{\dot{m}_a}{\rho_a V_d} \tag{1.9}$$

It is to be noted that irrespective of the engine whether SI, CI or gas engine, *volumetric rate of air flow is what to be taken into account* and not the mixture flow.

If ρ_a is taken as the atmospheric air density, then η_v represents the pumping performance of the entire inlet system. If it is taken as the air density in the inlet manifold, then η_v represents the pumping performance of the inlet port and valve only.

The normal range of volumetric efficiency at full throttle for SI engines is between 80 to 85% where as for CI engines it is between 85 to 90%. Gas engines have much lower volumetric efficiency since gaseous fuel displaces air and therefore the breathing capacity of the engine is reduced.

1.8.5 Relative Efficiency or Efficiency Ratio (η_{rel})

Relative efficiency or efficiency ratio is the ratio of thermal efficiency of an actual cycle to that of the ideal cycle. The efficiency ratio is a very useful criterion which indicates the degree of development of the engine.

$$\eta_{rel} = \frac{\text{Actual thermal efficiency}}{\text{Air-standard efficiency}} \tag{1.10}$$

1.8.6 Mean Effective Pressure (p_m)

Mean effective pressure is the average pressure inside the cylinders of an internal combustion engine based on the calculated or measured power output. It increases as manifold pressure increases. For any particular engine, operating at a given speed and power output, there will be a specific indicated mean effective pressure, *imep*, and a corresponding brake mean effective pressure, *bmep*. They are derived from the indicated and brake power respectively. For derivation see Chapter 17. Indicated power can be shown to be

$$ip = \frac{p_{im}LAnK}{60 \times 1000} \tag{1.11}$$

then, the indicated mean effective pressure can be written as

$$p_{im} = \frac{60000 \times ip}{LAnK} \tag{1.12}$$

Similarly, the brake mean effective pressure is given by

$$p_{bm} = \frac{60000 \times bp}{LAnK} \tag{1.13}$$

where ip = indicated power (kW)
p_{im} = indicated mean effective pressure (N/m^2)
L = length of the stroke (m)
A = area of the piston (m^2)
N = speed in revolutions per minute (rpm)
n = Number of power strokes
$N/2$ for 4-stroke and N for 2-stroke engines
K = number of cylinders

Another way of specifying the indicated mean effective pressure p_{im} is from the knowledge of engine indicator diagram (p-V diagram). In this case, p_{im}, may be defined as

$$p_{im} = \frac{\text{Area of the indicator diagram}}{\text{Length of the indicator diagram}}$$

where the length of the indicator diagram is given by the difference between the total volume and the clearance volume.

1.8.7 Mean Piston Speed ($\bar{s}_p$)

An important parameter in engine applications is the mean piston speed, $\bar{s}_p$. It is defined as

$$\bar{s}_p = 2LN$$

where L is the stroke and N is the rotational speed of the crankshaft in rpm. It may be noted that $\bar{s}_p$ is often a more appropriate parameter than crank rotational speed for correlating engine behaviour as a function of speed.

Resistance to gas flow into the engine or stresses due to the inertia of the moving parts limit the maximum value of $\bar{s}_p$ to within 8 to 15 m/s. Automobile engines operate at the higher end and large marine diesel engines at the lower end of this range of piston speeds.

1.8.8 Specific Power Output (P_s)

Specific power output of an engine is defined as the power output per unit piston area and is a measure of the engine designer's success in using the available piston area regardless of cylinder size. The specific power can be shown to be proportional to the product of the mean effective pressure and mean piston speed.

$$\text{Specific power output, } P_s = bp/A \tag{1.14}$$

$$= \text{constant} \times p_{bm} \times \bar{s}_p \tag{1.15}$$

As can be seen the specific power output consists of two elements, viz., the force available to work and the speed with which it is working. Thus, for the same piston displacement and *bmep*, an engine running at a higher speed will give a higher specific output. It is clear that the output of an engine can be increased by increasing either the speed or the *bmep*. Increasing the speed involves increase in the mechanical stresses of various engine components. For increasing the *bmep* better heat release from the fuel is required and this will involve more thermal load on engine cylinder.

1.8.9 Specific Fuel Consumption (sfc)

The fuel consumption characteristics of an engine are generally expressed in terms of specific fuel consumption in kilograms of fuel per kilowatt-hour. It is an important parameter that reflects how good the engine performance is. It is inversely proportional to the thermal efficiency of the engine.

$$\text{sfc} = \frac{\text{Fuel consumption per unit time}}{\text{Power}} \tag{1.16}$$

Brake specific fuel consumption and indicated specific fuel consumption, abbreviated as $bsfc$ and $isfc$, are the specific fuel consumptions on the basis of *bp* and *ip* respectively.

1.8.10 Inlet-Valve Mach Index (Z)

In a reciprocating engine the flow of intake charge takes place through the intake valve opening which is varying during the induction operation. Also, the maximum gas velocity through this area is limited by the local sonic velocity. Thus gas velocity is finally chosen by the following equation,

$$u = \frac{A_p}{C_i A_i} V_p \tag{1.17}$$

where u = gas velocity through the inlet valve at smallest flow area
A_p = piston area
A_i = nominal intake valve opening area
C_i = inlet valve flow co-efficient

and

$$\frac{u}{\alpha} = \frac{A_p}{A_i}\frac{V_p}{C_i\alpha} = \left(\frac{b}{D_i}\right)^2 \frac{V_p}{C_i\alpha} = Z \tag{1.18}$$

where b = cylinder diameter
D_i = inlet valve diameter
V_p = mean piston speed
α = inlet sonic velocity
C_i = inlet valve average flow co-efficient
Z = inlet valve Mach index.

Large number of experiments have been conducted on CFR single cylinder engine with gaseous mixtures and short induction pipe lengths, at fixed valve timing and fixed compression ratio, but varying inlet valve diameter and lift. The results are quite revealing as regards the relationship of volumetric efficiency versus Mach index are concerned. From Fig.1.16, it could be seen that the maximum volumetric efficiency is obtainable for an inlet Mach number of 0.55. Therefore, engine designers must take care of this factor to get the maximum volumetric efficiency for their engines.

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

The relative proportions of the fuel and air in the engine are very important from the standpoint of combustion and the efficiency of the engine. This is expressed either as a ratio of the mass of the fuel to that of the air or vice versa.

In the SI engine the fuel-air ratio practically remains a constant over a wide range of operation. In CI engines at a given speed the air flow does not vary with load; it is the fuel flow that varies directly with load. Therefore, the term fuel-air ratio is generally used instead of air-fuel ratio.

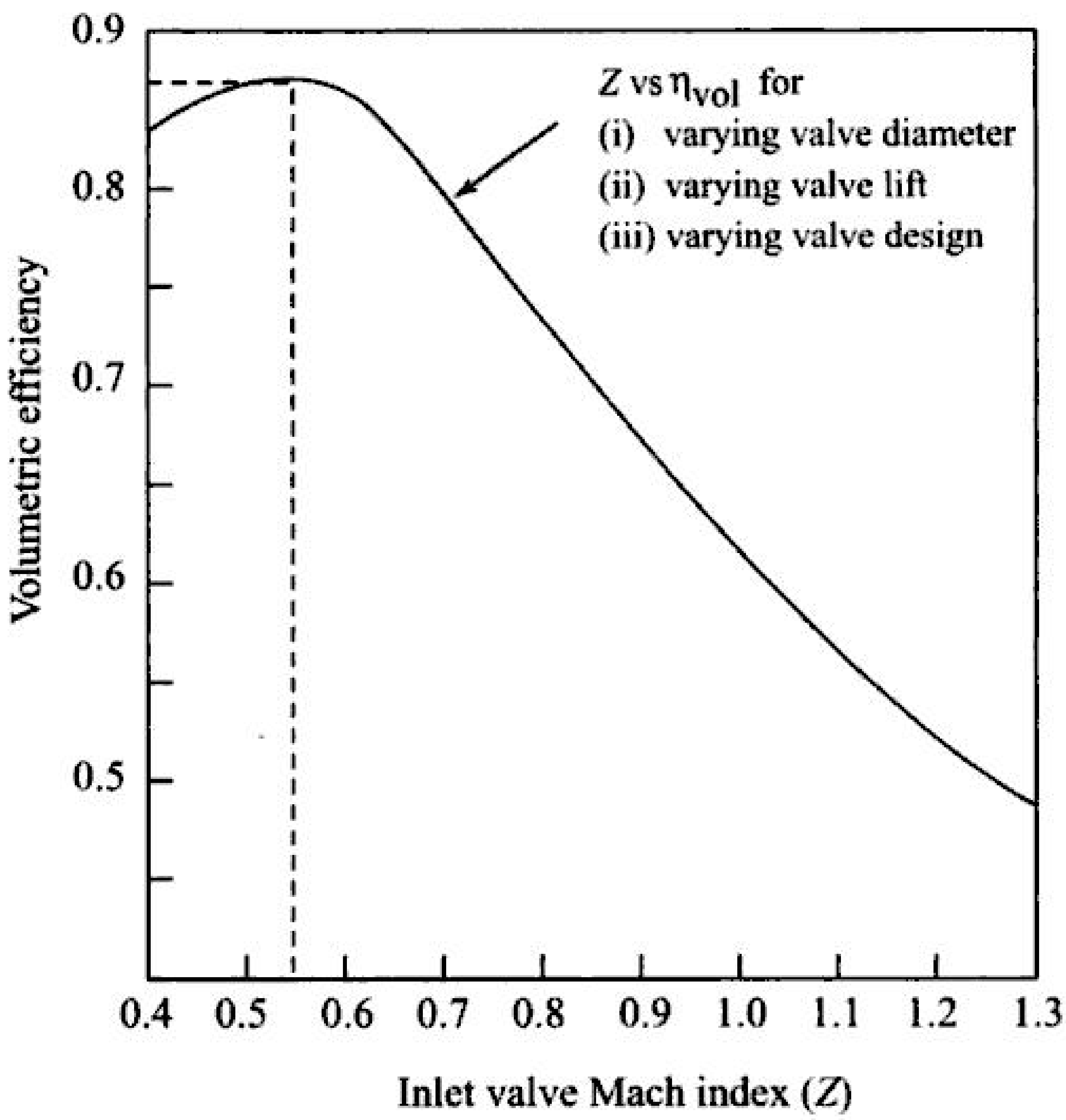


Fig. 1.16 Inlet-Valve Mach Index

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

$$\phi = \frac{\text{Actual fuel-air ratio}}{\text{Stoichiometric fuel-air ratio}} \tag{1.19}$$

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

1.8.12 Calorific Value (CV)

Calorific value of a fuel is the thermal energy released per unit quantity of the fuel when the fuel is burned completely and the products of combustion are cooled back to the initial temperature of the combustible mixture. Other terms used for the calorific value are heating value and heat of combustion.

When the products of combustion are cooled to 25 °C practically all the water vapour resulting from the combustion process is condensed. The heating value so obtained is called the higher calorific value or gross calorific value of the fuel. The lower or net calorific value is the heat released when water vapour in the products of combustion is not condensed and remains in the vapour form.

1.9 DESIGN AND PERFORMANCE DATA

Engine ratings usually indicate the highest power at which manufacturers expect their products to give satisfactory economy, reliability, and durability under service conditions. Maximum torque, and the speed at which it is achieved, is also usually given. Since both of these quantities depend on displaced volume, for comparative analysis between engines of different displacements in a given engine category normalized performance parameters are more useful.

Typical design and performance data for SI and CI engines used in different applications are summarized in Table 1.4. The four-stroke cycle dominates except in the smallest and largest engines. The larger engines are turbocharged or supercharged. The maximum rated engine speed decreases as engine size increases, maintaining the maximum mean piston speed in the range of about 8 to 15 m/s. The maximum brake mean effective pressure for turbocharged and supercharged engines is higher than for naturally aspirated engines. Because the maximum fuel-air ratio for SI engines is higher than for CI engines, their naturally aspirated maximum *bmep* levels are higher. As the engine size increases, brake specific fuel consumption decreases and fuel conversion efficiency increases due to the reduced heat losses and friction. For the large CI engines, brake thermal efficiencies of about 40% and indicated thermal efficiencies of about 50% can be obtained in modern engines.

Table 1.4 *Typical Design and Performance Data for Modern Internal Combustion Engines*

	Operating cycle (Stroke)	Compression ratio	Bore (m)	Stroke/bore ratio	Rated Maximum Speed (rev/min)	Rated Maximum *bmep* (atm)	Weight/Power ratio (kg/kW)	Approx. best *bsfc* (g/kW h)
Spark-ignition engines								
Small (e.g. motorcycles)	2/4	6–10	0.05–0.085	1.2–0.9	4500–7500	4–10	5.5–2.5	350
Passenger cars	4	8–10	0.07–0.1	1.1–0.9	4500–6500	7–10	4–2	270
Trucks	4	7–9	0.09–0.13	1.2–0.7	3600–5000	6.5–7	6.5–2.5	300
Large gas engines	2/4	8–12	0.22–0.45	1.1–1.4	300–900	6.8–12	23–35	200
Wankel engines	4	≈ 9	0.57 dm^3 per chamber		6000–8000	9.5–10.5	1.6–0.9	300
Compression-ignition engines								
Passenger cars	4	16–20	0.075–0.1	1.2–0.9	4000–5000	5–7.5	5–2.5	250
Trucks	4	16–20	0.1–0.15	1.3–0.8	2100–4000	6–9	7–4	210
Locomotive	4/2	16–18	0.15–0.4	1.1–1.3	425–1800	7–23	6–18	190
Large engines	2	10–12	0.4–1	1.2–3.0	110–400	9–17	12–50	180

Worked out Examples

1.1 The cubic capacity of a four-stroke over-square spark-ignition engine is 245 cc. The over-square ratio is 1.1. The clearance volume is 27.2 cc. Calculate the bore, stroke and compression ratio of the engine.

Solution

$$\text{Cubic capacity, } V_s = \frac{\pi}{4}d^2L = \frac{\pi}{4}\frac{d^3}{1.1} = 245$$

$$d^3 = 343$$

$$\text{Bore, } d = \mathbf{7\ cm} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Stroke, } L = \frac{7}{1.1} = \mathbf{6.36\ cm} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Compression ratio, } r = \frac{V_s + V_c}{V_c}$$

$$= \frac{245 + 27.2}{27.2} = \mathbf{10} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.2 The mechanical efficiency of a single-cylinder four-stroke engine is 80%. The frictional power is estimated to be 25 kW. Calculate the indicated power (ip) and brake power (bp) developed by the engine.

Solution

$$\frac{bp}{ip} = 0.8$$

$$ip - bp = 25$$

$$ip - 0.8 \times ip = 25$$

$$ip = \frac{25}{0.2} = \mathbf{125\ kW} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$bp = ip - fp = 125 - 25 = \mathbf{100\ kW} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.3 A 42.5 kW engine has a mechanical efficiency of 85%. Find the indicated power and frictional power. If the frictional power is assumed to be constant with load, what will be the mechanical efficiency at 60% of the load?

Solution

$$\text{Indicated power, } ip = \frac{bp}{\eta_m} = \frac{42.5}{0.85} = \mathbf{50\ kW} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Frictional power, } fp = ip - bp = 50 - 42.5$$

$$= \mathbf{7.5\ kW} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Brake power at 60\% load} = 42.5 \times 0.6 = 25.5\ \text{kW}$$

$$\text{Mechanical efficiency } \eta_m = \frac{bp}{bp + fp} = \frac{25.5}{25.5 + 7.5}$$

$$= 0.773 = \mathbf{77.3\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.4 Find out the speed at which a four-cylinder engine using natural gas can develop a brake power of 50 kW working under following conditions. Air-gas ratio 9:1, calorific value of the fuel = 34 MJ/m^3, Compression ratio 9:1, volumetric efficiency = 70%, indicated thermal efficiency = 35% and the mechanical efficiency = 80% and the total volume of the engine is 2 litres.

Solution

$$\text{Total volume/cylinder, } V_{tot} = \frac{2000}{4} = 500\ \text{cc}$$

$$\text{Swept volume/cylinder, } V_s = \frac{9}{10} \times 500 = 450\ \text{cc}$$

$$\text{Volume of air taken in/cycle} = \eta_v \times V_s$$

$$= 0.7 \times 450 = 315\ \text{cc}$$

$$\text{Volume of gas taken in/cycle} = \frac{315}{9} = 35\ \text{cc}$$

$$\text{Energy supplied/cylinder, } E = 35 \times 10^{-6} \times 34 \times 10^3$$

$$= 1.19\ \text{kJ} \qquad (1)$$

$$\text{Indicated thermal efficiency, } \eta_{ith} = \frac{bp/\eta_m}{\text{Energy supplied/cylinder/s}}$$

$$\text{Energy supplied/cylinder/s}, E_1 = \frac{50/0.8}{0.35 \times 4} = 44.64 \text{ kJ}$$

Now, energy supplied per cylinder in kJ $= \dfrac{E_1}{N/120}$

$$= \frac{44.64 \times 120}{N}$$

$$= \frac{5356.8}{N} \qquad (2)$$

$$\text{Equating (1) and (2) } \frac{5356.8}{N} = 1.19$$

$$N \approx \mathbf{4500 \text{ rpm}}$$ Ans

1.5 A four-stroke, four-cylinder diesel engine running at 2000 rpm develops 60 kW. Brake thermal efficiency is 30% and calorific value of fuel (CV) is 42 MJ/kg. Engine has a bore of 120 mm and stroke of 100 mm. Take $\rho_a = 1.15$ kg/m^3, air-fuel ratio = 15:1 and $\eta_m = 0.8$. Calculate (i) fuel consumption (kg/s); (ii) air consumption (m^3/s); (iii) indicated thermal efficiency; (iv) volumetric efficiency; (v) brake mean effective pressure and (vi) mean piston speed

Solution

$$\text{Fuel consumption, } \dot{m}_f = \frac{bp}{\eta_{bth} \times CV} = \frac{60}{0.3 \times 42000}$$

$$= \mathbf{4.76 \times 10^{-3} \text{ kg/s}}$$ Ans

$$\text{Air consumption} = \frac{\dot{m}_f}{\rho_a} \frac{A}{F} = \frac{4.76 \times 10^{-3}}{1.15} \times 15$$

$$= \mathbf{62.09 \times 10^{-3} \text{ m}^3/\text{ s}}$$ Ans

$$\text{Air flow rate/cylinder} = \frac{62.09 \times 10^{-3}}{4} = 15.52 \times 10^{-3} \text{ m}^3/\text{s}$$

$$\text{Indicated power} = \frac{bp}{\eta_m} = \frac{60}{0.8} = 75 \text{ kW}$$

$$\eta_{ith} = \frac{75}{4.76 \times 10^{-3} \times 42000}$$

$$= 0.37515 = \mathbf{37.51\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Volumetric efficiency =

$$\frac{\text{Actual volume flow rate of air}}{\text{Volume flow rate of air corresponding to displacement volume}} \times 100$$

$$\eta_v = \frac{15.52 \times 10^{-3}}{\frac{\pi}{4} \times 0.12^2 \times 0.10 \times \frac{2000}{120}} \times 100$$

$$= \mathbf{82.3\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Brake mean effective pressure,

$$p_{bm} = \frac{bp}{LAnK}$$

$$= \frac{60}{0.1 \times \frac{\pi}{4} \times 0.12^2 \times \frac{2000}{2 \times 60} \times 4} \times 10^3$$

$$= 7.96 \times 10^5 \text{ N/m}^2 = \mathbf{7.96 \text{ bar}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.6 A single-cylinder, four-stroke hydrogen fuelled spark-ignition engine delivers a brake power of 20 kW at 6000 rpm. The air-gas ratio is 8:1 and the calorific value of fuel is 11000 kJ/m^3. The compression ratio is 8:1. If volumetric efficiency is 70%, indicated thermal efficiency is 33% and the mechanical efficiency is 90%, calculate the cubic capacity of the engine.

Solution

$$\text{Energy input} = \frac{bp/\eta_m}{\eta_{ith}} = \frac{20}{0.8 \times 0.33}$$

$$= 75.76 \text{ kJ/s}$$

$$\text{Number of power strokes/s} = \frac{N}{2 \times 60} = \frac{6000}{120} = 50$$

$$\text{Energy input/power stroke} = \frac{75.76}{50} = 1.52 \text{ kJ}$$

$$\text{Actual volume of H}_2 \times CV = 1.52$$

Actual volume of hydrogen taken in

$$= \frac{1.52 \times 10^6}{11000} = 138.18 \text{ cc}$$

$$\text{Actual volume of air take in} = \frac{A}{F} \times 138.18 = 8 \times 138.18$$

$$= 1105.44 \text{ cc}$$

$$\text{Swept volume, } V_s = \frac{\text{Actual volume of air taken in}}{\eta_v}$$

$$= \frac{1105.44}{0.7} = 1579.2 \text{ cc}$$

$$\text{Cubic capacity of the engine} = V_s \times K = 1579.2 \times 1$$

$$= \mathbf{1579.2 \text{ cc}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.7 Consider two engines with the following details:

Engine I: Four-stroke, four-cylinder, SI engine, indicated power is 40 kW, mean piston speed 10 m/s.

Engine II: Two-stroke, two-cylinder, SI engine, indicated power is 10 kW.

Assume that mean effective pressure of both the engine to be same. Ratio of bore of the engine I:II = 2:1. Show that the mean piston speed of engine II is same as that of engine I.

Solution

$$ip = \frac{P_m LAnK}{60000}$$

$n = \frac{N}{2}$ *for four-stroke engine and* $n = N$ *for two-stroke engine.*

$$\bar{s}_p = 2LN$$

$$\text{For engine I: } 40 = \frac{P_{mI} \times A_I \times \frac{\bar{s}_{pI}}{4} \times 4}{60000}$$

$$\text{For engine II: } 10 = \frac{P_{mII} \times A_{II} \times \frac{\bar{s}_{pII}}{2} \times 2}{60000}$$

$$\frac{40}{10} = \frac{A_I}{A_{II}} \times \frac{10}{\bar{s}_{pII}}$$

$$\bar{s}_{pII} = \frac{A_I}{A_{II}} \times \frac{10}{4} = \left(\frac{d_1}{d_2}\right)^2 \times 2.5$$

$$= \left(\frac{2}{1}\right)^2 \times 2.5 = 10 \text{ m/s}$$

$$\bar{s}_{pII} = \bar{s}_{pI} = \mathbf{10 \text{ m/s}} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

1.8 An one-litre cubic capacity, four-stroke, four-cylinder SI engine has a brake thermal efficiency of 30% and indicated power is 40 kW at full load. At half load, it has a mechanical efficiency of 65%. Assuming constant mechanical losses, calculate: (i) brake power (ii) frictional power (iii) mechanical efficiency at full load (iv) indicated thermal efficiency. If the volume decreases by eight-fold during the compression stroke, calculate the clearance volume.

Solution

Let the brake power at full load be bp and the frictional power be fp.

$$bp + fp = 40 \text{ kW} \qquad (1)$$

$$\text{At half load, } bp = 0.5 \times bp \text{ at full load}$$

$$\eta_m = 0.65 = \frac{0.5\ bp}{0.5\ bp + fp}$$

$$0.5\ bp = 0.65 \times (0.5 \times bp + fp)$$

$$= 0.325 \times bp + 0.65 \times fp$$

$$fp = \frac{0.175}{0.65} \times bp = 0.27bp \qquad (2)$$

$$\text{Using (2) in (1)} \quad bp = \frac{40}{1.27} = \mathbf{31.5 \text{ kW}} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

$$fp = 31.5 \times 0.27 = \mathbf{8.5 \text{ kW}} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

$$\eta_m \text{ at full load} = \frac{31.5}{40} = 0.788 = \mathbf{78.8\%} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

$$\eta_{ith} = \frac{\eta_{bth}}{\eta_m} = \frac{30}{78.8} \times 100 = \mathbf{38\%} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

$$\text{Swept volume/cylinder} = \frac{1000}{4} = 250 \text{ cc}$$

$$r = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c} = 8$$

$$V_c = \frac{250}{7} = \mathbf{35.71 \text{ cc}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.9 A four-stroke petrol engine at full load delivers 50 kW. It requires 8.5 kW to rotate it without load at the same speed. Find its mechanical efficiency at full load, half load and quarter load?

Also find out the volume of the fuel consumed per second at full load if the brake thermal efficiency is 25%, given that calorific value of the fuel = 42 MJ/kg and specific gravity of petrol is 0.75. Estimate the indicated thermal efficiency.

Solution

Mechanical efficiency at full load $= \dfrac{bp}{bp + fp}$

$$= \frac{50}{50 + 8.5} = 0.8547 = \mathbf{85.47\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Mechanical efficiency at half load

$$= \frac{25}{25 + 8.5} = 0.7462 = \mathbf{74.62\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Mechanical efficiency at quarter load

$$= \frac{12.5}{12.5 + 8.5} = 0.5952 = \mathbf{59.52\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Mass flow rate of fuel $\dot{m}_f = \dfrac{bp}{\eta_{bth} \times CV}$

$$= \frac{50}{0.25 \times 42000} = 4.76 \times 10^{-3} \text{ kg/s}$$

Volume flow rate of fuel

$$= \frac{4.76 \times 10^{-3}}{750} = \mathbf{6.34 \times 10^{-6} \text{ m}^3/\text{s}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Indicated thermal efficiency at full load

$$\eta_{ith} = \frac{\eta_{bth}}{\eta_m} = \frac{0.25}{0.8547} = 0.2925 = \mathbf{29.25\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

1.10 The indicated thermal efficiency of four-stroke engine is 32% and its mechanical efficiency is 78%. The fuel consumption rate is 20 kg/h running at a fixed speed. The brake mean pressure developed is 6 bar and the mean piston speed is 12 m/s. Assuming it to be a single cylinder square engine, calculate the crank radius and the speed of the engine. Take $CV = 42000$ kJ/kg.

Solution

$$\text{Brake thermal efficiency, } \eta_{bth} = \eta_{ith} \times \eta_m = 0.32 \times 0.78$$
$$= 0.2496 = 24.96\%$$

Rate of energy input from fuel

$$= \frac{20}{3600} \times 42000 = 233.33 \text{ kW}$$

$$\text{Brake power } bp = \eta_{bth} \times 233.33$$
$$= 0.2496 \times 233.33 = 58.24 \text{ kW}$$

Since it is a square engine, $d = L$.

$$p_{bm} = \frac{bp \times 60000}{LAnK}$$
$$= \frac{58.24 \times 60000}{\frac{\pi}{4}L^3 \times n \times 1} = 6 \times 10^5$$
$$L^3 n = 7.415 \qquad (1)$$

Note L *is in m and* N *in per minute. Now,*

$$\bar{s}_p = 12 = \frac{2LN}{60}$$
$$LN = 360 \qquad (2)$$

Dividing (1) by (2), gives,

$$L^2 \frac{n}{N} = 0.0206$$

For a four-stroke engine $n/N = \frac{1}{2}$.

$$L = \sqrt{0.0206 \times 2} = 0.203 \text{ m}$$
$$= 203 \text{ mm}$$

$$\text{Crank radius} = \frac{203}{2} = 101.5 \text{ mm}$$

$$\text{Speed, } N = \frac{360}{L} = \frac{360}{0.203}$$

$$= \mathbf{1773.4 \text{ rpm}} \quad \underleftarrow{\text{Ans}}$$

Review Questions

1.1 Define the following : (i) engine and (ii) heat engine.

1.2 How are heat engines classified?

1.3 Explain the basic difference in their work principle?

1.4 Give examples of EC and IC engines.

1.5 Compare EC and IC engines.

1.6 What are the important basic components of an IC engine? Explain them briefly.

1.7 Draw the cross-section of a single cylinder spark-ignition engine and mark the important parts.

1.8 Define the following :

(i)	*bore*	*(iv)*	*clearance volume*
(ii)	*stroke*	*(v)*	*compression ratio*
(iii)	*displacement volume*	*(vi)*	*cubic capacity*

Mention the units in which they are normally measured.

1.9 What is meant by TDC and BDC? In a suitable sketch mark the two dead centres.

1.10 What is meant by cylinder row and cylinder bank?

1.11 With neat sketches explain the working principle of four-stroke spark-ignition engine.

1.12 Classify the internal combustion engine with respect to

(i)	cycle of operation	(iv)	type of ignition
(ii)	cylinder arrangements	(v)	type of fuels used
(iii)	method of charging the cylinder	(vi)	type of cooling

1.13 In what respects four-stroke cycle CI engine differ from that of an SI engine?

1.14 What is the main reason for the development of two-stroke engines and what are the two main types of two-stroke engines?

1.15 Describe with a neat sketch the working principle of a crankcase scavenged two-stroke engine.

1.16 Draw the ideal and actual indicator diagrams of a two-stroke SI engine. How are they different from that of a four-stroke cycle engine?

1.17 Compare four-stroke and two-stroke cycle engines. Bring out clearly their relative merits and demerits.

1.18 Compare SI and CI engines with respect to

(i)	*basic cycle*	*(v)*	*compression ratio*
(ii)	*fuel used*	*(vi)*	*speed*
(iii)	*introduction of fuel*	*(vii)*	*efficiency*
(iv)	*ignition*	*(viii)*	*weight*

1.19 Discuss in detail the application of various types of internal combustion engines.

1.20 Give an account of the first law analysis of an internal combustion engine.

1.21 Show by means of a diagram the energy flow in a reciprocating internal combustion engine.

1.22 What is meant by mean piston speed? Explain its importance.

1.23 Discuss briefly the design performance data of SI and CI engines.

1.24 Define the following efficiencies :

(i)	*indicated thermal efficiency*	*(iv) relative efficiency*
(ii)	*brake thermal efficiency*	*(v) volumetric efficiency*
(iii)	*mechanical efficiency*	

1.25 Explain briefly

(i) mean effective pressure
(ii) specific output
(iii) specific fuel consumption
(iv) fuel-air ratio
(v) heating value of the fuel

Exercise

1.1 A diesel engine has a brake thermal efficiency of 30 per cent. If the calorific value of the fuel is 42000 kJ/kg. Find its brake specific fuel consumption. *Ans:* 0.2857 kg/kW h

1.2 A gas engine having a cylinder 250 mm bore and 450 mm stroke has a volumetric efficiency of 80%. Air-gas ratio equals 9:1, calorific value of fuel 21000 kJ per m^3 at NTP. Calculate the heat supplied to the engine per working cycle. If the compression ratio is 5:1, what is the heat value of the mixture per working stroke per m^3 of total cylinder volume? *Ans:* (i) 36.08 kJ (ii) 1306.8 kJ/m^3

1.3 A certain engine at full load delivers a brake power of 36.8 kW at certain speed. It requires 7.36 kW to overcome the friction and to rotate the engine without load at the same speed. Calculate its mechanical efficiency. Assuming that the mechanical losses remain constant, what will be the mechanical efficiency at half load and quarter load. *Ans:* (i)83.33% (ii) 71.42% (iii) 55.55%

1.4 An engine used for pumping water develops a brake power of 3.68 kW. Its indicated thermal efficiency is 30%, mechanical efficiency is 80%, calorific value of the fuel is 42,000 kJ/kg and its specific gravity = 0.875. Calculate (i) the fuel consumption of the engine in (a) kg/h (b) litres/h (ii) indicated specific fuel consumption and (iii) brake specific fuel consumption. *Ans:* (i) (a) 1.31 kg/h (b) 1.5017 litres/h (ii) 0.2856 kg/kW h (iii) 0.3571 kg/kW h

1.5 A two-stroke CI engine develops a brake power of 368 kW while its frictional power is 73.6 kW. Its fuel consumption is 180 kg/h and works with an air-fuel ratio of 20:1. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) mechanical efficiency (iii) air consumption per hour (iv) indicated thermal efficiency and (v) brake thermal efficiency. *Ans:* (i) 441.6 kW (ii) 83.3% (iii) 3600 kg/h (iv) 21% (v) 17.5%

1.6 Compute the brake mean effective pressure of a four-cylinder, four-stroke diesel engine having 150 mm bore and 200 mm stroke which develops a brake power of 73.6 kW at 1200 rpm. *Ans:* 5.206 bar

1.7 Compute the brake mean effective pressure of a four-cylinder, two-stroke engine, 100 mm bore 125 mm stroke when it develops a torque of 490 Nm. *Ans:* 7.84 bar

1.8 Find the brake thermal efficiency of an engine which consumes 7 kg of fuel in 20 minutes and develops a brake power of 65 kW. The fuel has a heating value of 42000 kJ/kg. *Ans:* 26.53%

1.9 Find the mean piston speed of a diesel engine running at 1500 rpm. The engine has a 100 mm bore and L/d ratio is 1.5. *Ans:* 7.5 m/s

1.10 An engine is using 5.2 kg of air per minute while operating at 1200 rpm. The engine requires 0.2256 kg of fuel per hour to produce an indicated power of 1 kW. The air-fuel ratio is 15:1. Indicated thermal efficiency is 38% and mechanical efficiency is 80%. Calculate (i) brake power and (ii) heating value of the fuel. *Ans:* (i) 73.7 kW (ii) 41992.89 kJ/kg

1.11 A four-cylinder, four-stroke, spark-ignition engine has a bore of 80 mm and stroke of 80 mm. The compression ratio is 8. Calculate the cubic capacity of the engine and the clearance volume of each cylinder. What type of engine is this? *Ans:* (i) 1608.4 cc (ii) 57.4 cc (iii) Square engine

1.12 A four-stroke, compression-ignition engine with four cylinders develops an indicated power of 125 kW and delivers a brake power of 100 kW. Calculate (i) frictional power (ii) mechanical efficiency of the engine. *Ans:* (i) 25 kW (ii) 80%

1.13 An engine with 80 per cent mechanical efficiency develops a brake power of 30 kW. Find its indicated power and frictional power. If frictional power is assumed to be constant, what will be the mechanical efficiency at half load. *Ans:* (i) 37.5 kW (ii) 7.5 kW (iii) 66.7%

1.14 A single-cylinder, compression-ignition engine with a brake thermal efficiency of 30% uses high speed diesel oil having a calorific value of 42000 kJ/kg. If its mechanical efficiency is 80 per cent, calculate (i) $bsfc$ in kg/kW h (ii) $isfc$ in kg/kW h *Ans:* (i) 0.286 kg/kW h (ii) 0.229 kg/kW h

1.15 A petrol engine uses a fuel of calorific value of 42000 kJ/kg and has a specific gravity of 0.75. The brake thermal efficiency is 24 per cent and mechanical efficiency is 80 per cent. If the engine develops a brake power of 29.44 kW, calculate (i) volume of the fuel consumed per second (ii) indicated thermal efficiency

Ans: (i) 2.92×10^{-3} kg/s (ii) 30%

1.16 A single-cylinder, four-stroke diesel engine having a displacement volume of 790 cc is tested at 300 rpm. When a braking torque of 49 Nm is applied, analysis of the indicator diagram gives a mean effective pressure of 980 kPa. Calculate the brake power and mechanical efficiency of the engine.

Ans: (i) 1.54 kW (ii) 79.4%

1.17 A four-stroke SI engine delivers a brake power of 441.6 kW with a mechanical efficiency of 85 per cent. The measured fuel consumption is 160 kg of fuel in one hour and air consumption is 410 kg during one sixth of an hour. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) frictional power (iii) air-fuel ratio (iv) indicated thermal efficiency (v) brake thermal efficiency.

Ans: (i) 519.5 kW (ii) 77.9 kW (iii) 15.5 (iv) 28.1% (v) 23.9%

1.18 A two-stroke CI engine develops a brake power of 368 kW while 73.6 kW is used to overcome the friction losses. It consumes 180 kg/h of fuel at an air-fuel ratio of 20:1. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) mechanical efficiency; (iii) Air consumption (iv) indicated thermal efficiency (v) brake thermal efficiency.

Ans: (i) 441.6 kW (ii) 83.3% (iii) 1 kg/s (iv) 21% (v) 17.5%

1.19 A four-stroke petrol engine delivers a brake power of 36.8 kW with a mechanical efficiency of 80%. The air-fuel ratio is 15:1 and the fuel consumption is 0.4068 kg/kW h. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) frictional power(iii) brake thermal efficiency (iv) indicated thermal efficiency (v) total fuel consumption (vi) air consumption/second.

Ans: (i) 46 kW (ii) 9.2 kW (iii) 21% (iv) 26.25% (v) 0.0042 kg/s (vi) 0.063 kg/s.

1.20 A spark-ignition engine has a fuel-air ratio of 0.067. How many kg of air per hour is required for a brake power output of 73.6 kW at an overall brake thermal efficiency of 20%? How many m^3 of air is required per hour if the density of air

is 1.15 kg/m^3. If the fuel vapour has a density four times that of air, how many m^3 per hour of the mixture is required? The calorific value of the fuel is given as 42000 kJ/kg.
Ans: (i) 470.75 kg/h (ii) 409.35 m^3/h (iii) 416.21 m^3/h

1.21 A four-stroke CI engine having a cylinder diameter of 39 cm and stroke of 28 cm has a mechanical efficiency of 80%. Assume the frictional power to be 80 kW. Its fuel consumption is 86 kg/h with an air-fuel ratio of 18:1. The speed of the engine is 2000 rpm. Calculate (i) indicated power (ii) if η_{ith} is 40%, calculate the calorific value of the fuel used (iii) p_{im} (iv) $\dot{m}_a$/hour (v) $\bar{s}_p$.
Ans: (i) 400 kW (ii) 41860 kJ (iii) 12.13 bar (iv) 1548 kg/h (v) 18.7 m/s

1.22 A four-stroke LPG engine having a cylinder 250 mm diameter and stroke of 300 mm has a volumetric efficiency of 70% at atmospheric conditions. Gas to air ratio is 8:1. Calorific value of the fuel is 100 MJ/m^3 at atmospheric conditions. Find the heat supplied to the engine per working cycle. If the compression ratio is 10, what is the heat value of the mixture per working stroke per m^3 of the total cylinder volume?
Ans: (i) 128.8 kJ (ii) 7.8 MJ

1.23 A four-cylinder spark-ignition engine has the following dimensions: bore = 680 mm and a crank radius = 375 mm. If the compression ratio is 8:1, determine the (i) stroke length (ii) swept volume (iii) cubic capacity (iv) clearance volume and (v) total volume. If the volumetric efficiency is 80% determine the (vi) actual volume of air aspirated/stroke in each cylinder?
Ans: (i) 750 mm (ii) 1.088 m^3 (iii) 0.272 m^3 (iv) 0.039 m^3 (v) 0.0311 m^3 (vi) 0.2176 m^3

1.24 An engine with an indicated thermal efficiency of 25% and mechanical efficiency of 75% consumes 25 kg/h of fuel at a fixed speed. The brake mean effective pressure is 5 bar and the mean piston speed is 15 m/s. Assuming it is a single cylinder square engine determine the crank radius and the speed in rpm. Take CV of the fuel = 42 MJ/kg. *Ans:* (i) 68.2 mm (ii) 3300 rpm

1.25 A four-cylinder SI engine running at 1200 rpm gives 18.87 kW as brake power. When one cylinder missed firing the average torque was 100 Nm. Calculate the indicated thermal efficiency if the calorific value of fuel is 42000 kJ/kg. The engine uses 0.335 kg of fuel per kW/h. What is the mechanical efficiency of the engine? *Ans:* (i) 34.2% (ii) 74.9%

1.26 A certain engine with a bore of 250 mm has an indicated thermal efficiency of 30%. The $bsfc$ and specific power output are 0.35 kg/kW h and 90 kW/m^2. Find the mechanical efficiency and brake thermal efficiency of the engine. Take the calorific value of the fuel as 42 MJ/kg. *Ans:* (i) 81.7% (ii) 24.5%

1.27 A single-cylinder, four-stroke engine having a cubic capacity of 0.7 litre was tested at 200 rpm. From the indicator diagram the mean effective pressure was found to be 10^6 N/m^2 and the mechanical efficiency is 75%. Find the frictional power of the engine if the engine is an over-square engine with a over-square ratio of 0.8. Calculate the bore and stroke.
Ans: (i) 0.29 kW (ii) 41.5 mm (iii) 89.34 mm (iv) 111.68 mm

1.28 In a performance test on a four-stroke engine, the indicator diagram area was found to be 5×10^{-4} m^2 and the length of the indicator diagram was 0.05 m. If the y-axis has a scale of 1 m = 50 MPa, find the $imep$ of the engine given that bore = 150 mm, stroke = 200 mm. The measured engine speed was 1200 rpm. Also calculate the ip and $isfc$ of the engine if the fuel injected per cycle is 0.5 cc with the specific gravity of 0.8.
Ans: (i) 5 bar (ii) 70.68 kW (iii) 203 g/kW h

1.29 A four-stroke, four-cylinder automotive engine develops 150 Nm brake torque at 3000 rpm. Assuming *bmep* to be 0.925 bar, find (i) brake power (ii)displacement volume (iii) stroke (iv) bore. Take $\bar{s}_p$ = 12 m/s.
Ans: (i) 47.124 kW (ii) 5.1 $\times 10^{-3}$ m^3 (iii) 120 mm (iv) 233 mm

1.30 A single-cylinder, four-stroke, engine has a $bsfc$ of 1.13×10^{-5} kg/kW h and a fuel consumption rate of 0.4068 kg/h. The specific power output of the engine is 0.33 kW/cm^2. If the engine runs at 3000 rpm find the displacement volume of the cylinder and if the $\bar{s}_p$ is 15 m/s, find the *bmep*.

Ans: (i) 900 cc (ii) 4.44 bar

2

REVIEW OF BASIC PRINCIPLES

2.1 INTRODUCTION

The main objectives in studying the theory of IC engines can be summarized as

(i) To have a better understanding of the various processes taking place and the conditions prevailing in the engine cylinder.

(ii) To predict the changes in power, fuel consumption and reliability resulting from the changes in the operating conditions or changes in the design features of a given engine.

(iii) To predict the operating characteristics of a new design from test results on a similar engine of a different size.

Basic knowledge and the familiarity of the principles of thermodynamics, physics and chemistry is a must for achieving the above objectives. However, mere familiarity alone will not serve the purpose. The familiarity must be supplemented by the ability to apply these principles correctly. This requires an understanding of fundamentals used to derive the important formulae. The main aim of this chapter is to place before the reader the basic principles of thermodynamics, physics, chemistry and other relevant information for the understanding of the various details discussed in the subsequent chapters. Although a fairly exact treatment has been attempted, the reader is advised to go through other relevant textbooks to get more detailed account of the various basic principles.

2.2 BASIC AND DERIVED QUANTITIES

In this section we will briefly recall the definition of basic quantities like length, time, mass, weight, acceleration due to gravity, force and pressure. These quantities will be used to derive other quantities.

2.2.1 Length

The unit of length is metre. Originally, metal bar standards were used to define length. They have been abandoned now in favour of a standard length which can be reproduced in any laboratory in the world. Length is now defined in terms of the atomic standards. The orange light emitted by the gas krypton under special experimental conditions is used to define length. One metre is defined to be 1,650,763.73 wavelengths of the orange light Krypton-86. However, for convenience in engine applications, millimeter (mm) and centimeter (cm) may also be used to represent length.

2.2.2 Time

The unit of time is second. Time is also defined in terms of atomic standards. The time standard is based on the duration of the periods of the radiation of the atom cesium-133.

$$1 \text{ second} = 9,192,631,770 \text{ periods}$$

The second, minute, hour, day and year are used throughout the world. Their interrelation requires no description here.

2.2.3 Mass and Weight

The mass of a body, m is the amount of matter in the body. The mass of a given body therefore is the same anywhere in the universe. The unit of mass is kilogram (kg). However, for engine applications, gram (gm) may also be used as the unit of mass.

The weight of a body, w is the force with which the earth attracts the body. The weight of a given body will change with the value of acceleration due to gravity, g.

The weight, w is always equal to the number of kg mass, m times the local value of g or

$$w = mg \tag{2.1}$$

2.2.4 Gravitational Constant

The force of attraction by the earth experienced by any body is called the gravitational force. Since, the mass of most of the bodies under consideration are negligible compared to that of earth this force effects only the other bodies. This results in bodies not able to float and a body without support experiences a motion in the direction of the gravitational force. The acceleration of such freely falling body is called acceleration due to gravity. This is found to vary from place to place on the surface of the earth and hence an average value is used for engineering calculations, viz., 9.81 m/s^2 or 32 ft/s^2.

In the derivation of unit of force it is required to prescribe mass and acceleration. In prescribing the acceleration, two conventions are used, one to prescribe unit acceleration and the other to prescribe the acceleration due to gravity. In the four methods of systems of units so far evolved two use the former and two later.

$$\begin{array}{llll} \text{F.P.S.} & : & 1 \text{ pound} & = 1 \text{ slug} \times 32 \text{ ft/s}^2 \\ \text{C.G.S.} & : & 1 \text{ dyne} & = 1 \text{ gm} \times 1 \text{ m/s}^2 \\ \text{M.K.S.} & : & 1 \text{ kgf} & = 1 \text{ kg} \times 9.81 \text{ m/s}^2 \\ \text{SI} & : & 1 \text{ Newton} & = 1 \text{ kg} \times 1 \text{ m/s}^2 \end{array}$$

In using M.K.S. system care should be taken to differentiate between kgf and kg since 1 kgf is actually equal to 9.81 Newton. This is done by defining a gravitational constant, g_c which is numerically equal to the acceleration due to gravity but without dimensions. Since it is proposed to use SI units in this book use of g_c is not elaborated further.

2.2.5 Force

In this book the unit of force will be Newton and is abbreviated as N. A Newton is the force required to give an acceleration of 1 m/s^2 to a mass of one kg of matter. This definition is adopted because it is convenient to metrologists; its adoption is rendered possible by the universal validity of Newton's Second Law of Motion.

2.2.6 Pressure

Fluids, gas or liquid exert forces on its boundaries. These forces are not concentrated at particular points, but are distributed. It is therefore useful to define the quantity pressure as the force exerted on a surface which is normal to the force, divided by the area of the surface. The units of pressure is N/m^2. However, it may also be expressed in bar, Pascal and standard atmosphere, (atm).

The inter relation between bar, Pascal and standard atmosphere with N/m^2 is given below.

1 Pascal	(Pa)	$=$	$1\ N/m^2$
1 bar	(bar)	$=$	10^5 Pascal
1 standard atmosphere	(atm)	$\approx$	$1.01325 \times 10^5\ N/m^2$

2.2.7 Temperature

The unit of temperature is Kelvin. The Kelvin is 1/273.16 of the thermodynamic temperature of the triple point of water, viz., the point at which liquid water, water vapour and ice are in equilibrium.

2.3 THE GAS LAWS

During the various strokes of the engine different pressures and temperatures exist in the engine cylinder and all gases encountered in engine operation are assumed to obey at all these pressures and temperatures the simple gas law,

$$pV = N\overline{R}T \tag{2.2}$$

where	p	$=$	Pressure of the gas	(N/m^2)
	V	$=$	Volume occupied by the gas	(m^3)
	N	$=$	Number of moles of gas	(kmol)
	$\overline{R}$	$=$	Universal gas constant	(8.314 kJ/kmol K)
	T	$=$	Absolute temperature	(K)

The above equation is an approximation sufficiently close for engineering purposes. This law is very simple and 'always' applies to 'any' gas, regardless of the process to which the gas is subjected, as long as the gas is not near its liquefaction.

2.3.1 The Mole

A kilomol of any gas is a quantity of the gas whose weight is equal to the molecular weight in kg. Thus

$$N = \frac{m}{M} \tag{2.3}$$

where	m	$=$	Mass of gas (kg)
	M	$=$	Molecular weight of the gas, (g/mol)

Combining Eqs.2.2 and 2.3, the gas law may be written as

$$pV = \frac{m}{M}\overline{R}T = m\left(\frac{\overline{R}}{M}\right)T = mRT \qquad (2.4)$$

The quantity $\overline{R}/M$ is called the characteristic gas constant and is usually denoted by R. Since M is different for different gases, the characteristic gas constant will also be different for different gases.

2.3.2 Specific Volume and Density

The volume occupied by unit mass of a substance is known as its specific volume (m^3/kg) and is denoted by v. The inverse of the specific volume is the density (kg/m^3) and is denoted by ρ. The specific volume and density of the gas depend upon the pressure and temperature.

2.3.3 Simplification of the Gas Laws

When the number of moles m/M of gas under consideration is fixed, both N and R are constant. Then from Eqs.2.2 and 2.4,

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} = \cdots = \frac{p_n v_n}{T_n} = \text{Constant} \qquad (2.5)$$

This simple relation is always true when applied to a constant number of moles of a gas and makes it possible to calculate the effect of changing any two of the variables (pressure, volume or temperature) upon the remaining one.

2.3.4 Mixture of Gases

During expansion and exhaust strokes (also during suction and compression) the cylinder is occupied by two or more gases. Each of these gases diffuses and fills up the entire space, obeying the gas law just as though the other gases were not present. This principle is called Dalton's Law. For example, let three gases 1, 2 and 3 occupy the cylinder at a particular instant. Let the volume of the cylinder at that instant be V. If the temperature of the mixture is T, then the pressure exerted on the walls of the cylinder by gas 1 is, by the gas law,

$$p_1 = \frac{N_1\overline{R}T}{V}$$

and the pressure exerted by gas 2 is

$$p_2 = \frac{N_2 \overline{R} T}{V}$$

The total pressure within the container is

$$p_t = p_1 + p_2 + p_3 + \ldots = \frac{(N_1 + N_2 + N_3 + \ldots)\overline{R}T}{V} = \frac{N_t \overline{R} T}{V} \quad (2.6)$$

From the above expression for p_1 and p_2, it is seen that

$$\frac{p_1}{p_2} = \frac{N_1}{N_2}$$

and from Eq.2.6

$$\frac{p_1}{p_t} = \frac{N_1}{N_t} \quad (2.7)$$

The pressures p_1, p_2 etc. which go to make up the total pressure are called partial pressure.

2.4 FORMS OF ENERGY

An engine takes in chemical energy in the fuel and changes most of this into sensible energy during the combustion process, and then transforms part of the sensible energy into mechanical work on the piston head.

During the suction stroke the atmosphere does work on the mixture as it forces it into the cylinder. During the exhaust stroke the products of combustion do work on the atmosphere as they leave the cylinder. In order to understand these processes, it is necessary to study the various forms of energy in some detail.

Energy is defined as the capacity to do work. If a machine or body, because of its position, temperature or velocity is capable of doing work on another body, it is said to have energy. Work is done whenever motion takes place against a resistance. The amount of work done is the distance moved, times the magnitude of the resisting force in the direction of motion.

$$\text{Work} = \text{Force} \times \text{distance}$$

Work and energy are expressed in the same units, viz., Nm or Joule.

When energy is being transferred from one body to another by virtue of a temperature difference, it is called heat and is usually measured in Joules.

Work and heat are related by the mechanical equivalent of heat J. The relation is

$$W = JH \tag{2.8}$$

where W is the work, H is the heat and $J = 427$ kgf m/kcal. Thus work and heat can be expressed in any units of convenience with the help of J.

2.4.1 Transfer of Energy

Heat and work are both energy in transit. When two bodies at different temperatures are brought in contact, heat flows from a body at higher temperature to a body at lower temperature. Similarly, when work is done, the system doing work loses energy and the system upon which the work is being done gains exactly this amount of energy.

2.4.2 Stored Energy

In the two cases above, when the process is completed the energy gained by the second body is not called heat or work, since, heat is no longer flowing and work is no longer being done. The additional energy which is now stored in the second body may be called stored energy, E, which includes many types of energies like kinetic energy, potential energy, surface tension, electrical energy etc. depending upon the manner in which they are stored.

2.4.3 Potential Energy

It is the energy contained in the system by virtue of its elevation with reference to an arbitrarily chosen datum, usually the sea level. Or alternatively it is equivalent to energy required to raise the system from an arbitrary datum.

2.4.4 Kinetic Energy

Any body of mass, m kg, moving with a velocity of C metres per second will do $\frac{1}{2}mC^2$ Joules of work before coming to rest. In this state of motion, the body is said to have kinetic energy equal to $\frac{1}{2}mC^2$ Joules.

2.4.5 Internal Energy

A body may possess energy due to the motion or position or the attraction between the particles of which it is made. In a permanent gas (i.e. a gas which is far from its liquefaction point), the internal energy, E, will usually be in the form of

(i) translational motion of the molecules of the gas and

(ii) motion of the atoms within the molecules

These two kinds of internal energy are known as sensible internal energy, although only that part which is due to molecular translation can actually be felt as warmth. For a given quantity of a particular permanent gas, the amount of sensible internal energy present is fixed by the temperature of the gas alone. The sensible internal energy is denoted by U_s.The gas under consideration may also contain chemical energy which is usually denoted by U_c.

The sum of the sensible internal energy and the chemical energy of the fuel present and is called the total internal energy, U, i.e.,

$$U = U_s + U_c \tag{2.9}$$

If during any combustion process no heat is allowed to escape from the gases and no work is permitted to be done by the gases, then at any time during the process the increase in sensible energy due to combustion is exactly equal to the loss in chemical energy as the fuel is used up. The total internal energy therefore remains unchanged.

2.5 FIRST LAW OF THERMODYNAMICS

Whenever a system undergoes a cyclic change, the algebraic sum of work transfer is proportional to the algebraic sum of heat transfer as work and heat are mutually convertible from one form into the other.

For a closed system, a change in the energy content is the algebraic difference between the heat supply, Q, and the work done, W, during any change in the state. Mathematically,

$$dE = \delta Q - \delta W \tag{2.10}$$

The energy E may include many types of energies like kinetic energy, potential energy, electrical energy, surface tension etc., but from the thermodynamic point of view these energies are ignored and the energy due to rise in temperature alone is considered. It is denoted by U and the first law is written as:

$$dU = \delta Q - \delta W \tag{2.11}$$

or during a process 1-2, dU can be denoted as

$$U_2 - U_1 = \int_1^2 (\delta Q - \delta W) \tag{2.12}$$

2.6 PROCESS

A change in the condition or state of a substance is called a process. The process may consist of heating, flow from one place to another, expansion etc. In general, a process may be divided into non-flow or flow processes.

2.6.1 Non-flow Processes

If there is no flow of material into or out of a system during a process, it is called a non-flow process. This is the simplest kind of process, and much can be learned about it by applying the principle of conservation of energy.

2.7 ANALYSIS OF NON-FLOW PROCESSES

The purpose of the analysis is to apply the First Law of Thermodynamics to process in which a non-flow system changes from one state to the other and also to develop some useful relations.

2.7.1 Constant Volume or Isochoric Process

This process is usually encountered in the analysis of air-standard Otto, Diesel and Dual cycles. Figure 2.1 shows the constant volume process on a p-V diagram.

As there is no change in volume, the work $\int(pdV)$ is zero. Hence, according to the first law for the constant volume process the change in internal energy is equal to the heat transfer, i.e.,

$$dU = \delta Q = mC_v dT = mC_v(T_2 - T_1) \tag{2.13}$$

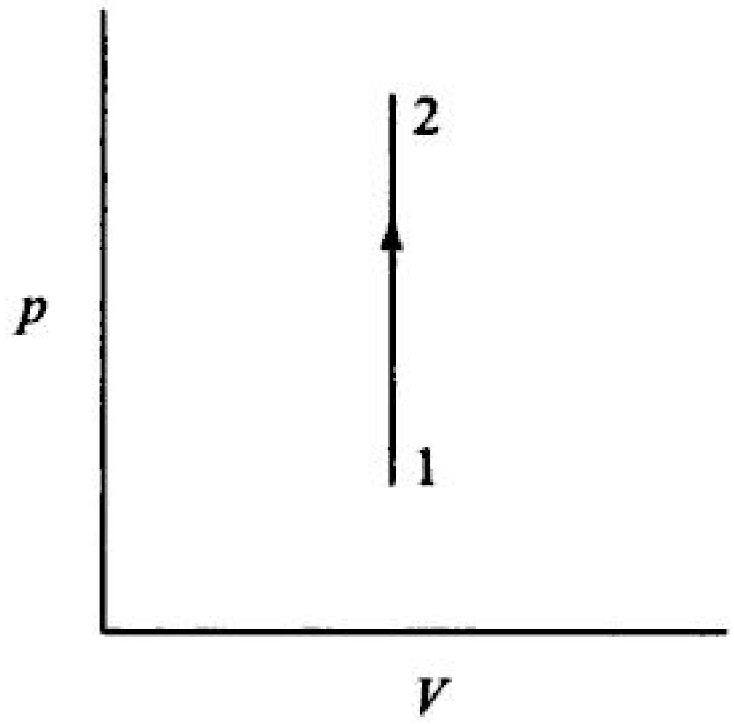


Fig. 2.1 Constant Volume Process

For unit mass,

$$du = \delta q = C_v dT$$

$$C_v = \left(\frac{du}{dT}\right)_V \tag{2.14}$$

i.e., specific heat at constant volume is the rate of change of internal energy with respect to absolute temperature.

2.7.2 Constant Pressure or Isobaric Process

Figure 2.2 shows a system that changes from state 1 to state 2 at constant pressure. Application of first law yields,

$$\delta Q = dU + pdV = d(U + pV) = dH \tag{2.15}$$

where H is known as the enthalpy. Thus, during constant pressure process, heat transfer is equal to change in enthalpy or

$$dH = \delta Q = mC_p dT \tag{2.16}$$

For unit mass,

$$\delta q = dh = C_p dT \tag{2.17}$$

$$C_p = \left(\frac{dh}{dT}\right)_p \tag{2.18}$$

i.e., specific heat at constant pressure is the rate of change of specific enthalpy with respect to absolute temperature.

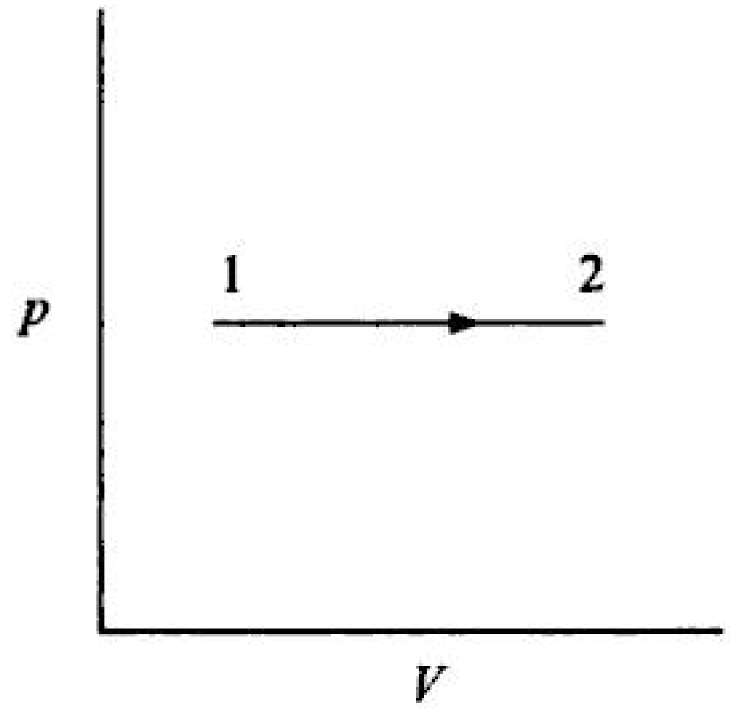


Fig. 2.2 Constant Pressure Process

2.7.3 Constant Temperature or Isothermal Process

The isothermal process on a p-V diagram is illustrated in Fig.2.3. As there is no temperature change during this process, there will not be any change in internal energy i.e., $dU = 0$, then according to the first law

$$\delta Q = \delta W \tag{2.19}$$

or

$$Q_{1-2} = \int_1^2 pdV = p_1 V_1 \log_e \left(\frac{V_2}{V_1}\right) \tag{2.20}$$

2.7.4 Reversible Adiabatic or Isentropic Process

If a process occurs in such a way that there is no heat transfer between the surroundings and the system, but the boundary of the system moves giving displacement work, the process is said to be adiabatic. Such a process is possible if the system is thermally insulated from the surroundings. Hence, $\delta Q = 0$, therefore,

$$\delta W = -\delta U = -mC_v dT \tag{2.21}$$

Reversible adiabatic process is also known as isentropic process. Let $pV^\gamma = C$ be the law of the isentropic process. For unit mass flow,

$$q_{1-2} = 0 = w_{1-2} + u_2 - u_1 \tag{2.22}$$

or

$$w_{1-2} = -(u_2 - u_1)$$

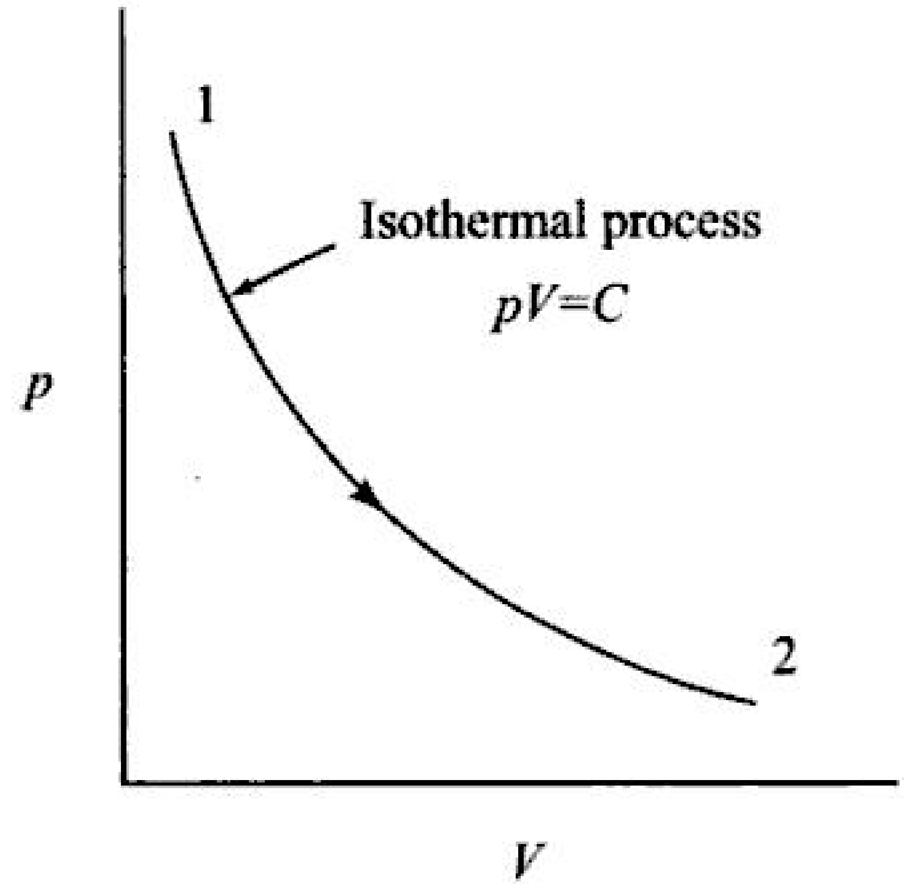


Fig. 2.3 Constant Temperature Process

In other words, work is done at the expense of internal energy

$$W_{1-2} = \int_1^2 pdV = \int_1^2 \frac{C}{V^\gamma}dV \tag{2.23}$$

$$= \frac{[CV^{1-\gamma}]_{V1}^{V2}}{1-\gamma} = \frac{CV_2^{1-\gamma} - CV_1^{1-\gamma}}{1-\gamma}$$

when $C = p_1V_1^\gamma = p_2V_2^\gamma$,

$$W_{1-2} = \frac{p_2V_2^\gamma V_2^{1-\gamma} - p_1V_2^\gamma V_2^{1-\gamma}}{1-\gamma}$$

$$= \frac{p_1V_1 - p_2V_2}{\gamma - 1}$$

Using $pV = RT$ for unit mass flow, we have,

$$w_{1-2} = \frac{R(T_1 - T_2)}{\gamma - 1}$$

$$= C_v(T_1 - T_2) = -(u_2 - u_1)$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

therefore,

$$\frac{T_2}{T_1} = \frac{p_2V_2}{p_1V_1} = \left(\frac{V_1}{V_2}\right)^\gamma \left(\frac{V_2}{V_1}\right)$$

$$= \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} \tag{2.24}$$

$$\frac{T_2}{T_1} = \frac{p_2V_2}{p_1V_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}\left(\frac{p_2}{p_1}\right)$$

$$= \left(\frac{p_2}{p_1}\right)^{\left(1-\frac{1}{\gamma}\right)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \tag{2.25}$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \tag{2.26}$$

2.7.5 Reversible Polytropic Process

In polytropic process, both heat and work transfers take place. It is denoted by the general equation $pV^n = C$, where n is the polytropic index. The following equations can be written by analogy to the equations for the reversible adiabatic process which is only a special case of polytropic process with $n = \gamma$. Hence, for a polytropic process

$$p_1V_1^n = p_2V_2^n \tag{2.27}$$

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1}$$

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\left(\frac{n-1}{n}\right)} \tag{2.28}$$

and

$$w_{1-2} = \frac{p_1V_1 - p_2V_2}{n-1} \tag{2.29}$$

2.7.6 Heat Transfer During Polytropic Process

Heat transfer per unit mass,

$$q_{1-2} = (u_2 - u_1) + \int_1^2 pdV \tag{2.30}$$

$$= C_v(T_2 - T_1) + \frac{R(T_2 - T_1)}{1-n}$$

$$= \left(C_v + \frac{R}{1-n}\right)(T_2 - T_1)$$

$$= \left(C_v + \frac{C_p - C_v}{1-n}\right)(T_2 - T_1)$$

$$
\begin{aligned}
&= \left(\frac{C_p - nC_v}{1-n}\right)(T_2 - T_1) \\
&= \left(\frac{C_v}{1-n}\right)\left(\frac{C_p}{C_v} - n\right)(T_2 - T_1) \\
&= \left(\frac{C_v}{1-n}\right)(\gamma - n)(T_2 - T_1) \\
&= \left(\frac{\gamma - n}{1-n}\right) C_v(T_2 - T_1) \qquad (2.31)
\end{aligned}
$$

Hence,

$$q_{1-2} = C_n(T_2 - T_1)$$

where,

$$C_n = \left(\frac{\gamma - n}{1-n}\right) C_v \qquad (2.32)$$

Table 2.1 gives the formulae for various process relations for easy reference.

2.8 ANALYSIS OF FLOW PROCESS

Consider a device shown in the Fig.2.4 through which a fluid flows at uniform rate and which absorbs heat and does work, also at a uniform rate. In engines, the gas flow, heat flow and work output vary throughout each cycle, but if a sufficiently long time interval (such as a minute) be chosen, then, even an engine can be considered to be operating under steady-flow conditions.

Applying the principle of conservation of energy to such a system during the chosen time interval, the total energy in the mass of fluid which enters the machine across boundary 1-1 plus the heat added to the fluid through the walls of the machine, minus the work done by the machine, must equal the total energy left in the equal mass of fluid crossing boundary 2-2 and leaving the machine. Expressed in the mathematical form,

$$Q - W_x = m\left[\left(h_2 + \frac{C_2^2}{2} + gz_2\right) - \left(h_1 + \frac{C_1^2}{2} + gz_1\right)\right]$$

or

$$\frac{Q - W_x}{m} = \Delta\left(h + \frac{C^2}{2} + gz\right) \qquad (2.33)$$

Table 2.1 Summary of Process Relations for a Perfect Gas

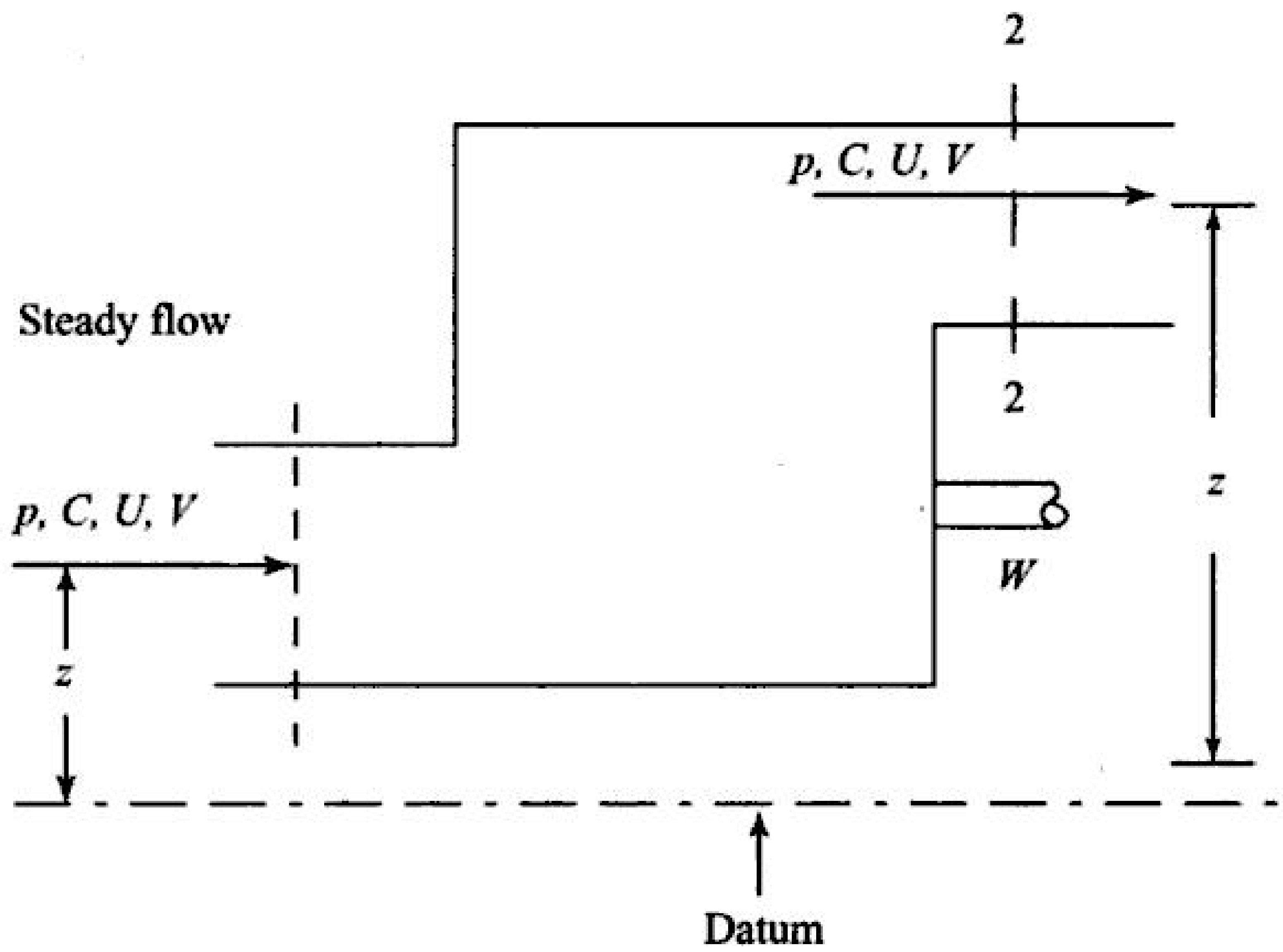
Process	Index n	p, V, T relation	Heat Transfer	Work $\int p\,dV$	Work $\int V\,dp$	ΔU
Constant Volume	∞	$T_1/T_2 = p_1/p_2$	$mC_v(T_2 - T_1)$	0	$V(p_2 - p_1)$	$mC_v(T_2 - T_1)$
Constant Pressure	0	$T_1/T_2 = V_1/V_2$	$mC_p(T_2 - T_1)$	$p(V_2 - V_1)$	0	$mC_v(T_2 - T_1)$
Isothermal	1	$p_1V_1 = p_2V_2$	$p_1V_1 \log_e\left(\frac{V_2}{V_1}\right)$	$p_1V_1 \log_e\left(\frac{V_2}{V_1}\right)$	$p_1V_1 \log_e\left(\frac{V_1}{V_2}\right)$	0
Isentropic	γ	$p_1V_1^\gamma = p_2V_2^\gamma$ $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$	0	$\frac{p_1V_1 - p_2V_2}{\gamma - 1}$	$\frac{\gamma}{\gamma-1}(p_2V_2 - p_1V_1)$	$mC_v(T_2 - T_1)$
Polytropic	n	$p_1V_1^n = p_2V_2^n$ $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}$	$\left(\frac{\gamma-n}{1-n}\right)mC_v(T_2 - T_1)$	$\frac{p_1V_1 - p_2V_2}{n - 1}$	$\frac{n}{n-1}(p_2V_2 - p_1V_1)$	$mC_v(T_2 - T_1)$

In the above equation, m denotes mass. It is known as the general energy equation and is the one which illustrates the first law of thermodynamics.

Here, W_x is the external work and given by

$$W = W_x - mp_1V_1 + mp_2V_2 \tag{2.34}$$

where W is the net work and V stands for volume.



U	$\rightarrow$	Internal energy	H	$\rightarrow$	Enthalpy	z	$\rightarrow$	Datum height
p	$\rightarrow$	Pressure	V	$\rightarrow$	Volume	C	$\rightarrow$	Velocity
U	$\rightarrow$	Internal energy	H	$\rightarrow$	Enthalpy	z	$\rightarrow$	Datum height

Fig. 2.4 Illustration of Steady Flow Process

2.8.1 Flow Work

The term pV in the above equation is called flow work and must be considered whenever flow takes place. It is the work necessary either to push certain quantity of mass into or out of the system.

2.9 WORK, POWER AND EFFICIENCY

In this section let us define work, power and efficiency and also the relation between them.

2.9.1 Work

Work might be defined as the energy expended when motion takes place in opposition to a force. Work thus represents energy being transferred from one place to another.

2.9.2 Power

Power is the rate of doing work. It is generally useless to do a certain piece of work unless it can be done within a reasonable length of time. When work is done at the rate of 736 Nm/s or 75 kgf m/s, the quantity of power being developed is known as one horse power. In SI units power is expressed in kW = 1 kN m/s.

2.9.3 Efficiency

It is common practice in engineering to establish a figure of merit for a device by comparing the actual performance of the device with the performance it would have had under some arbitrary set of ideal conditions. The ratio of actual performance to the ideal performance is called the efficiency of the device. Since the ideal performance is usually unattainable, the efficiency is usually less than 1. In internal combustion engines, one of the most important efficiencies is the thermal efficiency, which is defined as

$$\begin{aligned}\text{Thermal efficiency} = \eta_{th} &= \frac{\text{Work delivered by the engine}}{\text{Chemical energy in the fuel}} \\ &= \frac{\text{Work output}}{\text{Fuel energy input}}\end{aligned}$$

Obviously, the output and input may be measured over any convenient interval such as one cycle or one minute. The output and input energies must be expressed in same kind of units, since η_{th} is a number without units.

Let W be the work delivered by the engine (Nm) and m_f be the mass of fuel (kg) required and CV be the calorific value of the fuel (kJ/kg), then the efficiency η_{th} is given by

$$\eta_{th} = \frac{W}{m_f\ CV} \tag{2.35}$$

If W is the indicated work then η_{th} is called the indicated thermal efficiency, η_{ith}, and if W is the work delivered at the output shaft then it is called the brake thermal efficiency, η_{bth}. The ratio of the brake thermal efficiency to the indicated thermal efficiency is called the mechanical efficiency, η_m. These efficiencies have already been defined in Chapter 1.

Worked out Examples

2.1 A closed system undergoes a thermodynamic cycle consisting of four separate and distinct processes. The work and heat transfer for each of these processes is given below

Process	Heat Transfer (J/s)	Work Transfer (Nm/s)
1–2	7000	5300
2–3	−3500	9100
3–4	17500	8700
4–1	0	−2100

Show that the above data is in accordance with the first law of thermodynamics and determine : (i) Net rate of work output (ii) Efficiency of the cycle (iii) Change in internal energy in each process

Solution

For the above closed cycle 1–2–3–4–1,

$$\oint \delta Q = 7000 - 3500 + 17500 + 0$$

$$= 21000 \ \ \text{J/s}$$

$$\oint \delta W = 5300 + 9100 + 8700 - 2100$$

$$= 21000 \ \ \text{Nm/s} = 21000 \ \ \text{J/s}$$

As $\oint \delta Q = \oint \delta W$, the above cycle satisfies the first law of thermodynamics

$$\textit{Power} = \textit{Net rate of work} = 21000 \ \ \text{J/s}$$

$$= \mathbf{21 \ N/m^2kW} \qquad \underleftarrow{\text{Ans}}$$

$$\textit{Thermal efficiency} = \frac{\textit{Net work}}{\textit{Heat input}} = \frac{21000}{7000 + 17500} \times 100$$

$$= \mathbf{85.71\%} \qquad \underleftarrow{\text{Ans}}$$

Change in internal energy of each process can be estimated by using the relation, $\delta Q = du + \delta W$

Process	δQ J/s	δW Nm/s	$du = \delta Q - \delta W$ J/s
1–2	7000	5300	1700
2–3	−3500	9100	−12600
3–4	17500	8700	8800
4–1	0	−2100	2100

2.2 Air at the rate of 10 kg/min flows steadily through an air compressor in which the inlet and discharge pipe lines are at the same level. Following data regarding the fluid are available:

	At inlet	At outlet
Fluid velocity	5 m/s	10 m/s
Fluid pressure	1 bar	8 bar
Specific volume	0.5 m^3/kg	0.20 m^3/kg

The internal energy of the air leaving the compressor was 250 kJ/kg greater than that entering and during the process the system lost 140 kJ/s of energy dissipated as heat to the cooling water and to the environment, Find (i) Rate of shaft work input to the air (ii) Ratio of inlet pipe diameter to outlet pipe diameter

Solution

Consider 1 kg/min of mass flow :

$$\textit{Heat lost to cooling water and environment} \quad = \quad \frac{140 \times 60}{10}$$

$$= \quad 840 \ \text{kJ/kg}$$

Applying steady flow energy equation, (the potential energy changes are neglected as the inlet and outlet pipes are at the same level).

$$U_1 + p_1V_1 + \frac{C_1^2}{2} + q \quad = \quad U_2 + p_2V_2 + \frac{C_2^2}{2} + w$$

therefore,

$$w \quad = \quad (U_1 - U_2) + (p_1V_1 - p_2V_2) + \frac{C_1^2 - C_2^2}{2} + q$$

$$
\begin{aligned}
&= -250 \times 10^3 + (1 \times 10^5 \times 0.5 - 8 \times 10^5 \times 0.2) \\
&\quad + \frac{5^2 - 10^2}{2} - 140 \times 10^3 \\
&= -500037.5 \ \text{J/kg} \approx -500 \ \text{kJ/kg} \\
W &= mw = 10 \times (-500) \\
&= -5000 \ \text{kJ/min} = -83.33 \ \text{kJ/s}
\end{aligned}
$$

Power input $=$ **83.33 N/m²kW** $\stackrel{\text{Ans}}{\Longleftarrow}$

From continuity equation, $A_1C_1\rho_1 = A_2C_2\rho_2$. *Since, the specific volume,* $v_1 = \frac{1}{\rho_1}$. *We have,*

$$
\begin{aligned}
\frac{A_1C_1}{v_1} &= \frac{A_2C_2}{v_2} \\
\frac{A_1}{A_2} &= \frac{v_1C_2}{v_2C_1} = \frac{0.5 \times 10}{0.2 \times 5} = 5 \\
\frac{D1}{D2} &= \sqrt{5} = \mathbf{2.236}
\end{aligned}
$$

$\stackrel{\text{Ans}}{\Longleftarrow}$

2.3 A tank of volume 0.1 m³ contains 4 kg nitrogen, 1.5 kg oxygen and 0.75 kg carbon dioxide. If the temperature of the mixture is 20 °C, determine : (i) the total pressure of the mixture, (ii) the gas constant of the mixture. (Given that $R_{N_2} = 296.8$ J/kg K, $R_{O_2} = 259.83$ J/kg K and $R_{CO_2} = 188.9$ J/kg K).

Solution

N_2 : $m_1 = 4.00$ kg; $t_1 = t = 20$ °C; $V_1 = V = 0.1$ m³
O_2 : $m_2 = 1.50$ kg; $t_2 = t = 20$ °C; $V_2 = V = 0.1$ m³
CO_2 : $m_3 = 0.75$ kg; $t_3 = t = 20$ °C; $V_3 = V = 0.1$ m³

Assumptions :

(i) the component gases and the mixture behave like an ideal gas

(ii) the mixture obeys the Gibbs-Dalton law

For N_2 *:*

$$p_1 = \frac{m_1 R_{N_2} T}{V}$$

$$= \frac{4 \times 296.8 \times (273 + 20)}{0.1} = 34.78 \times 10^5 \ \text{N/m}^2$$

For O_2 *:*

$$p_2 = \frac{m_2 R_{O_2} T}{V}$$

$$= \frac{1.5 \times 259.83 \times (273 + 20)}{0.1} = 11.42 \times 10^5 \ \text{N/m}^2$$

For CO_2 *:*

$$p_3 = \frac{m_3 R_{CO_2} T}{V}$$

$$= \frac{0.75 \times 188.9 \times (273 + 20)}{0.1} = 4.15 \times 10^5 \ \text{N/m}^2$$

Then, total pressure, P

$$p = p_1 + p_2 + p_3 = (34.78 + 11.42 + 4.15) \times 10^5$$

$$= \mathbf{50.35 \times 10^5 \ N/m^2} \qquad \underleftarrow{\text{Ans}}$$

$$R = \frac{(m_{N_2} \ R_{N_2} + m_{O_2} \ R_{O_2} + m_{CO_2} \ R_{CO_2})}{m}$$

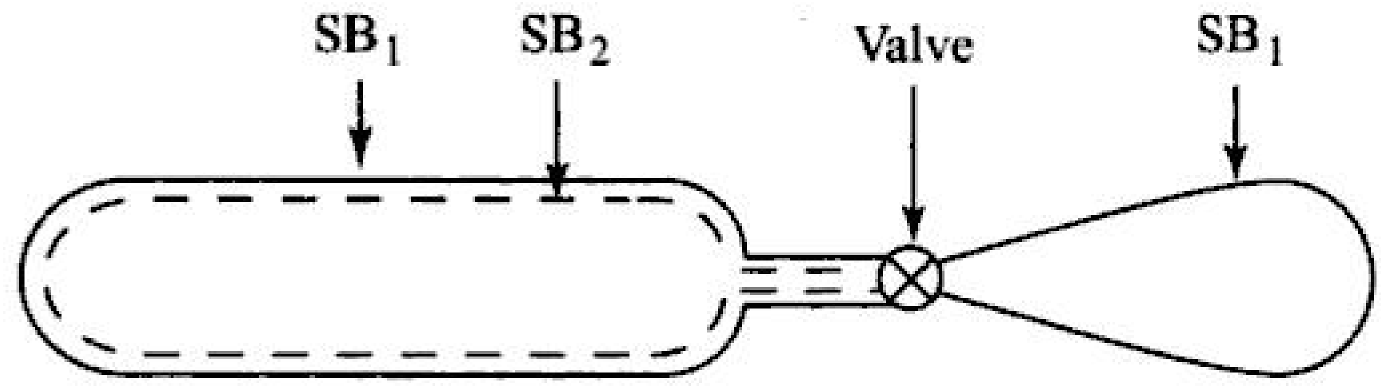
$$= \frac{(4 \times 296.8 + 1.5 \times 259.83 + 0.75 \times 188.92)}{4 + 1.5 + 0.75}$$

$$= \mathbf{274.98 \ N/m^2 J/kg \ K} \qquad \underleftarrow{\text{Ans}}$$

2.4 Determine the work done by the air which enters an evacuated bottle from the atmosphere when the cock is opened. The atmospheric pressure is 1.013×10^5 N/m^2 and 0.3 m^3 of air (measured at atmospheric conditions) enter.

Solution

No work is done by the part of the boundary in contact with the bottle. Only the moving external part need to be considered. Over



this part pressure is uniform at 1.013×10^5 N/m^2, therefore,

$$W_d = \int_{\substack{free\text{-}air\\boundary}} pdV + \int_{Bottle} pdV$$

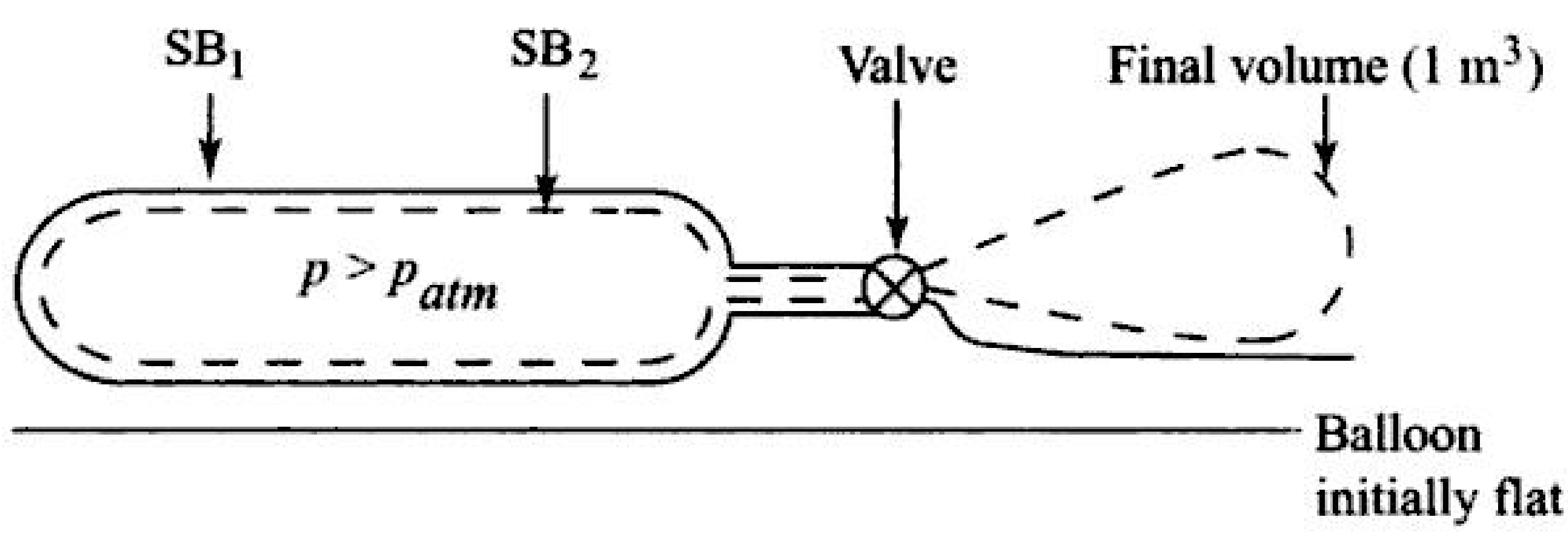
$$= P_{atm} \int dV + 0 = 1.013 \times 10^5(-0.3)$$

$$= \mathbf{-30390\ N/m^2Nm} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

The work is negative as the boundary is contracting.

2.5 A balloon of flexible material is to be filled with air from a storage bottle until it has a volume of 1 m^3. The atmospheric pressure is 1.01×10^5 N/m^2. Determine the work done by the system comprising the air initially in the bottle, given that the balloon is light and requires no stretching.

Solution



Initially the system boundary coincides with the inner surface of the storage bottle. At the end of the process the boundary also encloses the 1 m^3 content of the balloon.

The displacement work which is the only work in this process, is obtained by taking the summation of the values of $\int pdV$ for each part of the boundary. As there is no change in the volume of the bottle, dV is zero for the part of the boundary which is in contact with the bottle surface. Hence the pressure inside the cylinder is not necessary in the calculation. Therefore,

$$W_{displacement} = \int_{Balloon} pdV + \int_{Bottle} pdV$$

$$= p_{atm} \int dV + 0 = 1.01 \times 10^5 \times 1.0$$

$$= \mathbf{1.01 \times 10^5\ N/m^2Nm} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Review Questions

2.1 *What are the main objectives in studying the theory of I.C. engines?*

2.2 *What are the important fundamental quantities used in the application and analysis of I.C. engines? Explain them briefly.*

2.3 *What is meant by mole?*

2.4 *For a mixture of gases show that* $P_1/P_t = N_1/N_t$

2.5 *What are the various forms of energy normally used in engine applications. Briefly explain them.*

2.6 *What are isochoric and isobaric processes? Explain them.*

2.7 *For an isentropic process, show that (i)* $T_2/T_1 = (V_1/V_2)^{\gamma-1}$ *and (ii)* $T_2/T_1 = (p_1/p_2)^{(\gamma-1)/\gamma}$

2.8 *Derive an expression for heat transfer during a polytropic process.*

2.9 *Explain how an internal combustion engine can be analyzed from the point of view of steady flow process.*

2.10 *Define: (i) work (ii) power and (iii) efficiency*

Exercise

2.1 The piston of an oil engine, of area 50 cm^2, moves downwards 10 cm and draws in 300 cc of fresh air from the atmosphere. The pressure in the cylinder is uniform during the process at 0.8 bar, while the atmospheric pressure is 1.013 bar. The difference in the pressure is accounted for by flow resistance in the induction pipe and inlet valve. Determine the displacement work done by the air finally in the cylinder. *Ans:* −9.61 J

2.2 Air at 600 K and 1.2 bar undergoes an adiabatic expansion. The new value of pressure is 0.6 bar. The mass of the air, which may be treated as an ideal gas, is 0.12 kg. Determine (i) the work done by the air and (ii) the change in the internal energy of the air. *Ans:* (i) 9.285 kJ (ii) −9.285 kJ

2.3 An open tank is filled to the brim with a liquid of density 1100 kg/m^3. A spherical balloon 0.5 m in diameter is immersed in the liquid with its centre 2.5 m below the free liquid surface. Gas from a storage vessel is used to inflate the balloon thereby causing the tank to overflow. The atmospheric pressure is 10^5 N/m^2. Evaluate the work done by the gas in the balloon on the storage vessel as the balloon diameter increases to 1 m. Assume $g = 9.81$ m/s^2. *Ans:* 58.2×10^3 Nm

2.4 A gas flows steadily through a rotary compressor. The gas enters the compressor at a temperature of 16°, a pressure of 10^5 N/m^2 and an enthalpy of 391.2 KJ/kg. The gas leaves the compressor at a temperature of 245 °C, a pressure of 6×10^5 N/m^2 and an enthalpy of 534.5 KJ/kg. There is no net heat transfer to or from the gas as it flows through the compressor.

(i) Evaluate the external work done per unit mass of gas assuming the gas velocities at entry and exit to be negligible.

(ii) Evaluate the external work done per unit mass of gas when the gas velocity at entry is 80 m/s and at exit is 160 m/s.

Ans: (i) 143.3 KJ/kg (ii) 152.9 KJ/kg

2.5 The relation between the properties of gaseous oxygen gas may be expressed over a restricted range by :

$$Pv = 260t + 71 \times 10^3 \quad \text{and} \quad t = 1.52u - 273$$

where P is in N/m^2, v in m^3/kg, t in °C and u in kJ/kg.

(i) Evaluate the specific heat at constant volume and specific heat at constant pressure in kJ/kg K.

(ii) Show that for any process executed by unit mass of oxygen, $\Delta u = C_v \Delta t$ and $\Delta h = C_p \Delta t$

Ans: (i) 0.658 kJ/kg K (ii) 0.918 kJ/kg K

2.6 Two kg of a gas at 10 bar expands adiabatically and reversibly till the pressure falls to 5 bar. During the process 170 kJ of non-flow work is done by the system, and the temperature drops from 377 °C to 257 °C. Calculate (i) the value of the index of expansion and (ii) characteristic gas constant.

Ans: (i) 1.416 (ii) 294 J/kg K

2.7 Show that for a Vander Waals gas which has the equation of state

$$\left(P + \frac{a}{V_2} \quad = \quad RT\right)$$

The work done at constant temperature per unit mass of gas is

$$RT \ln \frac{V_2 - b}{V_1 - b} - a\left(\frac{1}{V_1} - \frac{1}{V_2}\right)$$

where V_1 and V_2 denote the initial and final specific volumes respectively.

2.8 In an air-standard cycle, heat is supplied at constant volume resulting in an increase in temperature of air from T_1 to T_2. The air is then expanded isentropically till its temperature falls to T_1. Finally, it is returned to its original state by a reversible isothermal compression process.

Show that the efficiency of the cycle is given by

$$\eta \quad = \quad 1 - \frac{T_1}{T_2 - T_1} \ln \frac{T_2}{T_1}$$

2.9 A volume of 5 m^3 of air at a pressure of 1 bar and 27 °C is compressed adiabatically to 5 bar. The compressed air is then expanded isothermally to original volume. Find :

(i) The final pressure of the air after expansion

(ii) The quantity of heat added from the beginning of compression to the end of expansion

(iii) The quantity of heat that must be added or subtracted to reduce the air after expansion to the original state of pressure, volume and temperature.

Ans: (i) 1.58 bar (ii) 910.09 kJ (iii) −729.8 kJ

2.10 A perfect gas of molecular weight 30 has the ratio of specific heats 1.3. One kg of the gas at 1 bar and 27 °C is compressed adiabatically inside a cylinder to 7 times its initial pressure. Heat is now extracted from the system at constant pressure until it reaches a state (temperature 27 °C) such that the gas when expanded isothermally from this state will reach the state before adiabatic compression. Calculate the heat transfer during constant pressure cooling and the work done during isothermal expansion. What is the amount of work transfer during the cycle and in which direction does it take place. R = 0.287 kJ/kg K.

Ans: (i) -204 kJ/kg (ii) 161.8 kJ/kg (iii) −42.2 kJ/kg

2.11 A perfect gas flows steadily through a horizontal cooler. The mass flow rate through the cooler is 1 kg/s. The pressure and temperature are 2 bar and 127 °C at entry and 1.5 bar and 7 °C at exit. The cross sectional areas at entry and exit are each 80 cm^2. Calculate the velocities of the gas at entry and exit of the cooler and the heat transfer rate in the cooler. Take for the gas C_v = 0.70 kJ/kg K and R = 160 J/kg K.

Ans: (i) 40 m/s (ii) 37.5 m/s (iii) −103.3 kJ/s

2.12 Air inside a cylinder is compressed from the same initial state, without friction such that the compression ratio, $\frac{V_2}{V_1}$ is 15. In one case it is compressed isothermally and in the other case it is compressed polytropically with polytropic index, $n = 1.3$. Calculate the ratios of work done and heat transfer in two cases. What will be the ratio of final pressure in the two cases.

Ans: (i) 0.649 (ii) 2.59 (iii) 0.444

3

AIR-STANDARD CYCLES AND THEIR ANALYSIS

3.1 INTRODUCTION

The operating cycle of an internal combustion engine can be broken down into a sequence of separate processes viz., intake, compression, combustion, expansion and exhaust. The internal combustion engine does not operate on a thermodynamic cycle as it involves an open system i.e., the working fluid enters the system at one set of conditions and leaves at another. However, it is often possible to analyze the open cycle as though it were a closed one by imagining one or more processes that would bring the working fluid at the exit conditions back to the condition of the starting point.

The accurate analysis of internal combustion engine processes is very complicated. In order to understand them it is advantageous to analyze the performance of an idealized closed cycle that closely approximates the real cycle. One such approach is the air-standard cycle, which is based on the following assumptions:

(i) The working medium is assumed to be a perfect gas and follows the relation $pV = mRT$ or $p = \rho RT$.

(ii) There is no change in the mass of the working medium.

(iii) All the processes that constitute the cycle are reversible.

(iv) Heat is assumed to be supplied from a constant high temperature source and not from chemical reactions during the cycle.

(v) Some heat is assumed to be rejected to a constant low temperature sink during the cycle.

(vi) It is assumed that there are no heat losses from the system to the surroundings.

(vii) The working medium has constant specific heats throughout the cycle.

(viii) The physical constants viz., C_p, C_v, γ and M of working medium are the same as those of air at standard atmospheric conditions. For example in SI units,

$$\begin{array}{llllll} C_p & = & 1.005 \text{ kJ/kg K} & M & = & 29 \text{ kg/kmol} \\ C_v & = & 0.717 \text{ kJ/kg K} & \gamma & = & 1.4 \end{array}$$

Due to these assumptions, the analysis becomes over-simplified and the results do not agree with those of the actual engine. Work output, peak pressure, peak temperature and thermal efficiency based on air-standard cycles will be the maximum that can be attained and will differ considerably from those of the actual engine. It is often used, mainly because of the simplicity in getting approximate answers to the complicated processes in internal combustion engines.

In this chapter, we will review the various cycles and also derive the equations for work output, mean effective pressure, efficiency etc. Also, comparison will be made between Otto, Dual and Diesel cycles to see which cycle is more efficient under a set of given operating conditions.

3.2 THE CARNOT CYCLE

Sadi Carnot, a French engineer, proposed a reversible cycle in 1824, in which the working medium receives heat at a higher temperature and rejects heat at a lower temperature. The cycle will consist of two isothermal and two reversible adiabatic processes as shown in Fig.3.1. Carnot cycle is represented as a standard of perfection and engines can be compared with it to judge the degree of perfection. It gives the concept of maximizing work output between two temperature limits.

The working of an engine based on the Carnot cycle can be explained referring to Fig.3.2 which shows a cylinder and piston arrangement working without friction. The walls of cylinder are assumed to be perfect insulators. The cylinder head is so arranged that it can be a perfect heat conductor as well as a perfect heat insulator.

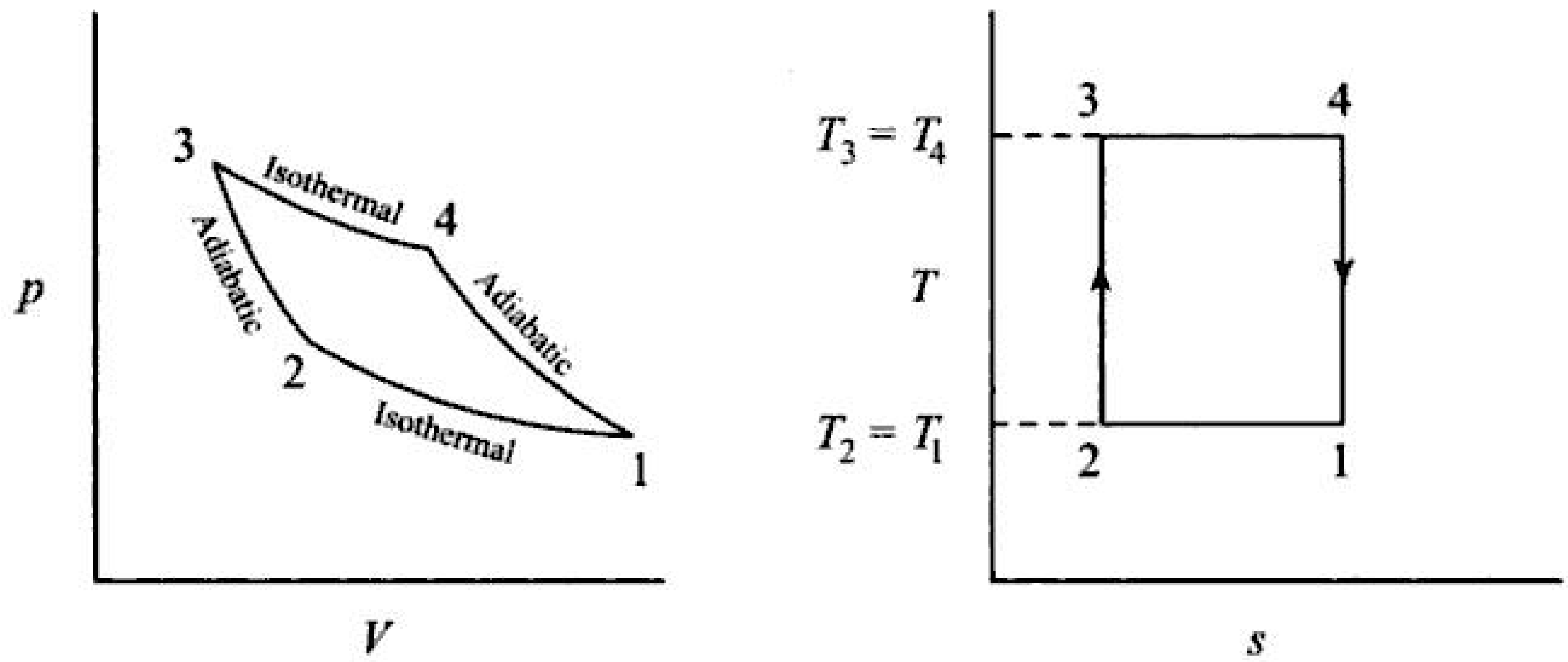


Fig. 3.1 Carnot Engine

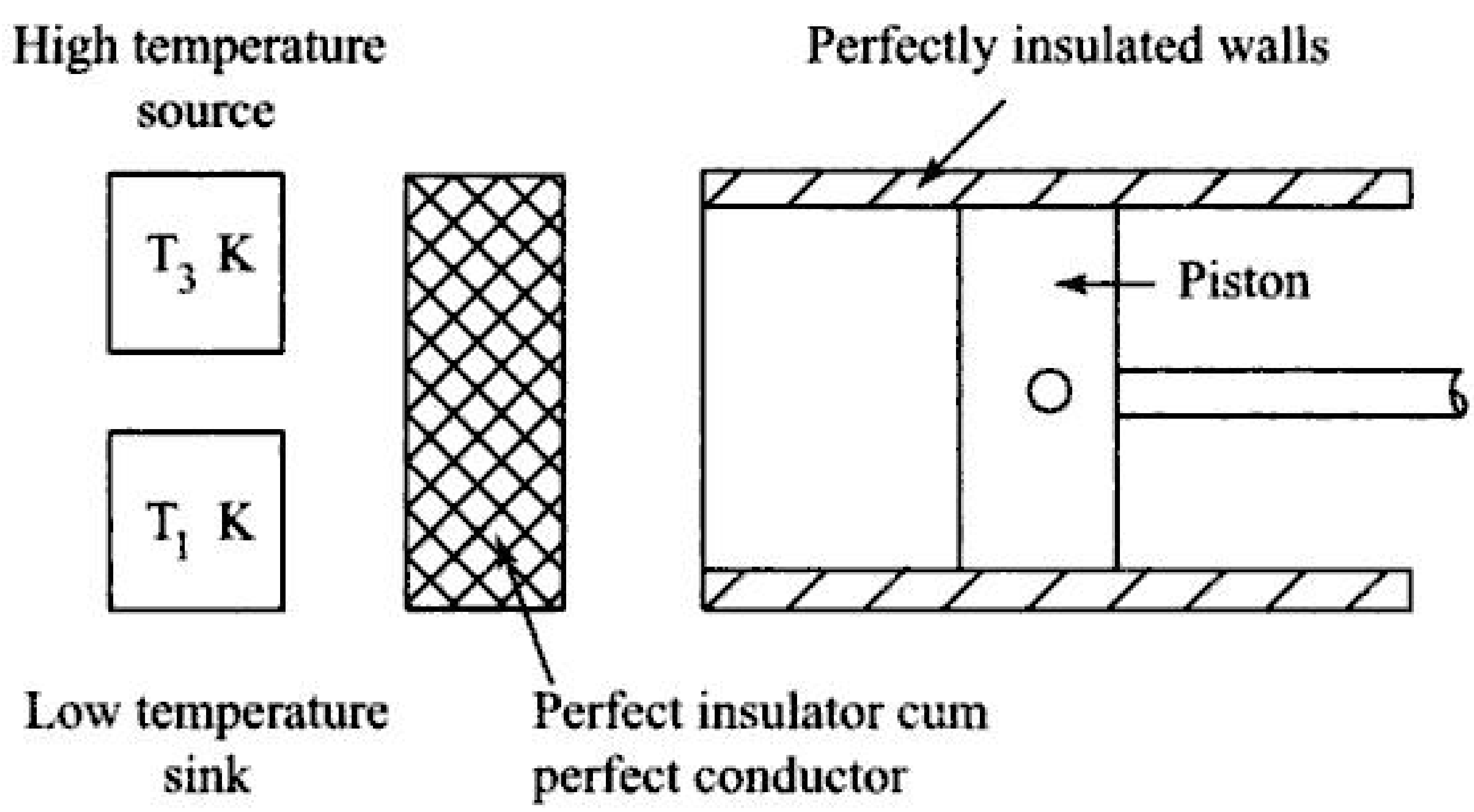


Fig. 3.2 Working Principle of a Carnot Engine

First the heat is transferred from a high temperature source, (T_3), to the working medium in the cylinder and as a result the working medium expands. This is represented by the isothermal process 3→4 in Fig.3.1. Now the cylinder head is sealed and it acts as a perfect insulator. The working medium in the cylinder is now allowed to expand further from state 4 to state 1 and is represented by reversible adiabatic process 4→1 in p-V and T-s diagrams in Fig.3.1. Now the system is brought into contact with a constant low temperature sink, (T_1), as the cylinder head is now made to act as a perfect heat conductor. Some heat is rejected to the sink without altering the temperature of sink and as a result the working medium is compressed from state 1 to 2 which is represented by isothermal line $1 \rightarrow 2$. Finally the cylinder head is made again to act as a perfect insulator and the working medium is compressed adiabatically from state 2 to 3 which is represented by process $2 \rightarrow 3$. Thus the cycle is completed.

Analyzing the cycle thermodynamically the efficiency of the cycle can be written as

$$\eta_{Carnot} = \frac{\text{Work done by the system during the cycle } (W)}{\text{Heat supplied to the system during the cycle } (Q_S)}$$

According to the first law of thermodynamics,

$$\text{Work done} = \text{Heat supplied} - \text{Heat rejected}$$

$$W = Q_S - Q_R \tag{3.1}$$

Considering the isothermal processes 1→2 and 3→4, we get

$$Q_R = mRT_1 \log_e \frac{V_1}{V_2} \tag{3.2}$$

$$Q_S = mRT_3 \log_e \frac{V_4}{V_3} \tag{3.3}$$

Considering adiabatic processes 2→3 and 4→1

$$\frac{V_3}{V_2} = \left(\frac{T_2}{T_3}\right)^{\left(\frac{1}{\gamma-1}\right)} \tag{3.4}$$

and

$$\frac{V_4}{V_1} = \left(\frac{T_1}{T_4}\right)^{\left(\frac{1}{\gamma-1}\right)} \tag{3.5}$$

Since $T_1 = T_2$ and $T_4 = T_3$ we have,

$$\frac{V_4}{V_1} = \frac{V_3}{V_2}$$

or

$$\frac{V_4}{V_3} = \frac{V_1}{V_2} = r \qquad (say) \tag{3.6}$$

then,

$$\eta_{Carnot} = \frac{mRT_3 \log_e r - mRT_1 \log_e r}{mRT_3 \log_e r} \tag{3.7}$$

$$= \frac{T_3 - T_1}{T_3} = 1 - \frac{T_1}{T_3} \tag{3.8}$$

The lower temperature i.e., sink temperature, T_1, is normally the atmospheric temperature or the cooling water temperature and hence fixed. So the increase in thermal efficiency can be achieved only by increasing the source temperature. In other words, the upper temperature is required to be maintained as high as possible, to achieve maximum thermal efficiency. Between two fixed temperatures Carnot cycle (and other reversible cycles) has the maximum possible efficiency compared to other air-standard cycles. In spite of this advantage, Carnot cycle does not provide a suitable basis for the operation of an engine using a gaseous working fluid because the work output from this cycle will be quite low.

Mean effective pressure, p_m, is defined as that hypothetical constant pressure acting on the piston during its expansion stroke producing the same work output as that from the actual cycle. Mathematically,

$$p_m = \frac{\text{Work Output}}{\text{Swept Volume}} \tag{3.9}$$

It can be shown as

$$p_m = \frac{\text{Area of indicator diagram}}{\text{Length of diagram}} \times \text{constant} \tag{3.10}$$

The constant depends on the mechanism used to get the indicator diagram and has the units, bar/m. These formulae are quite often used to calculate the performance of an internal combustion engine. If the work output is the indicated output then it is called indicated mean effective pressure, p_{im}, and if the work output is the brake output then it is called brake mean effective pressure, p_{bm}.

3.3 THE STIRLING CYCLE

The Carnot cycle has a low mean effective pressure because of its very low work output. Hence, one of the modified forms of the cycle to produce higher mean effective pressure whilst theoretically achieving full Carnot cycle efficiency is the Stirling cycle. It consists of two isothermal and two constant volume processes. The heat rejection and addition take place at constant temperature. The p-V and T-s diagrams for the Stirling cycle are shown in Figs.3.3(a) and 3.3(b) respectively. It is clear from Fig.3.3(b) that the amount of heat addition and rejection during constant volume processes is same.

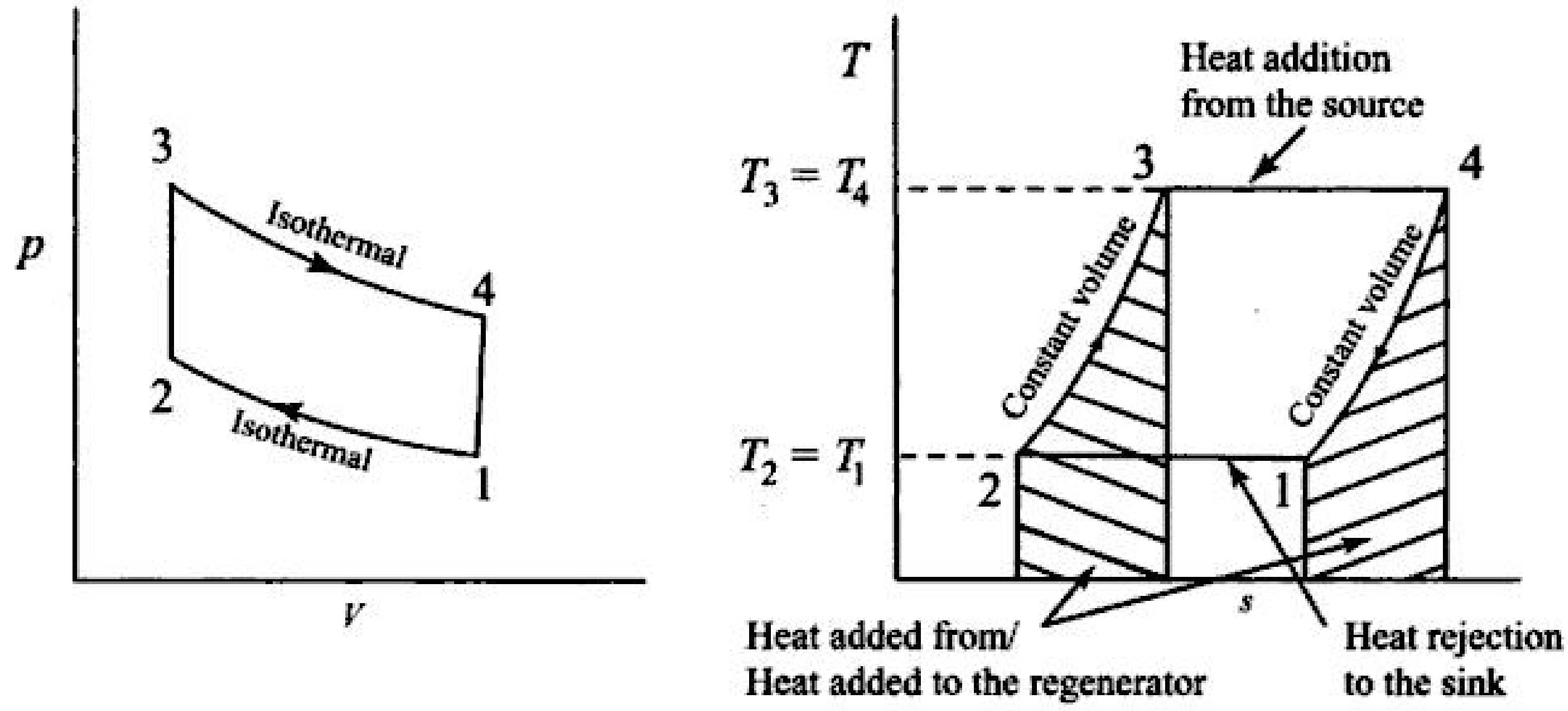


Fig. 3.3 Stirling Cycle

Hence, the efficiency of the cycle is given as

$$\eta_{Stirling} = \frac{RT_3 \log_e\left(\frac{V_4}{V_3}\right) - RT_1 \log_e\left(\frac{V_1}{V_2}\right)}{RT_3 \log_e\left(\frac{V_4}{V_3}\right)} \tag{3.11}$$

But $V_3 = V_2$ and $V_4 = V_1$

$$\eta_{Stirling} = \frac{T_3 - T_1}{T_3} \tag{3.12}$$

same as Carnot efficiency

The Stirling cycle was used earlier for hot air engines and became obsolete as Otto and Diesel cycles came into use. The design of Stirling engine involves a major difficulty in the design and construction of heat exchanger to operate continuously at very high temperatures. However, with the development in metallurgy and intensive research in this type of engine, the Stirling engine has staged a come back in practical appearance. In practice, the heat exchanger efficiency cannot be 100%. Hence the Stirling cycle efficiency will be less than Carnot efficiency and can be written as

$$\eta = \frac{R(T_3 - T_1)\log_e r}{RT_3 \log_e r + (1-\epsilon)C_v(T_3 - T_1)} \tag{3.13}$$

where ϵ is the heat exchanger effectiveness.

3.4 THE ERICSSON CYCLE

The Ericsson cycle consists of two isothermal and two constant pressure processes. The heat addition and rejection take place at constant pressure as well as isothermal processes. Since the process 2→3 and 3→4 are parallel to each other on the T-s diagram, the net effect is that the heat need be added only at constant temperature $T_3 = T_4$ and rejected at the constant temperature $T_1 = T_2$.

The cycle is shown on p-V and T-s diagrams in Fig.3.4(a) and 3.4(b) respectively. The advantage of the Ericsson cycle over the Carnot and Stirling cycles is its smaller pressure ratio for a given ratio of maximum to minimum specific volume with higher mean effective pressure.

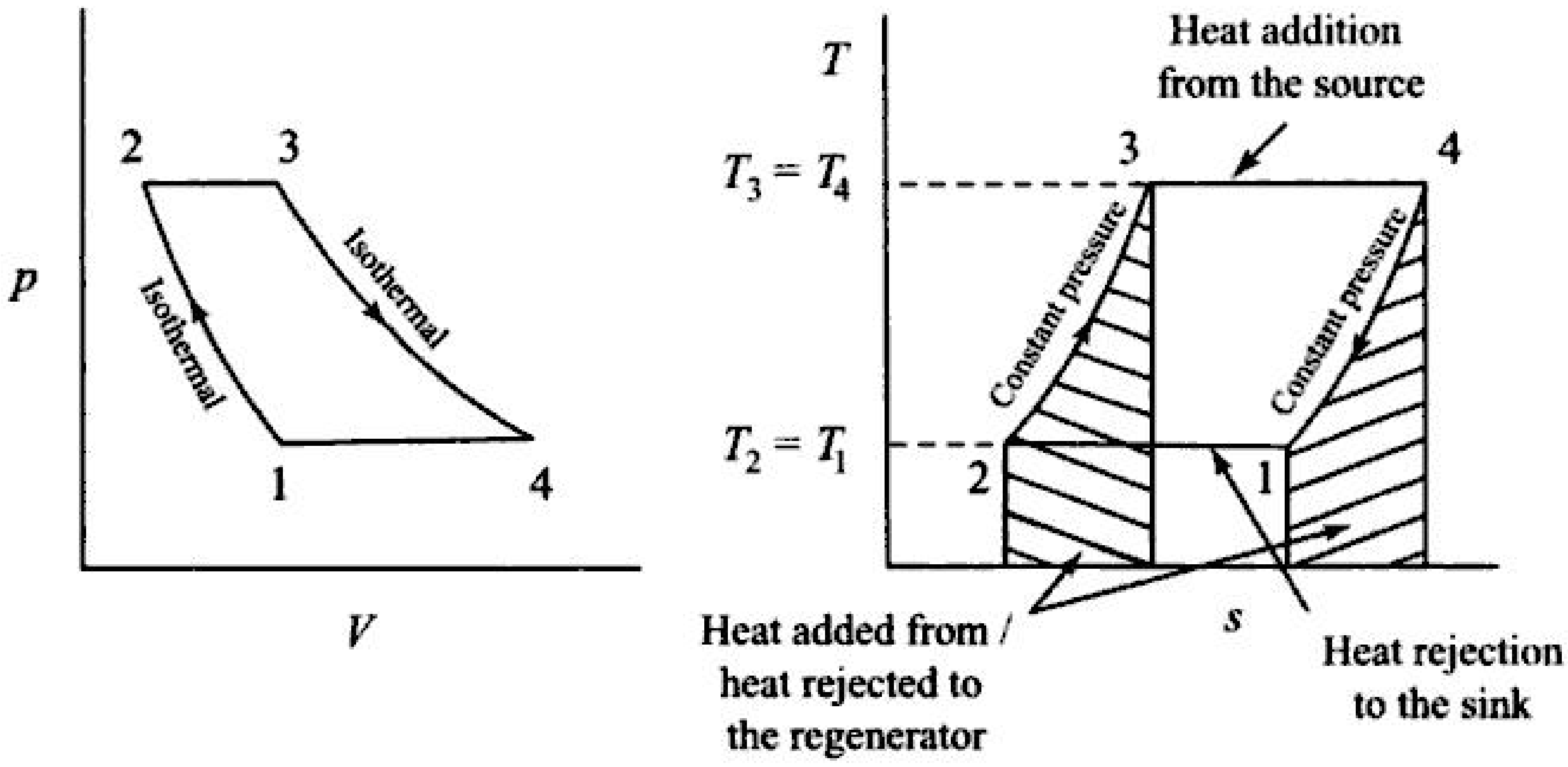


Fig. 3.4 Ericsson Cycle

The Ericsson cycle does not find practical application in piston engines but is approached by a gas turbine employing a large number of stages with heat exchangers, insulators and reheaters.

3.5 THE OTTO CYCLE

The main drawback of the Carnot cycle is its impracticability due to high pressure and high volume ratios employed with comparatively low mean effective pressure. Nicolaus Otto (1876), proposed a constant-volume heat addition cycle which forms the basis for the working of today's spark-ignition engines. The cycle is shown on p-V and T-s diagrams in Fig.3.5(a) and 3.5(b) respectively.

When the engine is working on full throttle, the processes 0→1 and 1→0 on the p-V diagram represents suction and exhaust processes and their effect is nullified. The process 1→2 represents isentropic compression of the air when the piston moves from

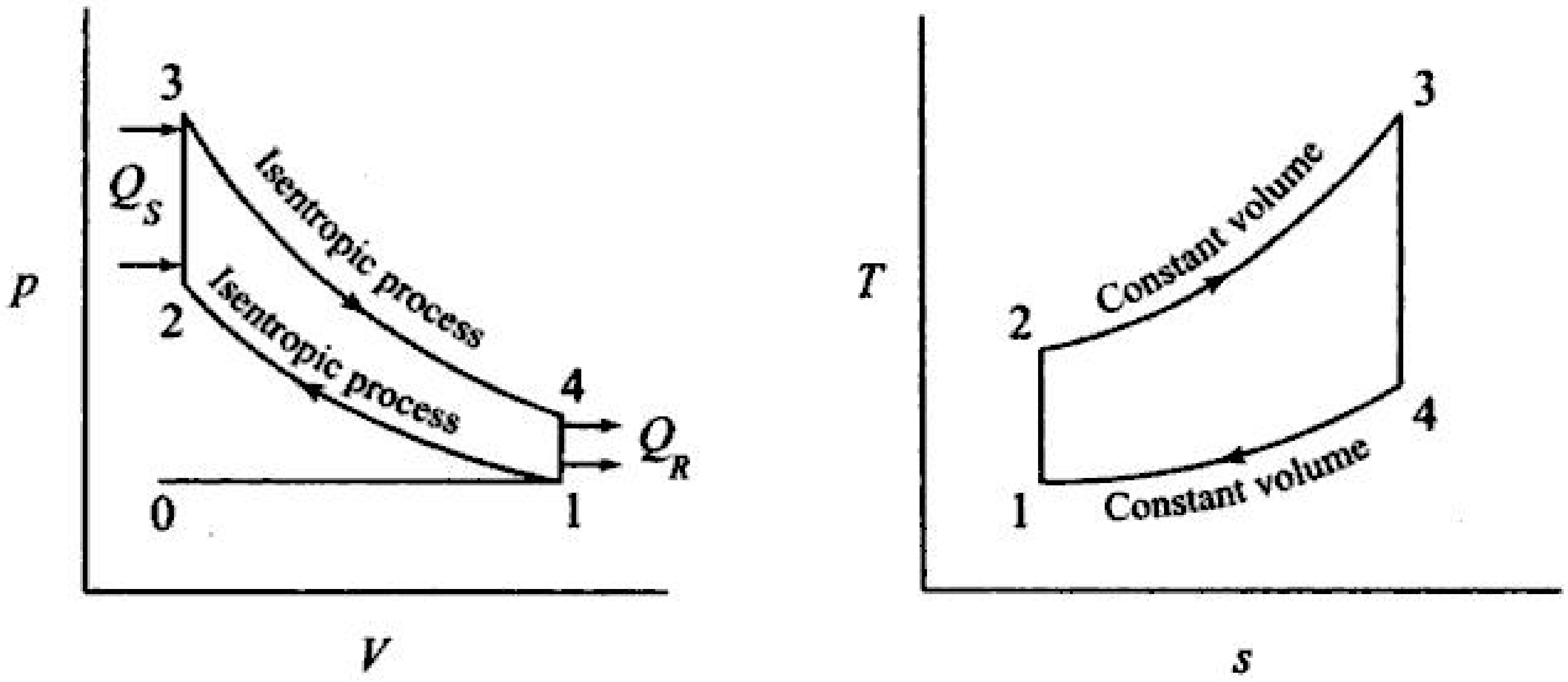


Fig. 3.5 Otto Cycle

bottom dead centre to top dead centre. During the process 2→3 heat is supplied reversibly at constant volume. This process corresponds to spark-ignition and combustion in the actual engine. The processes 3→4 and 4→1 represent isentropic expansion and constant volume heat rejection respectively.

3.5.1 Thermal Efficiency

The thermal efficiency of Otto cycle can be written as

$$\eta_{Otto} = \frac{Q_S - Q_R}{Q_S} \tag{3.14}$$

Considering constant volume processes 2→3 and 4→1, the heat supplied and rejected of air can be written as

$$Q_S = mC_v(T_3 - T_2) \tag{3.15}$$

$$Q_R = mC_v(T_4 - T_1) \tag{3.16}$$

$$\eta_{Otto} = \frac{m(T_3 - T_2) - m(T_4 - T_1)}{m(T_3 - T_2)}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{3.17}$$

Considering isentropic processes 1→2 and 3→4, we have

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} \tag{3.18}$$

and

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{(\gamma-1)} \tag{3.19}$$

But the volume ratios V_1/V_2 and V_4/V_3 are equal to the compression ratio, r. Therefore,

$$\frac{V_1}{V_2} = \frac{V_4}{V_3} = r \tag{3.20}$$

therefore,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \tag{3.21}$$

From Eq.3.21, it can be easily shown that

$$\frac{T_4}{T_3} = \frac{T_1}{T_2} = \frac{T_4 - T_1}{T_3 - T_2} \tag{3.22}$$

$$\eta_{Otto} = 1 - \frac{T_1}{T_2} \tag{3.23}$$

$$= 1 - \frac{1}{\left(\frac{V_1}{V_2}\right)^{(\gamma-1)}} \tag{3.24}$$

$$= 1 - \frac{1}{r^{(\gamma-1)}} \tag{3.25}$$

Note that the thermal efficiency of Otto cycle is a function of compression ratio r and the ratio of specific heats, γ. As γ is assumed to be a constant for any working fluid, the efficiency is increased by increasing the compression ratio. Further, the efficiency is independent of heat supplied and pressure ratio. The use of gases with higher γ values would increase efficiency of Otto cycle. Fig.3.6 shows the effect of γ and r on the efficiency.

3.5.2 Work Output

The net work output for an Otto cycle can be expressed as

$$W = \frac{p_3V_3 - p_4V_4}{\gamma - 1} - \frac{p_2V_2 - p_1V_1}{\gamma - 1} \tag{3.26}$$

Also

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = r^{\gamma}$$

$$\frac{p_3}{p_2} = \frac{p_4}{p_1} = r_p \quad \text{(say)} \tag{3.27}$$

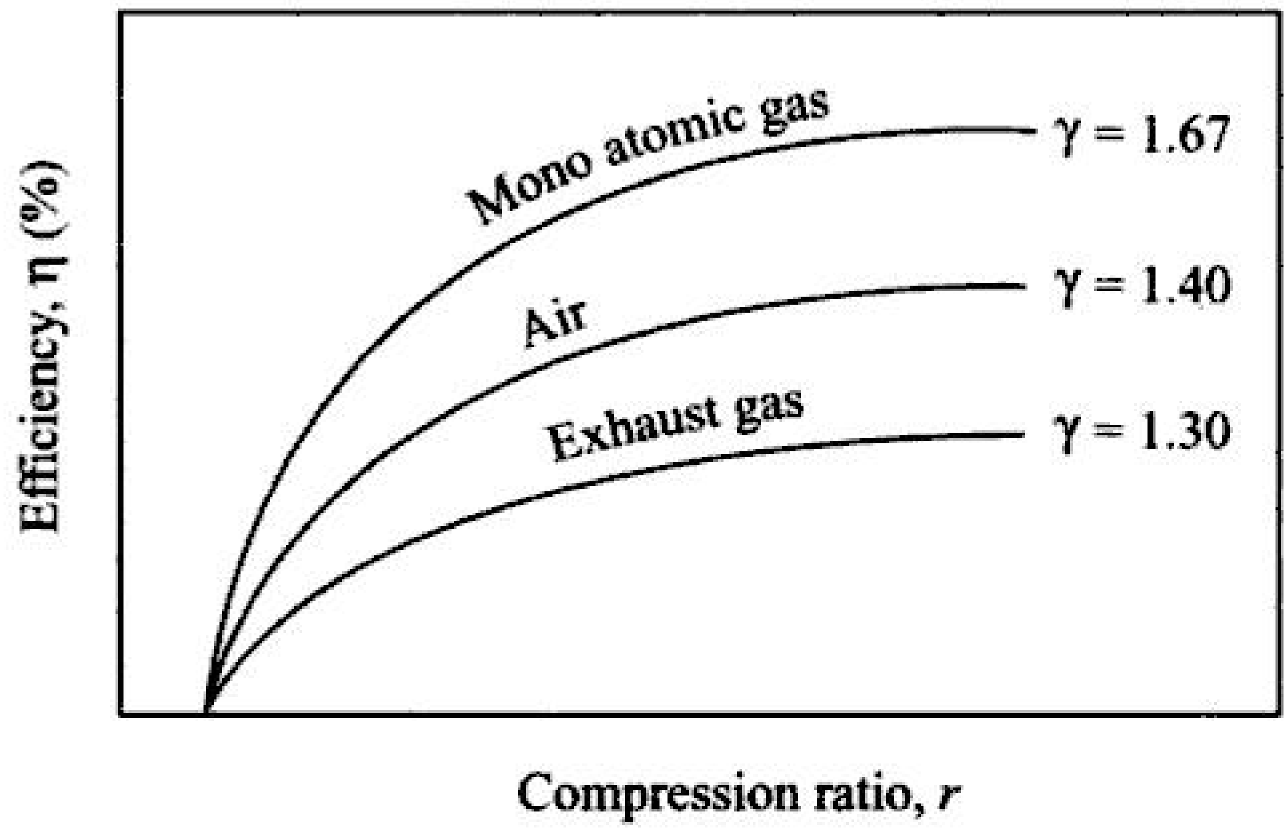


Fig. 3.6 Effect of r and γ on Efficiency for Otto Cycle

$$V_1 = rV_2 \quad \text{and} \quad V_4 = rV_3$$

therefore,

$$W = \frac{p_1V_1}{\gamma-1}\left(\frac{p_3V_3}{p_1V_1} - \frac{p_4V_4}{p_1V_1} - \frac{p_2V_2}{p_1V_1} + 1\right) \tag{3.28}$$

$$= \frac{p_1V_1}{\gamma-1}\left(\frac{r_pr^\gamma}{r} - r_p - \frac{r^\gamma}{r} + 1\right)$$

$$= \frac{p_1V_1}{\gamma-1}\left(r_pr^{\gamma-1} - r_p - r^{\gamma-1} + 1\right)$$

$$= \frac{p_1V_1}{\gamma-1}(r_p-1)\left(r^{\gamma-1} - 1\right) \tag{3.29}$$

3.5.3 Mean Effective Pressure

The mean effective pressure of the cycle is given by

$$p_m = \frac{\text{Work output}}{\text{Swept volume}} \tag{3.30}$$

$$\text{Swept volume} = V_1 - V_2 = V_2(r-1)$$

$$p_m = \frac{\frac{1}{\gamma-1}p_1V_1(r_p-1)\left(r^{(\gamma-1)} - 1\right)}{V_2(r-1)}$$

$$= \frac{p_1r(r_p-1)\left(r^{(\gamma-1)} - 1\right)}{(\gamma-1)(r-1)} \tag{3.31}$$

Thus, it can be seen that the work output is directly proportional to pressure ratio, r_p. The mean effective pressure which is an indication of the internal work output increases with a pressure ratio at a fixed value of compression ratio and ratio of specific heats. For an Otto cycle, an increase in the compression ratio leads to an increase in the mean effective pressure as well as the thermal efficiency.

3.6 THE DIESEL CYCLE

In actual spark-ignition engines, the upper limit of compression ratio is limited by the self-ignition temperature of the fuel. This limitation on the compression ratio can be circumvented if air and fuel are compressed separately and brought together at the time of combustion. In such an arrangement fuel can be injected into the cylinder which contains compressed air at a higher temperature than the self-ignition temperature of the fuel. Hence the fuel ignites on its own accord and requires no special device like an ignition system in a spark-ignition engine. Such engines work on heavy liquid fuels. These engines are called compression-ignition engines and they work on a ideal cycle known as Diesel cycle. The difference between Otto and Diesel cycles is in the process of heat addition. In Otto cycle the heat addition takes place at constant volume whereas in the Diesel cycle it is at constant pressure. For this reason, the Diesel cycle is often referred to as the constant-pressure cycle. It is better to avoid this term as it creates confusion with Joules cycle. The Diesel cycle is shown on p-V and T-s diagrams in Fig.3.7(a) and 3.7(b) respectively.

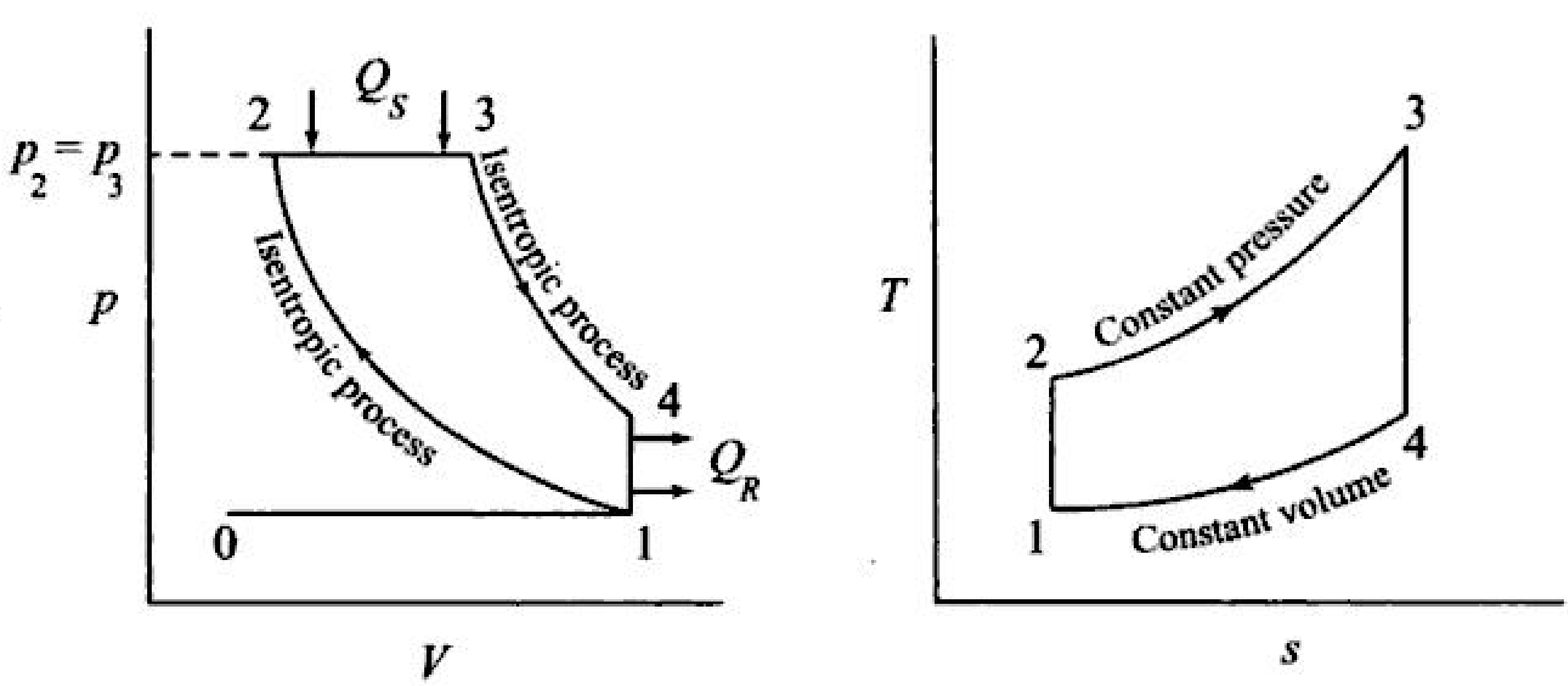


Fig. 3.7 Diesel Cycle

To analyze the diesel cycle the suction and exhaust strokes, represented by 0→1 and 1→0, are neglected as in the case of the Otto cycle. Here, the volume ratio $\frac{V_1}{V_2}$ is the compression ratio, r. The volume ratio $\frac{V_3}{V_2}$ is called the cut-off ratio, r_c.

3.6.1 Thermal Efficiency

The thermal efficiency of the Diesel cycle is given by

$$\eta_{Diesel} = \frac{Q_S - Q_R}{Q_S}$$

$$= \frac{mC_p(T_3 - T_2) - mC_v(T_4 - T_1)}{mC_p(T_3 - T_2)} \tag{3.32}$$

$$= 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$= 1 - \frac{1}{\gamma}\left(\frac{T_4 - T_1}{T_3 - T_2}\right) \tag{3.33}$$

Considering the process 1→2

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = T_1 r^{(\gamma-1)} \tag{3.34}$$

Considering the constant pressure process 2→3, we have

$$\frac{V_2}{T_2} = \frac{V_3}{T_3}$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c \quad \text{(say)}$$

$$T_3 = T_2 r_c \tag{3.35}$$

From Eqs.3.34 and 3.35, we have

$$T_3 = T_1 r^{(\gamma-1)} r_c \tag{3.36}$$

Considering process 3→4, we have

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{(\gamma-1)} \tag{3.37}$$

$$= T_3\left(\frac{V_3}{V_2} \times \frac{V_2}{V_4}\right)^{(\gamma-1)}$$

$$= \quad T_3\left(\frac{r_c}{r}\right)^{(\gamma-1)} \tag{3.38}$$

From Eqs.3.36 and 3.37, we have

$$T_4 \quad = \quad T_1 r^{(\gamma-1)} r_c \left(\frac{r_c}{r}\right)^{(\gamma-1)} \quad = \quad T_1 r_c^{\gamma}$$

$$\eta_{Diesel} \quad = \quad 1 - \frac{1}{\gamma}\left[\frac{T_1(r_c^{\gamma}-1)}{T_1\left(r^{(\gamma-1)}r_c - r^{(\gamma-1)}\right)}\right]$$

$$= \quad 1 - \frac{1}{\gamma}\left[\frac{(r_c^{\gamma}-1)}{r^{(\gamma-1)}r_c - r^{(\gamma-1)}}\right]$$

$$= \quad 1 - \frac{1}{r^{(\gamma-1)}}\left[\frac{r_c^{\gamma}-1}{\gamma(r_c-1)}\right] \tag{3.39}$$

It may be noted that the efficiency of the Diesel cycle is different from that of the Otto cycle only in the bracketed factor. This factor is always greater than unity. Hence for a given compression ratio, the Otto cycle is more efficient. In diesel engines the fuel cut-off ratio, r_c, depends on output, being maximum for maximum output. Therefore, unlike the Otto cycle the air-standard efficiency of the Diesel cycle depends on output. The higher efficiency of the Otto cycle as compared to the Diesel cycle for the same compression ratio is of no practical importance. In practice the operating compression ratios of diesel engines are much higher compared to spark-ignition engines working on Otto cycle. The normal range of compression ratio for diesel engine is 16 to 20 whereas for spark-ignition engines it is 6 to 10. Due to the higher compression ratios used in diesel engines the efficiency of a diesel engine is more than that of the gasoline engine.

3.6.2 Work Output

The net work output for a Diesel cycle is given by

$$W \;=\; p_2(V_3 - V_2) + \frac{p_3V_3 - p_4V_4}{\gamma-1} - \frac{p_2V_2 - p_1V_1}{\gamma-1} \tag{3.40}$$

$$= \; p_2V_2(r_c - 1) + \frac{p_3r_cV_2 - p_4rV_2}{\gamma-1} - \frac{p_2V_2 - p_1rV_2}{\gamma-1}$$

$$= \; V_2\left[\frac{p_2(r_c-1)(\gamma-1) + p_3r_c - p_4r - (p_2 - p_1r)}{\gamma-1}\right]$$

$$= V_2\left[\frac{p_2(r_c-1)(\gamma-1)+p_3\left(r_c-\frac{p_4}{p_3}r\right)-p_2\left(1-\frac{p_1}{p_2}r\right)}{\gamma-1}\right]$$

$$= p_2V_2\left[\frac{(r_c-1)(\gamma-1)+\left(r_c-r_c^{\gamma}r^{(1-\gamma)}\right)-\left(1-r^{(1-\gamma)}\right)}{\gamma-1}\right]$$

$$= \frac{p_1V_1r^{(\gamma-1)}\left[\gamma(r_c-1)-r^{(1-\gamma)}(r_c^{\gamma}-1)\right]}{\gamma-1} \quad (3.41)$$

3.6.3 Mean Effective Pressure

The expression for mean effective pressure can be shown to be

$$p_m = \frac{p_1V_1\left[r^{(\gamma-1)}\gamma(r_c-1)-(r_c^{\gamma}-1)\right]}{(\gamma-1)V_1\left(\frac{r-1}{r}\right)} \quad (3.42)$$

$$= \frac{p_1[\gamma r^{\gamma}(r_c-1)-r(r_c^{\gamma}-1)]}{(\gamma-1)(r-1)} \quad (3.43)$$

3.7 THE DUAL CYCLE

In the Otto cycle, combustion is assumed at constant volume while in Diesel cycle combustion is at constant pressure. In practice they are far from real. Since, some time interval is required for the chemical reactions during combustion process, the combustion cannot take place at constant volume. Similarly, due to rapid uncontrolled combustion in diesel engines, combustion does not occur at constant pressure. The Dual cycle, also called a mixed cycle or limited pressure cycle, is a compromise between Otto and Diesel cycles. Figures 3.8(a) and 3.8(b) show the Dual cycle on p-V and T-s diagrams respectively.

In a Dual cycle a part of the heat is first supplied to the system at constant volume and then the remaining part at constant pressure.

3.7.1 Thermal Efficiency

The efficiency of the cycle may be written as

$$\eta_{Dual} = \frac{Q_S-Q_R}{Q_S} \quad (3.44)$$

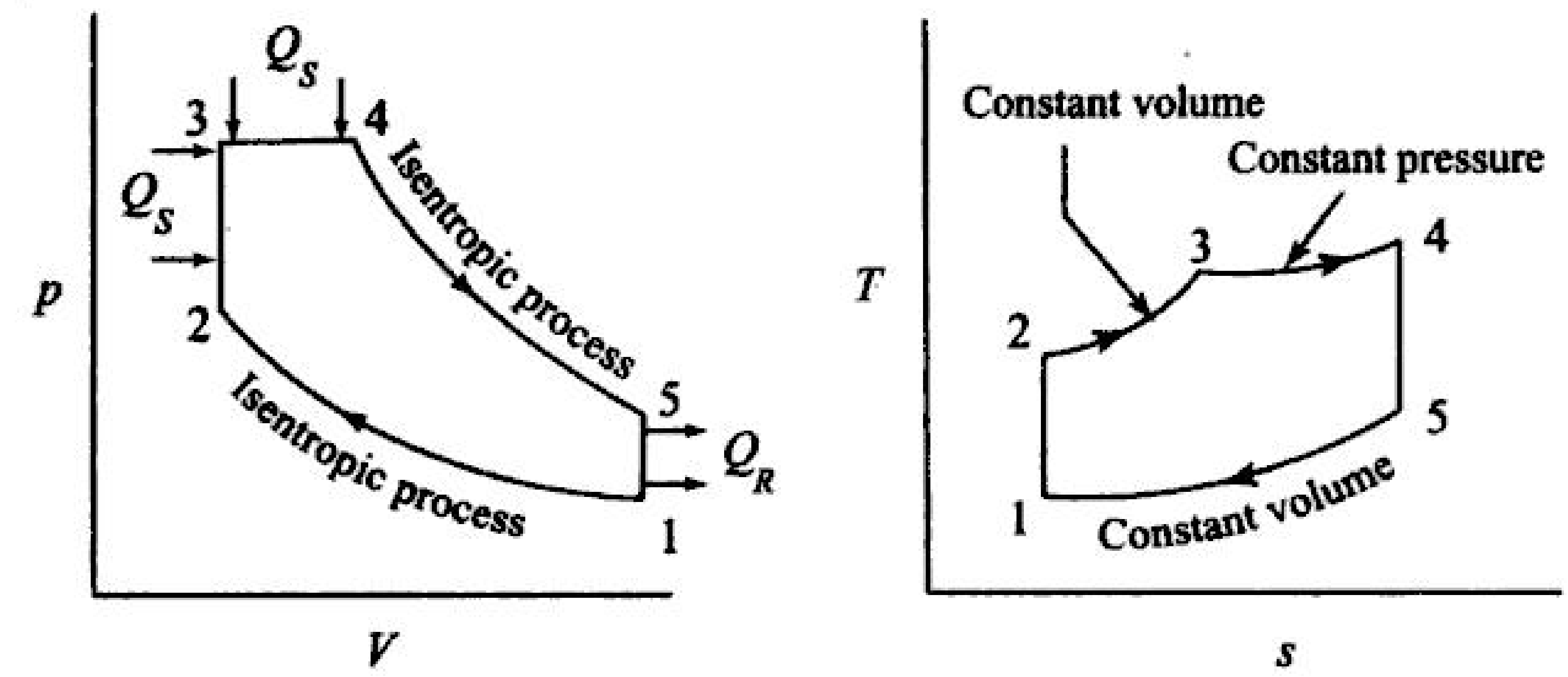


Fig. 3.8 Dual Cycle

$$= \frac{mC_v(T_3 - T_2) + mC_p(T_4 - T_3) - mC_v(T_5 - T_1)}{mC_v(T_3 - T_2) + mC_p(T_4 - T_3)}$$

$$= 1 - \frac{T_5 - T_1}{(T_3 - T_2) + \gamma(T_4 - T_3)} \qquad (3.45)$$

Now,

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = T_1 r^{(\gamma-1)} \qquad (3.46)$$

$$T_3 = T_2\left(\frac{p_3}{p_2}\right) = T_1 r_p r^{(\gamma-1)} \qquad (3.47)$$

where r_p is the pressure ratio in the constant volume heat addition process and is equal to $\frac{p_3}{p_2}$.

Cut-off ratio r_c is given by $(\frac{V_4}{V_3})$

$$T_4 = T_3\frac{V_4}{V_3} = T_3 r_c$$

Substituting for T_3 from Eq.3.47

$$T_4 = T_1 r_c r_p r^{(\gamma-1)} \qquad (3.48)$$

and

$$T_5 = T_4\left(\frac{V_4}{V_5}\right)^{(\gamma-1)} = T_1 r_p r_c r^{(\gamma-1)}\left(\frac{V_4}{V_5}\right)^{(\gamma-1)} \qquad (3.49)$$

Now

$$\frac{V_4}{V_5} = \frac{V_4}{V_1} = \frac{V_4}{V_3} \times \frac{V_3}{V_1} \qquad (3.50)$$

$$= \frac{V_4}{V_3} \times \frac{V_2}{V_1} \qquad (\text{since} V_2 = V_3)$$

Therefore,

$$\frac{V_4}{V_5} = \frac{r_c}{r} \tag{3.51}$$

where $\frac{V_4}{V_5}$ is the expansion ratio. Now,

$$\begin{aligned} T_5 &= T_1 r_p r_c r^{\gamma-1} \left(\frac{r_c}{r}\right)^{\gamma-1} \\ &= T_1 r_p r_c^{\gamma} \end{aligned} \tag{3.52}$$

Substituting for T_2, T_3, T_4 and T_5 into Eq.3.45 and simplifying

$$\eta = 1 - \frac{1}{r^{(\gamma-1)}} \left[\frac{r_p r_c^{\gamma} - 1}{(r_p - 1) + r_p \gamma (r_c - 1)}\right] \tag{3.53}$$

It can be seen from the above equation that a value of $r_p > 1$ results in an increased efficiency for a given value of r_c and γ. Thus the efficiency of Dual cycle lies between that of the Otto cycle and the Diesel cycle having same compression ratio.

With $r_c = 1$, it becomes an Otto cycle, and with $r_p = 1$, it becomes a Diesel cycle.

3.7.2 Work Output

The net work output of the cycle is given by

$$\begin{aligned} W &= p_3(V_4 - V_3) + \frac{p_4 V_4 - p_5 V_5}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1} \\ &= \frac{p_1 V_1}{\gamma - 1}\left[(\gamma - 1)\left(\frac{p_4 V_4}{p_1 V_1} - \frac{p_3 V_3}{p_1 V_1}\right) + \frac{p_4 V_4}{p_1 V_1} - \frac{p_5 V_5}{p_1 V_1} - \frac{p_2 V_2}{p_1 V_1} + 1\right] \\ &= \frac{p_1 V_1}{\gamma - 1}\Big[(\gamma - 1)\left(r_c r_p r^{\gamma-1} - r_p r^{\gamma-1}\right) + r_c r_p r^{\gamma-1} \\ &\qquad - r_p r_c^{\gamma} - r^{\gamma-1} + 1\Big] \\ &= \frac{p_1 V_1}{\gamma - 1}\left[\gamma r_c r_p r^{\gamma-1} - \gamma r_p r^{\gamma-1} + r_p r^{\gamma-1} - r_p r_c^{\gamma} - r^{\gamma-1} + 1\right] \\ &= \frac{p_1 V_1}{\gamma - 1}\left[\gamma r_p r^{\gamma-1}(r_c - 1) + r^{\gamma-1}(r_p - 1) - (r_p r_c^{\gamma} - 1)\right] \end{aligned} \tag{3.54}$$

3.7.3 Mean Effective Pressure

The mean effective pressure is given by

$$p_m = \frac{\text{Work output}}{\text{Swept volume}} = \frac{W}{V_s}$$

$$= \frac{1}{V_1 - V_2}\frac{p_1 V_1}{\gamma - 1}\left[\gamma r_p r^{\gamma-1}(r_c - 1) + r^{\gamma-1}(r_p - 1) - (r_p r_c^\gamma - 1)\right]$$

$$= \frac{1}{\left(1 - \frac{V_2}{V_1}\right)}\frac{p_1}{(\gamma - 1)}\Big[\gamma r_p r^{\gamma-1}(r_c - 1) + r^{\gamma-1}(r_p - 1)$$

$$-(r_p r_c^\gamma - 1)\Big]$$

$$= p_1 \frac{[\gamma r_p r^\gamma (r_c - 1) + r^\gamma (r_p - 1) - r(r_p r_c^\gamma - 1)]}{(\gamma - 1)(r - 1)} \qquad (3.55)$$

3.8 COMPARISON OF THE OTTO, DIESEL AND DUAL CYCLES

The important variable factors which are used as the basis for comparison of the cycles are compression ratio, peak pressure, heat addition, heat rejection and the net work. In order to compare the performance of the Otto, Diesel and Dual combustion cycles some of the variable factors must be fixed. In this section, a comparison of these three cycles is made for the same compression ratio, same heat addition, constant maximum pressure and temperature, same heat rejection and net work output. This analysis will show which cycle is more efficient for a given set of operating conditions.

3.8.1 Same Compression Ratio and Heat Addition

The Otto cycle 1→2→3→4→1, the Diesel cycle 1→2→3′→4′→1 and the Dual cycle 1→2→2″→3″→4″→1 are shown in p-V and T-s diagrams in Fig.3.9(a) and 3.9(b) respectively for the same compression ratio and heat input.

From the T-s diagram, it can be seen that Area 5236 = Area 523′6′ = Area 522″3″6″ as this area represents the heat input which is the same for all cycles.

All the cycles start from the same initial state point 1 and the air is compressed from state 1 to 2 as the compression ratio is same. It is seen from the T-s diagram for the same heat input, the heat rejection in Otto cycle (area 5146) is minimum and heat rejection

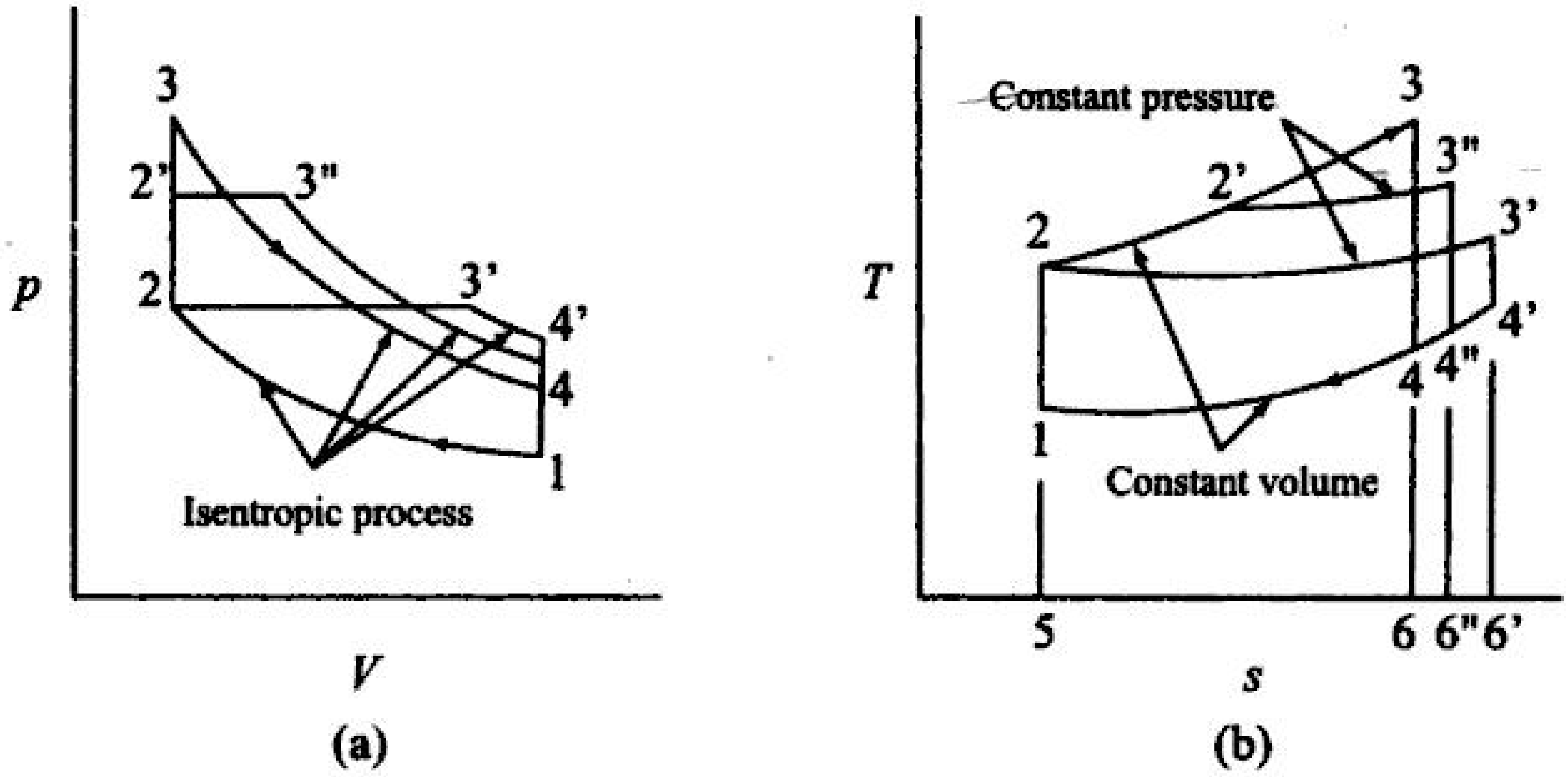


Fig. 3.9 Same Compression Ratio and Heat Addition

in Diesel cycle ($514'6'$) is maximum. *Consequently Otto cycle has the highest work output and efficiency. Diesel cycle has the least efficiency and Dual cycle having the efficiency between the two.*

One more observation can be made i.e., Otto cycle allows the working medium to expand more whereas Diesel cycle is least in this respect. The reason is heat is added before expansion in the case of former (Otto cycle) and the last portion of heat supplied to the fluid has a relatively short expansion in case of the latter (Diesel cycle).

3.8.2 Same Compression Ratio and Heat Rejection

Efficiency of Otto cycle is given by [Figs.3.10(a) and 3.10(b)]

$$\eta_{Otto} = 1 - \frac{Q_R}{Q_S}$$

where Q_S is the heat supplied in the Otto cycle and is equal to the area under the curve $2 \rightarrow 3$ on the T-s diagram [Fig.3.10(b)]. The efficiency of the Diesel cycle is given by

$$\eta_{Diesel} = 1 - \frac{Q_R}{Q'_S}$$

where Q'_S is heat supplied in the Diesel cycle and is equal to the area under the curve $2 \rightarrow 3'$ on the T-s diagram [Fig.3.10(b)].

From the T-s diagram in Fig.3.10 it is clear that $Q_S > Q'_S$ i.e., heat supplied in the Otto cycle is more than that of the Diesel cycle. *Hence, it is evident that, the efficiency of the Otto cycle is greater*

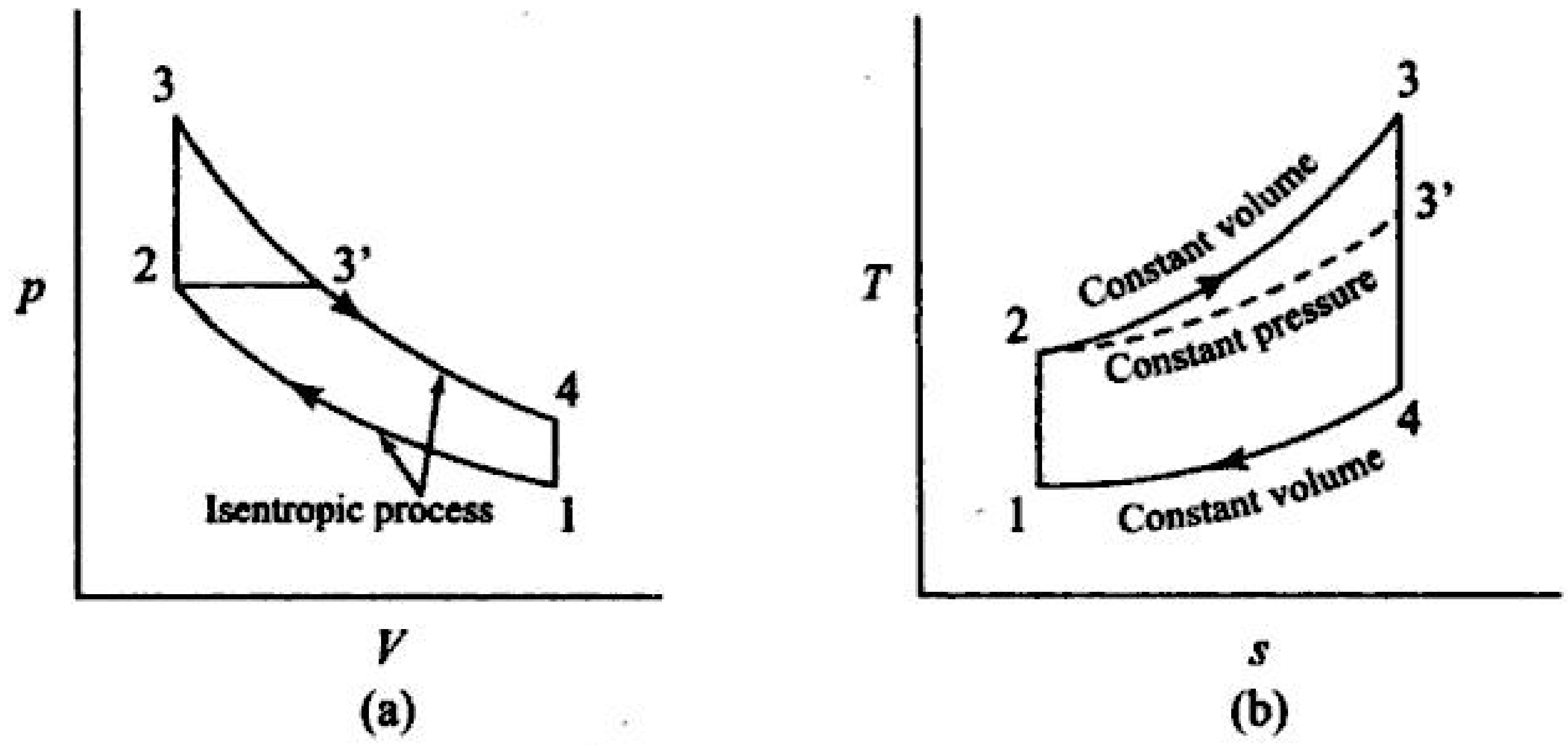


Fig. 3.10 Same Compression Ratio and Heat Rejection

than the efficiency of the Diesel cycle for a given compression ratio and heat rejection.

3.8.3 Same Peak Pressure, Peak Temperature and Heat Rejection

Figures 3.11(a) and 3.11(b) show the Otto cycle 1→2→3→4 and Diesel cycle 1→2′→3→4 on *p*-*V* and *T*-*s* coordinates, where the peak pressure and temperature and the amount of heat rejected are the same.

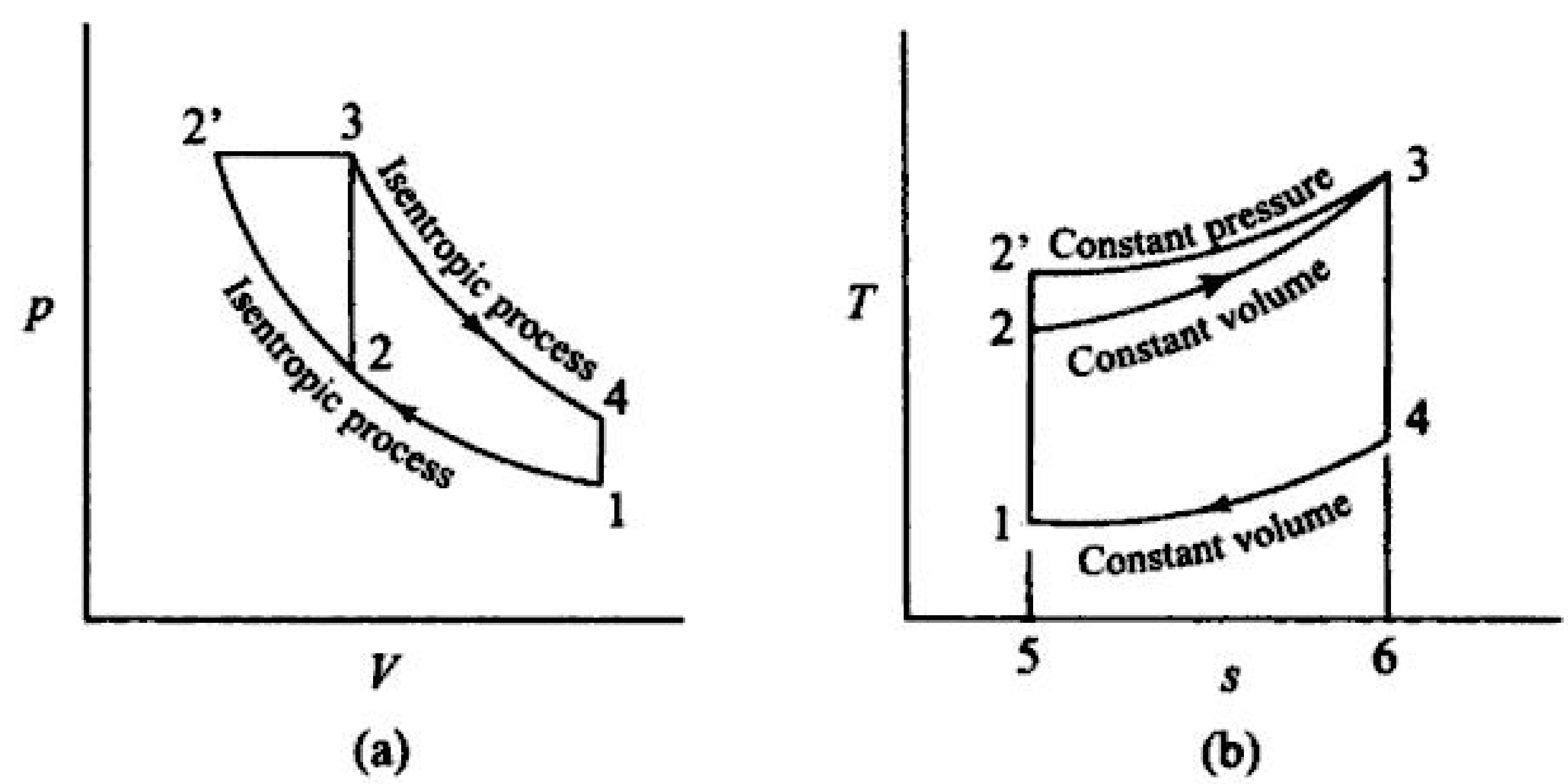


Fig. 3.11 Same Peak Pressure and Temperature

The efficiency of the Otto cycle 1→2→3→4 is given by

$$\eta_{Otto} = 1 - \frac{Q_R}{Q_S}$$

where Q_S in the area under the curve 2→3 in Fig.3.11(b).
The efficiency of the Diesel cycle, 1→2→3′→3→4 is

$$\eta_{Diesel} = 1 - \frac{Q_R}{Q'_S}$$

where Q'_S is the area under the curve 2′→3 in Fig.3.11(b).

It is evident from Fig.3.11 that $Q'_S > Q_S$. Therefore, the Diesel cycle efficiency is greater than the Otto cycle efficiency when both engines are built to withstand the same thermal and mechanical stresses.

3.8.4 Same Maximum Pressure and Heat Input

For same maximum pressure and same heat input the Otto cycle (12341) and Diesel cycle (12′3′4′1) are shown on *p-V* and *T-s* diagrams in Figs.3.12(a) and 3.12(b) respectively.

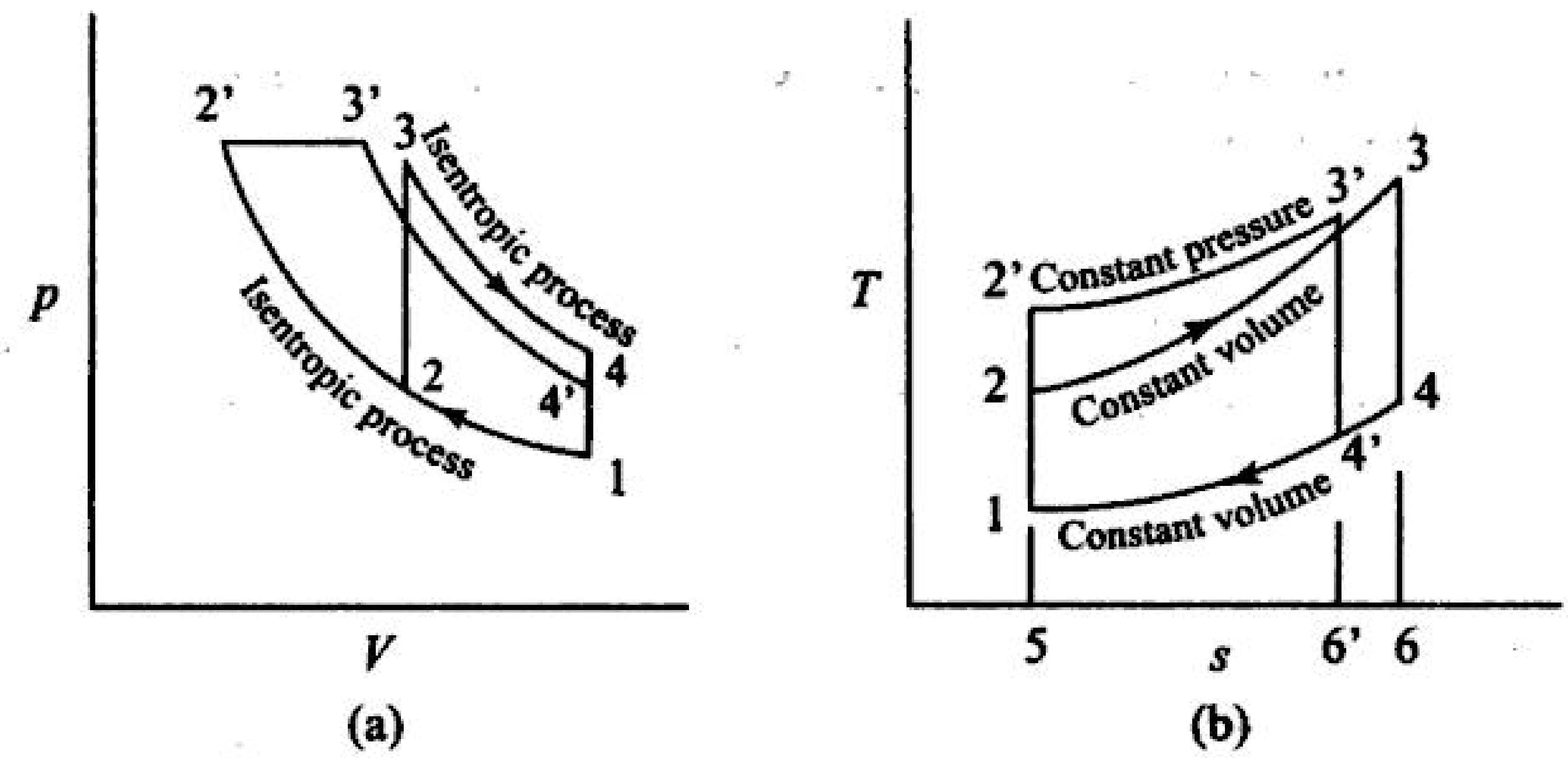


Fig. 3.12 Same Maximum Pressure and Heat Input

It is evident from the figure that the heat rejection for Otto cycle (area 1564 on *T-s* diagram) is more than the heat rejected in Diesel cycle (156′4′). Hence Diesel cycle is more efficient than Otto cycle for the condition of same maximum pressure and heat input. One can make a note that with these conditions the Diesel cycle has higher compression ratio $\frac{V_1}{V_{2'}}$ than that of Otto cycle $\frac{V_1}{V_2}$. One should also note that the cycle which is having higher efficiency allows maximum expansion. The Dual cycle efficiency will be between these two.

3.8.5 Same Maximum Pressure and Work Output

The efficiency, η, can be written as

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{\text{Work done}}{\text{Work done} + \text{Heat rejected}}$$

Refer to T-s diagram in Fig.3.12(b). For same work output the area 1234 (work output of Otto cycle) and area 12′3′4′ (work output of Diesel cycle) are same. To achieve this, the entropy at 3 should be greater than entropy at 3′. It is clear that the heat rejection for Otto cycle is more than that of Diesel cycle. Hence, for these conditions the Diesel cycle is more efficient than the Otto cycle. The efficiency of Dual cycle lies between the two cycles.

3.9 THE LENOIR CYCLE

The Lenoir cycle consists of the following processes [see Fig.3.13(a)]. Constant volume heat addition (1→2); isentropic expansion (2→3); constant pressure heat rejection (3→1). The Lenoir cycle is used for pulse jet engines.

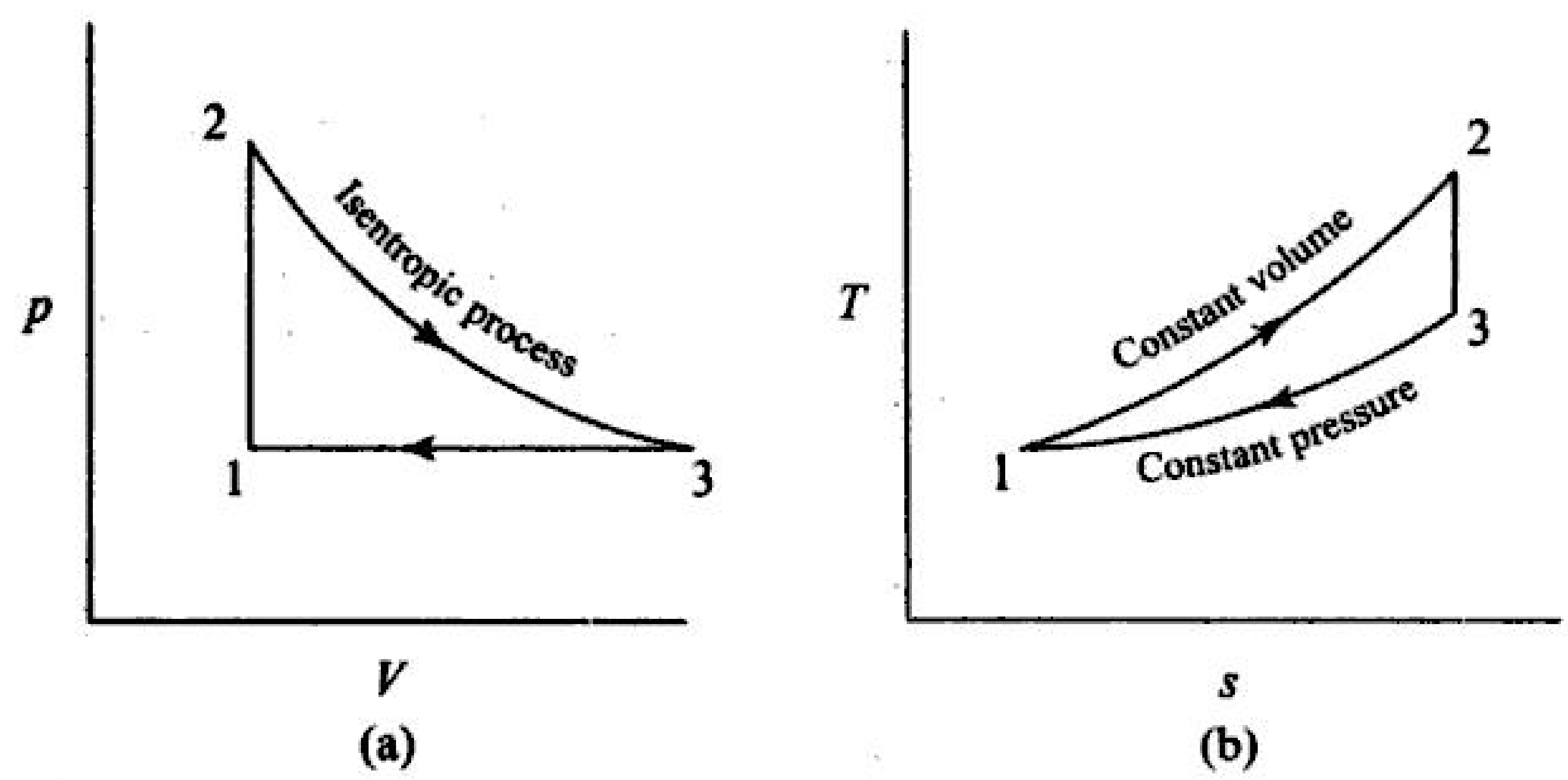


Fig. 3.13 Lenoir Cycle

$$\eta_{Lenoir} = \frac{Q_S - Q_F}{Q_S}$$

$$Q_S = mC_v(T_2 - T_1) \tag{3.56}$$

$$Q_R = mC_p(T_3 - T_1) \tag{3.57}$$

$$\eta_{Lenoir} = \frac{mC_v(T_2 - T_1) - mC_p(T_3 - T_1)}{mC_v(T_2 - T_1)} \tag{3.58}$$

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

Taking $p_2/p_1 = r_p$, we have $T_2 = T_1 r_p$ and

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \tag{3.59}$$

$$T_3 = T_2\left(\frac{1}{r_p}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} = T_1 r_p\left(\frac{1}{r_p}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} = T_1 r_p^{(1/\gamma)}$$

$$\eta = 1 - \gamma\left(\frac{T_1 r_p^{(1/\gamma)} - T_1}{T_1 r_p - T_1}\right) = 1 - \gamma\left(\frac{r_p^{(1/\gamma)} - 1}{r_p - 1}\right) \tag{3.60}$$

Thus the efficiency of the Lenoir cycle depends upon the pressure ratio as well as the ratio of specific heats, viz., γ.

3.10 THE ATKINSON CYCLE

Atkinson cycle is an ideal cycle for Otto engine exhausting to a gas turbine. In this cycle the isentropic expansion (3→4) of an Otto cycle (1234) is further allowed to proceed to the lowest cycle pressure so as to increase the work output. With this modification the cycle is known as Atkinson cycle. The cycle is shown on p-V and T-s diagrams in Figs.3.14(a) and 3.14(b) respectively.

$$\eta_{Atkinson} = \frac{Q_S - Q_R}{Q_S} \tag{3.61}$$

$$= \frac{mC_v(T_3 - T_2) - mC_p(T_{4'} - T_1)}{mC_v(T_3 - T_2)} \tag{3.62}$$

$$= 1 - \gamma\left(\frac{T_{4'} - T_1}{T_3 - T_2}\right) \tag{3.63}$$

the compression ratio, $r = \frac{V_1}{V_2}$ and the expansion ratio $e = \frac{V_{4'}}{V_3}$. Now,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = r^{(\gamma-1)} \tag{3.64}$$

Therefore,

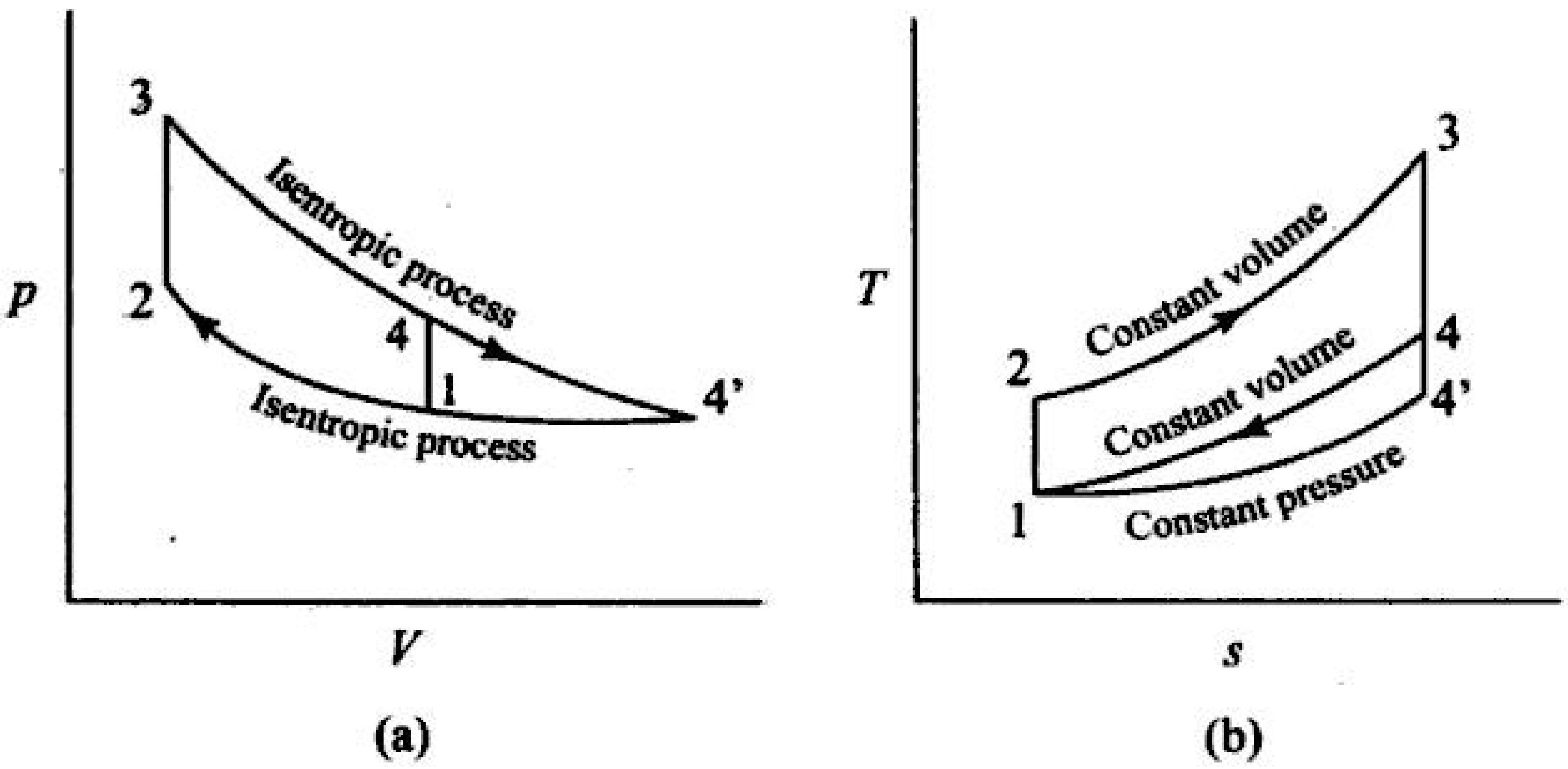


Fig. 3.14 Atkinson Cycle

$$T_2 = T_1 r^{(\gamma-1)} \tag{3.65}$$

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \left(\frac{p_3}{p_{4'}} \times \frac{p_{4'}}{p_2}\right)$$

$$= \left(\frac{p_3}{p_{4'}} \times \frac{p_1}{p_2}\right) \tag{3.66}$$

$$\frac{p_3}{p_{4'}} = \left(\frac{V_{4'}}{V_3}\right)^{\gamma} = e^{\gamma} \tag{3.67}$$

$$\frac{p_1}{p_2} = \left(\frac{V_2}{V_1}\right)^{\gamma} = \frac{1}{r^{\gamma}} \tag{3.68}$$

Substituting Eqs.3.67 and 3.68 in Eq.3.66,

$$\frac{T_3}{T_2} = \frac{e^{\gamma}}{r^{\gamma}} \tag{3.69}$$

$$T_3 = T_2\frac{e^{\gamma}}{r^{\gamma}} = T_1 r^{(\gamma-1)}\frac{e^{\gamma}}{r^{\gamma}} = T_1\frac{e^{\gamma}}{r} \tag{3.70}$$

$$\frac{T_{4'}}{T_3} = \left(\frac{V_3}{V_{4'}}\right)^{(\gamma-1)} = \frac{1}{e^{(\gamma-1)}}$$

$$T_{4'} = T_3\frac{1}{e^{(\gamma-1)}} \tag{3.71}$$

$$= T_1\left(\frac{e^{\gamma}}{r}\right)\left(\frac{1}{e^{(\gamma-1)}}\right)$$

$$T_{4'} = T_1 \frac{e}{r} \tag{3.72}$$

Substituting the values of T_2, T_3, $T_{4'}$ in the Eq.3.63,

$$\eta_{Atkinson} = 1 - \gamma \left[\frac{T_1 e/r - T_1}{T_1 e^{\gamma}/r - T_1 r^{(\gamma-1)}} \right]$$

$$= 1 - \gamma \left[\frac{e - r}{e^{\gamma} - r^{\gamma}} \right] \tag{3.73}$$

3.11 THE BRAYTON CYCLE

The Brayton cycle is a theoretical cycle for gas turbines. This cycle consists of two reversible adiabatic or isentropic processes and two constant pressure processes.

Figure 3.15 shows the Brayton cycle on p-V and T-s coordinates. The cycle is similar to the Diesel cycle in compression and heat addition. The isentropic expansion of the Diesel cycle is further extended followed by constant pressure heat rejection.

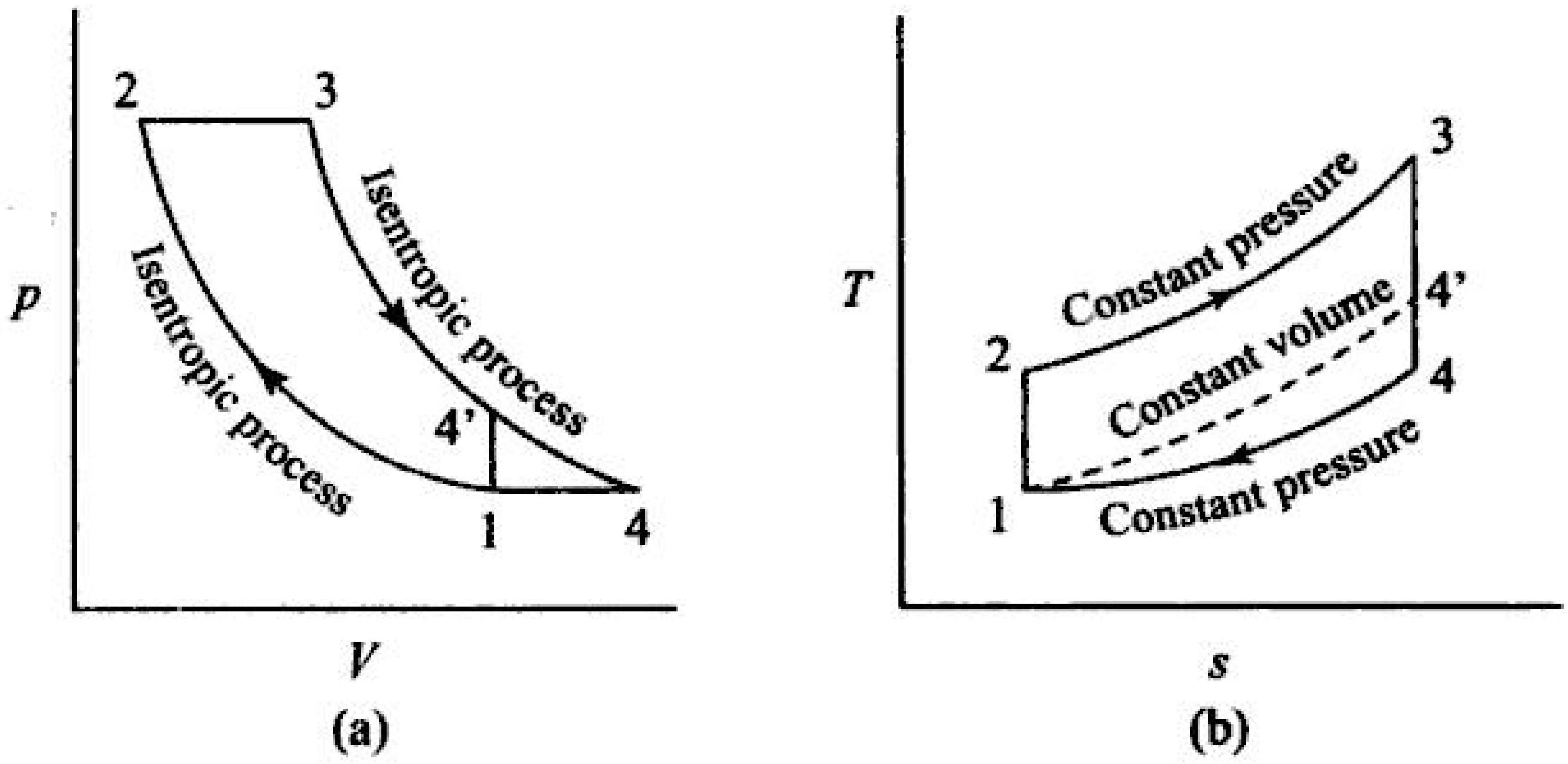


Fig. 3.15 Brayton Cycle

$$\eta_{Brayton} = \frac{Q_S - Q_R}{Q_S}$$

$$= \frac{mC_p(T_3 - T_2) - mC_p(T_4 - T_1)}{mC_p(T_3 - T_2)} \tag{3.74}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

If r is compression ratio i.e., (V_1/V_2) and r_p is the pressure ratio i.e., (p_2/p_1) then,

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \tag{3.75}$$

$$= \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = r^{(\gamma-1)} \tag{3.76}$$

$$T_4 = \frac{T_3}{r^{(\gamma-1)}} \tag{3.77}$$

$$T_1 = \frac{T_2}{r^{(\gamma-1)}} \tag{3.78}$$

$$\eta_{Brayton} = 1 - \frac{(T_3/r^{(\gamma-1)}) - (T_2/r^{(\gamma-1)})}{T_3 - T_2}$$

$$= 1 - \frac{1}{r^{(\gamma-1)}} \tag{3.79}$$

$$r = \frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} = r_p^{\frac{1}{\gamma}} \tag{3.80}$$

$$= 1 - \frac{1}{\left(r_p^{\frac{1}{\gamma}}\right)^{\gamma-1}} = 1 - \frac{1}{r_p^{\left(\frac{\gamma-1}{\gamma}\right)}} \tag{3.81}$$

From Eq.3.81, it is seen that the efficiency of the Brayton cycle depends only on the pressure ratio and the ratio of specific heat, γ.

$$\text{Network output} = \text{Expansion work} - \text{Compression work}$$

$$= C_p(T_3 - T_4) - C_p(T_2 - T_1)$$

$$= C_pT_1\left(\frac{T_3}{T_1} - \frac{T_4}{T_1} - \frac{T_2}{T_1} + 1\right)$$

$$\frac{W}{C_pT_1} = \frac{T_3}{T_1} - \frac{T_4}{T_3}\frac{T_3}{T_1} - \frac{T_2}{T_1} + 1 \tag{3.82}$$

It can be easily seen from the Eq.3.82 (work output) that, the work output of the cycle depends on initial temperature, T_1, the ratio of the maximum to minimum temperature, $\frac{T_3}{T_1}$, pressure ratio, r_p and γ which are used in the calculation of $\frac{T_2}{T_1}$. Therefore, for the same pressure ratio and initial conditions work output depends on the maximum temperature of the cycle.

Worked out Examples

OTTO CYCLE

3.1 An engine working on Otto cycle has the following conditions : Pressure at the beginning of compression is 1 bar and pressure at the end of compression is 11 bar. Calculate the compression ratio and air-standard efficiency of the engine. Assume $\gamma = 1.4$.

Solution

$$r = \frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{1}{\gamma}\right)} = 11^{\frac{1}{1.4}}$$

$$= \mathbf{5.54} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

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

$$= 1 - \left(\frac{1}{5.54}\right)^{0.4} = 0.496$$

$$= \mathbf{49.6\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

3.2 In an engine working on ideal Otto cycle the temperatures at the beginning and end of compression are 50 °C and 373 °C. Find the compression ratio and the air-standard efficiency of the engine.

Solution

$$r = \frac{V_1}{V_2} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{\gamma-1}} = \left(\frac{646}{323}\right)^{\frac{1}{0.4}}$$

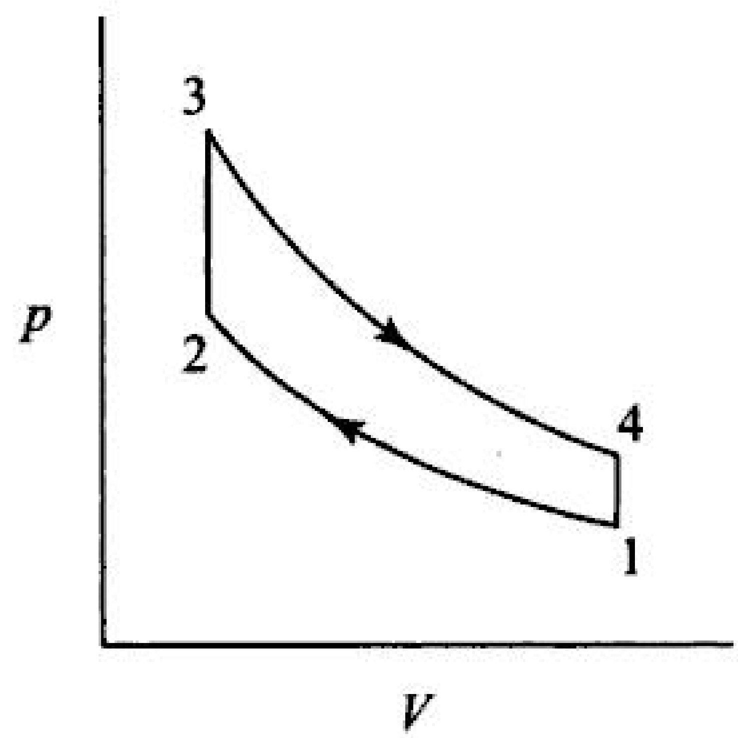
$$= \mathbf{5.66} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{T_1}{T_2}$$

$$= 1 - \frac{323}{646} = 0.5 = \mathbf{50\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

3.3 In an Otto cycle air at 17 °C and 1 bar is compressed adiabatically until the pressure is 15 bar. Heat is added at constant volume until the pressure rises to 40 bar. Calculate the air-standard efficiency, the compression ratio and the mean effective pressure for the cycle. Assume $C_v = 0.717$ kJ/kg K and $R = 8.314$ kJ/kmol K.

Solution



Consider the process 1 – 2

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

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

$$= \left(\frac{15}{1}\right)^{\frac{1}{1.4}} = 6.91$$

$$\eta = 1-\left(\frac{1}{r}\right)^{\gamma-1} = 1-\left(\frac{1}{6.91}\right)^{0.4}$$

$$= 0.539 = \mathbf{53.9\%} \qquad \underleftarrow{\text{Ans}}$$

$$T_2 = \frac{p_2V_2}{p_1V_1}T_1 = \frac{15}{1}\times\frac{1}{6.91}\times 290$$

$$= 629.5 \text{ K}$$

Consider the process 2 – 3

$$T_3 = \frac{p_3T_2}{p_2} = \frac{40}{15}\times 629.5 = 1678.7$$

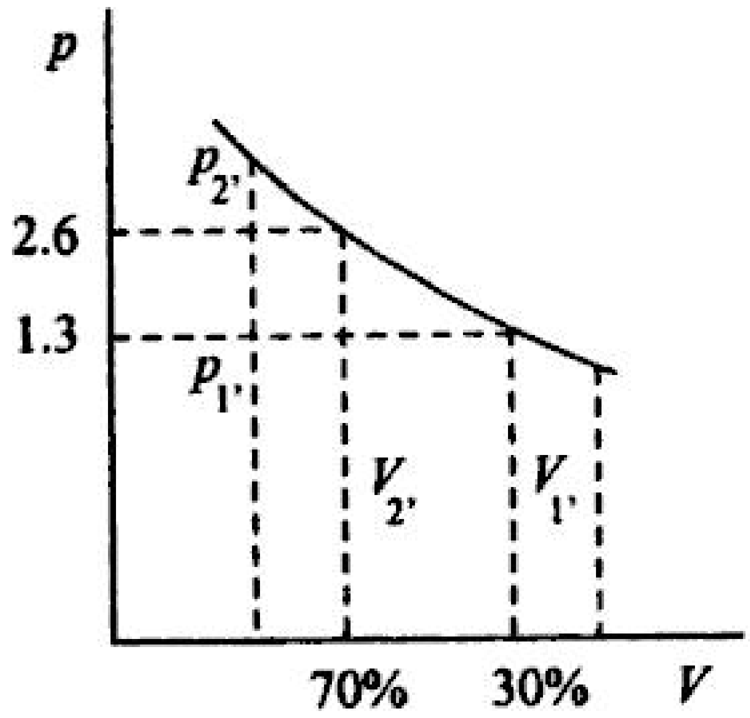
$$\begin{aligned}
\textit{Heat supplied} &= C_v(T_3 - T_2) \\
&= 0.717 \times (1678.7 - 629.5) \\
&= 752.3 \text{ kJ/kg} \\
\textit{Work done} &= \eta \times q_s \\
&= 0.539 \times 752.3 \quad = \quad 405.5 \text{ kJ/kg} \\
p_m &= \frac{\textit{Work done}}{\textit{Swept volume}} \\
v_1 &= \frac{V_1}{m} \quad = \quad M\frac{RT_1}{p_1} \\
&= \frac{8314 \times 290}{29 \times 1 \times 10^5} \quad = \quad 0.8314 \text{ m}^3\text{/kg} \\
v_1 - v_2 &= \frac{5.91}{6.91} \times 0.8314 \quad = \quad 0.711 \text{ m}^3\text{/kg} \\
p_m &= \frac{405.5}{0.711} \times 10^3 \quad = \quad 5.70 \times 10^5 \text{ N/m}^2 \\
&= \mathbf{5.70 \ bar}
\end{aligned}$$

$\stackrel{\text{Ans}}{\Longleftarrow}$

3.4 Fuel supplied to an SI engine has a calorific value 42000 kJ/kg. The pressure in the cylinder at 30% and 70% of the compression stroke are 1.3 bar and 2.6 bar respectively. Assuming that the compression follows the law $pV^{1.3}$ = constant. Find the compression ratio. If the relative efficiency of the engine compared with the air-standard efficiency is 50%. Calculate the fuel consumption in kg/kW h.

Solution

$$\begin{aligned}
V_2 &= 1 \\
V_{1'} &= 1 + 0.7(r - 1) \quad = \quad 0.7r + 0.3 \\
V_{2'} &= 1 + 0.3(r - 1) \quad = \quad 0.3r + 0.7 \\
\frac{V_{1'}}{V_{2'}} &= \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}
\end{aligned}$$



$$= \left(\frac{2.6}{1.3}\right)^{\frac{1}{1.3}} = 1.7$$

$$\frac{0.7r + 0.3}{0.3r + 0.7} = 1.7$$

$$r = \mathbf{4.68} \quad \underleftarrow{\text{Ans}}$$

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

$$= 1 - \frac{1}{4.68^{0.4}} = 0.46$$

$$= 46\%$$

$$\text{Relative efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Air-standard efficiency}}$$

$$\eta_{ith} = 0.5 \times 0.46 = 0.23$$

$$\eta_{th} = \frac{ip}{CV \times \dot{m}}$$

where $\dot{m}$ is in kg/s

$$\frac{\dot{m}}{ip} = \frac{1}{42000 \times 0.23}$$

$$= 1.035 \times 10^{-4} \text{ kg/kW s}$$

$$= 1.035 \times 10^{-4} \times 3600 \text{ kg/kW h}$$

$$isfc = \mathbf{0.373\ kg/kW\ h} \quad \underleftarrow{\text{Ans}}$$

3.5 A gas engine working on the Otto cycle has a cylinder of diameter 200 mm and stroke 250 mm. The clearance volume is 1570 cc. Find the air-standard efficiency. Assume $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K for air.

Solution

$$\text{Stroke volume, } V_s = \frac{\pi}{4}d^2L = \frac{\pi}{4} \times 20^2 \times 25$$

$$= 7853.98 \text{ cc}$$

$$\text{Compression ratio, } r = 1 + \frac{V_s}{V_c} = 1 + \frac{7853.98}{1570}$$

$$= 6.00$$

$$\gamma = \frac{C_p}{C_v} = \frac{1.004}{0.717} = 1.4$$

$$\text{Air-standard efficiency} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{6^{(0.4)}}$$

$$= 0.512 = \mathbf{51.2\%} \qquad \xLeftarrow{\text{Ans}}$$

3.6 In a S.I. engine working on the ideal Otto cycle, the compression ratio is 5.5. The pressure and temperature at the beginning of compression are 1 bar and 27 °C respectively. The peak pressure is 30 bar. Determine the pressure and temperatures at the salient points, the air-standard efficiency and the mean effective pressure. Assume ratio of specific heats to be 1.4 for air.

Solution

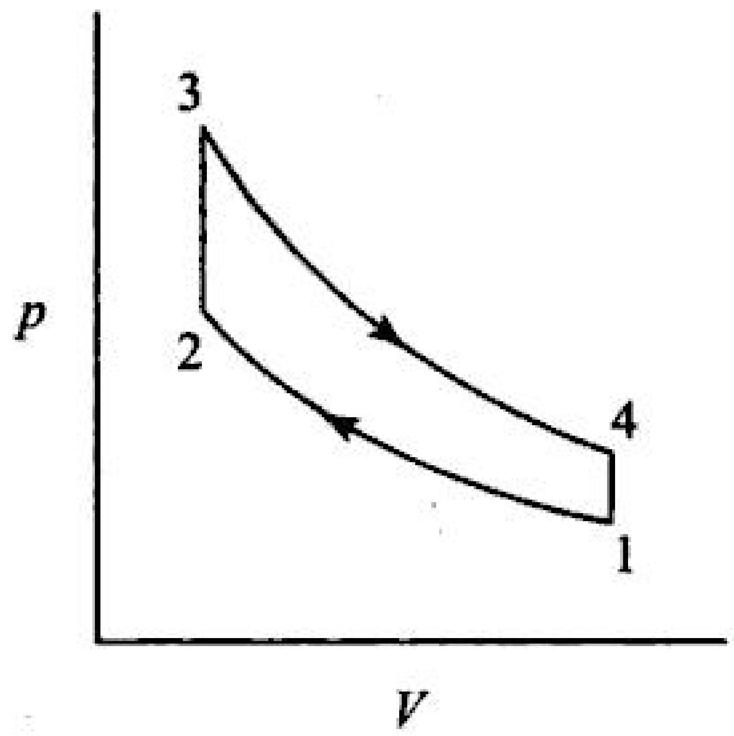
$$\text{Since } V_2 = V_3 = V_c$$
$$V_1 = rV_2 = rV_c,$$

Consider the process 1 – 2,

$$\frac{p_2}{p_1} = r^{\gamma} = 5.5^{1.4} = 10.88$$

$$p_2 = 10.88 \times 1 \times 10^5 = \mathbf{10.88 \times 10^5\ N/m^2} \qquad \xLeftarrow{\text{Ans}}$$

$$\frac{T_2}{T_1} = r^{\gamma-1} = 5.5^{0.4} = 1.978$$



$$T_2 = 1.978 \times 300 = 593.4 \text{ K} = \mathbf{320.4^\circ\ C} \quad \xLeftarrow{\text{Ans}}$$

Consider the process 2 – 3,

$$p_3 = 30 \times 10^5 \text{ N/m}^2$$

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \frac{30}{10.88} = 2.757$$

$$T_3 = 2.757 \times 593.4 = 1636 \text{ K}$$

$$= \mathbf{1363^\circ\ C} \quad \xLeftarrow{\text{Ans}}$$

Consider the process 3 – 4,

$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^\gamma = \left(\frac{V_1}{V_2}\right)^\gamma$$

$$= r^\gamma = 5.5^{1.4} = 10.88$$

$$p_4 = \frac{p_3}{10.88} = \mathbf{2.76 \times 10^5\ N/m^2} \quad \xLeftarrow{\text{Ans}}$$

$$\frac{T_3}{T_4} = r^{\gamma-1} = 5.5^{0.4} = 1.978$$

$$T_4 = \frac{T_3}{1.978} = \frac{1636}{1.978} = 827.1 \text{ K}$$

$$= \mathbf{554.1^\circ\ C} \quad \xLeftarrow{\text{Ans}}$$

$$\eta_{Otto} = 1 - \frac{1}{r^{(\gamma-1)}} = 1 - \frac{1}{5.5^{0.4}} = 0.4943$$

$$= \mathbf{49.43\%} \quad \xLeftarrow{\text{Ans}}$$

$$p_m = \frac{\text{Indicated work/cycle}}{V_s}$$

$$= \frac{\text{Area of } p\text{-}V \text{ diagram } 1234}{V_s}$$

$$\text{Area } 1234 = \text{Area under 3-4} - \text{Area under 2-1}$$

$$= \frac{p_3V_3 - p_4V_4}{\gamma - 1} - \frac{p_2V_2 - p_1V_1}{\gamma - 1}$$

$$= \frac{30 \times 10^5 \times V_c - 2.76 \times 10^5 \times 5.5V_c}{0.4}$$

$$- \frac{10.88 \times 10^5 \times V_c - 1 \times 10^5 \times 5.5V_c}{0.4}$$

$$= 23.6 \times 10^5 \times V_c = p_m \times V_s$$

$$p_m = \frac{23.6 \times 10^5 \times V_c}{V_s}$$

$$= \frac{23.6 \times 10^5 \times V_c}{4.5 \times V_c} = 5.24 \times 10^5 \text{ N/m}^2$$

$$= \mathbf{5.24 \text{ bar}}$$ Ans

3.7 A gas engine operating on the ideal Otto cycle has a compression ratio of 6:1. The pressure and temperature at the commencement of compression are 1 bar and 27 °C. The heat added during the constant volume combustion process is 1170 kJ/kg. Determine the peak pressure and temperature, work output per kg of air and air-standard efficiency. Assume C_v = 0.717 kJ/kg K and $\gamma = 1.4$ for air.

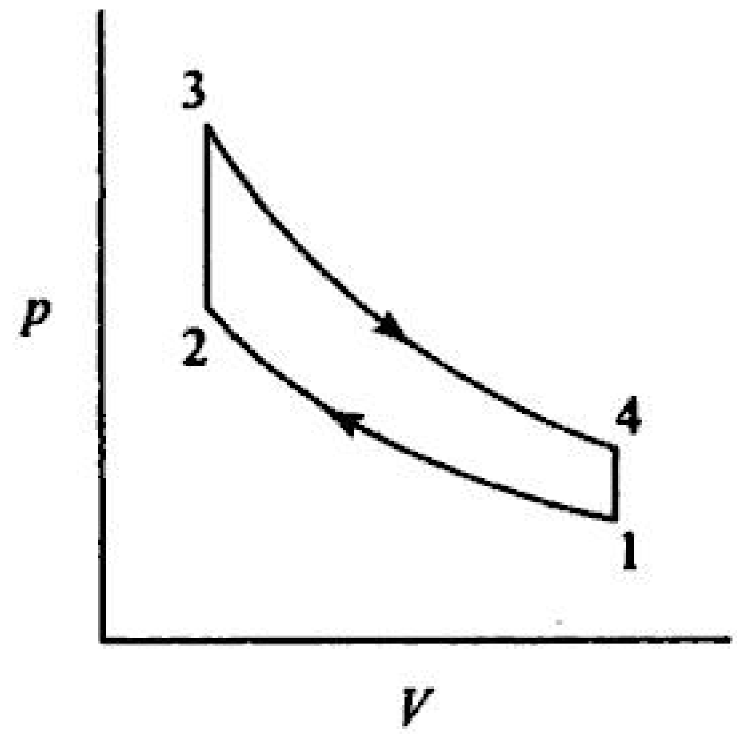
Solution

Consider the process 1-2

$$\frac{p_2}{p_1} = r^{\gamma}$$

$$= 6^{1.4} = 12.28$$

$$p_2 = 12.28 \times 10^5 \text{ N/m}^2$$



$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r^{\gamma-1} = 6^{0.4} = 2.05$$

$$T_2 = 2.05 \times 300 = 615K = 342^\circ \text{ C}$$

Consider the process 2-3

For unit mass flow

$$q_s = q_{2-3} = C_v(T_3 - T_2)$$

$$= 1170 \text{ kJ/kg}$$

$$T_3 - T_2 = \frac{1170}{0.717} = 1631.8$$

$$T_3 = 1631.8 + 615 = 2246.8 \text{ K}$$

$$= \mathbf{1973.8^\circ \text{ C}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\frac{p_3}{p_2} = \frac{T_3}{T_2} = \frac{2246.8}{615} = 3.65$$

$$\text{Peak pressure}, p_3 = 3.65 \times 12.28 \times 10^5$$

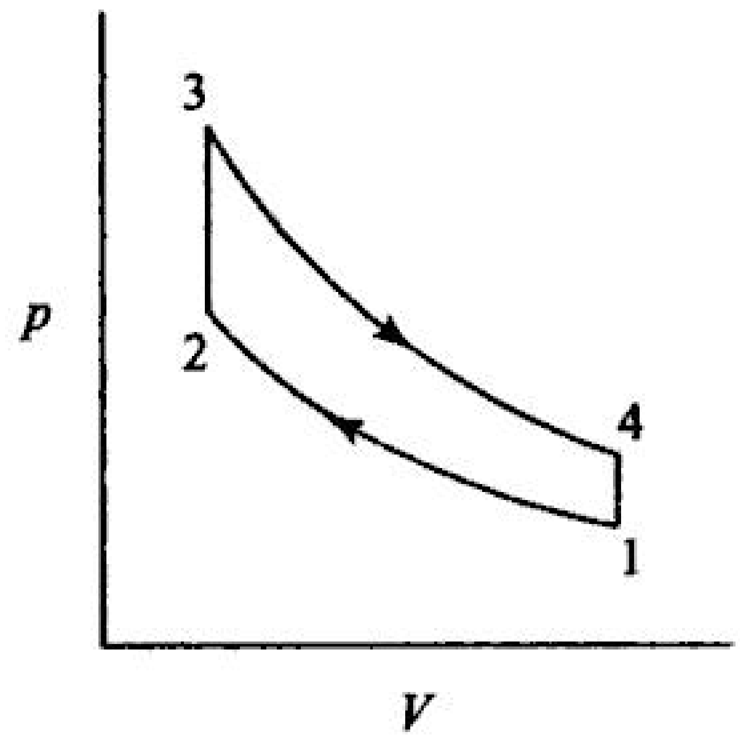
$$= 44.82 \times 10^5 \text{ N/m}^2 = \mathbf{44.82 \text{ bar}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Work output = *Area of p-V diagram*

= *Area under (3 - 4) - Area under (2 - 1)*

$$= \frac{p_3V_3 - p_4V_4}{\gamma - 1} - \frac{p_2V_2 - p_1V_1}{\gamma - 1}$$

$$= \frac{mR}{\gamma - 1}[(T_3 - T_4) - (T_2 - T_1)]$$



Consider the process 2 – 3

For unit mass flow

$$\text{Heat supplied, } q_s = C_v(T_3 - T_2)$$

$$= 0.717 \times (2000 - 672)$$

$$= \mathbf{952.2\ kJ/kg} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\eta = \frac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}} = \frac{w}{q_s}$$

$$w = \eta q_s = 0.539 \times 952.2$$

$$= \mathbf{513.2\ kJ/kg} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$p_3 = p_2\left(\frac{T_3}{T_2}\right) = 15 \times \frac{2000}{672}$$

$$= 44.64 \text{ bar}$$

Consider the process 3 – 4

$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^{\gamma}$$

$$= \left(\frac{V_1}{V_2}\right)^{\gamma} = \frac{p_2}{p_1}$$

$$p_4 = p_3\left(\frac{p_1}{p_2}\right) = \frac{44.64 \times 1}{15}$$

$$= \mathbf{2.98\ bar} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

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$$= \left(\frac{35}{1}\right)^{\frac{1}{1.4}} = 12.674$$

$$\text{Cut-off ratio} = \frac{V_3}{V_2} = \frac{V_3}{V_1} \times \frac{V_1}{V_2}$$

$$= \frac{\text{Compression ratio}}{\text{Expansion ratio}}$$

$$= \frac{12.674}{5} = 2.535$$

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

$$= \left(\frac{35}{1}\right)^{0.286} = 2.76$$

$$T_2 = 2.76 \times 290 = 801.7 \text{ K}$$

Consider the process 2 – 3

$$T_3 = T_2\frac{V_3}{V_2} = 801.7 \times \frac{V_3}{V_2}$$

$$= 801.7 \times 2.535 = 2032.3 \text{ K}$$

Consider the process 3 – 4

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{\gamma-1} = 2032.3 \times \left(\frac{1}{5}\right)^{0.4}$$

$$= 1067.6 \text{ K}$$

$$\text{Heat added} = C_p(T_3 - T_2) = 1.004 \times (2032.3 - 801.7)$$

$$= \mathbf{1235.5 \ kJ/kg} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Heat rejected} = C_v(T_4 - T_1) = 0.717 \times (1067.6 - 290)$$

$$= \mathbf{557.5 \ kJ/kg} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Efficiency} = \frac{\text{Heat supplied - Heat rejected}}{\text{Heat supplied}}$$

$$= \frac{1235.5 - 557.5}{1235.5} = 0.549$$

$$= \mathbf{54.9\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

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$$\text{Cut-off ratio, } r_c = \frac{V_3}{V_2} = \frac{T_3}{T_2}$$

Consider the process 1 – 2

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

$$= r^{(\gamma-1)} = 14^{0.4} = 2.874$$

$$T_2 = 2.874 \times 333 = 957.04 \text{ K}$$

Consider the process 2 – 3

$$\text{Heat added/kg of air} = C_p(T_3 - T_2) = F/A \times CV$$

$$T_3 - T_2 = \frac{F/A \times CV}{C_p} = \frac{42000}{50 \times 1.004}$$

$$= 836.6$$

$$T_3 = 1793.64K$$

$$r_c = \frac{T_3}{T_2} = \frac{1793.64}{957.04} = 1.874$$

$$\eta = 1 - \frac{1}{1.4 \times 14^{0.4}} \times \left(\frac{1.874^{1.4} - 1}{0.874}\right)$$

$$= 0.60 = \mathbf{60\%}$$ $\stackrel{\text{Ans}}{\Longleftarrow}$

3.18 In an ideal Diesel cycle, the pressure and temperature are 1.03 bar and 27 °C respectively. The maximum pressure in the cycle is 47 bar and the heat supplied during the cycle is 545 kJ/kg. Determine (i) the compression ratio (ii) the temperature at the end of compression (iii) the temperature at the end of constant pressure combustion and (iv) the air-standard efficiency. Assume $\gamma = 1.4$ and $C_p = 1.004$ kJ/kg K for air.

Solution

$$p_2 = p_3 = 47 \times 10^5 \text{ N/m}^2$$

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

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$$V_s = 6 \times \frac{\pi}{4} d^2\ L = 6 \times \frac{\pi}{4} \times 10^2 \times 12$$

$$= 5654.8 \text{ cc} = 5.65 \times 10^{-3} \text{ m}^3$$

$$V_1 = 5.65 \times 10^{-3} \times \frac{9}{8} = 6.36 \times 10^{-3} \text{ m}^3$$

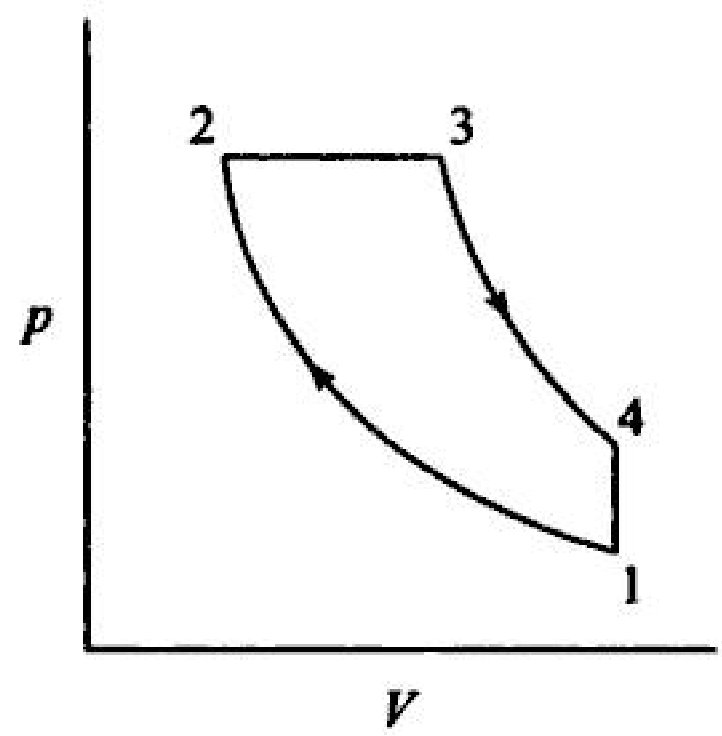
$$\dot{m}_a = \frac{1.03 \times 10^5 \times 6.36 \times 10^{-3} \times 30}{287 \times 308 \times 2}$$

$$= 0.111 \text{ kg/s}$$

Power output $= 508.6 \times 0.111 = \mathbf{56.45\ kW}$ $\stackrel{\text{Ans}}{\Longleftarrow}$

3.20 The mean effective pressure of an ideal Diesel cycle is 8 bar. If the initial pressure is 1.03 bar and the compression ratio is 12, determine the cut-off ratio and the air-standard efficiency. Assume ratio of specific heats for air to be 1.4.

Solution



Work output $= p_m \times V_s$

$= $ *Area 1234*

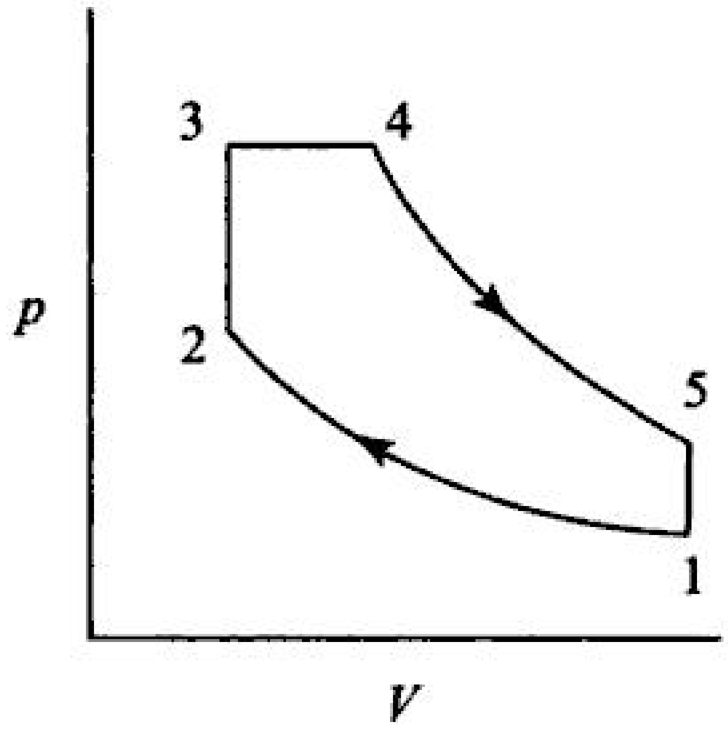
$= $ *Area under 2 – 3 + Area under 3 – 4 –*

Area under 2 – 1

$$= p_2 (V_3 - V_2) + \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1}$$

various salient points of the cycle and the cycle efficiency. The pressure and temperature of air at the commencement of compression are 1 bar and 100 °C respectively. Assume $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K for air.

Solution



$$\frac{V_s}{V_c} = r - 1 = 9$$

$$\gamma = \frac{C_p}{C_v} = \frac{1.004}{0.717} = 1.4$$

Consider the process 1 – 2

$$\frac{p_2}{p_1} = r^{\gamma} = 10^{1.4} = 25.12$$

$$p_2 = 25.12 \times 10^5 \text{ N/m}^2 = \mathbf{25.12 \text{ bar}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\frac{T_2}{T_1} = r^{(\gamma-1)} = 10^{0.4} = 2.512$$

$$T_2 = 2.512 \times 373 = 936.9 \text{ K} = \mathbf{663.9^\circ C} \stackrel{\text{Ans}}{\Longleftarrow}$$

Consider the process 2 – 3 and 3 – 4

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \frac{70}{25.12} = 2.787$$

$$T_3 = 2.787 \times 936.9 = 2611.1 \text{ K}$$

$$= \mathbf{2338^\circ\ C} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\frac{T_4}{T_5} = r_e^{(n-1)} = 5.3^{0.3} = 1.649$$

$$T_5 = \frac{1312.65}{1.649} = 796.03 \text{ K}$$

$$\frac{p_4}{p_5} = r_e^n = 5.3^{1.3} = 8.741$$

$$p_5 = \frac{p_4}{8.741} = \frac{23.2 \times 10^5}{8.741}$$

$$= 2.654 \times 10^5 \text{ N/m}^2$$

$$\text{Area } 12345 = \left[\frac{23.2 \times 1.509V_c - 2.654 \times 8V_c}{0.3} + 23.2 \times (1.509V_c - V_c) - \frac{14.93 \times V_c - 1 \times 8V_c}{0.3}\right] \times 10^5$$

$$= 34.63 \times V_c \times 10^5 \text{ N/m}^2$$

$$V_c = \frac{V_1}{8}$$

$$\text{Area } 12345 = p_m \times V_s = p_m \times 7 \times V_c$$

Therefore,

$$p_m = \frac{34.63}{7} = 4.95 \times 10^5 \text{ N/m}^2$$

$$= \mathbf{4.95 \text{ bar}}$$ Ans

$$\eta = \frac{w}{q_s}$$

$$v_1 = \frac{mRT_1}{p_1} = \frac{1 \times 287 \times 300}{1 \times 10^5}$$

$$= 0.861 \text{ m}^3\text{/kg}$$

$$w = 34.63 \times 10^5 \times \frac{v_1}{8}$$

$$= 34.63 \times 10^5 \times \frac{0.861}{8}$$

$$= 3.727 \times 10^5 \text{ J/kg} = 372.7 \text{ kJ/kg}$$

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$$\text{Power output} = \eta \times \text{total rate of heat added}$$

$$= 0.467 \times 66.5 = 31.1 \text{ kW}$$

$$\text{Power output/cylinder} = \frac{31.1}{4} = 7.76 \text{ kW}$$

Work done/cylinder/cycle

$$= \frac{7.76}{25} = 0.3104 \text{ kJ}$$

$$p_m = \frac{W}{V_s} = \frac{0.3104 \times 1000}{300 \times 10^{-6}}$$

$$= \mathbf{10.35 \times 10^5 \ N/m^2} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

As discussed in the text, this problem illustrates that for the same compression ratio and heat input Otto cycle is more efficient.

3.25 The compression ratio of an engine is 10 and the temperature and pressure at the start of compression is 37 °C and 1 bar. The compression and expansion processes are both isentropic and the heat is rejected at exhaust at constant volume. The amount of heat added during the cycle is 2730 kJ/kg. Determine the mean effective pressure and thermal efficiency of the cycle if (i) the maximum pressure is limited to 70 bar and heat is added at both constant volume and constant pressure and (ii) if all the heat is added at constant volume. In this case how much additional work per kg of charge would be obtained if it were possible to expand isentropically the exhaust gases to their original pressure of 1 bar. Assume that the charge has the same physical properties as that of air.

Solution

$$v_1 = \frac{V_1}{m} = \frac{RT_1}{p_1}$$

$$= \frac{287 \times 310}{1 \times 10^5} = 0.89 \text{ m}^3/\text{kg}$$

Consider the process 1-2

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = 310 \times 10^{0.4}$$

$$= 778.7 \text{ K}$$

3.18 Compare the Otto cycle for the same peak pressure and temperature. Illustrate the cycles on p-V and T-s diagrams.

3.19 Draw the p-V and T-s diagrams of a Dual cycle. Why this cycle is also called limited pressure or mixed cycle?

3.20 Derive the expressions for the efficiency and mean effective pressure of a Dual cycle.

3.21 Compare Otto, Diesel and Dual cycles for the

(i) *same compression ratio and heat input*

(ii) *same maximum pressure and heat input*

(iii) *same maximum pressure and temperature*

(iv) *same maximum pressure and work output*

3.22 Sketch the Lenoir cycle on p-V and T-s diagrams and obtain an expression for its air-standard efficiency.

3.23 Compare the Otto cycle and Atkinson cycle. Derive the expression for the efficiency of Atkinson cycle.

3.24 Derive an expression for the air-standard efficiency of the Joule cycle in terms of

(i) *compression ratio*

(ii) *pressure ratio.*

3.25 Where do the following cycles have applications

(i) *Otto cycle*

(ii) *Diesel cycle*

(iii) *Dual cycle*

(iv) *Stirling cycle*

(v) *Ericsson cycle*

(vi) *Atkinson cycle*

(vii) *Lenoir cycle*

(viii) *Joule cycle*

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ideal thermal efficiency. Assume $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K. *Ans:* 41.36%

3.21 The cycle of an internal combustion engine with isochoric heat supply is performed with the compression ratio equal to 8. Find heat supplied to the cycle and the useful work, if the removed heat is 500 kJ/kg and the working fluid is air.
Ans: (i) 1148.63 kJ/kg (ii) 648.63 kJ/kg

3.22 The initial parameters (at the beginning of compression) of the cycle of an internal combustion engine with isobaric heat supply are 0.1 MPa and 80 °C. The compression ratio is 16 and the heat supplied is 850 kJ/kg. Calculate the parameters at the characteristic points of the cycle and the thermal efficiency, if the working fluid is air.

Ans: (i) $p_2 = 48.5$ bar (ii) $p_3 = 48.5$ bar
(iii) $p_4 = 2.26$ bar (iv) $T_2 = 1070.1$ K
(v) $T_3 = 1916$ K (vi) $T_4 = 797.73$ K
(vii) $\eta_{th} = 62.5\%$

3.23 The pressure ratio $\lambda = 1.5$ in the process of isochoric heat supply for the cycle of an internal combustion engine with a mixed supply of heat = 1034 kJ/kg and the compression ratio = 13. Find the thermal efficiency and temperature at the characteristic points of the cycle if the initial parameters are 0.09 MPa and 70 °C and the working substance is air.

Ans: (i) $\eta_{th} = 57.5\%$ (ii) $T_2 = 956.9$ K
(iii) $T_3 = 1435.5$ K (iv) $T_4 = 2122.86$ K
(v) $T_5 = 890$ K

3.24 The parameters of the initial state of one kilogram of air in the cycle of an internal combustion engine are 0.095 MPa and 65 °C. The compression ratio is 11. Compare the values of the thermal efficiency for isobaric and isochoric heat supply in amounts of 800 kJ, assuming that $k = 1.4$.
Ans: $\eta_{t_p} = 55.7$ %, $\eta_{t_v} = 61.7\%$

3.25 Find the thermal efficiency of the cycle of an internal combustion engine with a mixed heat supply, if the minimum temperature of the cycle is 85 °C and the maximum temperature is 1700 K. The compression ratio is 15 and the pressure ratio in the process of heat supply is 1.3. The working fluid is air.
Ans: $\eta_{th} = 65.25$ %

close to the actual conditions than those used in the air-standard cycle analysis.

4.2 FUEL–AIR CYCLES AND THEIR SIGNIFICANCE

By air-standard cycle analysis, it is understood how the efficiency is improved by increasing the compression ratio. However, analysis cannot bring out the effect of air-fuel ratio on the thermal efficiency because the working medium was assumed to be air. In this chapter, the presence of fuel in the cylinder is taken into account and accordingly the working medium will be a mixture of fuel and air. By fuel-air cycle analysis it will be possible to bring out the effect of fuel-air ratio on thermal efficiency and also study how the peak pressures and temperatures during the cycle vary with respect to fuel-air ratio. In general, influence of many of the engine operating variables on the pressures and temperatures within the engine cylinder may be better understood by the examination of the fuel-air cycles. The fuel-air cycle analysis takes into account the following :

(i) *The actual composition of the cylinder gases :* The cylinder gases contains fuel, air, water vapour and residual gas. The fuel-air ratio changes during the operation of the engine which changes the relative amounts of CO_2, water vapour, etc.

(ii) *The variation in the specific heat with temperature* : Specific heats increase with temperature except for mono-atomic gases. Therefore, the value of γ also changes with temperature.

(iii) *The effect of dissociation* : The fuel and air do not completely combine chemically at high temperatures (above 1600 K) and this leads to the presence of CO, H_2, H and O_2 at equilibrium conditions.

(iv) *The variation in the number of molecules* : The number of molecules present after combustion depend upon fuel-air ratio and upon the pressure and temperature after the combustion.

Besides taking the above factors into consideration, the following assumptions are commonly made :

(i) There is no chemical change in either fuel or air prior to combustion.

(ii) Subsequent to combustion, the charge is always in chemical equilibrium.

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the value of C_v because of variable specific heat, which reduces the temperature as already explained.

The process, $2' \rightarrow 3'$ is heat addition with the variation in specific heat. From $3'$, if expansion takes place at constant specific heats, this would result in the process $3' \rightarrow 4''$ whereas actual expansion due to variable specific heat will result in $3' \rightarrow 4'$ and $4'$ is higher than $4''$. The magnitude in the difference between $4'$ and $4''$ is proportional to the reduction in the value of γ.

Consider the process $3'4''$

$$T_{4''} = T_{3'}\left(\frac{v_3}{v_4}\right)^{\gamma-1} \tag{4.7}$$

for the process $3'4'$

$$T_{4'} = T_{3'}\left(\frac{v_3}{v_4}\right)^{k-1} \tag{4.8}$$

Reduction in the value of k due to variable specific heat results in increase of temperature from $T_{4''}$ to $T_{4'}$.

4.5 DISSOCIATION

Dissociation process can be considered as the disintegration of combustion products at high temperature. Dissociation can also be looked as the reverse process to combustion. During dissociation the heat is absorbed whereas during combustion the heat is liberated. In IC engines, mainly dissociation of CO_2 into CO and O_2 occurs, whereas there is a very little dissociation of H_2O.

The dissociation of CO_2 into CO and O_2 starts commencing around 1000 °C and the reaction equation can be written as

$$CO_2 \rightleftharpoons 2CO + O_2 + \text{Heat}$$

Similarly, the dissociation of H_2O occurs at temperatures above 1300 °C and is written as

$$H_2O \rightleftharpoons 2H_2 + O_2 + \text{Heat}$$

The presence of CO and O_2 in the gases tends to prevent dissociation of CO_2; this is noticeable in a rich fuel mixture, which, by producing more CO, suppresses dissociation of CO_2. On the other hand, there is no dissociation in the burnt gases of a lean fuel-air mixture. This is mainly due to the fact that the temperature produced is too low for this phenomenon to occur. Hence, the maximum extent of dissociation occurs in the burnt gases of the chemically correct fuel-air mixture when the temperatures are expected to be high but decreases with the leaner and richer mixtures.

In case of internal combustion engines heat transfer to the cooling medium causes a reduction in the maximum temperature and pressure. As the temperature falls during the expansion stroke the separated constituents recombine; the heat absorbed during dissociation is thus again released, but it is too late in the stroke to recover entirely the lost power. A portion of this heat is carried away by the exhaust gases.

Figure 4.2 shows a typical curve that indicates the reduction in the temperature of the exhaust gas mixtures due to dissociation with respect to air-fuel ratio. With no dissociation maximum temperature is attained at chemically correct air-fuel ratio. With dissociation maximum temperature is obtained when mixture is slightly rich. Dissociation reduces the maximum temperature by about 300 °C even at the chemically correct air-fuel ratio. In the Fig.4.2, lean mixtures and rich mixtures are marked clearly.

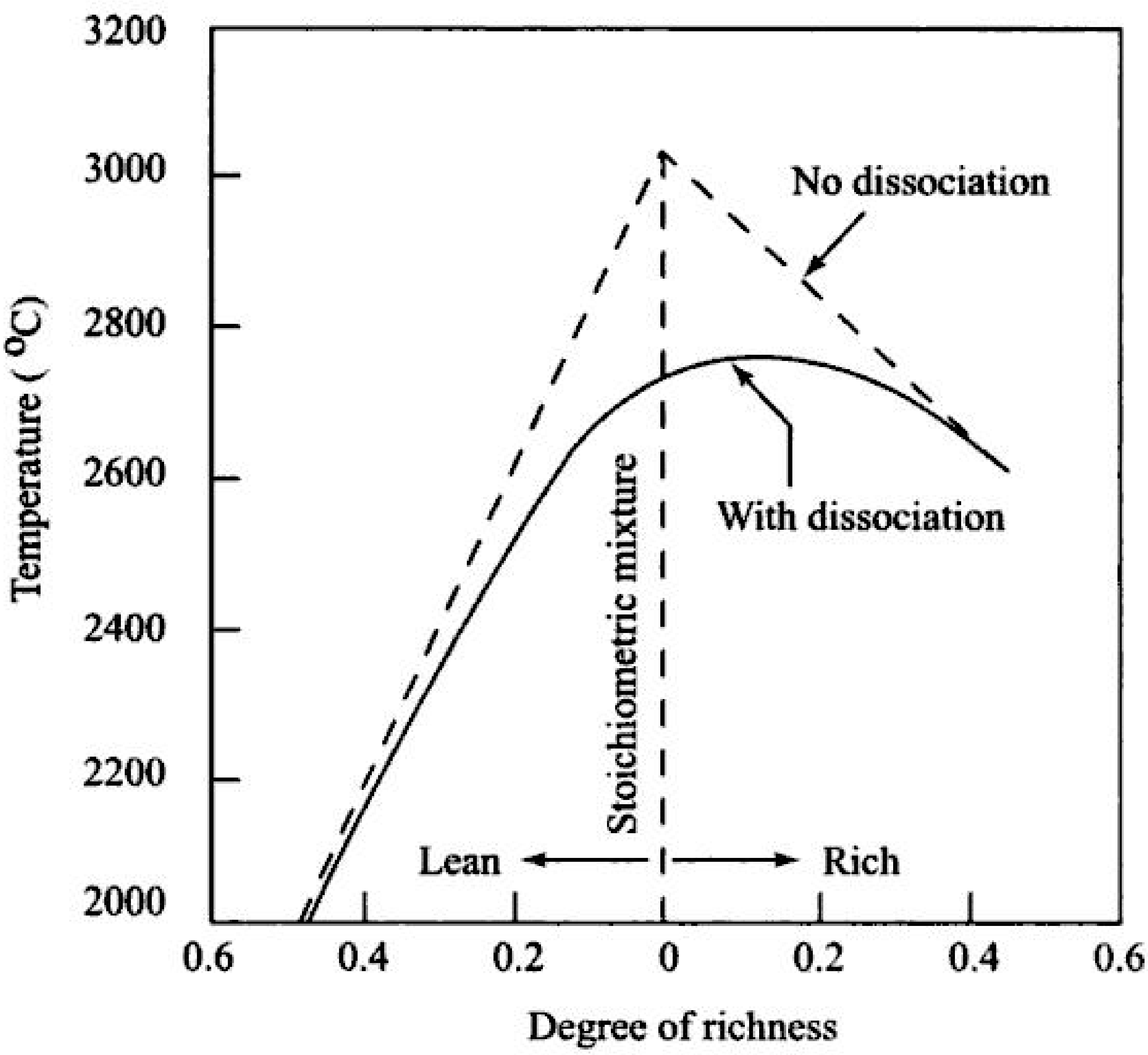

Fig. 4.2 Effect of Dissociation on Temperature

The effect of dissociation on output power is shown in Fig.4.3 for a typical four-stroke spark-ignition engine operating at constant speed. If there is no dissociation, the brake power output is maximum when the mixture ratio is stoichiometric. The shaded area between the brake power graphs shows the loss of power due to dissociation. When the mixture is quite lean there is no dissociation. As the air-fuel ratio decreases i.e., as the mixture becomes rich the maximum temperature rises and dissociation commences. The maximum dissociation occurs at chemically correct mixture strength. As the mixture becomes richer, dissociation effect tends to decline due to incomplete combustion.

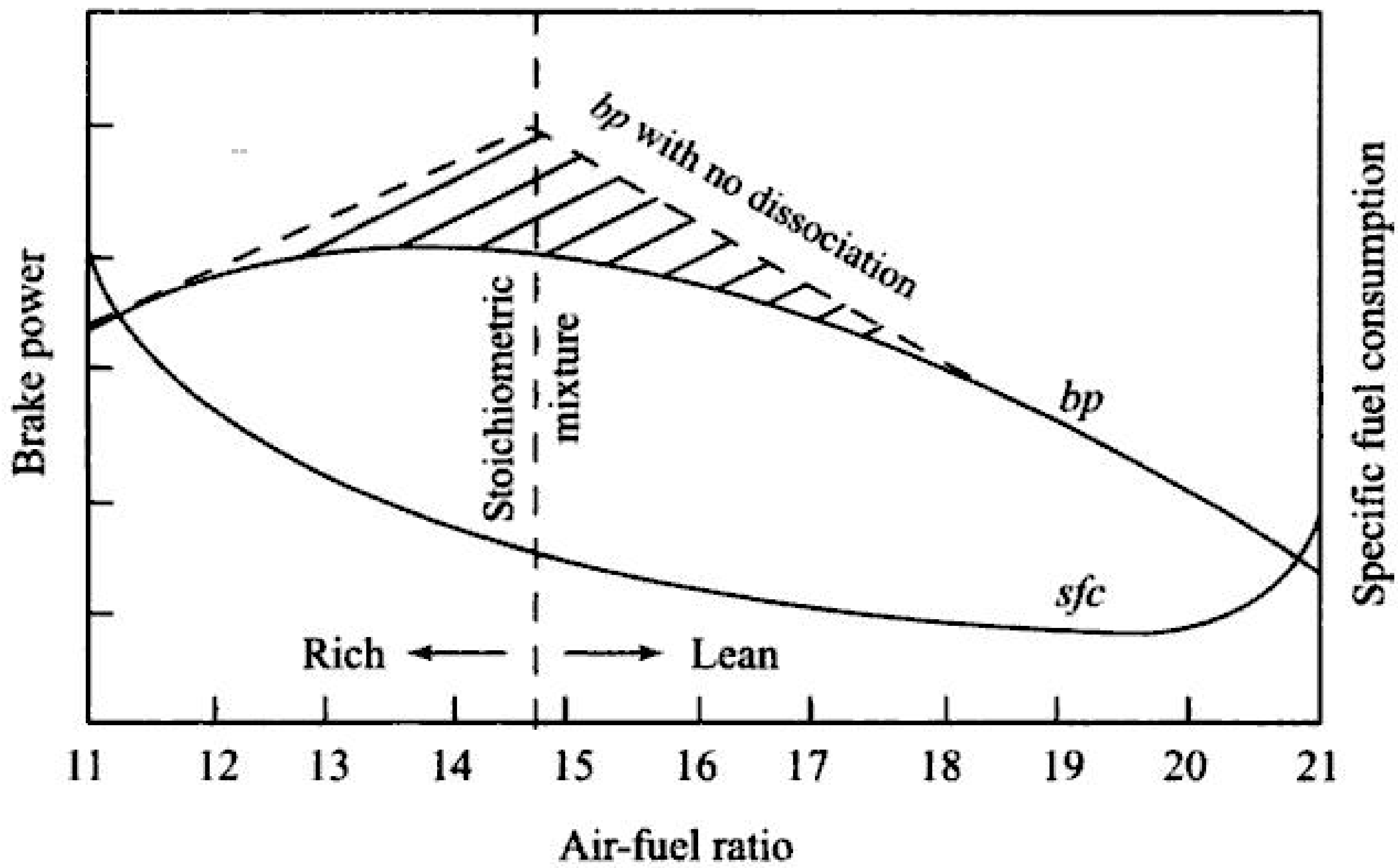

Fig. 4.3 Effect of Dissociation on Power

Dissociation effects are not so pronounced in a CI engine as in an SI engine. This is mainly due to

(i) the presence of a heterogeneous mixture and

(ii) excess air to ensure complete combustion.

Both these factors tend to reduce the peak gas temperature attained in the CI engine.

Figure 4.4 shows the effect of dissociation on p-V diagram of Otto cycle. Because of lower maximum temperature due to dissociation the maximum pressure is also reduced and the state after combustion will be represented by $3'$ instead of 3. If there was no reassociation due to fall of temperature during expansion the expansion process would be represented by $3' \to 4''$ but due to reassociation the expansion follows the path $3' \to 4'$. By comparing with the ideal expansion

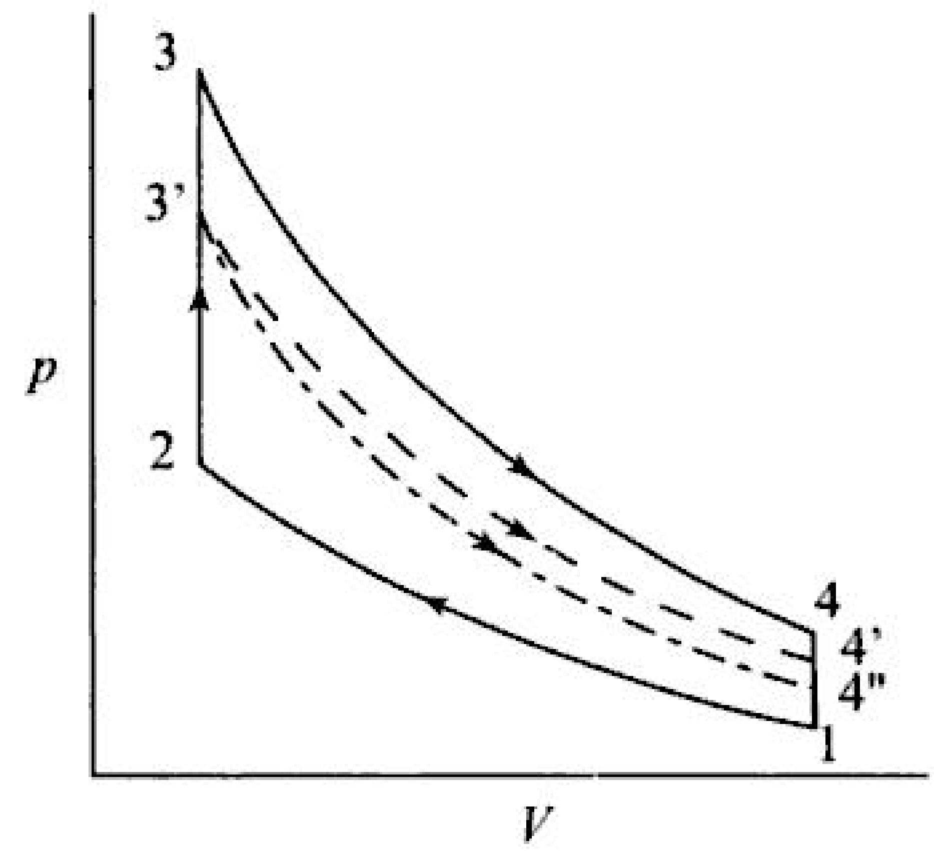

Fig. 4.4 Effect of Dissociation shown on a p-V Diagram

$3 \to 4$, it is observed that the effect of dissociation is to lower the temperature and consequently the pressure at the beginning of the expansion stroke. This causes a loss of power and also efficiency. Though during recombining the heat is given back it is too late to contribute a convincing positive increase in the output of the engine.

4.6 EFFECT OF NUMBER OF MOLES

As already mentioned the number of molecules present in the cylinder after combustion depends upon the fuel-air ratio, type and extend of reaction in the cylinder. According to the gas law

$$pV = N\overline{R}T$$

the pressure depends on the number of molecules or moles present. This has direct effect on the amount of work the cylinder gases can impart on the piston.

4.7 COMPARISON OF AIR–STANDARD AND FUEL–AIR CYCLES

In this section reasons for difference between air-standard cycles and fuel-air cycles is discussed. The magnitude of difference between the two cycles can be attributed to the following factors :

(i) character of the cycle (due to assumptions)

(ii) equivalence ratio (actual $F/A \div$ stoichiometric F/A)

(iii) chemical composition of the fuel

Figure 4.5 shows variation of efficiency with mixture strength of fuel-air cycle relative to that of air cycle showing the gain in efficiency as the mixture becomes leaner. It is seen from Fig.4.5 that the efficiency ratio (fuel-air cycle efficiency/air-standard cycle efficiency) increases as the mixture becomes leaner and leaner tending towards the air-standard cycle efficiency. It is to be noted that this, trend exists at all compression ratios.

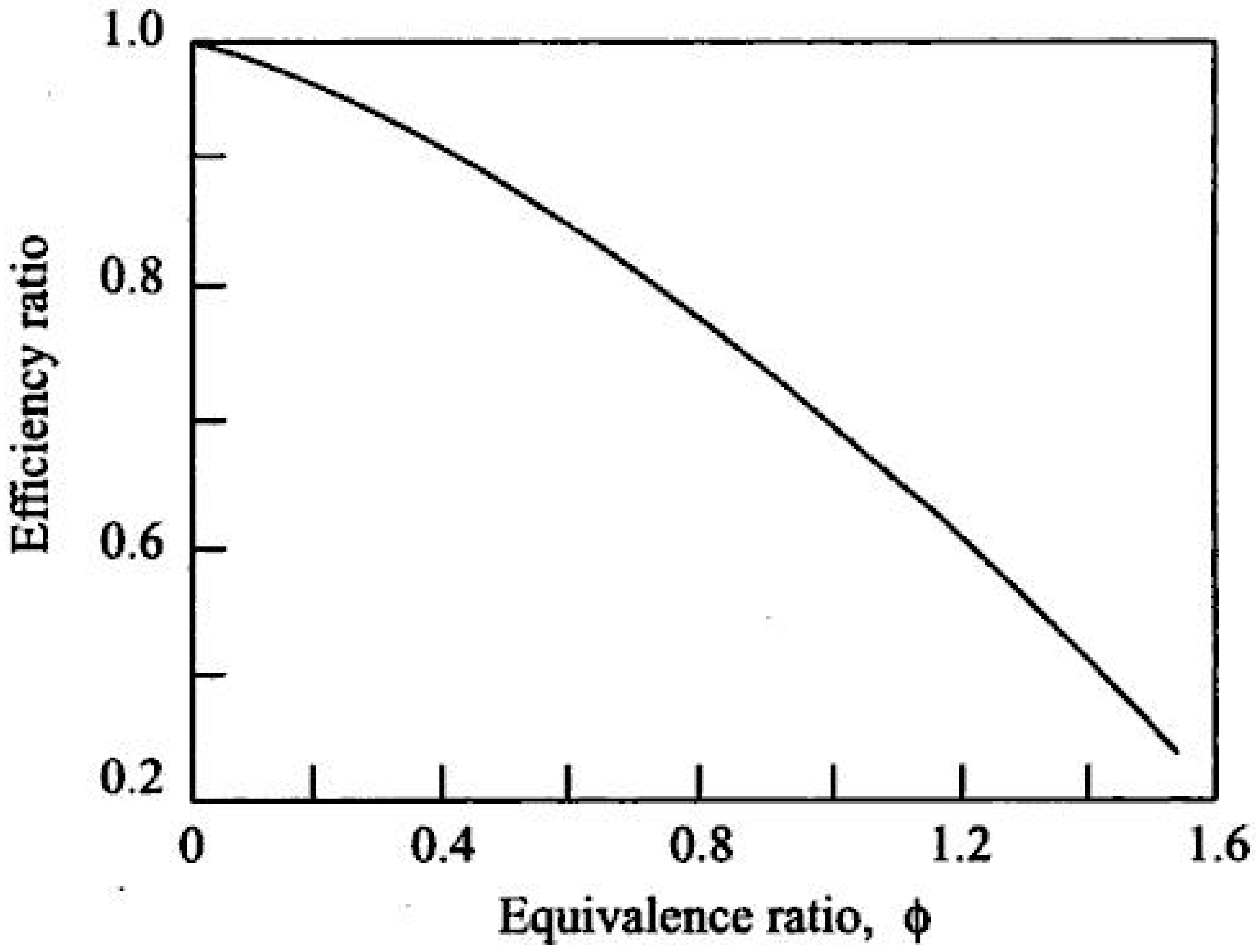

Fig. 4.5 Effect of Relative Fuel-Air ratio on Efficiency Ratio

At very low fuel-air ratio the mixture would tend to behave like a perfect gas with constant specific heat. Cycles with lean to very lean mixtures tend towards air-standard cycles. In such cycles the pressure and temperature rises. Some of the chemical reactions involved tend to be more complete as the pressure increases. These considerations apply to constant-volume as well as constant-pressure cycles.

The simple air-standard cycle analysis cannot predict the variation of thermal efficiency with mixture strength since air is assumed to be the working medium. However, fuel-air cycle analysis suggests that the thermal efficiency will deteriorate as the mixture supplied to an engine is enriched. This is explained by the increasing losses due to variable specific heats and dissociation as the mixture strength approaches chemically correct values. This is because, the gas temperature goes up after combustion as the mixture strength approaches chemically correct values. Enrichment beyond the chemically correct ratio will lead to incomplete combustion and loss in thermal efficiency. Therefore, it will appear that thermal efficiency will increase as the mixture is made leaner. However, beyond a certain leaning, the combustion becomes erratic with loss of efficiency. Thus the maximum efficiency is within the lean zone very near the stoichiometric ratio. This gives rise to combustion loop, as shown in Fig.4.6 which can be plotted for different mixture strengths for an engine running at constant speed and at a constant throttle setting. This loop gives an idea about the effect of mixture strength on the specific fuel consumption.

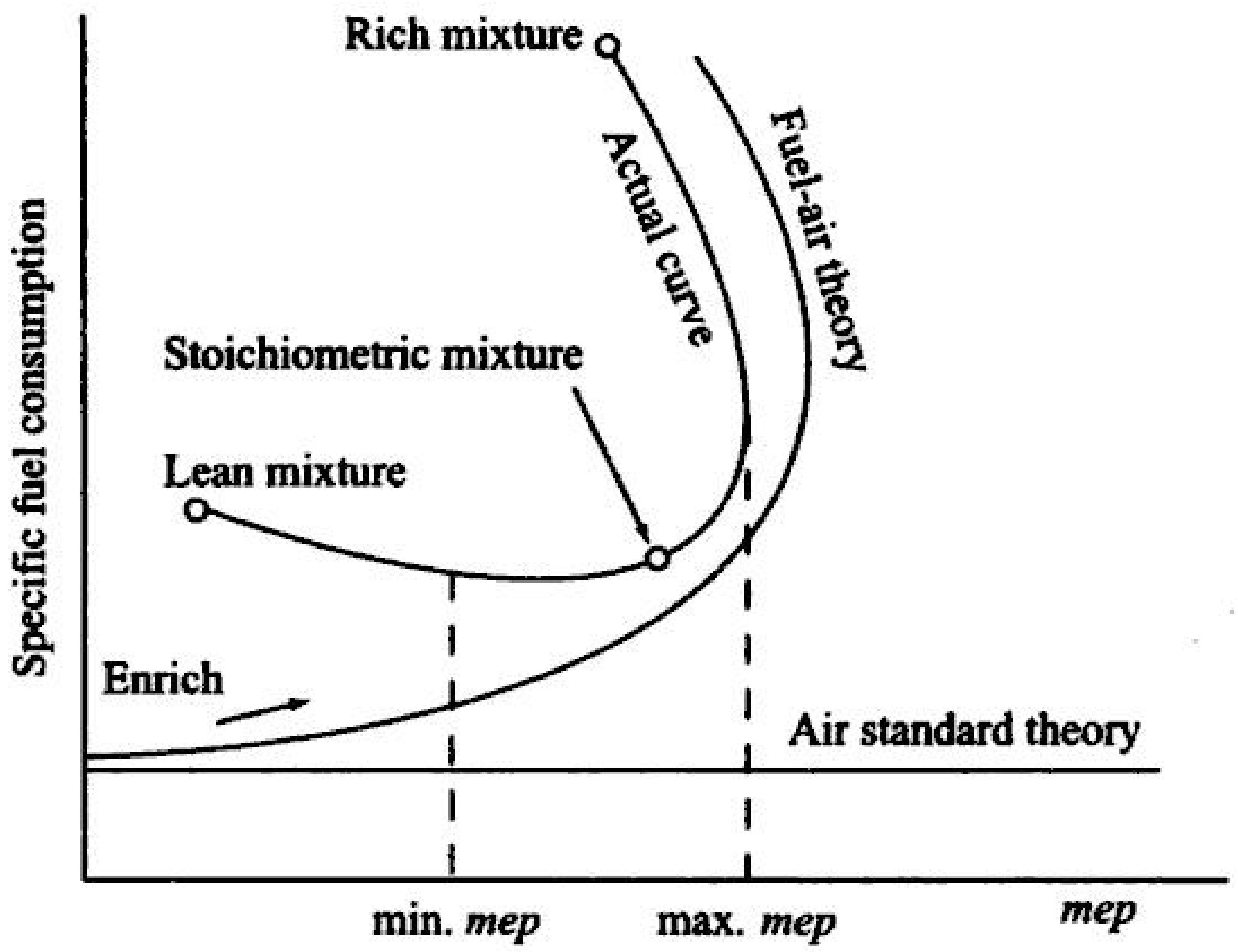

Fig. 4.6 Specific Fuel Consumption vs Mean Effective Pressure at Constant Speed and Constant Throttle Setting

4.8 EFFECT OF OPERATING VARIABLES

The effect of the common engine operating variables on the pressure and temperature within the engine cylinder is better understood by fuel-air cycle analysis. The details are discussed in the following sections.

4.8.1 Compression Ratio

The fuel-air cycle efficiency increases with the compression ratio in the same manner as the air-standard cycle efficiency, principally for the same reason (more scope of expansion work). This is shown in Fig.4.7.

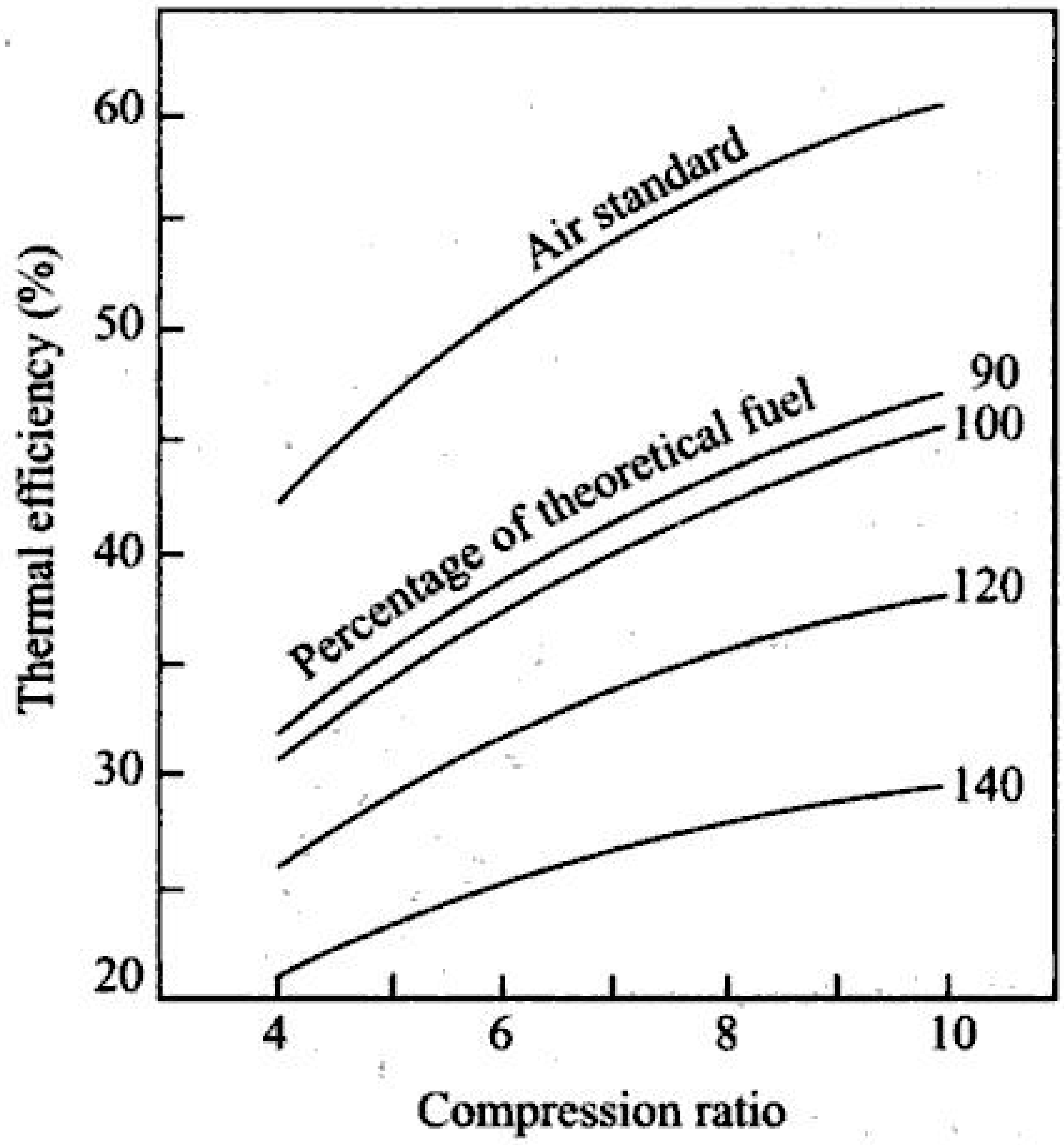

Fig. 4.7 Effect of Compression Ratio and Mixture Strength on Efficiency

The variation of indicated thermal efficiency with respect to the equivalence ratio for various compression ratios is given in Fig.4.8 The equivalence ratio, ϕ, is defined as ratio of actual fuel-air ratio to chemically correct fuel-air ratio on mass basis. The maximum pressure and maximum temperature increase with compression ratio since the temperature, T_2, and pressure, p_2, at the end of compression are higher. However, it can be noted from the experimental results (Fig.4.9) that the ratio of fuel-air cycle efficiency to air-standard efficiency is independent of the compression ratio for a given equivalence ratio for the constant-volume fuel-air cycle.

4.8.2 Fuel–Air Ratio

(i) *Efficiency :* As the mixture is made lean (less fuel) the temperature rise due to combustion will be lowered as a result of

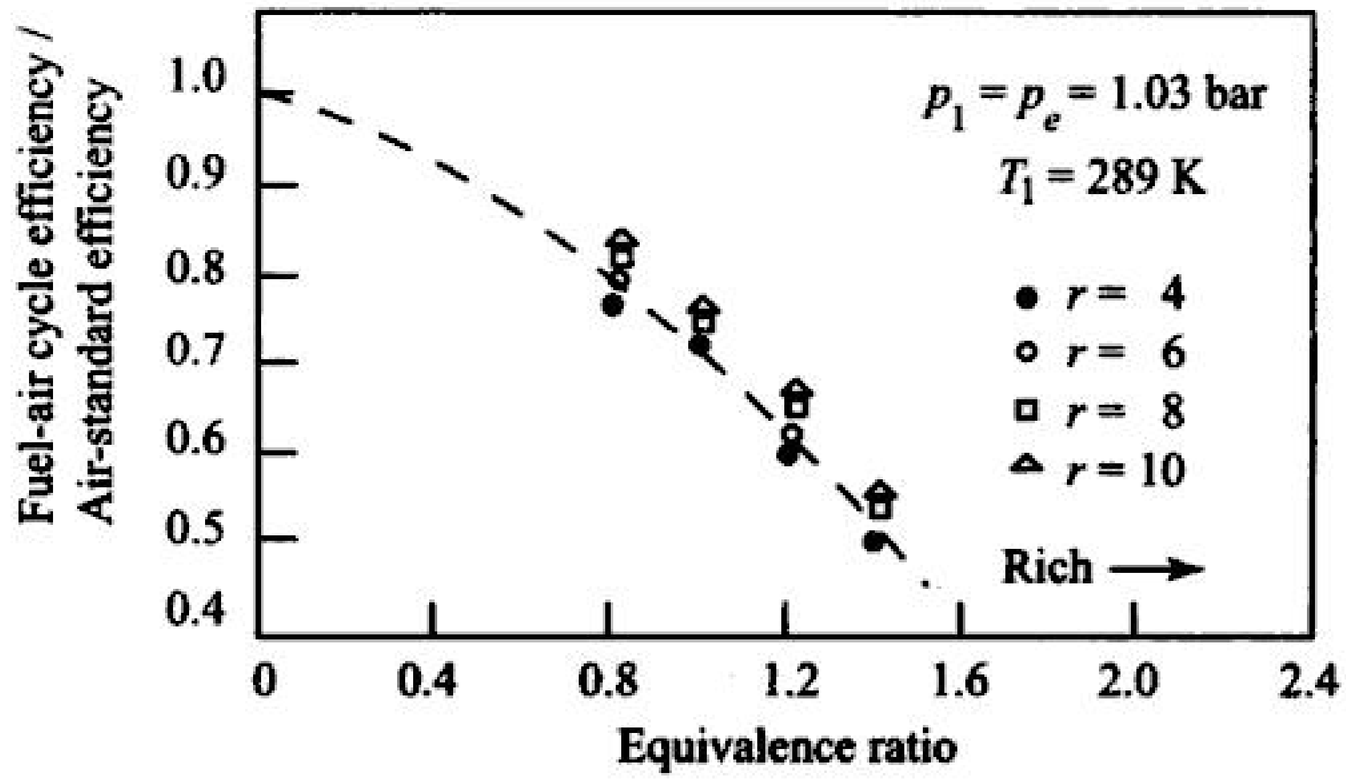

Fig. 4.9 Variation of Efficiency with Mixture Strength for a Constant Volume Fuel-Air Cycle

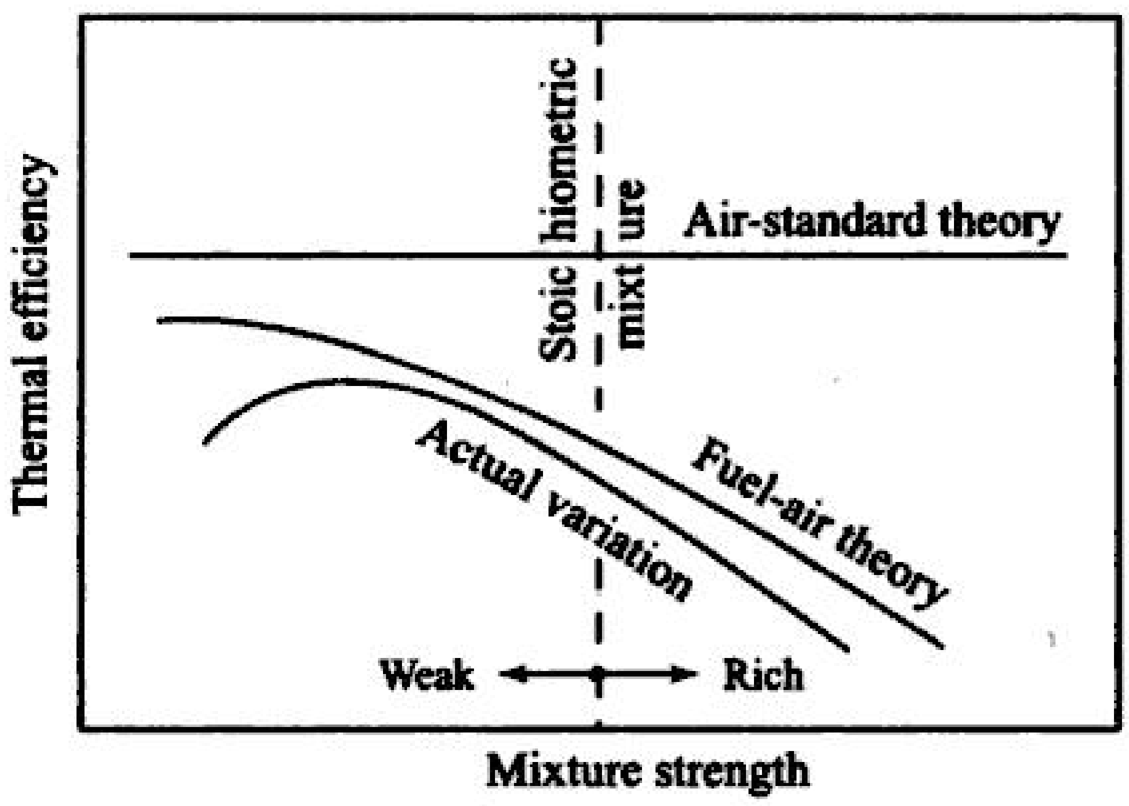

Fig. 4.10 Effect of Mixture Strength on Thermal Efficiency

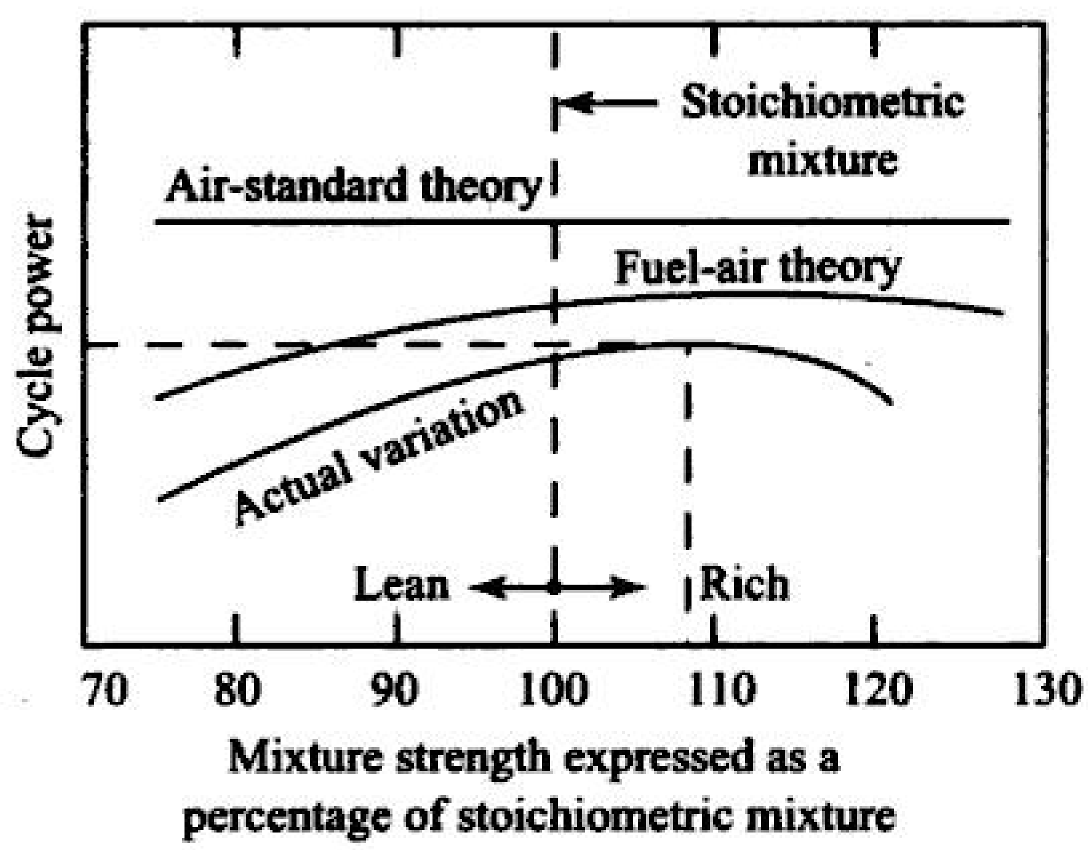

Fig. 4.11 Effect of Fuel-Air Ratio on Power

ture is slightly rich, i.e., around 6% or so ($F/A = 0.072$ or $A/F = 14 : 1$) as shown in Fig.4.12. At chemically correct ratio there is still some oxygen present at the point 3 (in the p-V diagram, refer Fig.4.1) because of chemical equilibrium effects a rich mixture will cause more fuel to combine with oxygen at that point thereby raising the temperature T_3. However, at richer mixtures increased formation of CO counters this effect.

(iv) *Maximum Pressure :* The pressure of a gas in a given space depends upon its temperature and the number of molecules The curve of p_3, therefore follows T_3, but because of the increasing number of molecules p_3 does not start to decrease until the mixture is some what richer than that for maximum T_3 (at $F/A = 0.083$ or A/F $12 : 1$), i.e. about 20 per cent rich (Fig.4.12).

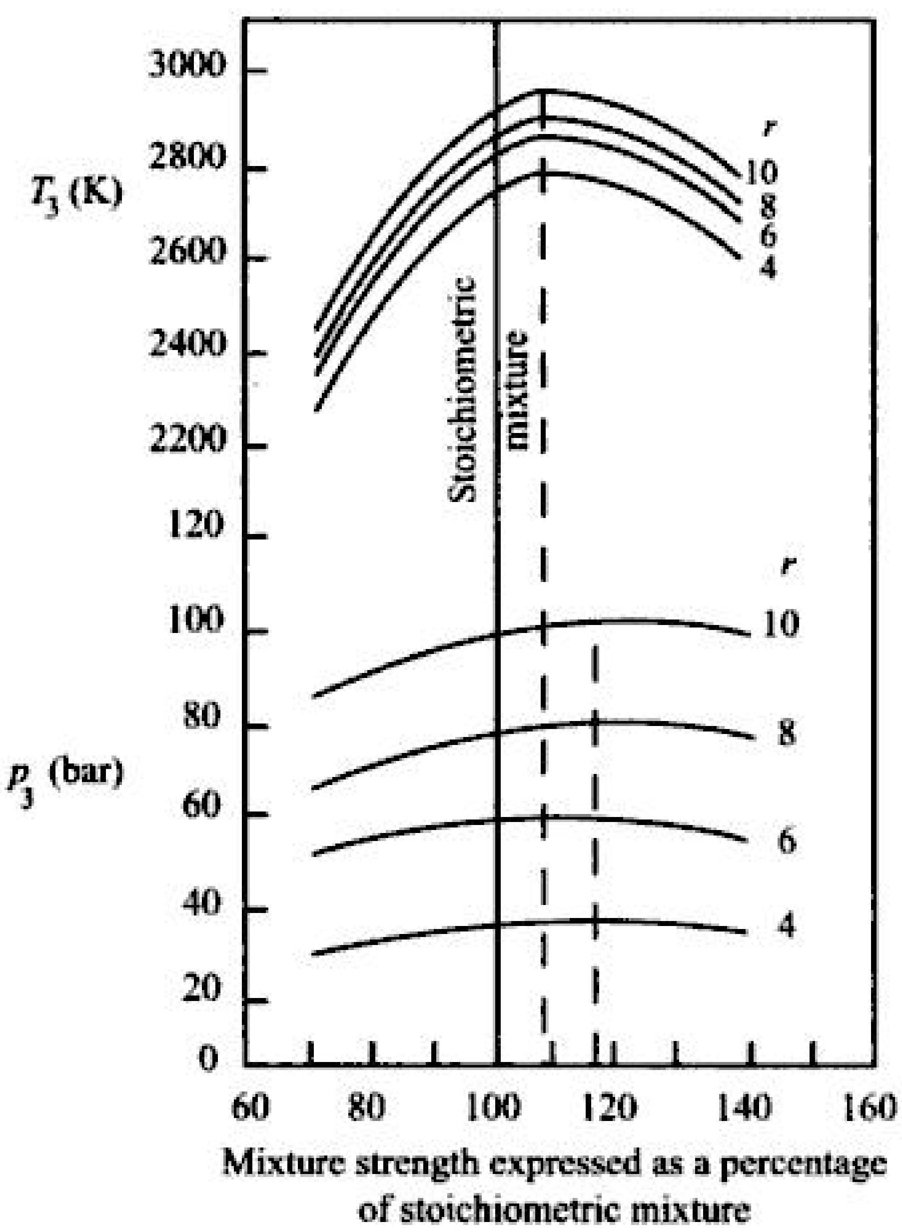

Fig. 4.12 Effect of Equivalence Ratio on T_3 and p_3

(v) *Exhaust Temperature :* The exhaust gas temperature, T_4 is maximum at the chemically correct mixture as shown in Fig.4.13. At this point the fuel and oxygen are completely used up, as the effect of chemical equilibrium is not significant. At lean mixtures, because of less fuel, T_3 is less and hence T_4 is less. At rich mixtures less sensible energy is developed and hence T_4

is less. That is, T_4 varies with fuel-air ratio in the same manner as T_3 except that maximum T_4 is at the chemically correct fuel-air ratio in place of slightly rich fuel-air ratio (6%) as in case of T_3. However, the behaviour of T_4 with compression ratio is different from that of T_3 as shown in Fig.4.13. Unlike T_3, the exhaust gas temperature, T_4 is lower at high compression ratios, because the increased expansion causes the gas to do more work on the piston leaving less heat to be rejected at the end of the stroke. The same effect is present in the case of air-cycle analysis also.

Table 4.1 Condition for Maximum Temperature and Pressure in a Constant Volume Fuel-Air Cycle

Variable	Maximum at	Reason
1. Temperature, T_3 (see Fig.4.12)	6% rich, $F/A = 0.072$; $A/F = 14:1, \phi = 1.06$	Because of chemical equilibrium some O_2 still present even at chemically correct F/A ratio. More fuel can be burnt. Limit is reached at 6% rich. If ¿ 6% rich CO formation.
2. Pressure, p_3 (see Fig.4.12)	20% rich, $F/A = 0.083$; $A/F = 12:1$	$pV = N\overline{R}T$. p depends on T and N
3. Temperature, T_4 (see Fig.4.13)	Chemically correct fuel-air ratio	No effect of chemical equilibrium due to low temperature and incomplete combustion at rich mixture.
4. Mean effective pressure (see Fig.4.14)	6% rich, $F/A = 0.0745$; $A/F = 13.5, \phi = 1.05$ to 1.1.	*mep* follows the trend of p_3 and p_4.

(vi) *Mean Effective Pressure* (*mep*) *:* The mean effective pressure increases with compression ratio. It follows the trend of p_3 and p_4 and hence it is maximum at a fuel-air ratio slightly richer than the chemically correct ratio as shown in Fig.4.14. Table 4.1 shows a summary of conditions which give maximum pressure and temperature in a constant-volume cycle assuming fuel-air cycle approximations.

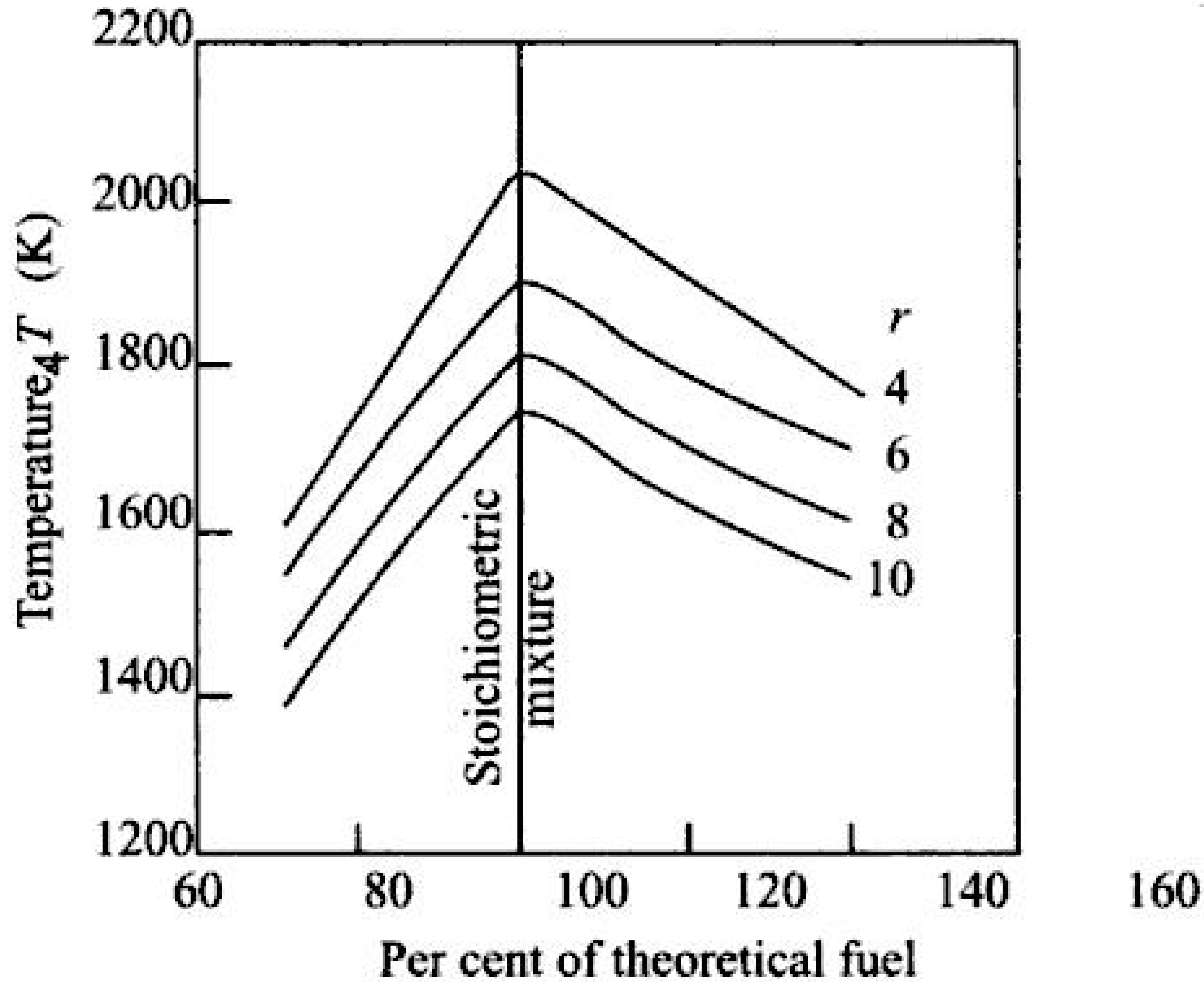

Fig. 4.13 Effect of Fuel-Air Ratio on the Exhaust Gas Temperature

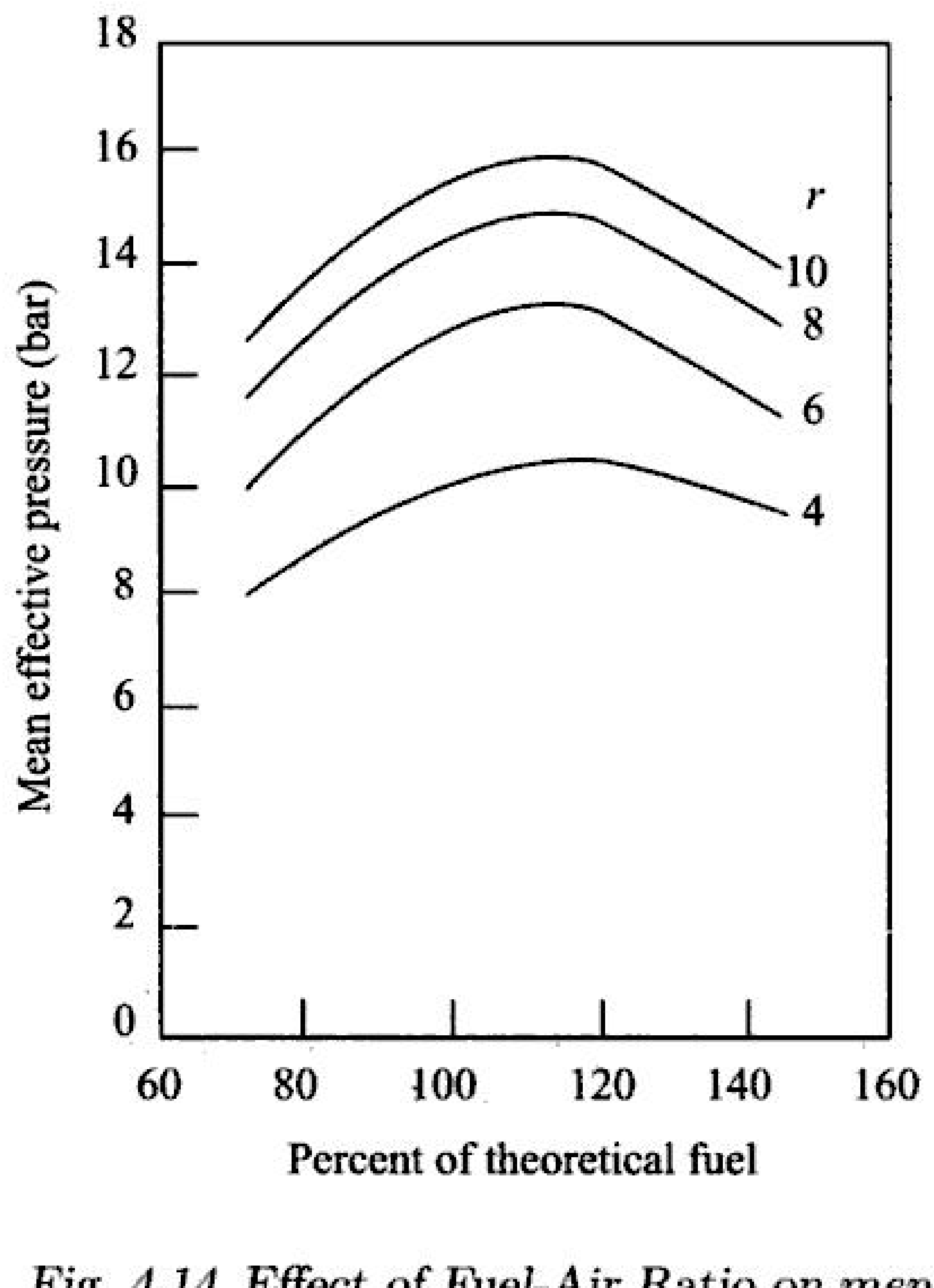

Fig. 4.14 Effect of Fuel-Air Ratio on mep

Worked out Examples

4.1 What will be the effect on the efficiency of an Otto cycle having a compression ratio of 8, if C_v increases by 1.6%?

Solution

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

$$C_p - C_v = R$$

$$\frac{C_p}{C_v} = \gamma$$

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

$$\eta = 1 - \left(\frac{1}{r}\right)^{R/C_v}$$

$$1 - \eta = r^{-R/C_v}$$

$$\ln(1-\eta) = \frac{-R}{C_v}\ln r$$

Differentiating

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

$$d\eta = -\frac{(1-\eta)R\ln r}{C_v^2}dC_v$$

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

Now,

$$\eta = 1 - \left(\frac{1}{8}\right)^{0.4} = 0.565 = 56.5\%$$

$$\frac{d\eta}{\eta} = -\frac{(1-0.565)\times(1.4-1)\times\ln 8}{0.565}\times\frac{1.6}{100}$$

$$= -1.025\% \qquad \underleftarrow{\text{Ans}}$$

4.2 What will be the effect on the efficiency of a diesel cycle having a compression ratio of 20 and a cut-off ratio is 5% of the swept volume, if the C_v increases by 1%. Take $C_v = 0.717$ and $R = 0.287$ kJ/kg K.

Solution

$$\eta_{Diesel} = 1 - \left(\frac{1}{r}\right)^{\gamma-1}\left(\frac{1}{\gamma}\,\frac{r_c^{\gamma}-1}{r_c-1}\right)$$

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

Taking logarithm

$$\ln(1-\eta) = -\ln\gamma + \ln(r_c^{\gamma}-1) - \ln(r_c-1) - (\gamma-1)\ln r$$

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

$$\gamma = \left(1+\frac{R}{C_v}\right)$$

Substituting this in the above equation

$$\ln(1-\eta) = -\ln\left(\frac{R}{C_v}+1\right) + \ln\left(r_c^{\left(\frac{R}{C_v}+1\right)}-1\right) - \ln(r_c-1) - \frac{R}{C_v}\ln r$$

Differentiating we get,

$$-\frac{d\eta}{\eta} = \frac{\frac{R}{C_v^2}dC_v}{\frac{R}{C_v}+1} - \frac{\frac{R}{C_v^2}\left(r_c^{\left(\frac{R}{C_v}+1\right)}\right)\ln r_c dC_v}{r_c^{\left(\frac{R}{C_v}+1\right)}-1} + \frac{R}{C_v^2}\ln r dC_v$$

$$\frac{d\eta}{\eta} = -\frac{dC_v}{C_v}\frac{R}{C_v}\left(\frac{1-\eta}{\eta}\right) \times \left(\frac{1}{\frac{R}{C_v}+1} + \ln r - \frac{r_c^{\frac{R}{C_v}+1}\ln(r_c)}{r_c^{\frac{R}{C_v}+1}-1}\right)$$

$$\frac{d\eta}{\eta} = -\frac{dC_v}{C_v}\left(\frac{1-\eta}{\eta}\right)(\gamma-1)\left[\frac{1}{\gamma} + \ln r - \frac{r_c^{\gamma}\ln(r_c)}{r_c^{\gamma}-1}\right]$$

$$\gamma = 1.4$$

$$\frac{V_1}{V_2} = r = 20$$

$$V_1 = 20V_2$$

$$V_s = 20V_2 - V_2 = 19V_2$$

$$V_3 = 0.05V_s + V_2$$

$$= (0.05 \times 19V_2) + V_2 = 1.95V_2$$

$$r_c = \frac{V_3}{V_2} = \frac{1.95V_2}{V_2} = 1.95$$

$$\gamma = 1.4$$

$$\eta = 1 - \frac{1}{\gamma}\frac{1}{r^{\gamma-1}}\frac{r_c^{\gamma}-1}{r_c-1}$$

$$= 1 - \frac{1}{1.4} \times \left(\frac{1}{20}\right)^{0.4} \times \frac{1.95^{1.4}-1}{1.95-1} = 0.649$$

$$\frac{d\eta}{\eta} = -0.01 \times \frac{1-0.649}{0.649}$$

$$\times 0.4 \times \left[\frac{1}{1.4} + \ln(20) - \frac{1.95^{1.4} \times \ln(1.95)}{1.95^{1.4}-1}\right]$$

$$= \mathbf{-0.565\%}$$ $\stackrel{\text{Ans}}{\Longleftarrow}$

4.3 A petrol engine having a compression ratio of 6 uses a fuel with calorific value of 42 MJ/kg. The air-fuel ratio is 15:1. Pressure and temperature at the start of the suction stroke is 1 bar and 57 °C respectively. Determine the maximum pressure in the cylinder if the index of compression is 1.3 and the specific heat at constant volume is given by $C_v = 0.678 + 0.00013$ T, where T is in Kelvin. Compare this value with that obtained when $C_v = 0.717$ kJ/kg K.

Solution Consider the process 1-2

$$p_2V_2^n = p_1V_1^n$$

$$p_2 = p_1\left(\frac{V_1}{V_2}\right)^n$$

$$= 1 \times 6^{1.3} = 10.27 \text{ bar}$$

$$\frac{V_3}{V_s} = \frac{2.496 \times V_2}{15 \times V_2} \times 100 = \mathbf{16.64\%} \quad \underleftarrow{\text{Ans}}$$

4.5 An oil engine, working on the dual combustion cycle, has a compression ratio of 13:1. The heat supplied per kg of air is 2000 kJ, half of which is supplied at constant volume and the other half at constant pressure. If the temperature and pressure at the beginning of compression are 100 °C and 1 bar respectively, find (i) the maximum pressure in the cycle and (ii) the percentage of stroke when cut-off occurs. Assume $\gamma = 1.4$, $R = 0.287$ kJ/kg K and $C_v = 0.709 + 0.000028T$ kJ/kg K.

Solution

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

$$p_2 = p_2 \left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times 10^5 \times 13^{1.4}$$

$$= 36.27 \times 10^5 \text{ N/m}^2$$

$$T_1 V_1^{(\gamma-1)} = T_2 V_2^{\gamma-1}$$

$$T_2 = T_1 r^{(\gamma-1)}$$

$$= 373 \times 13^{0.4} = 1040.6 \text{ K}$$

For unit mass :
Consider the process 2-3,

$$Q_{2-3} = \frac{1}{2} \times 2000 = 1000 \text{ kJ}$$

$$Q_{2-3} = m \int_2^3 (0.709 + 0.000028T)dT$$

$$1000 = 0.709 \times (T_3 - 1040.6) + \frac{0.000028}{2} \times \left(T_3^2 - 1040.6^2\right)$$

$$T_3 = 2362.2 \text{ K}$$

$$p_3 = p_2 \left(\frac{T_3}{T_2}\right) = 36.27 \times \left(\frac{2362.2}{1040.6}\right) \times 10^5$$

$$= \mathbf{82.34 \times 10^5 \text{ N/m}^2} \quad \underleftarrow{\text{Ans}}$$

Consider the process 3-4,

$$Q_{3-4} = \frac{1}{2} \times 1000 = 500 \text{ kJ}$$

$$C_p = C_v + R = 0.996 + 0.000028\, T$$

$$Q_{3-4} = m \int_3^4 dT$$

$$500 = \int_3^4 (0.996 + 0.000028) dT$$

$$= 0.996 \times (T_4 - 2362.2) + \frac{0.000028}{2} \times \left(T_4^2 - 2362.2^2\right)$$

On solving,

$$T_4 = 2830.04 \text{ K}$$

$$V_4 = V_3 \left(\frac{T_4}{T_3}\right) = \frac{2830.04}{2362.2} V_3$$

$$= 1.198\, V_3$$

$$\text{Stroke volume, } V_s = V_1 - V_3 = V_3(r-1) = 12\, V_3$$

$$\text{Cut-off \% of stroke} = \frac{V_4 - V_3}{V_s} \times 100$$

$$= \frac{V_4 - V_3}{12\, V_3} \times 100$$

$$= \frac{1.198 - 1}{12} \times 100 = \mathbf{1.65\%}$$ $\xLeftarrow{\text{Ans}}$

4.6 A petrol engine with a compression ratio of 7 used a mixture of iso-octane and hexane as fuel. The pressure and temperature at the beginning of the compression process is 1 bar and 55.22 °C respectively. If the fuel-air mixture is 19.05% rich and the maximum pressure developed is 115.26 bar then evaluate the composition of the mixture (in percentage weight). Take C_v = 0.717 kJ/kg K, $(CV)_{\text{hexane}}$ = 43 MJ/kg, $(CV)_{\text{iso-octane}}$ = 42 MJ/kg and $pV^{1.31}$ is constant for the expansion and compression processes.

Solution

Suppose every kmol of iso-octane is mixed with x kmols of hexane. The stoichiometric equation is

$$xC_6H_{14} + C_8H_{18} + y(0.21\ O_2 + 0.79\ N_2) \rightarrow$$

$$(8 + 6x)CO_2 + (9 + 7x)\ H_2O + (0.79y)N_2$$

Equating number of oxygen atoms on both sides

$$0.42y = 2 \times (8 + 6x) + (9 + 7x)$$

$$y = \frac{1}{0.42} \times (25 + 19x)$$

As the fuel-air mixture is 19.05% rich, for x moles of hexane and 1 mole of iso-octane, number of moles of air present

$$= \frac{y}{1.1905} = (50 + 38x)$$

The combustion equation may now be written as

$$xC_6H_{14} + C_8H_{18} + (50 + 38x)(0.21\ O_2 + 0.79\ N_2) \rightarrow$$

$$aCO_2 + b\ CO + (9 + 7x)\ H_2O + 0.79(50 + 38x)N_2$$

Equating number of carbon atoms

$$a + b = 6x + 8 \qquad (1)$$

Equating number of oxygen atoms

$$2a + b = 8.96x + 12 \qquad (2)$$

Solving Eqs.(1) and (2) we get

$$a = 2.96x + 4$$

$$b = 3.04x + 4$$

$$\frac{n_f}{n_i} = \left(\frac{56.5 + 43.02x}{51 + 39x}\right)$$

State 2:

$$p_2 = p_1 r^n = 12.796 \text{ bar}$$

$$T_2 = T_1 r^{n-1} = 600 \text{ K}$$

Now, consider (50 + 38 x) kmols of air

Molecular weight of C_6H_{14} = *86 kg/kmol and molecular weight of* C_8H_{18} = *114 kg/kmol. Therefore,*

$$\text{Heat added} = 86x \times 43 \times 10^3 + 114 \times 42 \times 10^3$$

$$= (3698x + 4788) \times 10^3 \text{ kJ}$$

Total weight

$$= 28.97 \times (50 + 38x) + (86x + 114)$$

$$= (1186.86x + 1562.5) \text{ kg}$$

$$Q_s = mC_v(T_3 - T_2)$$

$$(3698x + 4788) \times 10^3 = (1186.86x + 1562.5) \times 0.717 \times (T_3 - T_2)$$

$$T_3 = 600 + \frac{3698x + 4788}{0.851x + 1.120}$$

$$= \frac{4208.6x + 5460}{0.851x + 1.120}$$

$$p_3 = p_2\left(\frac{T_3}{T_2}\right)\left(\frac{n_f}{n_i}\right)$$

$$115.26 = \frac{12.796}{600} \times \left(\frac{4208.6x + 5460}{0.851x + 1.120}\right) \times \left(\frac{43.02x + 56.5}{39x + 51}\right)$$

Solving this we get $x = 0.1$.

Percentage weight of iso-octane

$$= \left(\frac{114}{114 + 86x}\right) \times 100\%$$

$$= \mathbf{93\%}$$ Ans

Percentage weight of hexane

$$= \left(\frac{86x}{114 + 86x}\right) \times 100\%$$

$$= \mathbf{7\%}$$ Ans

Review Questions

4.1 Mention the various simplified assumptions used in fuel-air cycle analysis.

4.2 What is the difference between air-standard cycle and fuel-air cycle analysis? Explain the significance of the fuel-air cycle.

4.3 Explain why the fuel-air cycle analysis is more suitable for analyzing through a computer rather than through hand calculations.

4.4 How do the specific heats vary with temperature? What is the physical explanation for this variation?

4.5 Explain with the help of a p-V diagram the loss due to variation of specific heats in an Otto cycle.

4.6 Show with the help of a p-V diagram for an Otto cycle, that the effect of dissociation is similar to that of variation of specific heats.

4.7 Explain by means of suitable graphs the effect of dissociation on maximum temperature and brake power. How does the presence of CO affect dissociation?

4.8 Explain the effect of change of number of molecules during combustion on maximum pressure in the Otto cycle.

4.9 Compare the air-standard cycle and fuel-air cycles based on

(i) character of the cycle

(ii) fuel-air ratio

(iii) chemical composition of the fuel

4.10 Is the effect of compression ratio on efficiency the same in fuel-air cycles also? Explain.

4.11 From the point of view of fuel-air cycle analysis how does fuel-air ratio affect efficiency, maximum power, temperature and pressure in a cycle.

4.12 How do exhaust temperature and mean effective pressure affect the engine performance? Explain.

Exercise

4.1 Find the percentage change in the efficiency of an Otto cycle having a compression ratio of 10, if C_v decreases by 2%.
Ans: 1.22%

4.2 Find the percentage increase in the efficiency of a Diesel cycle having a compression ratio of 16 and cut-off ratio is 10% of the swept volume, if C_v decreases by 2%. Take C_v = 0.717 and γ = 1.4.
Ans: 1.23%

4.3 The air-fuel ratio of a Diesel engine is 31:1. If the compression ratio is 15:1 and the temperature at the end of compression is 1000 K, find at what percentage of stroke is the combustion complete if the combustion begins at TDC and continuous at constant pressure. Calorific value of the fuel is 40000 kJ/kg. Assume the variable specific heat, $C_p = a + bT$, where $a = 1$ and $b = 0.28 \times 10^{-4}$.
Ans: 15.68%

4.4 An engine working on the Otto cycle, uses hexane (C_6H_{14}) as fuel. The engine works on chemically correct air-fuel ratio and the compression ratio is 8. Pressure and temperature at the beginning of compression are 1 bar and 77 °C respectively. If the calorific value of the fuel is 43000 kJ/kg and C_v = 0.717 kJ/kg K, find the maximum temperature and pressure of the cycle. Assume the compression follows the law $pV^{1.3} = c$.

Ans: (i) 4343.6 K (ii) 99.28 bar

4.5 Find the percentage change in efficiency of a dual cycle having compression ratio = 16 and cut-off ratio of 10% of swept volume and if C_v increases by 2%. Given $\frac{T_3}{T_2} = 1.67$.
Ans: 0.68%

4.6 It is estimated that for air operating in a given engine the γ decreases by 2% from its original value of 1.4. Find the change in efficiency. The pressure at the end of compression is 18 bar.

Ans: 4.5%

5

ACTUAL CYCLES AND THEIR ANALYSIS

5.1 INTRODUCTION

The actual cycles for IC engines differ from the fuel-air cycles and air-standard cycles in many respects. The actual cycle efficiency is much lower than the air-standard efficiency due to various losses occurring in the actual engine operation. The major losses are due to:

(i) Variation of specific heats with temperature

(ii) Dissociation of the combustion products

(iii) Progressive combustion

(iv) Incomplete combustion of fuel

(v) Heat transfer into the walls of the combustion chamber

(vi) Blowdown at the end of the exhaust process

(vii) Gas exchange process

An estimate of these losses can be made from previous experience and some simple tests on the engines and these estimates can be used in evaluating the performance of an engine.

5.2 COMPARISON OF AIR-STANDARD AND ACTUAL CYCLES

The actual cycles for internal combustion engines differ from air-standard cycles in many respects. These differences are mainly due to:

(i) The working substance being a mixture of air and fuel vapour or finely atomized liquid fuel in air combined with the products of combustion left from the previous cycle.

(ii) The change in chemical composition of the working substance.

(iii) The variation of specific heats with temperature.

(iv) The change in the composition, temperature and actual amount of fresh charge because of the residual gases.

(v) The progressive combustion rather than the instantaneous combustion.

(vi) The heat transfer to and from the working medium.

(vii) The substantial exhaust blowdown loss, i.e., loss of work on the expansion stroke due to early opening of the exhaust valve.

(viii) Gas leakage, fluid friction etc., in actual engines.

Points (i) to (iv), being related to fuel-air cycles have already been dealt in detail in Chapter 4. Remaining points viz. (v) to (viii) are in fact responsible for the difference between fuel-air cycles and actual cycles.

Most of the factors listed above tend to decrease the thermal efficiency and power output of the actual engines. On the other hand, the analysis of the cycles while taking these factors into account clearly indicates that the estimated thermal efficiencies are not very different from those of the actual cycles.

Out of all the above factors, major influence is exercised by

(i) *Time loss factor* i.e. loss due to time required for mixing of fuel and air and also for combustion.

(ii) *Heat loss factor* i.e. loss of heat from gases to cylinder walls.

(iii) *Exhaust blowdown factor* i.e. loss of work on the expansion stroke due to early opening of the exhaust valve.

These major losses which are not considered in the previous two chapters are discussed in the following sections.

5.3 TIME LOSS FACTOR

In air-standard cycles the heat addition is assumed to be an instantaneous process whereas in an actual cycle it is over a definite period of time. The time required for the combustion is such that under all circumstances some change in volume takes place while it is in progress. The crankshaft will usually turn about 30 to 40° between the initiation of the spark and the end of combustion. There will be a time loss during this period and is called time loss factor.

The consequence of the finite time of combustion is that the peak pressure will not occur when the volume is minimum i.e., when the piston is at TDC; but will occur some time after TDC. The pressure, therefore, rises in the first part of the working stroke from b to c as shown in Fig.5.1. The point 3 represents the state of gases had the combustion been instantaneous and an additional amount of work equal to area shown hatched would have been done. This loss of work reduces the efficiency and is called *time loss due to progressive combustion* or merely *time losses.*

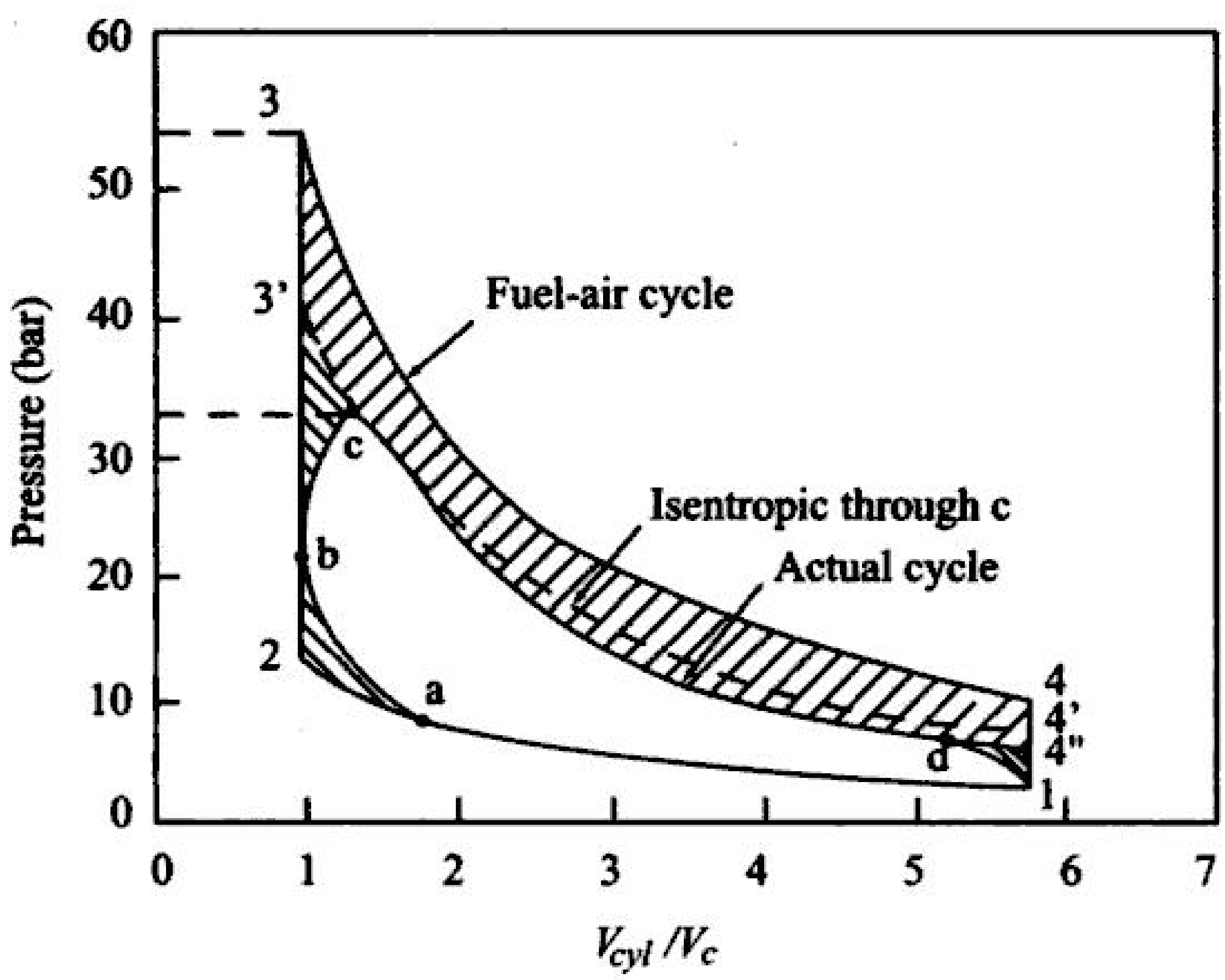

Fig. 5.1 The Effect of Time Losses shown on p-V Diagram

The time taken for the burning depends upon the flame velocity which in turn depends upon the type of fuel and the fuel-air ratio and also on the shape and size of the combustion chamber. Further, the distance from the point of ignition to the opposite side of the combustion space also plays an important role.

In order that the peak pressure is not reached too late in the expansion stroke, the time at which the combustion starts is varied by varying the spark timing or spark advance. Figures 5.2 and 5.3 show the effect of spark timing on *p-V* diagram from a typical trial. With spark at *TDC* (Fig.5.2) the peak pressure is low due to the expansion of gases. If the spark is advanced to achieve complete combustion close to *TDC* (Fig.5.3) additional work is required to compress the burning gases.

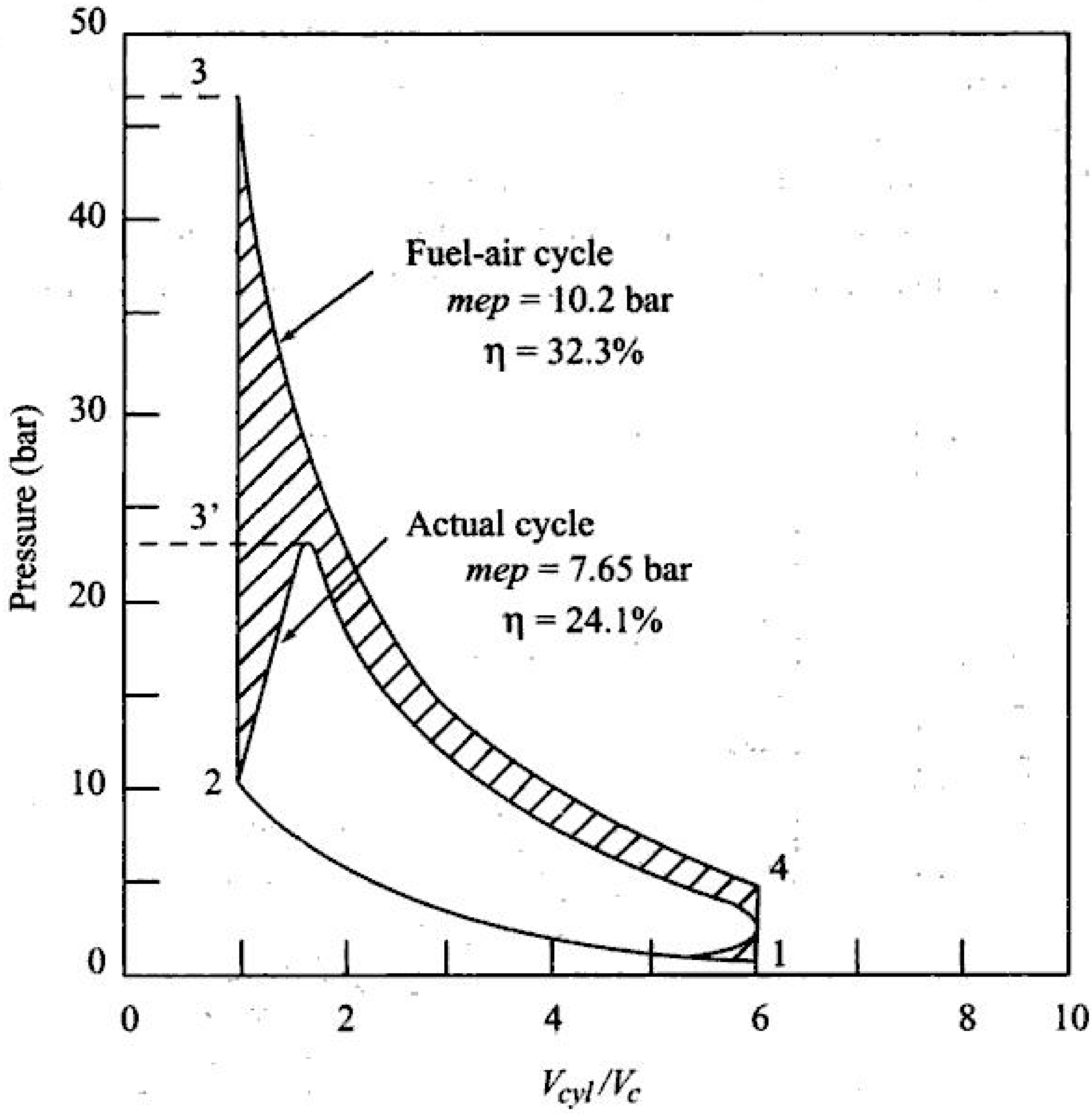

Fig. 5.2 Spark at TDC, Advance 0°

This represents a direct loss. In either case, viz., with or without spark advance the work area is less and the power output and efficiency are lowered. Therefore, a moderate or optimum spark advance (Fig.5.4) is the best compromise resulting in minimum losses on both the compression and expansion strokes.

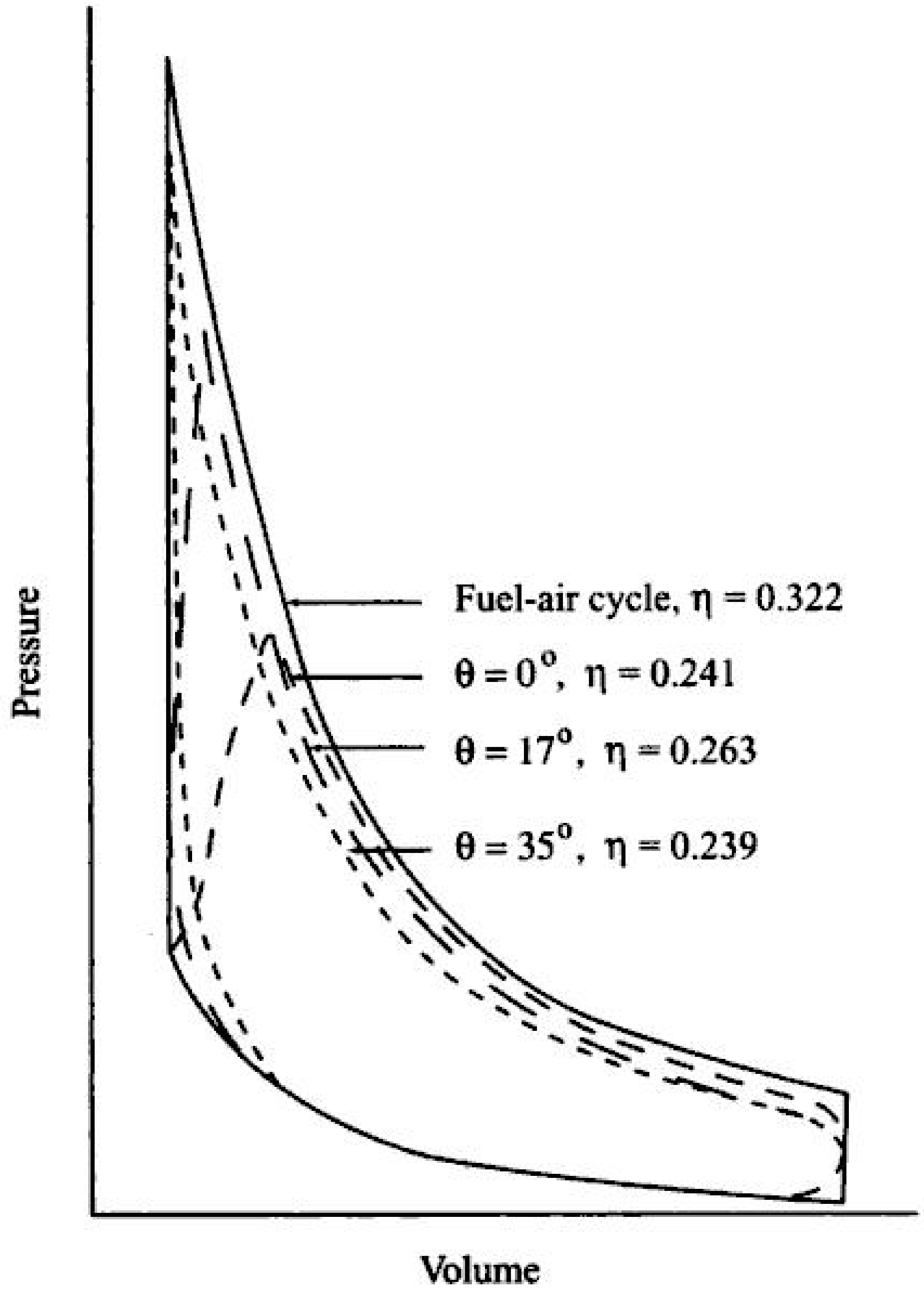

Fig. 5.5 p-V Diagram showing Power Loss due to Ignition Advance

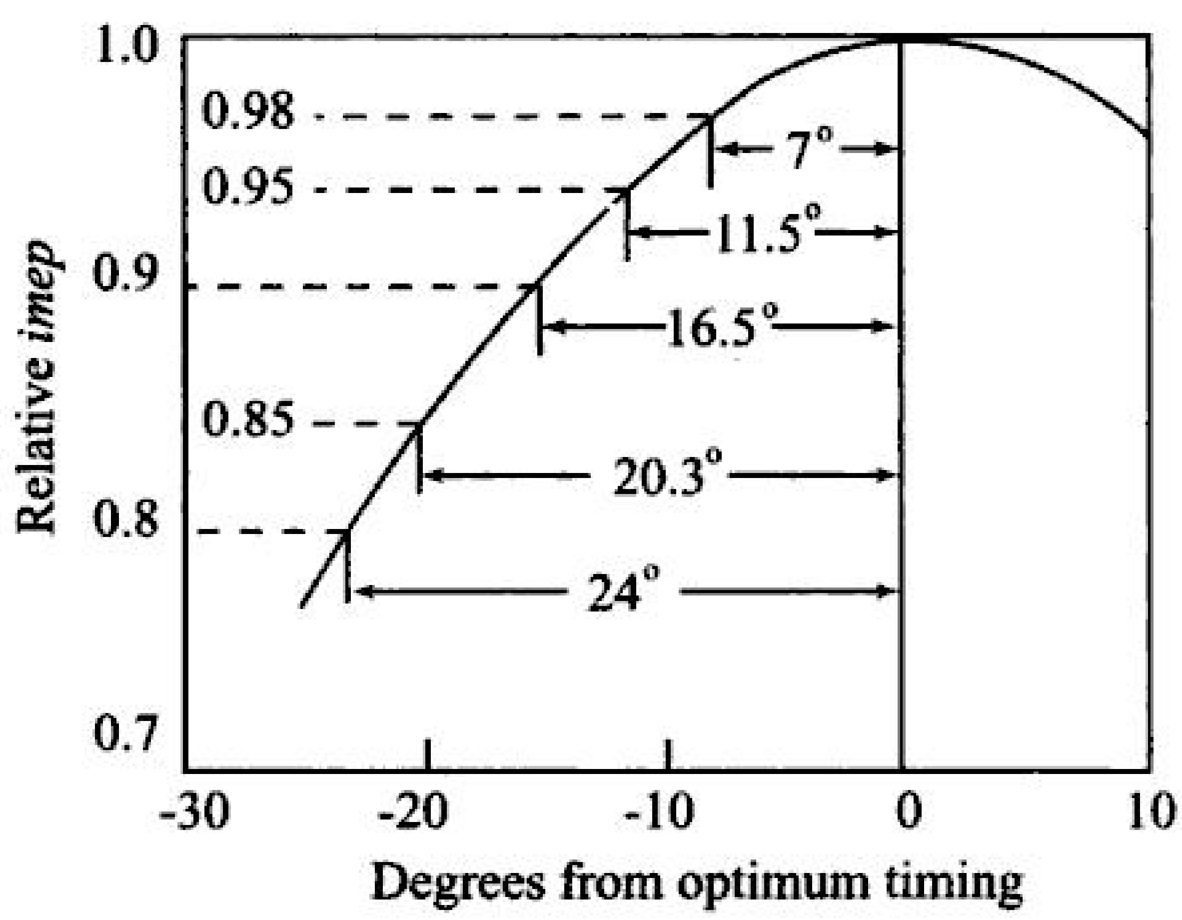

Fig. 5.6 Power Loss due to Ignition Advance

of the volume of air actually inducted at ambient condition to swept volume. However, it may also be defined on mass basis as the ratio of the actual mass of air drawn into the engine during a given period of time to the theoretical mass which should have been drawn in during that same period of time, based upon the total piston displacement of the engine, and the temperature and pressure of the surrounding atmosphere.

The above definition is applicable only to the naturally aspirated engine. In the case of the supercharged engine, however, the theoretical mass of air should be calculated at the conditions of pressure and temperature prevailing in the intake manifold. The volumetric efficiency is affected by many variables, some of the important ones are:

(i) *The density of the fresh charge :* As the fresh charge arrives in the hot cylinder, heat is transferred to it from the hot chamber walls and the hot residual exhaust gases, raising its temperature. This results in a decrease in the mass of fresh charge admitted and a reduction in volumetric efficiency. The volumetric efficiency is increased by low temperatures (provided there are no heat transfer effects) and high pressure of the fresh charge, since density is thereby increased, and more mass of charge can be inducted into a given volume.

(ii) *The exhaust gas in the clearance volume :* As the piston moves from TDC to BDC on the intake stroke, these products tend to expand and occupy a portion of the piston displacement greater than the clearance volume, thus reducing the space available to the incoming charge. In addition, these exhaust products tend to raise the temperature of the fresh charge, thereby decreasing its density and further reducing volumetric efficiency.

(iii) *The design of the intake and exhaust manifolds :* The exhaust manifold should be so designed as to enable the exhaust products to escape readily, while the intake manifold should be designed so as to bring in the maximum possible fresh charge. This implies minimum restriction is offered to the fresh charge flowing into the cylinder, as well as to the exhaust products being forced out.

(iv) *The timing of the intake and exhaust valves :* Valve timing is the regulation of the points in the cycle at which the valves are set to open and close. Since, the valves require a finite period of time to open or close for smooth operation, a slight "lead" time is necessary for proper opening and closing. The design of the valve operating cam provides for the smooth transition from one position to the other, while the cam setting determines the timing of the valve.

The effect of the *intake valve* timing on the engine air capacity is indicated by its effect on the air inducted per cylinder per cycle, i.e., the mass of air taken into one cylinder during one suction stroke. Figure 5.10 shows representative intake valve timing for both a low speed and high speed SI engine. In order to understand the effect of the intake valve timing on the charge inducted per cylinder per cycle, it is desirable to follow through the intake process, referring to the Fig.5.10.

While the intake valve should open, theoretically, at TDC, almost all SI engines employ an intake valve opening of a few degrees before TDC on the exhaust stroke. This is to ensure that the valve will be fully open and the fresh charge starts to flow into the cylinder as soon as the piston reaches TDC. In Fig.5.10, the intake valve starts to open 10° before TDC. It may be noted from Fig.5.10 that for a low speed engine, the intake valve closes 10° after BDC, and for a high speed engine, 60° after BDC.

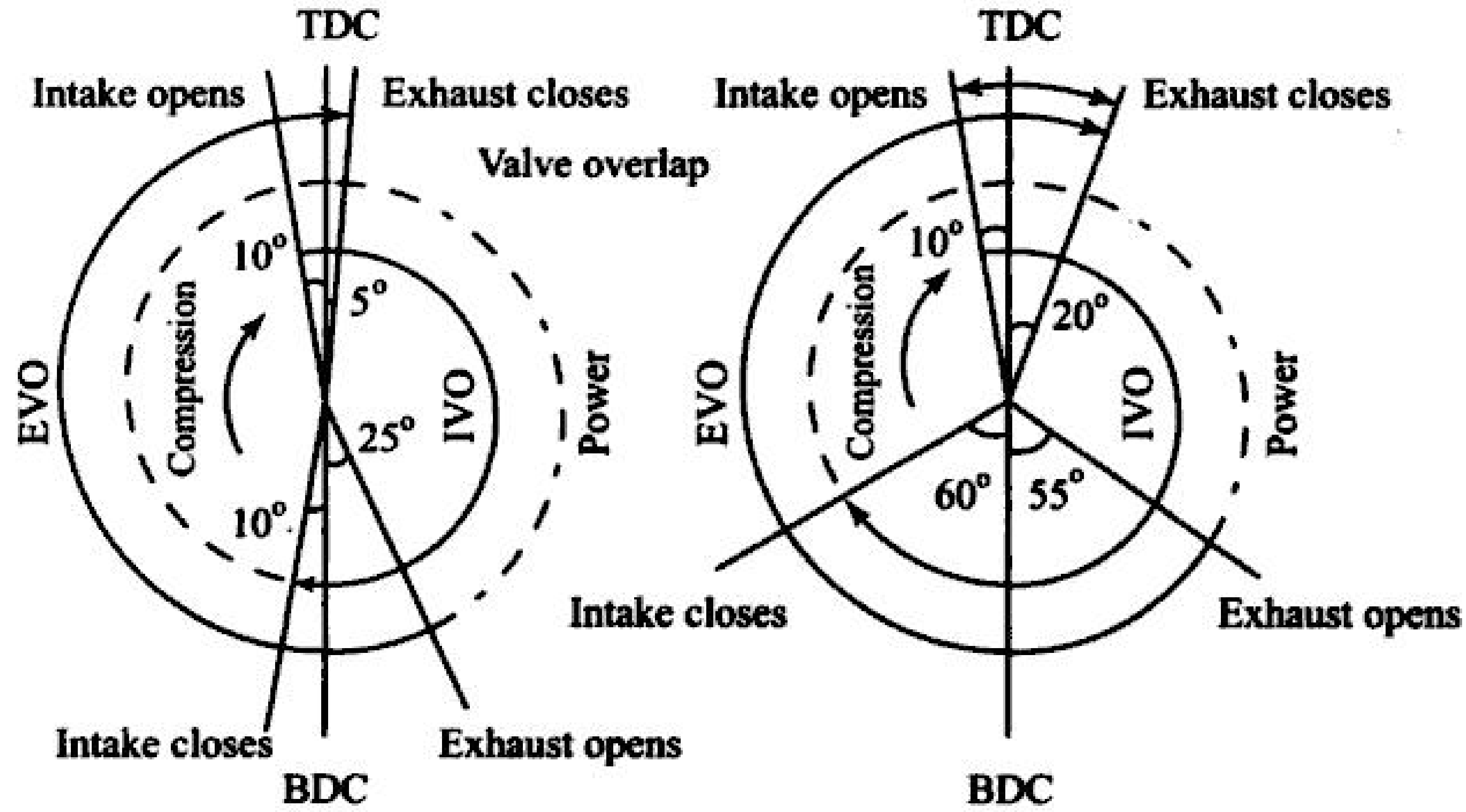

Fig. 5.10 Valve Timing Diagram of Four-Stroke Engines

As the piston descends on the intake stroke, the fresh charge is drawn in through the intake port and valve. When the piston reaches BDC and starts to ascend on the compression stroke, the inertia of the incoming fresh charge tends to cause it to continue to

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temperature, with exposure to sunlight and also on contact with metals. Gasoline specifications therefore limit both the gum content of the fuel and its tendency to form gum during storage.

(viii) *Sulphur Content :* Hydrocarbon fuels may contain free sulphur, hydrogen sulphide and other sulphur compounds which are objectionable for several reasons. The sulphur is a corrosive element of the fuel that can corrode fuel lines, carburettors and injection pumps and it will unite with oxygen to form sulphur dioxide that, in the presence of water at low temperatures, may form sulphurous acid. Since sulphur has a low ignition temperature, the presence of sulphur can reduce the self-ignition temperature, then promoting knock in the SI engine.

6.5.2 CI Engine Fuels

(i) *Knock Characteristics :* Knock in the CI engine occurs because of an ignition lag in the combustion of the fuel between the time of injection and the time of actual burning. As the ignition lag increases, the amount of fuel accumulated in the combustion chamber increases and when combustion actually takes place, abnormal amount of energy is suddenly released causing an excessive rate of pressure rise which results in an audible knock. Hence, a good CI engine fuel should have a short ignition lag and will ignite more readily. Furthermore, ignition lag affects the starting, warm up, and leads to the production of exhaust smoke in CI engines. The present day measure in the cetane rating, the best fuel in general, will have a cetane rating sufficiently high to avoid objectionable knock.

(ii) *Volatility :* The fuel should be sufficiently volatile in the operating range of temperature to produce good mixing and combustion. Figure 6.3 is a representative distillation curve of a typical diesel fuel.

(iii) *Starting Characteristics :* The fuel should help in starting the engine easily. This requirement demands high enough volatility to form a combustible mixture readily and a high cetane rating in order that the self-ignition temperature is low.

(iv) *Smoking and Odour :* The fuel should not promote either smoke or odour in the engine exhaust. Generally, good volatility is the

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90% gasoline). The data of these tests which include performance and emission levels are compared to pure gasoline (M0) and pure methanol (M100). Some smart flexible fuel (or variable-fuel) engines are capable of using any random mixture combination of methanol and gasoline ranging from pure methanol to pure gasoline. Two fuel tanks are used and various flow rates of the two fuels can be pumped to the engine, passing through a mixing chamber. Using information from sensors in the intake and exhaust, the electronic monitoring system (EMS) adjusts to the proper air-fuel ratio, ignition timing, injection timing, and valve timing (where possible) for the fuel mixture being used.

One problem with gasoline-alcohol mixtures as a fuel is the tendency for alcohol to combine with any water present. When this happens the alcohol separates locally from the gasoline, resulting in a non-homogeneous mixture. This causes the engine to run erratically due to the large air-fuel ratio differences between the two fuels.

Methanol can be obtained from many sources, both fossil and renewable. These include coal, petroleum, natural gas, biomass, wood, landfills, and even the ocean. However, any source that requires extensive manufacturing or processing raises the price of the fuel.

Emissions from an engine using M10 fuel are about the same as those using gasoline. The advantage (and disadvantage) of using this fuel is mainly the 10% decrease in gasoline use. With M85 fuel there is a measurable decrease in HC and CO exhaust emissions. However, there is an increase in NO_x and a large ($\approx 500\%$) increase in formaldehyde formation.

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

7.4.3 Ethanol

Ethanol has been used as automobile fuel for many years in various regions of the world. Brazil is probably the leading user, where in the early 1990s. About 5 million vehicles operated on fuels that were 93% ethanol. For a number of years gasohol (gasoline + alcohol) has been available at service stations in the United States. Gasohol is a mixture of 90% gasoline and 10% ethanol. As with methanol, the development of systems using mixtures of gasoline and ethanol

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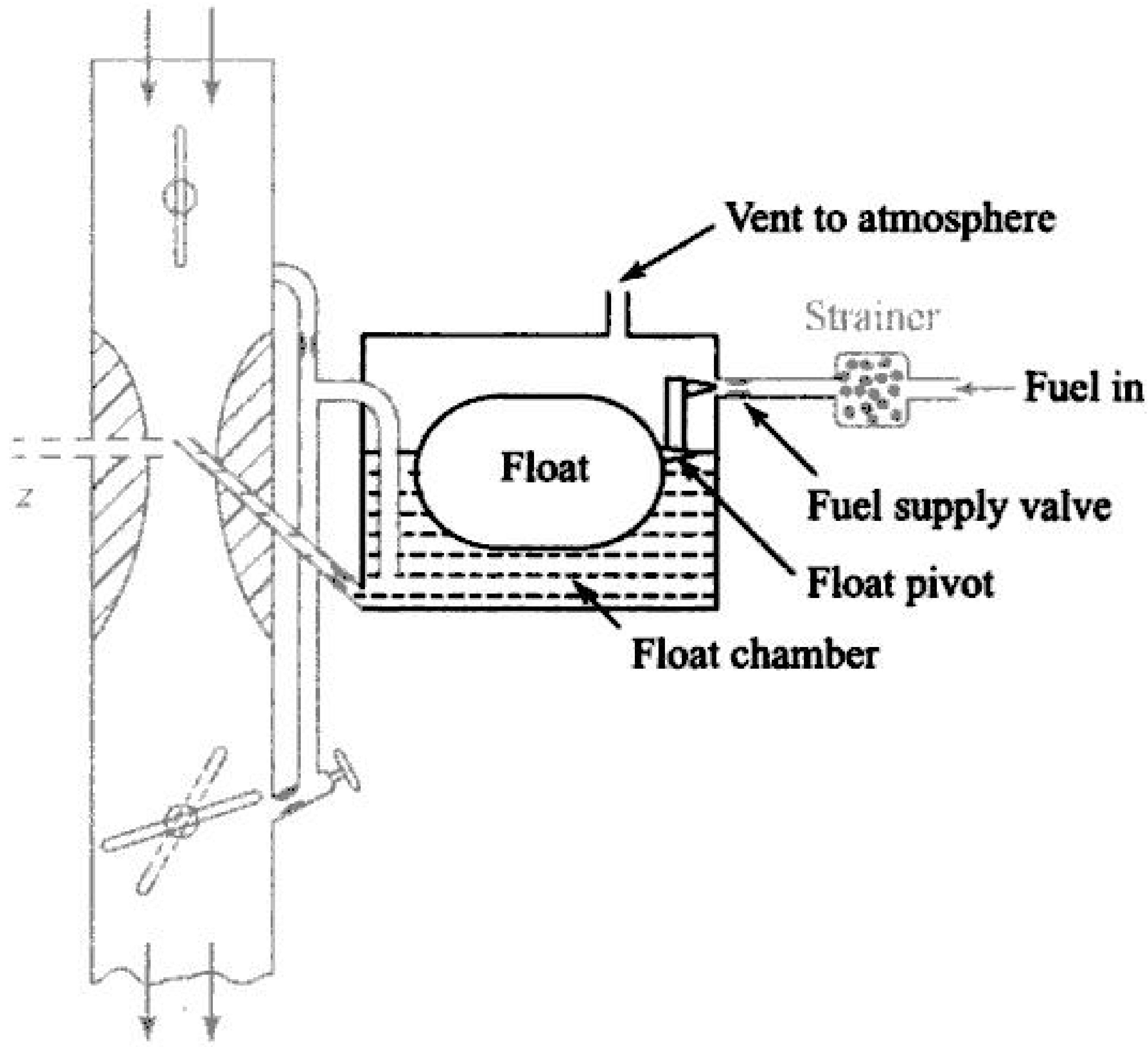

Fig. 8.9 Float Chamber

(iii) the passage leading to the idling system

The three functions of the main metering system are

(i) to proportion the fuel-air mixture

(ii) to decrease the pressure at the discharge nozzle exit

(iii) to limit the air flow at full throttle

The automobiles fitted with SI engine requires a rich mixture for idling and low speed operation (Fig.8.3). Figure 8.10 shows a schematic diagram of a carburetor highlighting the main metering and idling system. Usually air-fuel ratio of about 12:1 is required for idling. In order to provide such rich mixture, during idling, most of the modern carburetors incorporate special idling system is their construction. This consists of idling fuel passage and idling ports as shown in Fig.8.10. This system gets operational at starting, idling and very low speed running of the vehicle engine and is non-operational when throttle is opened beyond 15% to 20%. When the

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9.3.1 Air Injection System

In this system, fuel is forced into the cylinder by means of compressed air. This system is little used nowadays, because it requires a bulky multi-stage air compressor. This causes an increase in engine weight and reduces the brake power output further. One advantage that is claimed for the air injection system is good mixing of fuel with the air with resultant higher mean effective pressure. Another is the ability to utilize fuels of high viscosity which are less expensive than those used by the engines with solid injection systems. These advantages are off-set by the requirement of a multistage compressor thereby making the air-injection system obsolete.

9.3.2 Solid Injection System

In this system the liquid fuel is injected directly into the combustion chamber without the aid of compressed air. Hence, it is also called *airless mechanical injection* or *solid injection system.* Solid injection systems can be classified into four types.

(i) Individual pump and nozzle system

(ii) Unit injector system

(iii) Common rail system

(iv) Distributor system

All the above systems comprise mainly of the following components.

(i) fuel tank,

(ii) fuel feed pump to supply fuel from the main fuel tank to the injection system,

(iii) injection pump to meter and pressurize the fuel for injection,

(iv) governor to ensure that the amount of fuel injected is in accordance with variation in load,

(v) injector to take the fuel from the pump and distribute it in the combustion chamber by atomizing it into fine droplets,

(vi) fuel filters to prevent dust and abrasive particles from entering the pump and injectors thereby minimizing the wear and tear of the components.

Table 9.1 Comparison of Various Fuel-Injection Systems

Job	Air injection system	Solid injection system		
		Individual pump	Common rail	Distributor
Metering	Pump	Pump	Injection valve	Pump
Timing	Fuel cam	Pump cam	Fuel cam	Fuel cam
Injection rate	Spray valve	Pump cam	Spray valve	Fuel cam
Atomization	Spray valve	Spray tip	Spray tip	Spray tip
Distribution	Spray valve	Spray tip	Spray tip	Spray tip

9.4 FUEL FEED PUMP

A schematic sketch of fuel feed pump is shown in Fig.9.4. It is of spring loaded plunger type. The plunger is actuated through a push rod from the cam shaft.

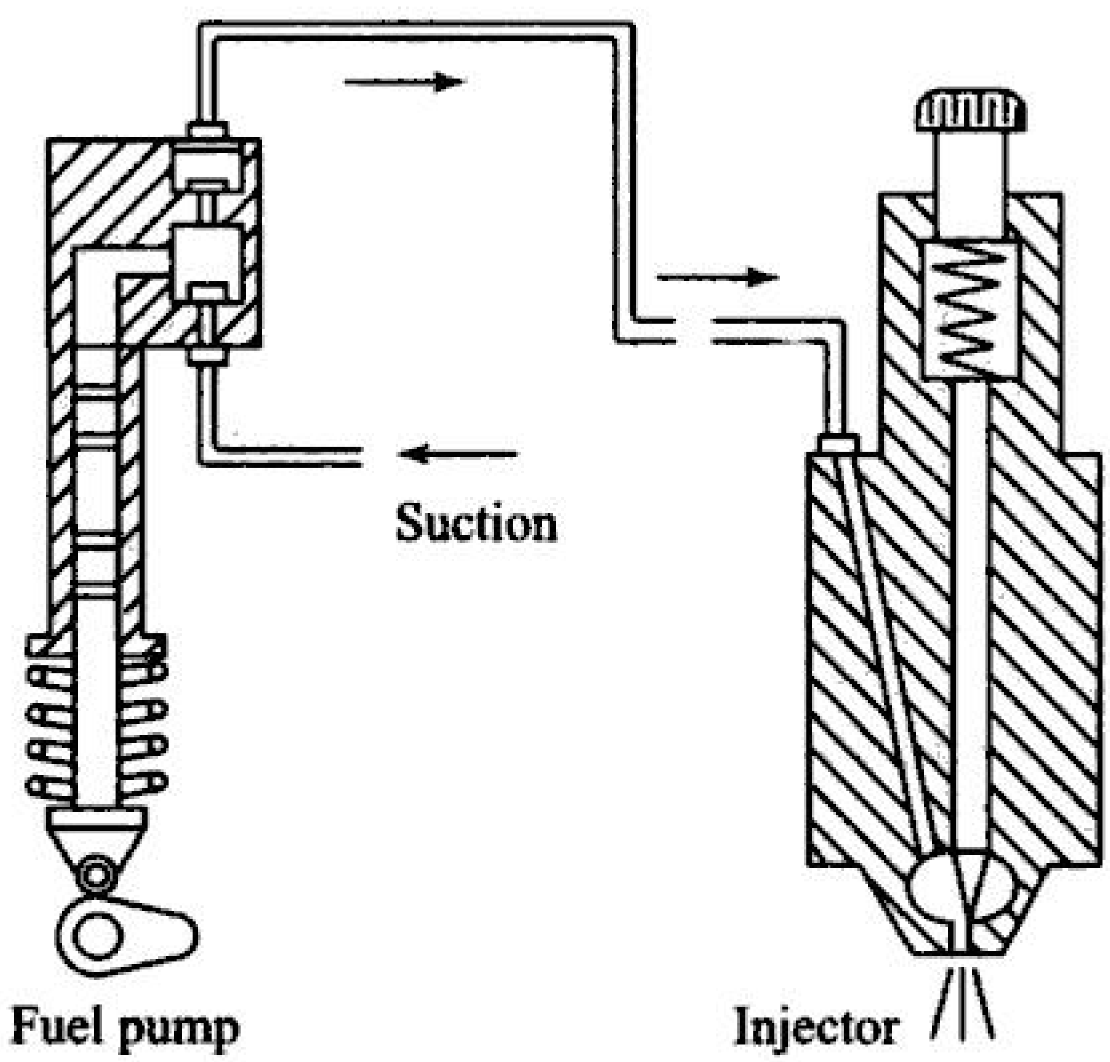

Fig. 9.4 Schematic Diagram of Fuel Feed Pump

At the minimum lift position of the cam the spring force on the plunger creates a suction which causes fuel flow from the main tank into the pump. When the cam is turned to its maximum lift position, the plunger is lifted upwards. At the same time the inlet valve is closed and the fuel is forced through the outlet valve. When the

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a central longitudinal passage in the rotor and also two sets of radial holes (each equal to the number of engine cylinders) located at different heights. One set is connected to pump inlet via central passage whereas the second set is connected to delivery lines leading to injectors of the various cylinders.

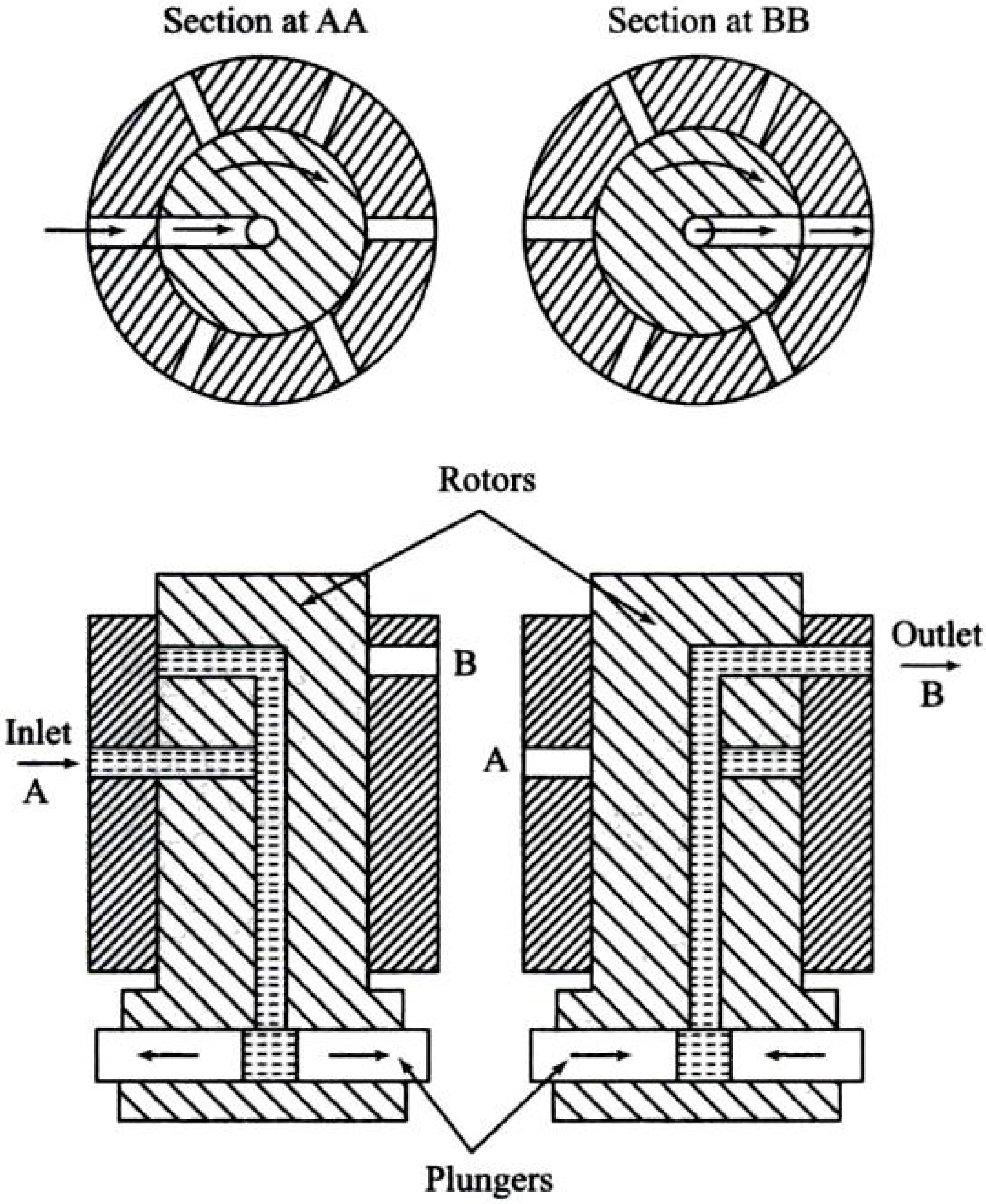

Fig. 9.7 Principle of Working of Distributor Type Fuel-Injection Pump

The fuel is drawn into the central rotor passage from the inlet port when the pump plunger move away from each other. Wherever, the radial delivery passage in the rotor coincides with the delivery port for any cylinder the fuel is delivered to each cylinder in turn.

Main advantages of this type of pump lies in its small size and its light weight. A schematic diagram of Roosa Master distributor pump is shown in Fig.9.8.

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(iii) a nozzle

(iv) an injector body

A cross sectional view of a typical Bosch fuel injector is shown in Fig.9.11.

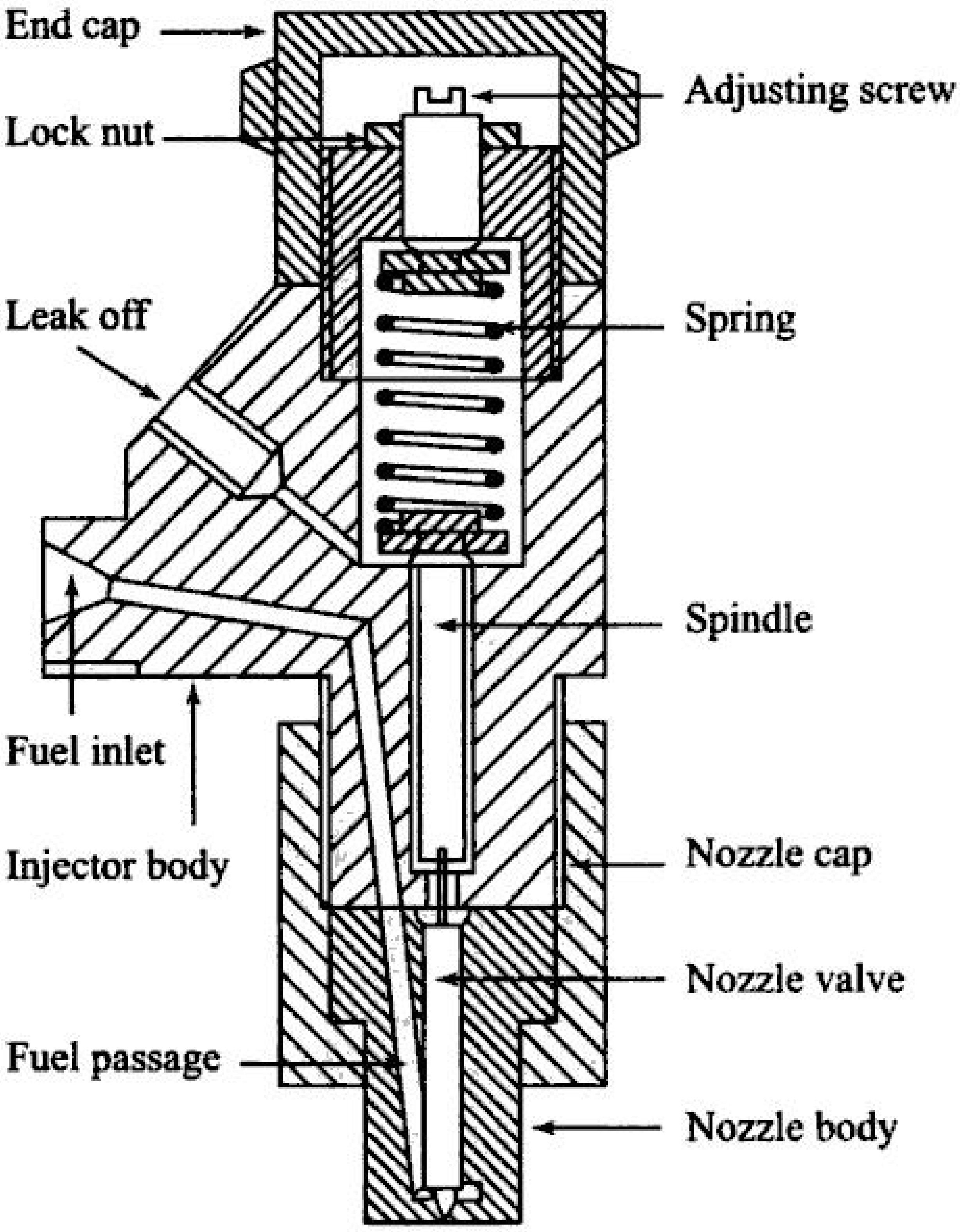

Fig. 9.11 Fuel Injector (Bosch)

When the fuel is supplied by the injection pump it exerts sufficient force against the spring to lift the nozzle valve, fuel is sprayed into the combustion chamber in a finely atomized particles. After, fuel from the delivery pump gets exhausted, the spring pressure pushes the nozzle valve back on its seat. For proper lubrication between nozzle valve and its guide a small quantity of fuel is allowed to leak through the clearance between them and then drained back to fuel tank through leak off connection. The spring tension and hence the valve opening pressure is controlled by adju. ing the screw provided at the top.

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400 m/s. The velocity of the fuel through nozzle orifice in terms of h can be given by

$$V_f = C_d\sqrt{2gh}$$

where h is the pressure difference between injection and cylinder pressure, measured in m of fuel column.

The volume of the fuel injected per second, Q, is given by

$$\begin{aligned} Q &= \text{Area of all orifices} \times \text{fuel jet velocity} \times \\ &\quad \text{time of one injection} \times \text{number of injections} \\ &\quad \text{per second for one orifice} \end{aligned}$$

$$Q = \left(\frac{\pi}{4}d^2 \times n\right) \times V_f \times \left(\frac{\theta}{360} \times \frac{60}{N}\right) \times \left(\frac{N_i}{60}\right)$$

where N_i for four-stroke engine is rpm/2 and for a two-stroke engine N_i is rpm itself and d is the diameter of one orifice in m, n is the number of orifices, θ is the duration of injection in crank angle degrees and N_i is the number of injections per minute. Usually the rate of fuel-injection is expressed in mm^3/degree crank angle/litre cylinder displacement volume to normalize the effect of engine size.

The rate of fuel injected/degree of crankshaft rotation is a function of injector camshaft velocity, the diameter of the injector plunger, and flow area of the tip orifices. Increasing the rate of injection decreases the duration of injection for a given fuel input and subsequently introduces a change in injection timing. A higher rate of injection may permit injection timing to be retarded from optimum value. This helps in maintaining fuel economy without excessive smoke emission. However, an increase in injection rate requires an increased injection pressure and increases the load on the injector push rod and the cam. This may affect the durability of the engine.

9.11 INJECTION IN SI ENGINE

Fuel-injection systems are commonly used in CI engines. Presently gasoline injection system is coming into vogue in SI engines because of the following drawbacks of the carburetion.

(i) Non uniform distribution of mixture in multicylinder engines.

(ii) Loss of volumetric efficiency due to restrictions for the mixture flow and the possibility of back firing.

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$$d = \sqrt{\frac{4 \times 0.4932 \times 10^{-6}}{\pi}}$$

$$= 0.792 \times 10^{-3} \text{ m}$$

$$= \mathbf{0.792 \text{ mm}}$$ Ans

9.3 A four-cylinder, four-stroke diesel engine develops a power of 180 kW at 1500 rpm. The $bsfc$ is 0.2 kg/kW h. At the beginning of injection pressure is 30 bar and the maximum cylinder pressure is 50 bar. The injection is expected to be at 200 bar and maximum pressure at the injector is set to be about 500 bar. Assuming the following:

C_d for injector	=	0.7
S.G. of fuel	=	0.875
Atmospheric pressure	=	1 bar
Effective pressure difference	=	Average pressure difference over the injection period

Determine the total orifice area required per injector if the injection takes place over 15° crank angles.

Solution

$$\text{Power output/cylinder} = \frac{180}{4} = 45 \text{ kW}$$

$$\text{Fuel consumption/cylinder} = 45 \times bsfc$$

$$= 45 \times 0.2 = 9 \text{ kg/h}$$

$$= 0.15 \text{ kg/min}$$

$$\text{Fuel injected/cycle} = \frac{0.15}{(rpm/2)} = \frac{0.15}{(1500/2)}$$

$$= 2 \times 10^{-4} \text{ kg}$$

$$\text{Time for injection} = \frac{\theta}{360 \times rpm/60}$$

$$= \frac{15 \times 60}{360 \times 1500}$$

$$= 1.667 \times 10^{-3} \text{ s}$$

$$= \quad 0.12345 \times 10^{-6} \text{ m}^3/\text{s}$$

$$= \quad 0.12345 \text{ cc/s}$$

Coefficient of compressibility is defined as

$$K_{comp} = \frac{\textit{Change in volume/unit volume}}{\textit{Difference in pressure causing the compression}}$$

$$= \frac{(V_1 - V_2)/V_1}{(p_2 - p_1)}$$

Here,

$$K_{comp} = 80 \times 10^{-6} \textit{ per bar}$$

$$V_1 = \textit{Fuel in pump barrel + Fuel inside the injector + Fuel in pipe line}$$

$$= 4 + 2 + 3 = 9 \; cc$$

$$V_1 - V_2 = K_{comp} \times V_1 \times (p_2 - p_1)$$

$$= 80 \times 10^{-6} \times 9 \times (300 - 1) = 0.21528 \; cc$$

Plunger/displacement volume

$$= 0.12345 + 0.21528$$

$$= \mathbf{0.339\ cc} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Pump work, } W_P = \frac{1}{2}(p_{inj} - p_o)(V_1 - V_2) + (p_{inj} - p_{cyl})V_f$$

$$= \frac{1}{2}(300 - 1) \times 10^5 \times 0.2153 \times 10^{-6} + (300 - 40) \times 10^5 \times 0.12345 \times 10^{-6}$$

$$= 3.22 + 3.21$$

$$= 6.43 \; J$$

Power lost for pumping the fuel

$$= \frac{6.43}{1000} \times \frac{1500}{2 \times 60}$$

$$= \mathbf{0.08\ kW} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

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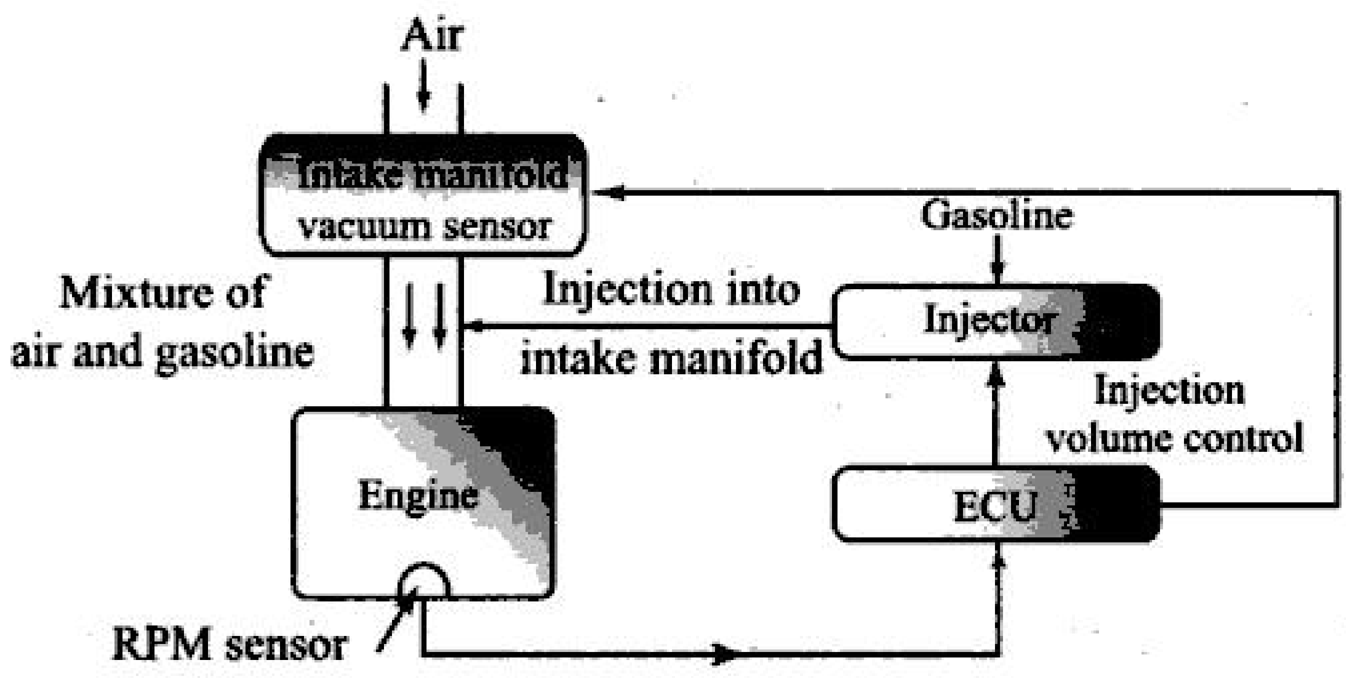

Fig. 10.6 D-MPFI Gasoline Injection System

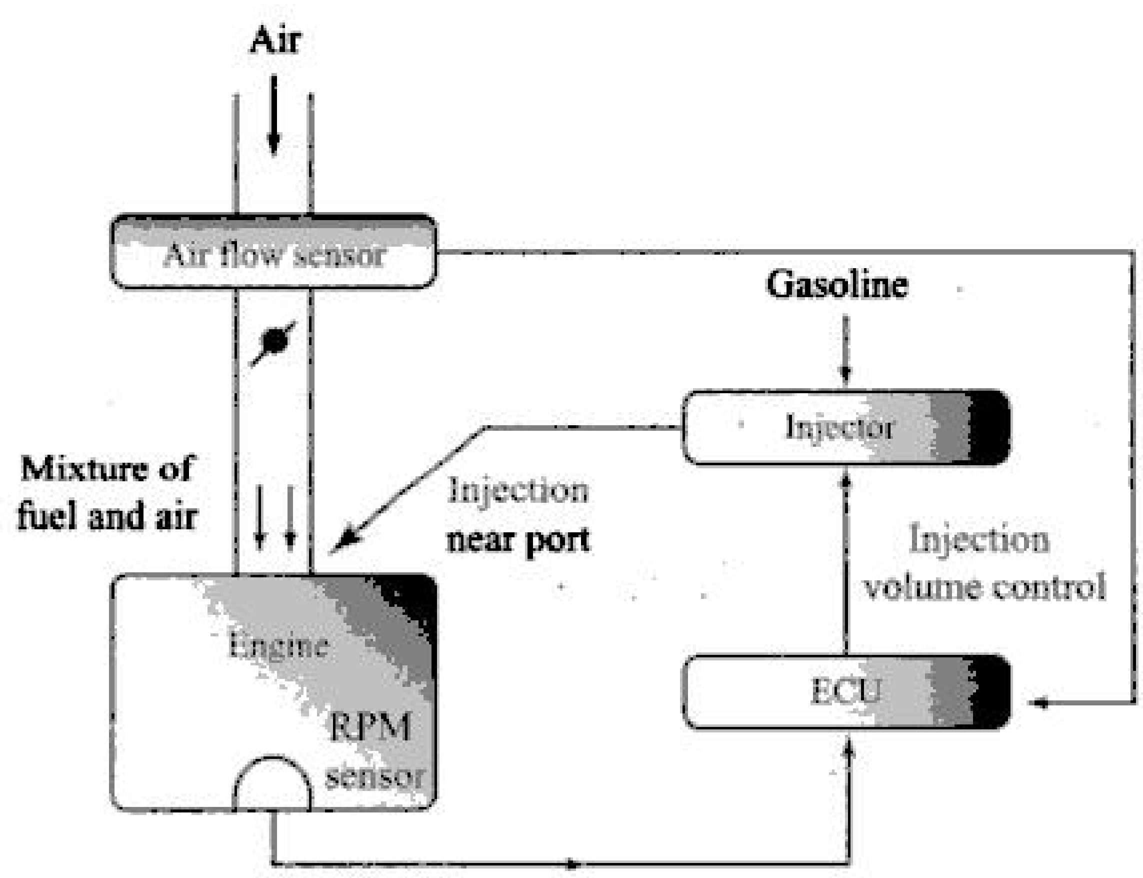

Fig. 10.7 L-MPFI Gasoline Injection System

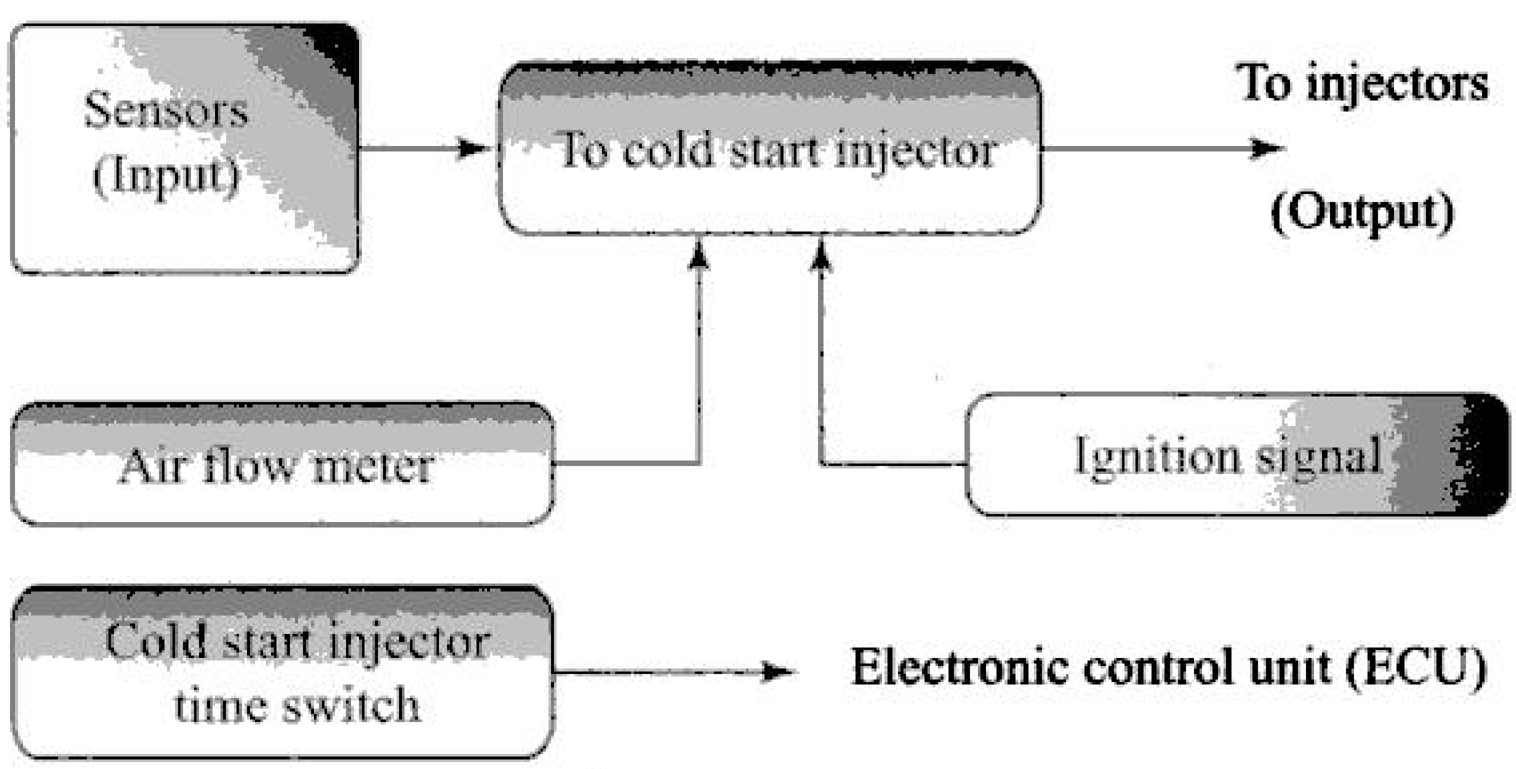

Fig. 10.8 MPFI-Electronic Control System

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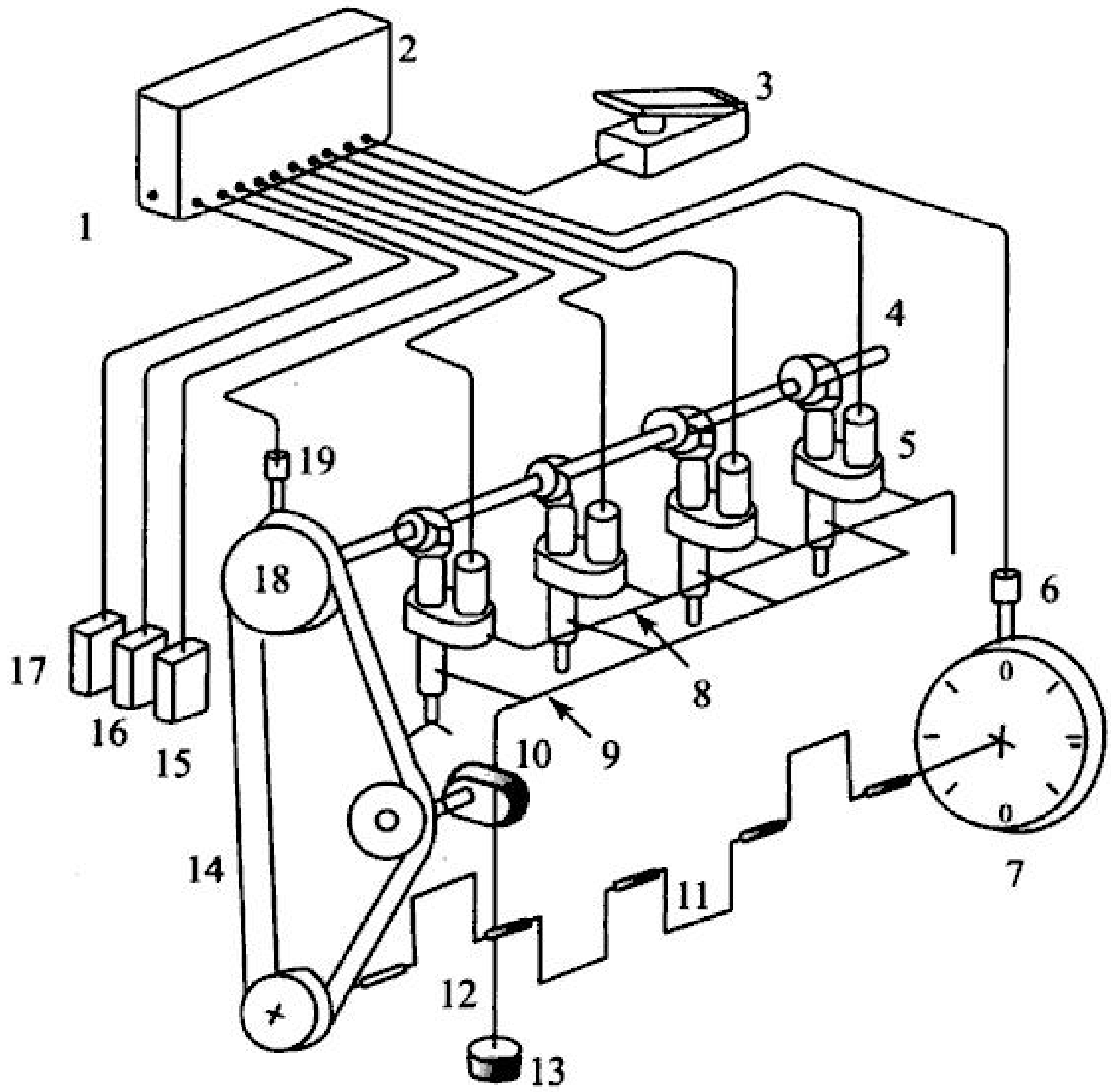

1	Battery supply (12 V or 24 V)	10	Fuel gear pump
2	Electronic control unit (ECU)	11	Crankshaft
3	Driver command (Accelerator pedal)	12	Fuel pickup
4	Injector camshaft	13	Gauze strainer
5	Injector unit	14	Timing belt
6	Magnetic transducer monitors engine speed and crankshaft position	15	Boost sensor connection
7	Sixty tooth flywheel	16	Coolant temperature connection
8	Fuel return gallery	17	Other sensor connection
9	Fuel supply gallery	18	Camshaft four-toothed gear wheel
		19	Magnetic transducer firing order sen

Fig. 10.15 A Typical Electronically Controlled Diesel Injection System

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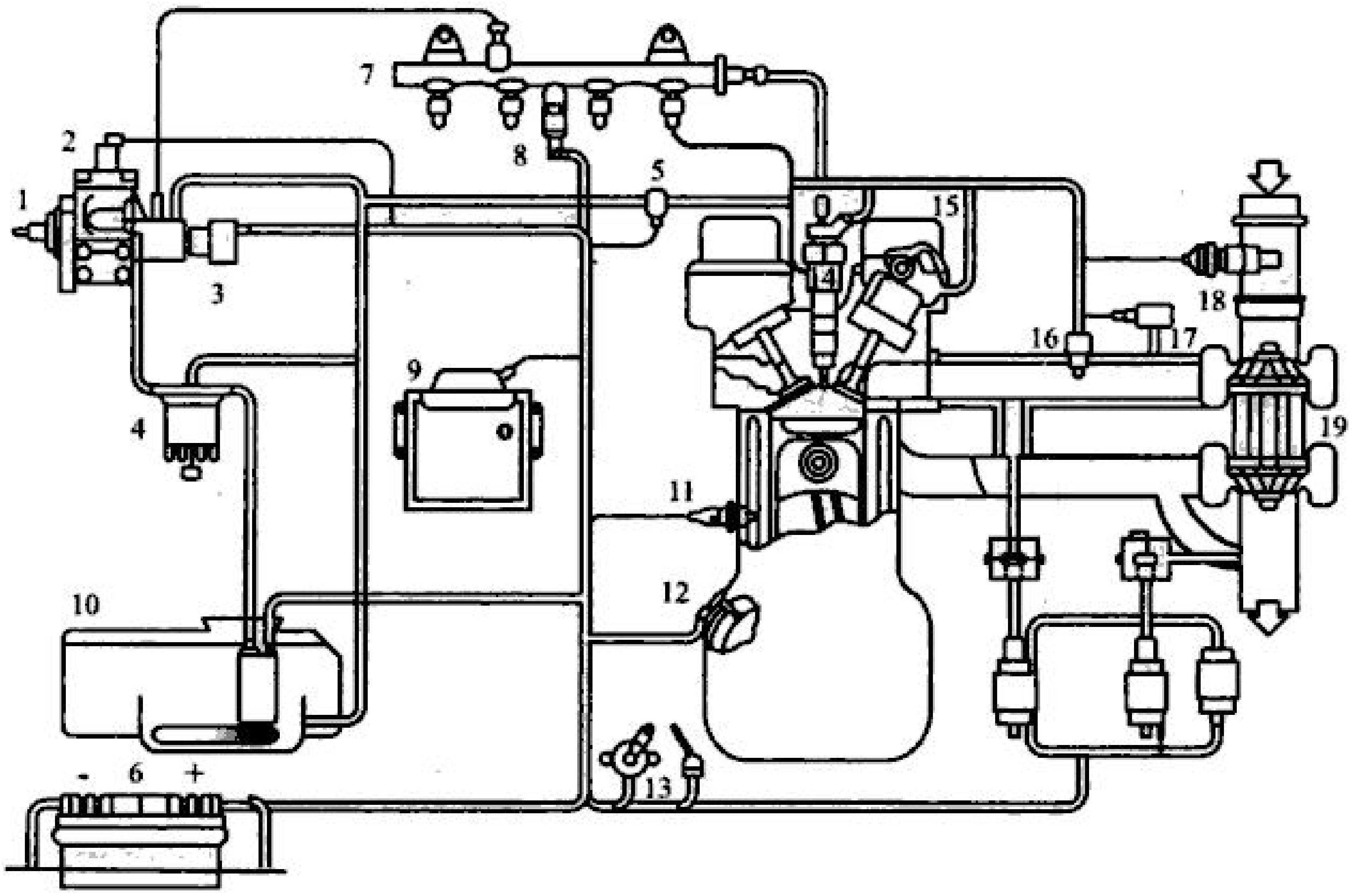

1	High pressure pump	11	Coolant temperature sensor
2	Element shutoff valve	12	Crankshaft speed
3	Pressure control valve	13	Accelerator pedal sensor
4	Fuel filter	14	Injector
5	Fuel temperature sensor	15	Camshaft speed sensor
6	Battery	16	Intake air temperature sensor
7	High pressure accumulator (rail)	17	Boost pressure sensor
8	Rail pressure sensor	18	Air mass meter
9	ECU	19	Turbocharger
10	Fuel tank with prefilter and presupply pump		

Fig. 10.18 Sensors of a Common Rail Injection System, together with Various System Components

11.2 ENERGY REQUIREMENTS FOR IGNITION

The total enthalpy required to cause the flame to be self sustaining and promote ignition, is given by the product of the surface area of the spherical flame and the enthalpy per unit area. It is reasonable to assume that the basic requirement of the ignition system is that it should supply this energy within a small volume. Further, ignition should occur in a time interval sufficiently short to ensure that only a negligible amount of energy is lost other than to establish the flame. In view of this last mentioned condition, it is apparent that the rate of supply of energy is as important a factor as the total energy supplied.

A small electric spark of short duration would appear to meet most of the requirements for ignition. A spark can be caused by applying a sufficiently high voltage between two electrodes separated by a gap, and there is a critical voltage below which no sparking occurs. This critical voltage is a function of the dimension of the gap between the electrodes, the fuel-air ratio and the pressure of the gas. Additionally, the manner in which the voltage is raised to the critical value and the configuration and the condition of the electrodes are important in respect of the energy required.

An ignition process obeys the law of conservation of energy. Hence, it can be treated as a balance of energy between:

(i) that provided by an external source

(ii) that released by chemical reaction and

(iii) that dissipated to the surroundings by means of thermal conduction, convection and radiation

11.3 THE SPARK ENERGY AND DURATION

With a homogeneous mixture in the cylinder, spark energy of the order of 1 mJ and a duration of a few micro-seconds would suffice to initiate the combustion process. However, in practice, circumstances are less than the ideal. The pressure, temperature and density of the mixture between the spark plug electrodes have a considerable influence on the voltage required to produce a spark. Therefore, the spark energy and duration are to be of sufficient order to initiate combustion under the most unfavourable conditions expected in the vicinity of the spark plug over the complete range of engine operation. Usually, if the spark energy exceeds 40 mJ and the duration is longer than 0.5 ms, reliable ignition is obtained. If the resistance of the

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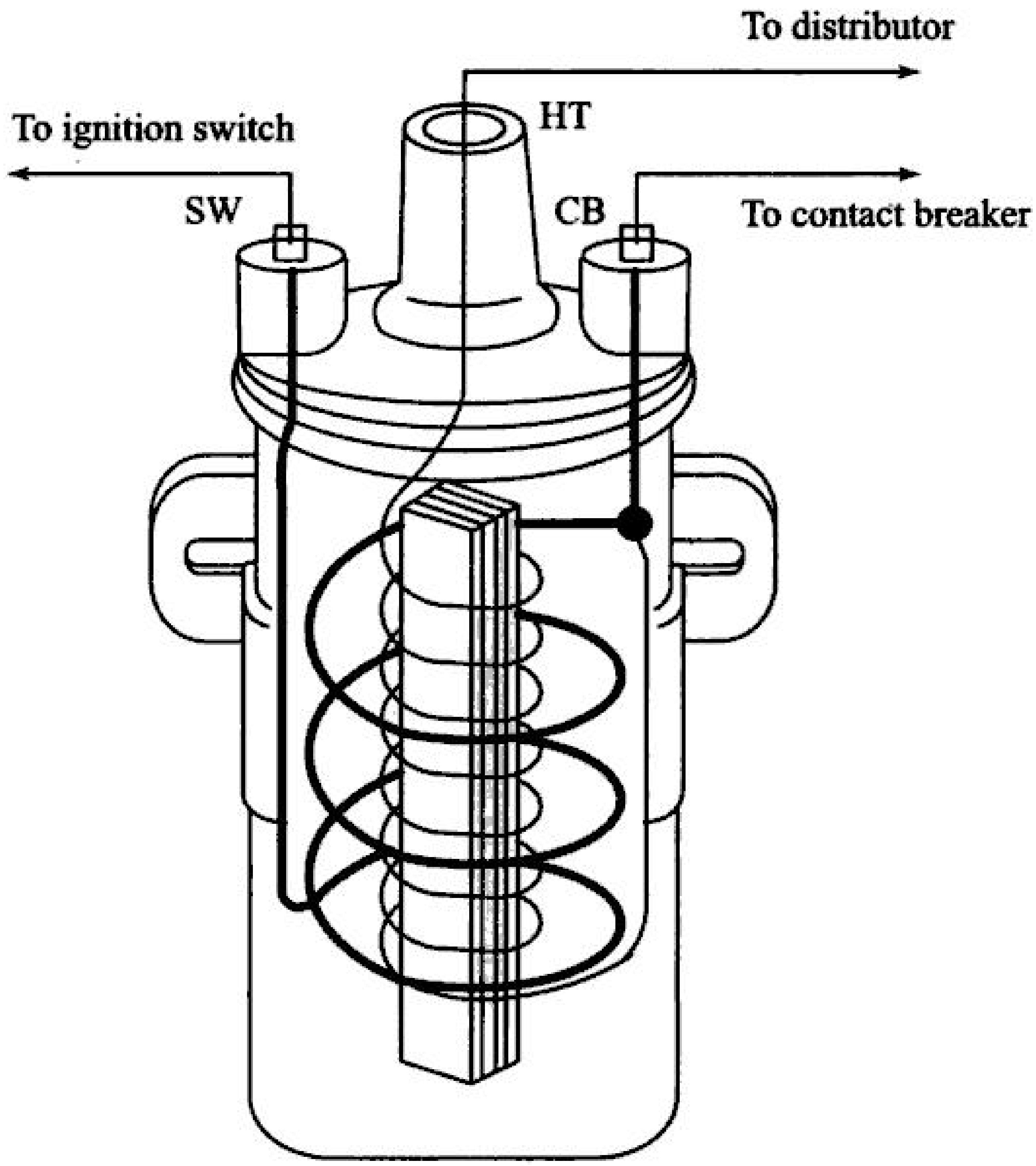

Fig. 11.3 Ignition Coil

pressure of the spring thereby closing the primary circuit. The condition and adjustment of the contact breaker points are important. The points are subjected to a very severe hammering during their period of service. Uneven wear of the points may require a refacing or replacing depending upon the condition of the points.

An eight cylinder engine running at 3000 rpm requires 12000 sparks per minute, i.e. 200 sparks per second. If the breaker is to operate satisfactorily at this speed, the travel of the breaker arm must be held down to the minimum to ensure a positive spark and the breaker arm must be made very light.

11.6.6 Capacitor

The principle of construction of the ignition capacitor is the same as that of every electrical capacitor, which is very simple: two metal plates – separated by an insulating material – are placed face to face.

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secondary windings all rotate between the poles of a stationary magnet, whilst, in the second type the magnet revolves and the windings are kept stationary. A third type of magneto called the *polar inductor type* is also in use. In the polar inductor type *magneto* both the magnet and the windings remain stationary but the voltage is generated by reversing the flux field with the help of soft iron polar projections, called inductors.

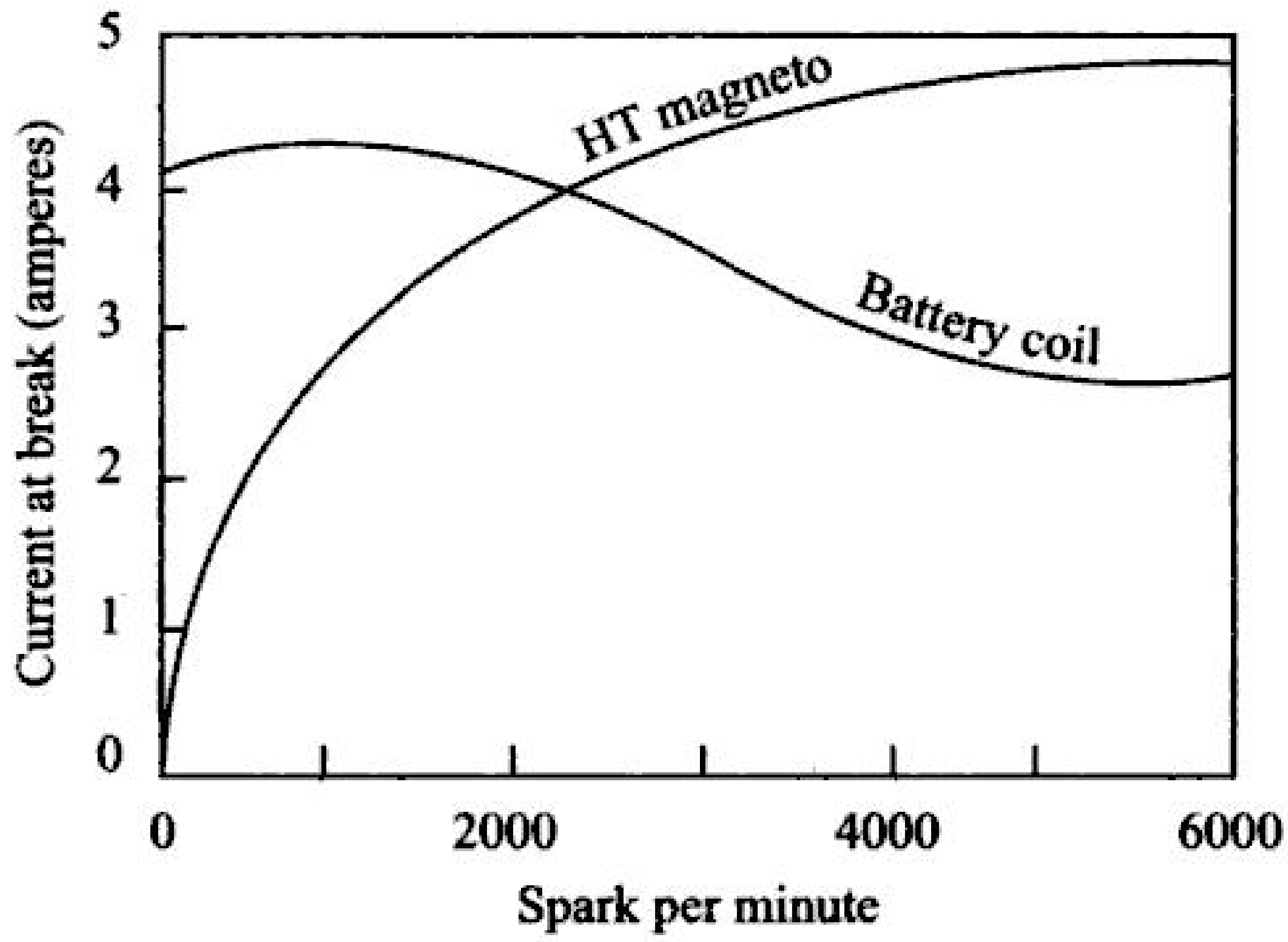

Fig. 11.10 Breaker Current versus Speed in Coil and Magneto Ignition Systems

The working principle of the magneto ignition system is exactly the same as that of the coil ignition system. With the help of a cam, the primary circuit flux is changed and a high voltage is produced in the secondary circuit.

The variation of the breaker current with speed for the coil ignition system and the magneto ignition system is shown in Fig.11.10. It can be seen that since the cranking speed at start is low the current generated by the magneto is quite small. As the engine speed increases the flow of current also increases. Thus, with magneto there is always a starting difficulty and sometimes a separate battery is needed for starting. The magneto is best at high speeds and therefore is widely used for sports and racing cars, aircraft engines etc.

In comparison, the battery ignition system is more expensive but highly reliable. Because of the poor starting characteristics of the magneto system invariably the battery ignition system is preferred to

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11.14.3 Part Load Operation

Part load operation of a spark ignition engine is affected by throttling the incoming charge. Due to throttling a smaller amount of charge enters the cylinder and the dilution due to residual gases is also greater. This results in the combustion process being slower. In order to overcome the problem of exhaust gas dilution and the low charge density at part load operation the spark advance must be increased.

11.14.4 Type of Fuel

Ignition delay will depend upon the type of the fuel used in the engine. For maximum power and economy a slow burning fuel needs a higher spark advance than a fast burning fuel.

11.15 SPARK ADVANCE MECHANISM

It is obvious from the above discussion that the point in the cycle where the spark occurs must be regulated to ensure maximum power and economy at different speeds and loads and this must be done automatically. The purpose of the spark advance mechanism is to assure that under every condition of engine operation, ignition takes place at the most favourable instant in time, i.e., most favourable from a standpoint of engine power, fuel economy, and minimum exhaust dilution. By means of these mechanisms the advance angle is accurately set so that ignition occurs before the top dead-center point of the piston. The engine speed and the engine load are the control quantities required for the automatic adjustment of the ignition timing. Most of the engines are fitted with mechanisms which are integral with the distributor and automatically regulate the optimum spark advance to account for change of speed and load. The two mechanisms used are:

(i) Centrifugal advance mechanism

(ii) Vacuum advance mechanism

These mechanisms are discussed in greater details in the following sections.

11.15.1 Centrifugal Advance Mechanism

The centrifugal advance mechanism controls the ignition timing for full-load operation. The adjustment mechanism is designed so that

12

COMBUSTION AND COMBUSTION CHAMBERS

12.1 INTRODUCTION

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.

12.2 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

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.

12.6 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.

Figure 12.3 shows the rate of flame propagation. In area I, (A$\rightarrow$B), the flame front progresses relatively slowly due to a low *transposition rate* and low turbulence. The transposition of the flame front is very little since there is a comparatively small mass of charge burned at the start. The low reaction rate plays a dominant role resulting in a slow advance of the flame. Also, since the spark plug is to be necessarily located in a quiescent layer of gas that is close to the cylinder wall, the lack of turbulence reduces the reaction rate and hence the flame speed. As the flame front leaves the quiescent zone and proceeds into more turbulent areas (area II) where it consumes a greater mass of mixture, it progresses more rapidly and at a constant rate (B$\rightarrow$C) as shown in Fig.12.3.

The volume of unburned charge is very much less towards the end of flame travel and so the *transposition rate* again becomes negligible thereby reducing the flame speed. The reaction rate is also reduced again since the flame is entering a zone (area III) of relatively low turbulence (C$\rightarrow$D) in Fig.12.3.

in the crankshaft rotation. It also tends to promote an undesirable occurrence known as knocking. 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.

12.9 ABNORMAL COMBUSTION

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 preignition and mechanical damage to the engine.

12.10 THE PHENOMENON OF KNOCK IN SI ENGINES

In a spark-ignition engine combustion which is initiated between the spark plug electrodes spreads across the combustible 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 autoignition occurs at various pin-point locations. This phenomenon is called knocking. The process of autoignition leads towards engine knock.

The phenomenon of knock may be explained by referring to Fig.12.6(a) which shows the cross-section of the combustion chamber with flame advancing from the spark plug location A without knock

valve are two hottest parts in the combustion chamber, the end gas should not be compressed against them.

Retarding the Spark Timing: By retarding the spark timing from the optimized timing, i.e., having the spark closer to TDC, the peak pressures are reached farther down on the power stroke and are thus of lower magnitude. This might reduce the knocking. However, the spark timing will be different from the MBT timing. This will affect the brake torque and power output of the engine.

Power Output of the Engine: A decrease in the output of the engine decreases the temperature of the cylinder and the combustion chamber walls and also the pressure of the charge thereby lowering mixture and end gas temperatures. This reduces the tendency to knock.

12.11.2 Time Factors

Increasing the flame speed or increasing the duration of the ignition ignition lag or reducing the time of exposure of the unburned mixture to autoignition condition will tend to reduce knocking. The following factors, in most cases, reduce the possibility of knocking.

Turbulence: Turbulence depends on the design of the combustion chamber and on engine speed. Increasing turbulence increases the flame speed and reduces the time available for the end charge to attain autoignition conditions thereby decreasing the tendency to knock.

Engine Speed: An increase in engine speed increases the turbulence of the mixture considerably resulting in increased flame speed, and reduces the time available for preflame reactions. Hence knocking tendency is reduced at higher speeds.

Flame Travel Distance: The knocking tendency is reduced by shortening the time required for the flame front to traverse the combustion chamber. Engine size (combustion chamber size), and spark plug position are the three important factors governing the flame travel distance.

Engine Size: The flame requires a longer time to travel across the combustion chamber of a larger engine. Therefore, a larger engine has a greater tendency for knocking than a smaller engine since there is more time for the end gas to autoignite. Hence, an SI engine is generally limited to size of about 150 mm bore.

Combustion Chamber Shape: Generally, the more compact the combustion chamber is, the shorter is the flame travel and the combustion time and hence better antiknock characteristics. Therefore,

leaving a slightly restricted passage communicating with the cylinder thereby creating additional turbulence during the compression stroke. This design reduces the knocking tendency by shortening the effective flame travel length by bringing that portion of the head which lay over the farther side of the piston into as close a contact as possible with the piston crown, forming a quench space. The thin layer of gas (entrapped between the relatively cool piston and also cooler head) loses its heat rapidly because of large enclosing surface thereby avoiding knocking. By placing the spark plug in the centre of the effective combustion space slightly towards the hot exhaust valve, the flame travel length is reduced.

I-Head Type or Overhead Valve: The I-head type is also called the overhead valve combustion chamber in which both the valves are located on the cylinder head. The overhead valve engine [Fig.12.8(d)] is superior to a side valve or an L-head engine at high compression ratios. Some of the important characteristics of this type of valve arrangement are:

(i) less surface to volume ratio and therefore less heat loss

(ii) less flame travel length and hence greater freedom from knock

(iii) higher volumetric efficiency from larger valves or valve lifts

(iv) confinement of thermal failures to cylinder head by keeping the hot exhaust valve in the head instead of the cylinder block.

F-Head Type: The F-head type of valve arrangement is a compromise between L-head and I-head types. Combustion chambers in which one valve is in the cylinder head and the other in the cylinder block are known as F-head combustion chambers [Fig.12.8(e)]. Modern F-head engines have exhaust valve in the head and inlet valve in the cylinder block. The main disadvantage of this type is that the inlet valve and the exhaust valve are separately actuated by two cams mounted on two camshafts driven by the crankshaft through gears.

12.13 COMBUSTION IN COMPRESSION-IGNITION ENGINES

There are certain basic differences existing between the combustion process in the SI and CI engines. In the SI engine, a homogeneous carburetted mixture of gasoline vapour and air, in a certain proportion, is compressed (compression ratio 6:1 to 10:1) and the mixture is ignited at one place before the end of the compression stroke by

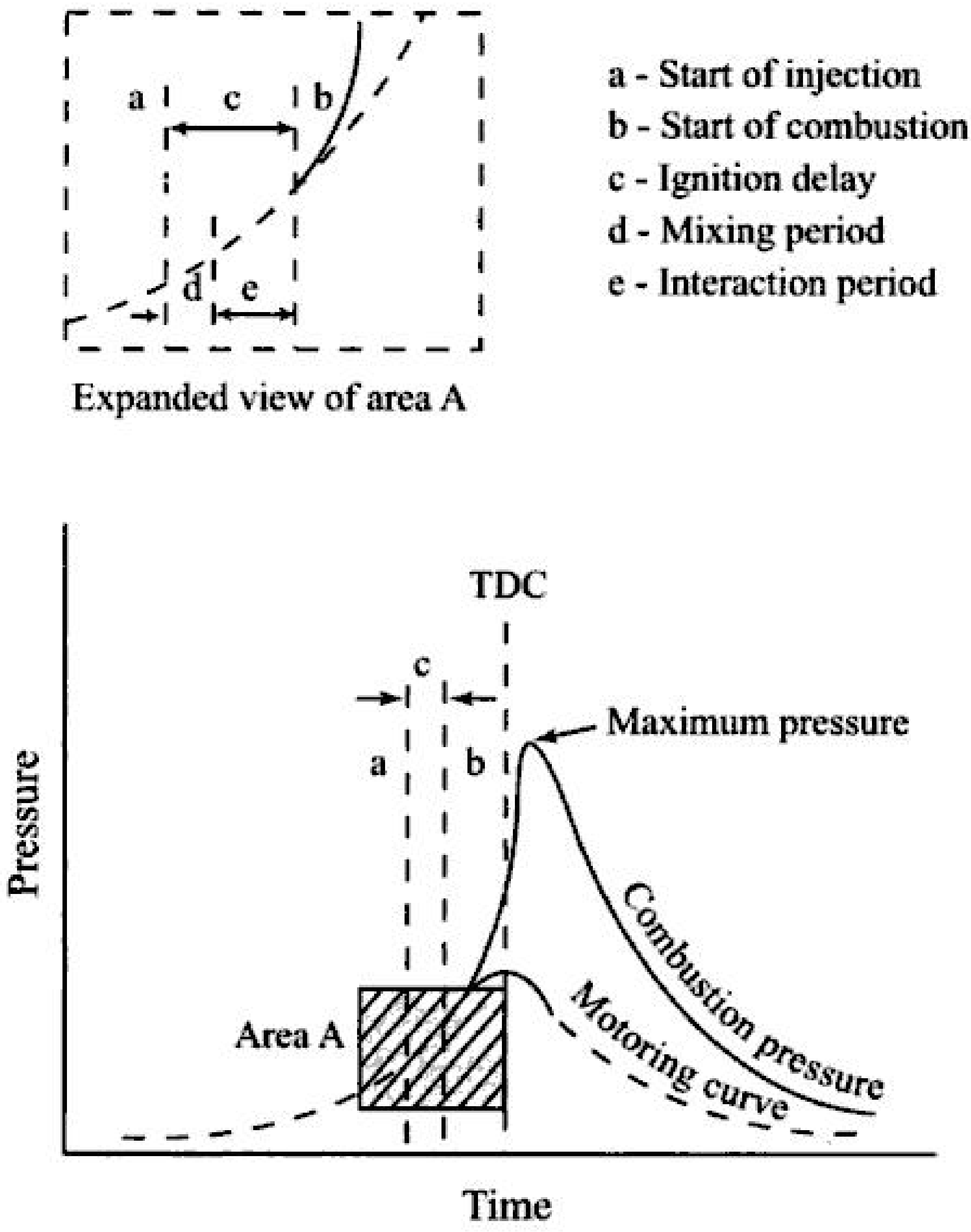

Fig. 12.12 Pressure-Time Diagram illustrating Ignition Delay

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.

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.

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increase in weight of the reciprocating parts. Therefore, in practice the engine designers always try to use a lower compression ratio which helps in easy cold starting and light load running at high speeds.

12.15.2 Engine Speed

The delay period could be given either in terms of absolute time (in milliseconds) or in terms of crank angle degrees. Fig.12.15 shows the decrease in delay period in terms of milliseconds with increase in engine speed in a variable speed operation with a given fuel.

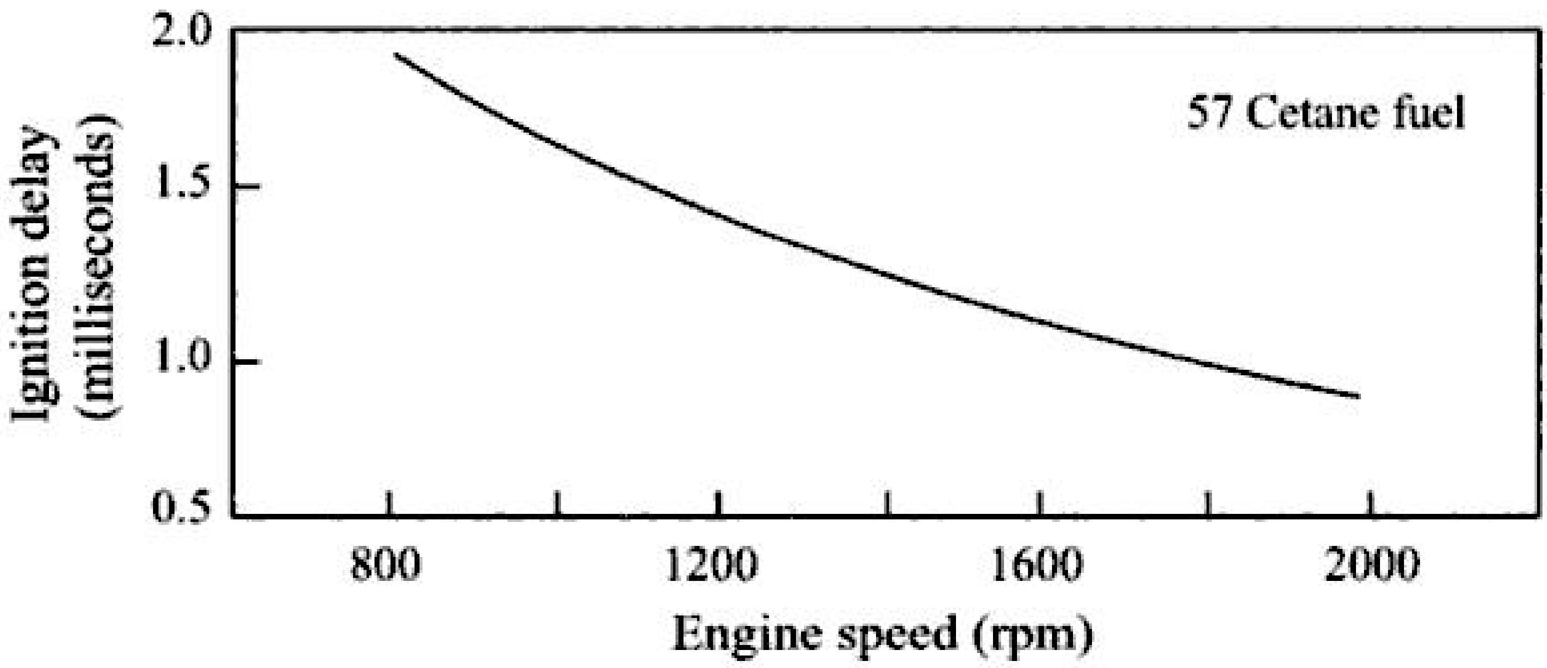

Fig. 12.15 Effect of Speed on Ignition Delay in a Diesel Engine

With increase in engine speed, the loss of heat during compression decreases, resulting in the rise of both the temperature and pressure of the compressed air thus reducing the delay period in milliseconds. However, in degrees of crank travel the delay period increases as the engine operates at a higher rpm. The fuel pump is geared to the engine, and hence the amount of fuel injected during the delay period depends on crank degrees and not on absolute time. Hence, at high speeds, there will be more fuel present in the cylinder to take part in the second stage of uncontrolled combustion resulting in high rate of pressure rise.

12.15.3 Output

With an increase in engine output the air-fuel ratio decreases, operating temperatures increase and hence delay period decreases. The rate of pressure rise is unaffected but the peak pressure reached may be high.

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If, on the other hand, the ignition delay is longer, the *actual burning* of the first few droplets is delayed and a greater quantity of fuel droplets gets accumulated in the chamber. When the *actual burning* commences, the additional fuel can cause too rapid a rate of pressure rise as shown in Fig.12.17(b), 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, as shown in Fig.12.17(c). Such a situation produces the extreme pressure differentials and violent gas vibrations known as knocking and is evidenced by audible knock. The phenomenon is similar to that in the SI engine. However, *in the SI engine, knocking occurs near the end of combustion whereas in the CI engine, knocking occurs near the beginning of combustion.*

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.

12.17 COMPARISON OF KNOCK IN SI AND CI ENGINES

It may be interesting to note that knocking in spark-ignition engines and compression-ignition engines is fundamentally due to the autoignition of the fuel-air mixture. In both the cases, the knocking depends on the autoignition lag of the fuel-air mixture. But careful examination of the knocking phenomenon in spark-ignition and the compression-ignition engines reveals the following differences. A comparison of the knocking process in SI and CI engines is shown on the pressure-time diagrams of Fig.12.18.

(i) In spark-ignition engines, the autoignition of the end gas away from the spark plug, most likely near the end of the combustion causes knocking. But in compression-ignition engines the autoignition of the charge causing knocking is at the start of combustion. It is the first charge that autoignites and causes knocking in the compression-ignition engines. This is illustrated in Fig.12.18. It is clear from Fig.12.18 that explosive auto- ignition is more or less over before the peak pressure for the compression-ignition engines. But for spark-ignition engines, the condition for explosive autoignition of the end charge is more favourable after the peak pressure. In order to avoid knocking in spark-ignition engines, it is necessary to prevent

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and the air. The effect of swirl has already been discussed in Section 12.13. The fuel is injected into the combustion chamber by an injector having a single or multihole orifices. The increase in the number of jets reduces the intensity of air swirl needed.

When the liquid fuel is injected into the combustion chamber, the spray cone gets disturbed due to the air motion and turbulence inside. The onset of combustion will cause an added turbulence that can be guided by the shape of the combustion chamber. Since the turbulence is necessary for better mixing, and the fact that it can be controlled by the shape of the combustion chamber, makes it necessary to study the combustion chamber design in detail.

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 located 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 effected 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) Precombustion chamber in which combustion swirl is induced.

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

The details of these chambers are discussed in the following sections.

12.18.1 Direct–Injection Chambers

An open combustion chamber is defined as one in which the combustion space is essentially a single cavity with little restriction from one part of the chamber to the other and hence with no large difference in pressure between parts of the chamber during the combustion process. There are many designs of open chamber some of which are shown in Fig.12.19.

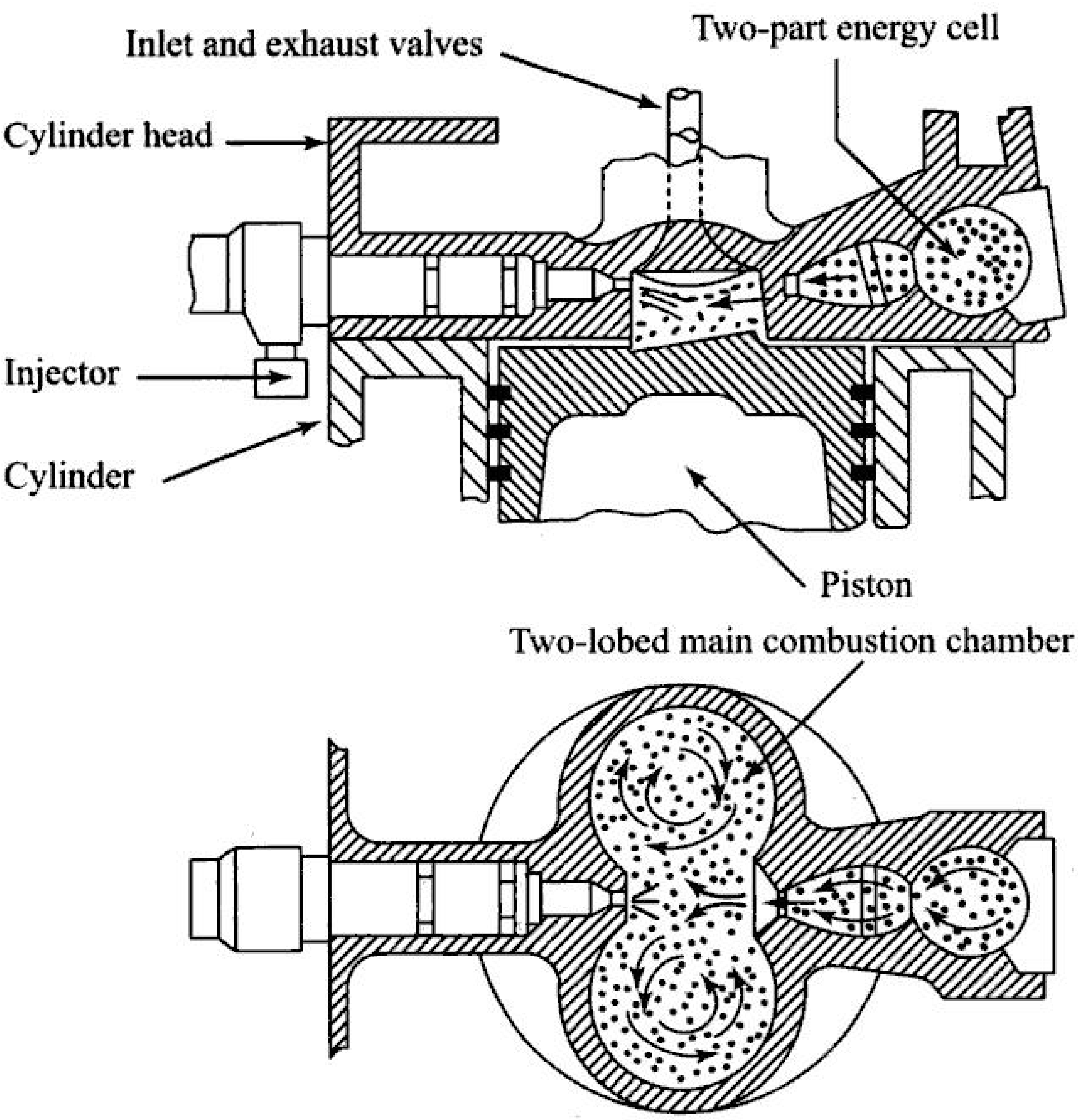

Fig. 12.22 Lanova Air-Cell Combustion Chamber

During compression, the pressure in the main chamber is higher than that inside the energy cell due to restricted passage area between the two. At the *TDC*, the difference in pressure will be high and air will be forced at high velocity through the opening into the energy cell and this moment the fuel-injection also begins. Combustion starts initially in the main chamber where the temperature is comparatively higher but the rate of burning is very slow due to absence of any air motion. In the energy cell, the fuel is well mixed with air and high pressure is developed due to heat-release and the hot burning gases blow out through the small passage into the main chamber. This high velocity jet produces swirling motion in the main chamber and thereby thoroughly mixes the fuel with air resulting in complete combustion. The design is not suitable for variable speed

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13.3.1 Fluid-film or Hydrodynamic Friction

The hydrodynamic friction is associated with the phenomena when a complete film of lubricant exists between the two bearing surfaces. In this case the friction force entirely depends on the lubricant viscosity. This type of friction is the main mechanical friction loss in the engine.

13.3.2 Partial-film Friction

When rubbing (metal) surfaces are not sufficiently lubricated, there is a contact between the rubbing surfaces in some regions. During normal engine operation there is almost no metallic contact except between the compression (top) piston ring and cylinder walls. This is mainly at the end of each stroke where the piston velocity is nearly zero. During starting of the engine, the journal bearings operate in partial-film friction. Thus, partial-film friction contributes very little to total engine friction and hence, it may be neglected.

13.3.3 Rolling Friction

The rolling friction is due to rolling motion between the two surfaces. Ball and roller bearings and tappet rollers are subjected to rolling friction. Bearings of this type have a coefficient of friction which is nearly independent of load and speed. This friction is partly due to local rubbing from distortion under load and partly due to continuous *climbing* of roller. Rolling friction coefficient is lower than journal bearing friction coefficient during starting and initial running of engine. The reason is that the oil viscosity is high and moreover, partial friction exists in journal bearing during starting where engine uses plain journal bearings on the crankshaft. Rolling friction is negligible compared to total friction.

13.3.4 Dry Friction

Even when an engine is not operated for a long time there is little possibility for direct metal to metal contact. Always some lubricant exists between the rubbing surfaces even after long periods of disuse. One can take the dry friction to be non-existent and hence, this can be safely neglected while considering engine friction.

13.3.5 Journal Bearing Friction

A circular cylindrical shaft called journal rotates against a cylindrical surface called the bearing. Journal bearings are called partial when

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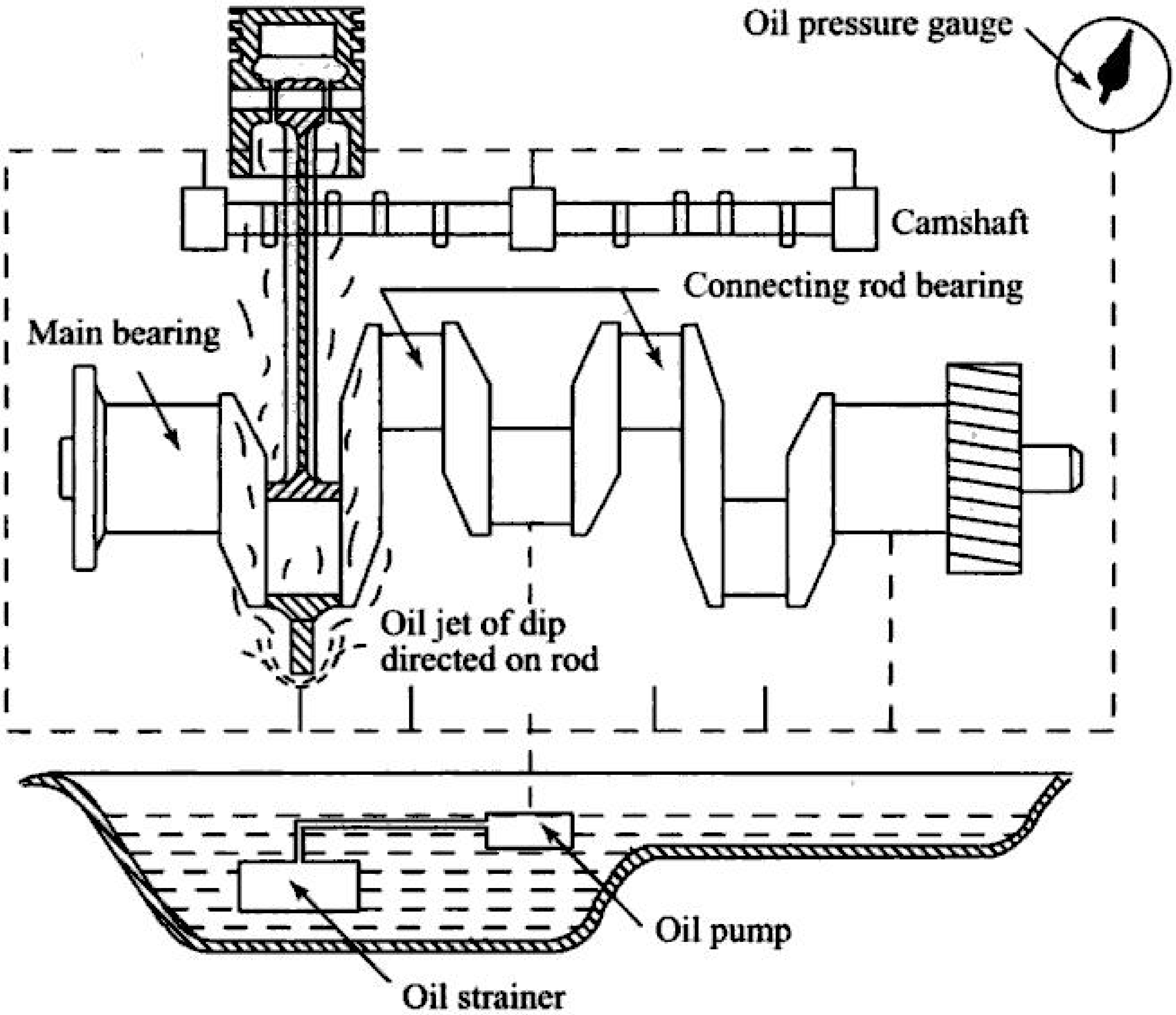

Fig. 13.10 Splash and Pressure Lubrication System

piston pins and the main and connecting rod bearings. The basic components of the wet sump lubrication systems are (i) pump (ii) strainer (iii) pressure regulator (iv) filter (v) breather.

A typical wet sump and its components are shown in Fig.13.12. Oil is drawn from the sump by a gear or rotor type of oil pump through an oil strainer. The strainer is a fine mesh screen which prevents foreign particles from entering the oil circulating systems. A pressure relief valve is provided which automatically keeps the delivery pressure constant and can be set to any value. When the oil pressure exceeds that for which the valve is set, the valve opens and allows some of the oil to return to the sump thereby relieving the oil pressure in the systems. Fig.13.13 shows a typical gear pump, pressure relief valve and by-pass. Most of the oil from the pump goes directly to the engine bearings and a portion of the oil passes through a cartridge filter which removes the solid particles from the oil.

neutralize acids formed by oxidation, or, it may be non-alkaline and protect the metal by forming a surface film.

Some additives may unite with oxygen, either preferentially to the oil or else with some already oxidized portion of the oil or fuel contaminant. Other additives might act as metal deactivators and as corrosion shields by chemically combining with the metal. Thus a thin sulphide or phosphide coating on the metal deactivates those metals that act as catalysts while protecting other metals from corrosive attack. Zinc ditinophosphate serves as an anti-oxidant and anticorrosive additive.

13.12.2 Detergent-Dispersant

This type of additives improve the detergent action of the lubricating oil. These additives might be metallic salts or organic acids. The action due to the additive may arise either from direct chemical reaction or from polar attraction. Thus the additives may chemically combine with the compounds in the oil that would otherwise form sludge and varnish. On the other hand, if the additive and the deposits in the engine are polar compounds, the detergent action may arise from neutralization of the electric moment of the deposit molecules with that of the additive. In this manner the deposit could be neutralized and would not tend to cling to other molecules or to the surface. Therefore, agglomeration of deposits would be prevented.

13.12.3 Extreme Pressure Additives

At high loads and speeds with high surface temperatures, an extreme pressure additive is necessary. Such additives interact with the metal surface to form a complex inorganic film containing iron, oxygen, carbon and hydrogen. Welding is prevented by the presence of the film.

13.12.4 Pour Point Depressors

In order to obtain fluidity or flow of oil at low temperatures, pour point depressants are added to the lubricating oils to lower the pour point. An engine having a lubricant with higher pour point will not get adequate lubrication during starting at low ambient temperatures and excessive wear would result. These additives tend to prevent the formation of wax at the low temperatures encountered in starting.

the cylinder gases takes place through the gas layer and through the cylinder walls to the cooling medium. A large temperature drop is produced in the stagnant layer adjacent to the walls.

The peak cylinder gas temperature may be 2800 K while the temperature of the cylinder inner wall surface may be only 450 K due to cooling, (Fig.14.3). Heat is transferred from the gases to the cylinder walls when the gas temperature is higher than the wall temperature. The rate and direction of flow of heat varies depending upon the temperature differential. If no cooling is provided, there could be no heat flow, so that the whole cylinder wall would soon reach an average temperature of the cylinder gases. By providing adequate cooling, the cylinder wall temperature can be maintained at optimum level.

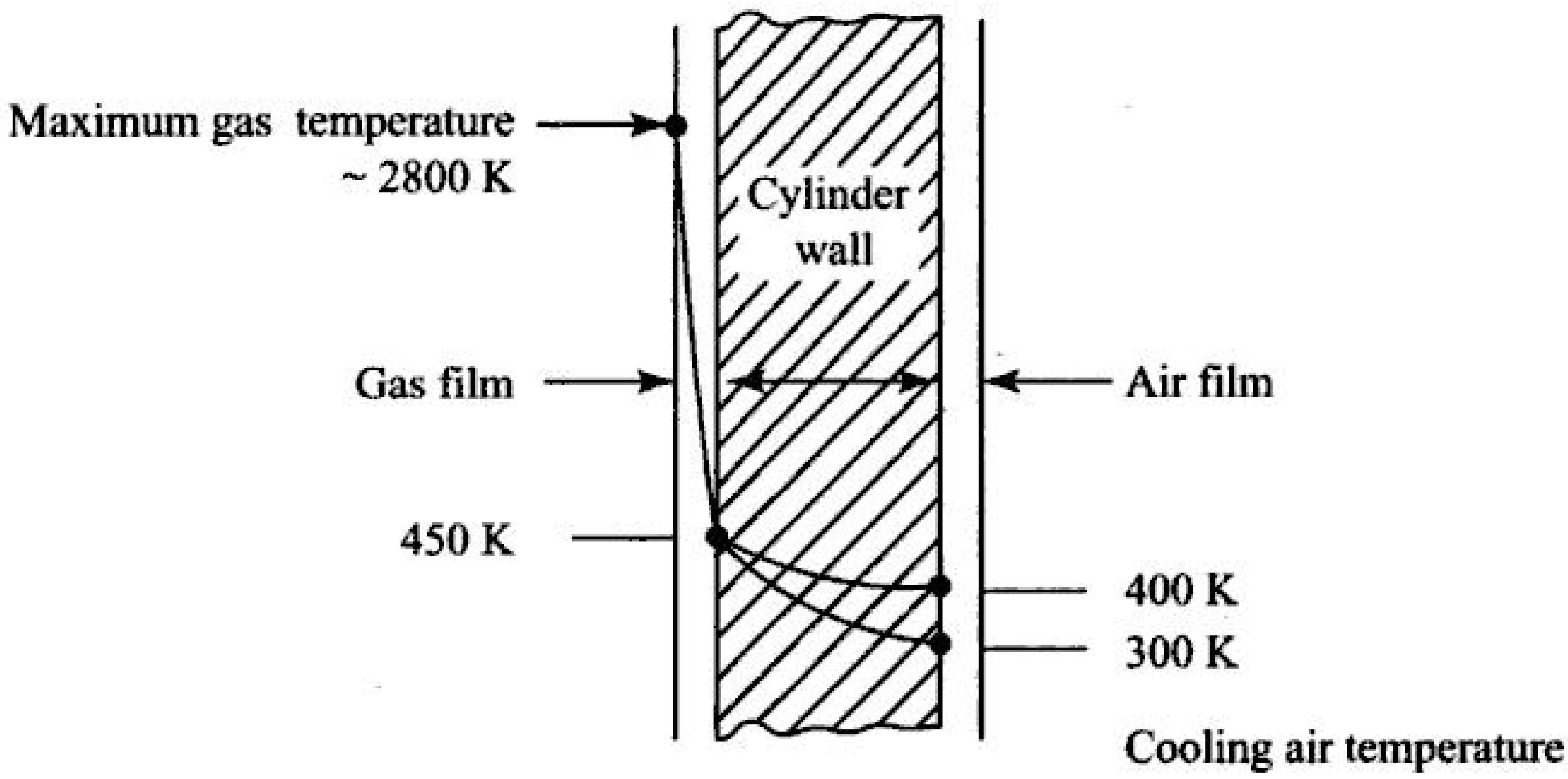

Fig. 14.3 Cylinder Wall Temperature Distribution of a Properly Cooled Cylinder

14.5 HEAT TRANSFER

Heat transfer occurs when a temperature difference exists. As a result of combustion, high temperatures are produced, inside the engine cylinder. Considerable heat flow occurs from the gases to the surrounding metal walls. In addition to this the shearing of the oil film (separating the bearing surfaces) transforms available energy into internal energy of the oil film. This increases the temperature of oil film and results in heat transfer from the oil to the bearing surfaces. However, the heat transfer on this account is quite small.

the engine as a result of the combustion process. A large portion of the heat from the gases of combustion is transferred to the cylinder head and walls, piston and valves. Unless this excess heat is carried away and these parts are adequately cooled, the engine will be damaged. A cooling system must be provided not only to prevent damage to the vital parts of the engine, but the temperature of these components must be maintained within certain limits in order to obtain maximum performance from the engine. Adequate cooling is then a fundamental requirement associated with reciprocating internal combustion engines. Hence, a cooling system is needed to keep the engine from not getting so hot as to cause problems and yet to permit it to run hot enough to ensure maximum efficiency of the engine. The duty of cooling system, in other words, is to keep the engine from getting not too hot and at the same time not to keep it too cool either!

14.10 CHARACTERISTICS OF AN EFFICIENT COOLING SYSTEM

The following are the two main characteristics desired of an efficient cooling system:

(i) It should be capable of removing about 30% of heat generated in the combustion chamber while maintaining the optimum temperature of the engine under all operating conditions of the engine.

(ii) It should remove heat at a faster rate when engine is hot. However, during starting of the engine the cooling should be minimum, so that the working parts of the engine reach their operating temperatures in a short time.

14.11 TYPES OF COOLING SYSTEMS

In order to cool the engine a cooling medium is required. This can be either air or a liquid. Accordingly there are two types of systems in general use for cooling the IC engines. They are

(i) liquid or indirect cooling system

(ii) air or direct cooling system

gets heated up quickly. When the preset temperature is reached the thermostat allows the water to flow through the radiator. Usually a Bellow type thermostat is used, the details of which are shown in Fig.14.12. In modern engines, a wax-element type thermostat is normally employed.

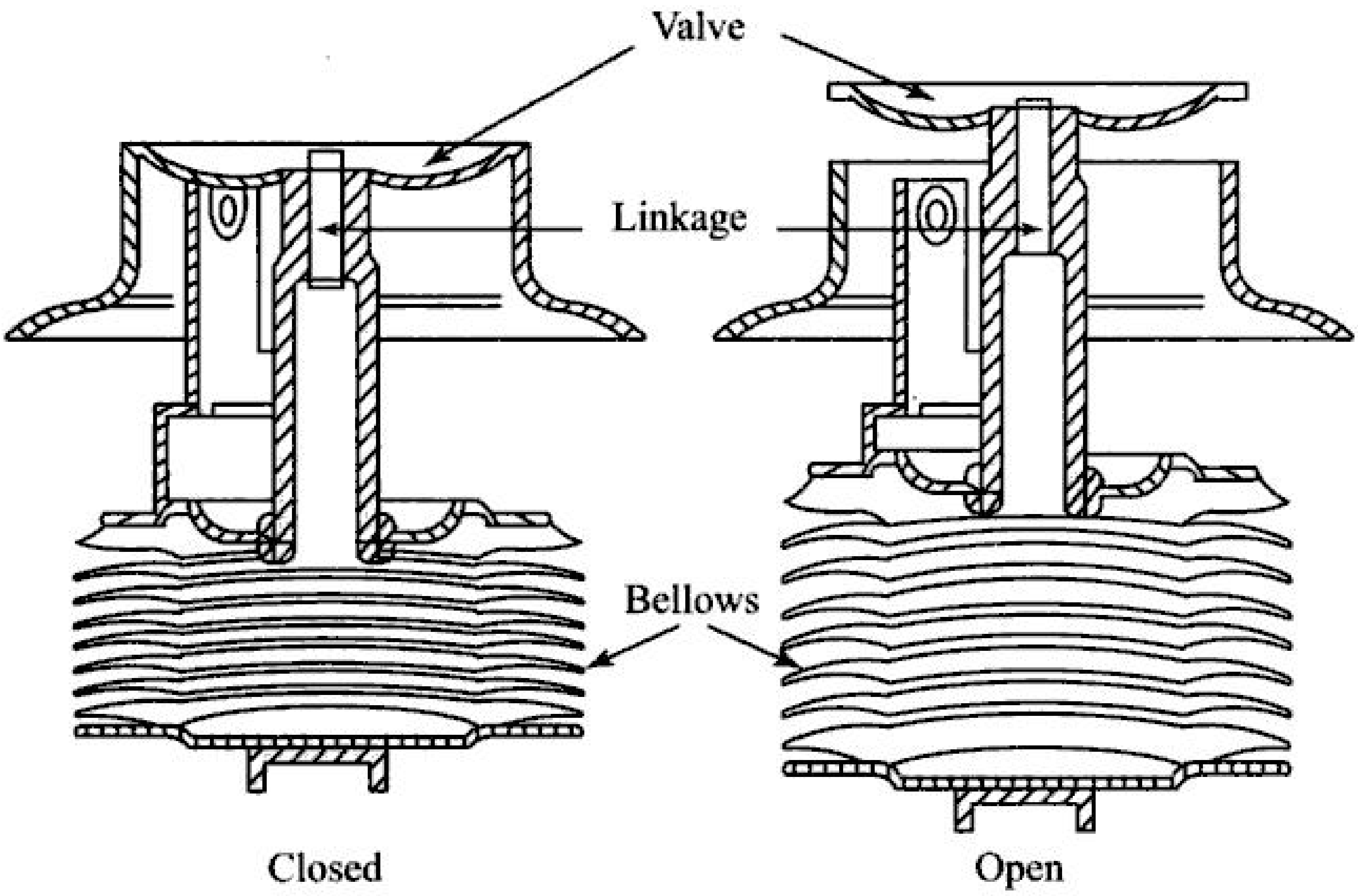

Fig. 14.12 Bellows Type Thermostat

The unit consists of a closed bellows with volatile liquid under reduced pressure. When the bellows is heated the liquid vaporizes and creates enough pressure to expand the bellows. The movement of bellows operates a linkage which opens the valve. When the unit is cooled, the gas condenses, the pressure is reduced and the bellows collapses to close the valve.

14.12.4 Evaporative Cooling System

This system is predominantly used in stationary engines. In this, the engine will be cooled because of the evaporation of the water in the cylinder jackets into steam. Here, the advantage is taken from the high latent heat of vapourization of water by allowing it to evaporate in the cylinder jackets. If the steam is formed at a pressure above atmospheric the temperature will be above the normal permissible temperature.

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15

ENGINE EMISSIONS AND THEIR CONTROL

15.1 INTRODUCTION

Internal combustion engines generate undesirable emissions during the combustion process. In this, both SI and CI engines are equally responsible for the same. The emissions exhausted into the surroundings pollute the atmosphere and causes the following problems

(i) global warming

(ii) acid rain

(iii) smog

(iv) odours

(v) respiratory and other health hazards

The major causes of these emissions are non-stoichiometric combustion, dissociation of nitrogen, and impurities in the fuel and air. The emissions of concern are: unburnt hydrocarbons (HC), oxides of carbon (CO_x), oxides of nitrogen (NO_x), oxides of sulphur (SO_x), and solid carbon particulates.

It is the dream of engineers and scientists to develop engines and fuels such that very few quantity of harmful emissions are generated, and these could be let into the surroundings without a major impact on the environment. However, with the present technology this is not possible, and after-treatment of the exhaust gases as well as in-cylinder reduction of emissions are very important. In case of after-treatment it consists mainly of the use of thermal or catalytic

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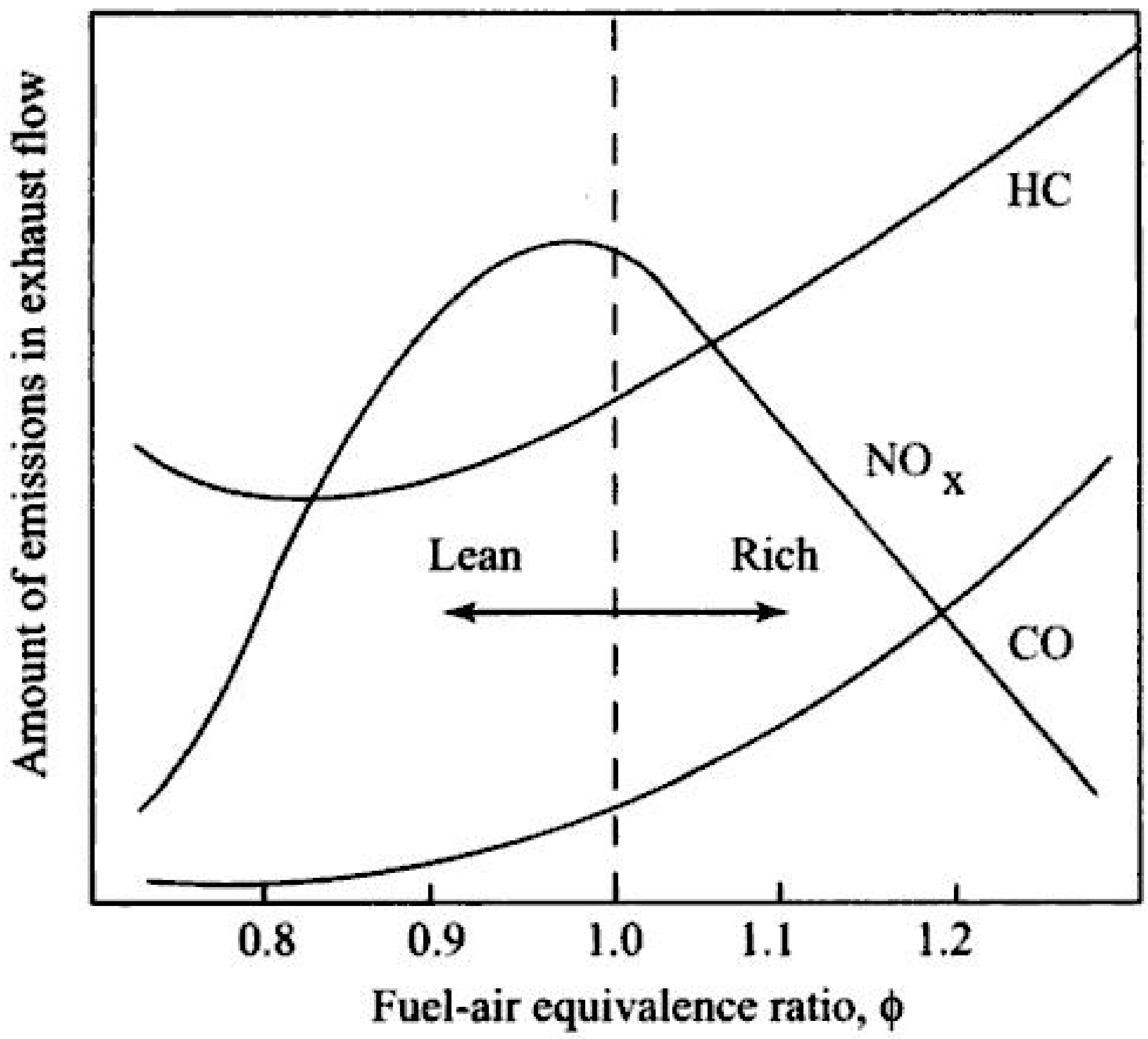

Fig. 15.1 Emissions as a Function of Equivalence Ratio for an SI Engine

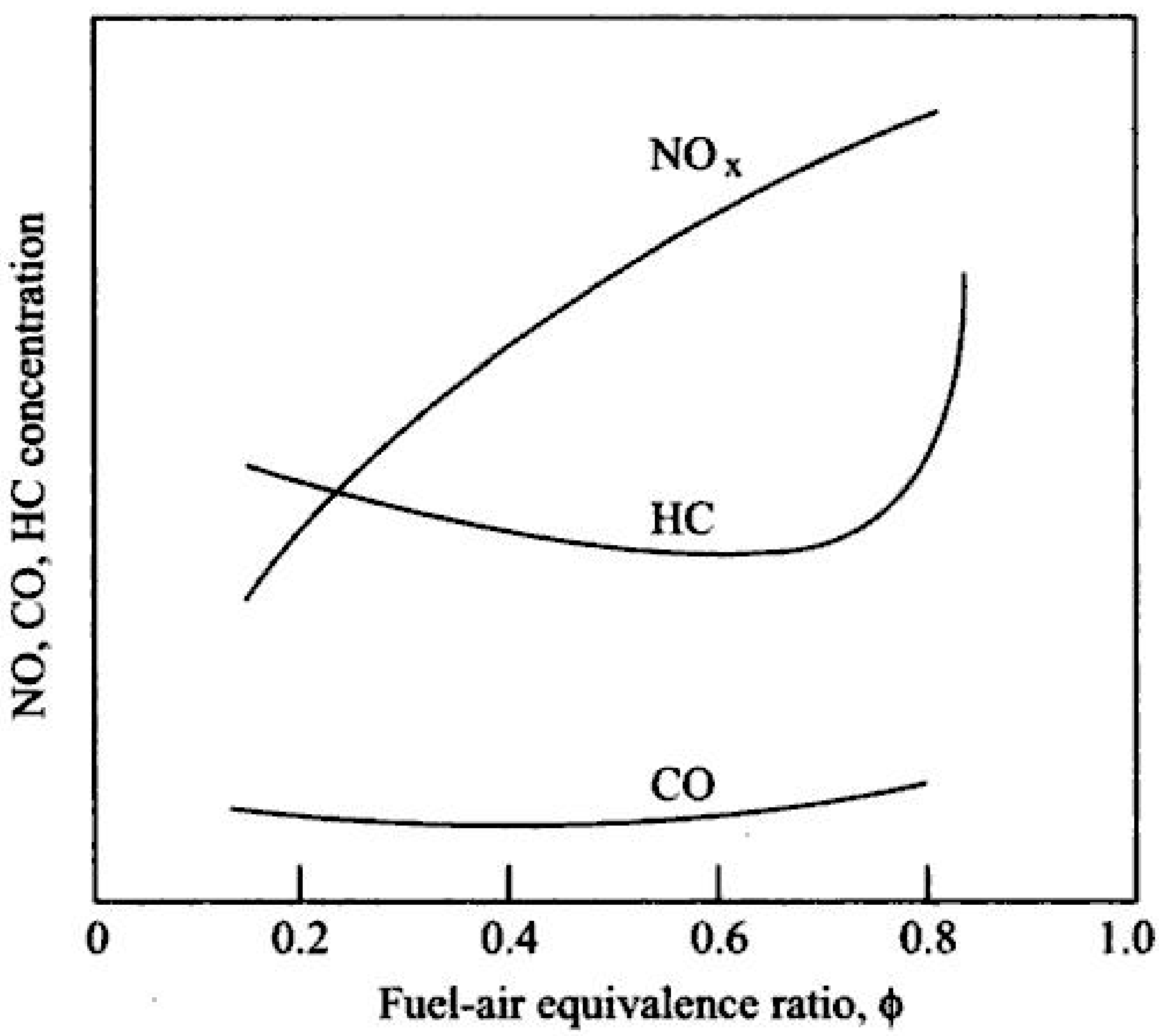

Fig. 15.2 Emissions as a Function of Equivalence Ratio for a CI Engine

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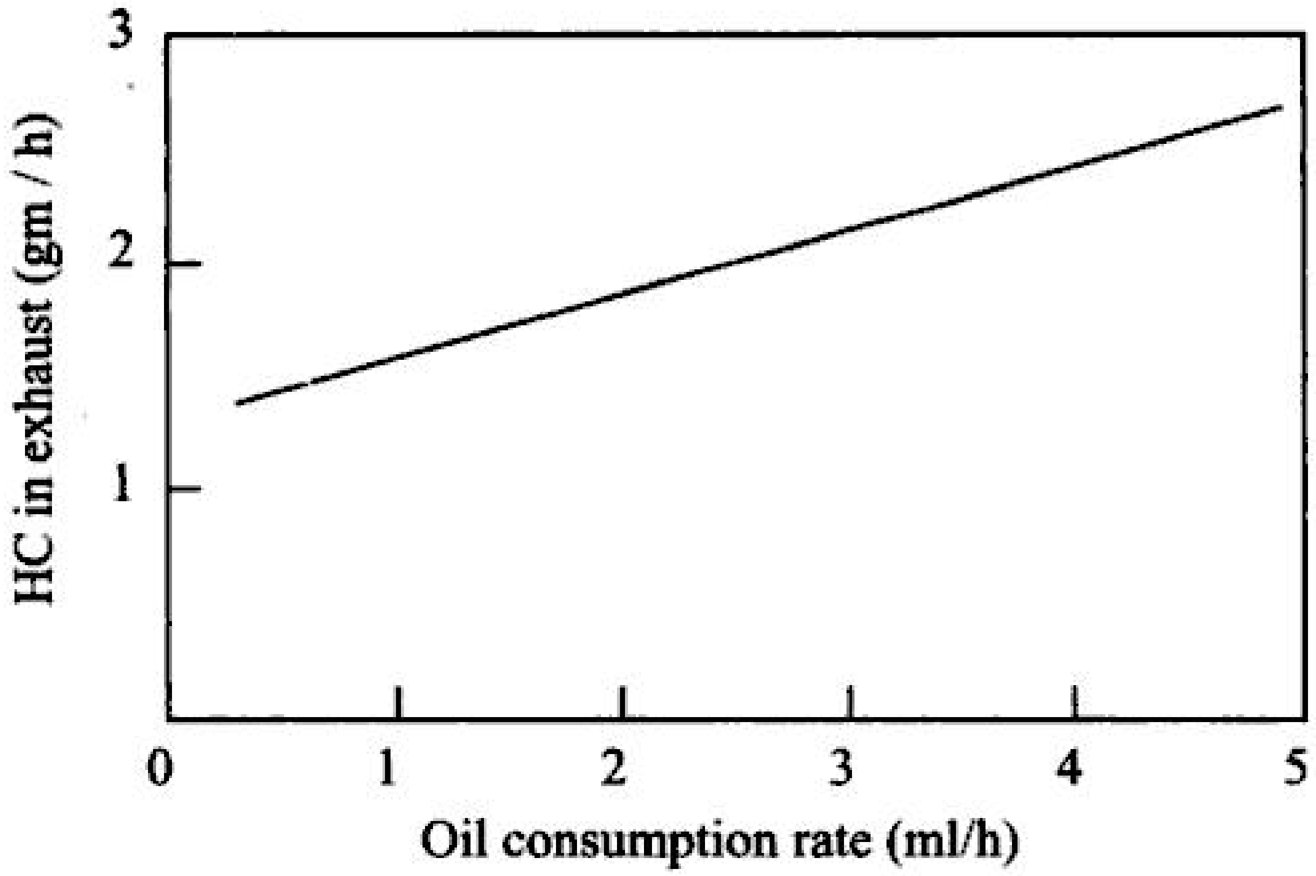


Fig. 15.4 HC Exhaust Emissions as a Function of Engine Oil Consumption

Often as an engine ages, due to wear, clearance between the pistons and cylinder walls increases. This increases oil consumption contributes to an increase in HC emissions in three ways:

(i) there is added crevice volume,

(ii) there is added absorption-desorption of fuel in the thicker oil film on cylinder walls, and

(iii) there is more oil burned in the combustion process.

15.7 HYDROCARBON EMISSION FROM TWO-STROKE ENGINES

Older two-stroke SI engines and many modern small two-stroke SI engines add HC emissions to the exhaust during the scavenging process. The intake air-fuel mixture is used to push exhaust residual out of the open exhaust port which is called scavenging. When this is done, some of the air and fuel mix with the exhaust gases and are expelled out of the cylinder before the exhaust port closes. This can be a major source of HC in the exhaust and is one of the major reasons why there have been no modern two-stroke cycle automobile engines.

Some experimental automobile two-stroke cycle engines and just about all small engines use crankcase compression, and this is a second source of hydrocarbon emissions. The crankcase area and pistons

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

$$O + O_2 \rightarrow O_3 \tag{15.11}$$

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

15.11 PARTICULATES

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

Soot particles are clusters of solid carbon spheres. These spheres have diameters from 9 nm to 90 nm (1 nm $= 10^{-9}$ m). But most of them are within the range of 15-30 nm. The spheres are solid carbon with HC and traces of other components absorbed on the surface. A single soot particle may contain up to 5000 carbon spheres.

Carbon spheres are generated in the combustion chamber in the fuel-rich zones where there is not enough oxygen to convert all carbon to CO_2 :

$$C_x\, H_y + z\, O_2 \rightarrow a\, CO_2 + b\, H_2\, O + cCO + dC(s) \tag{15.12}$$

Then, as turbulence and mass motion continue to mix the components in the combustion chamber, most of these carbon particles $[C(s)]$ find sufficient oxygen to further react and are converted to CO_2:

$$C(s) + O_2 \rightarrow CO_2 \tag{15.13}$$

More than 90 to 95% of carbon particles originally generated within an engine are thus converted to CO_2 and never comes out as carbon particles. It should be remembered that if CI engines are made to

temperature needed to sustain these oxidation processes is reduced to 250–300 °C, making for a much more attractive system.

A catalyst is a substance that accelerates a chemical reaction by lowering the energy needed for it to proceed. The catalyst is not consumed in the reaction and so functions indefinitely unless degraded by heat, age, contaminants, or other factors. Catalytic converters are chambers mounted in the flow system through which the exhaust gases pass through. These chambers contain catalytic material, which promotes the oxidation of the emissions contained in the exhaust flow.

Generally, catalytic converters are called three-way converters because they are used to reduce the concentration of CO, HC, and NO_x in the exhaust. Catalytic converter is usually a stainless steel container mounted somewhere along the exhaust pipe of the engine. Inside the container is a porous ceramic structure through which the exhaust gas flows. In most converters, the ceramic is a single honeycomb structure with many flow passages (Fig.15.7). Some converters use loose granular ceramic with the gas passing between the packed spheres. Volume of the ceramic structure of a converter is generally about half the displacement volume of the engine. This results in a volumetric flow rate of exhaust gas such that there are 5 to 30 changeovers of gas each second, through the converter. Catalytic converters for CI engines need larger flow passages because of the solid soot in the exhaust gases.

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

Palladium and platinum promote the oxidation of CO and HC as in Eqs.15.19 and 15.20, with platinum especially active in the hydrocarbon reaction. Rhodium promotes the reaction of NO_x in one or more of the following reactions:

$$NO + CO \rightarrow \frac{1}{2} N_2 + CO_2 \qquad (15.21)$$

$$2\,NO + 5\,CO + 3\,H_2O \rightarrow 2\,NH_3 + 5\,CO_2 \qquad (15.22)$$

$$2\,NO + CO \rightarrow N_2O + CO_2 \qquad (15.23)$$

Small amounts of lead impurities are found in some fuels, and 10 to 30% of this ends up in the catalytic converter. Up until the early 1990s leaded gasoline was quite common. Note that leaded gasoline cannot be used in engines equipped with catalytic converters. Use of leaded gasoline filled two times (full tank) would completely poison a converter and make it totally useless. To reduce the chances of

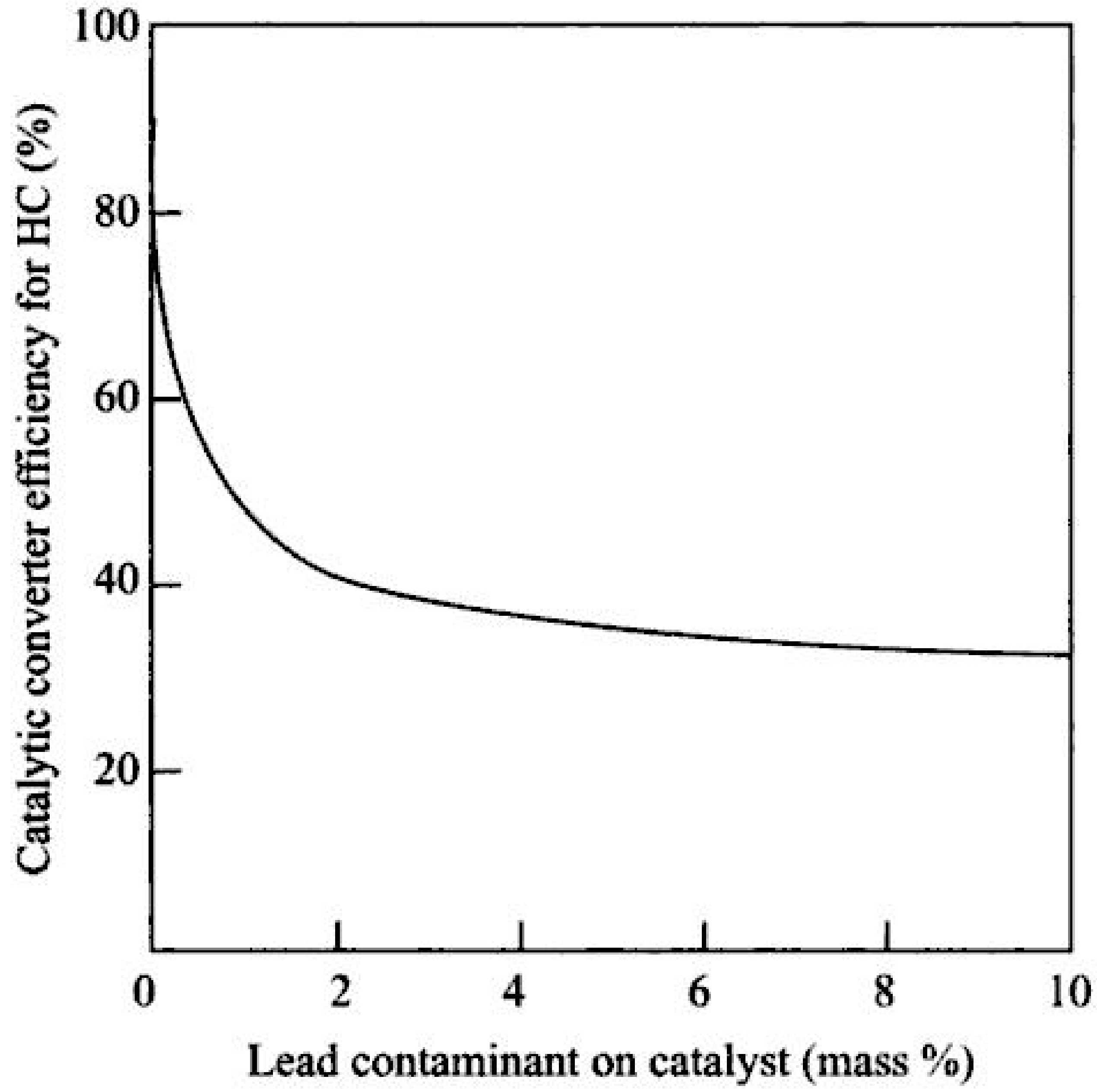


Fig. 15.10 Reduction of catalytic Converter Efficiency due to Contamination by Lead

accidently using leaded gasoline with a catalytic converter, the fuel pump nozzle size and the diameter of the fuel tank inlet are made smaller for unleaded gasoline.

15.14.1 Sulphur

Sulphur offers unique problems for catalytic converters. Some catalysts promote the conversion of SO_2 to SO_3, which eventually gets converted to sulphuric acid. This degrades the catalytic converter and contributes to acid rain. New catalysts are being developed that promote the oxidation of HC and CO but do not change SO_2 to SO_3. Some of these create almost no SO_3 if the temperature of the converter is kept less than 400 °C.

15.14.2 Cold Start-Ups

As can be seen from Fig.15.8 catalytic converters are not very efficient when they are cold. When an engine is started after not being operated for several hours, it takes several minutes for the converter to reach an efficient operating temperature. The temperature at which a converter becomes 50% efficient is defined as the light-off temperature, and this is in the range of about 250–300 °C.

A large percentage of automobile travel is for short distances where the catalytic converter never reaches efficient operating temperature, and therefore emissions are high. Unfortunately, most short trips occur in cities where high emissions are more harmful. Further, all engines use a rich mixture when starting. Otherwise cold start-ups pose a major problem.

It is estimated that cold start-ups are the source of 70-90% of all HC emissions. A major reduction in emissions is therefore possible if catalytic converters could be preheated, at least to light-off temperature, before engine startup. Preheating to full steady-state operating temperature would be even better. Several methods of pre-heating have been tried with varying success. Because of the time involved and amount of energy needed, most of these methods preheat only a small portion of the total converter volume. This small section is large enough to treat the low exhaust flow rate which usually occurs at startup and immediately following. By the time higher engine speeds are used, most of the catalytic converter has been heated by the hot exhaust gas, and the higher flow rates are fully treated. Methods of catalytic converter preheating include the following.

(i) by locating the converter close to the engine

(ii) by having superinsulation

(iii) by employing electric preheating

(iv) by using flame heating

(v) incorporating thermal batteries

15.15 CI ENGINES

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

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

15.15.1 Particulate Traps

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

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

Another way of lowering the ignition temperature of the carbon soot and promoting self-regeneration in traps is to use catalyst additives in the diesel fuel. These additives generally consist of copper compounds or iron compounds, with about 6 to 8 grams of additive in 1000 liters of fuel is usually normal. To keep the temperatures high enough to self-regenerate in a catalytic system, traps can be mounted as close to the engine as possible, even before the turbocharger.

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

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

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

15.15.2 Modern Diesel Engines

Carbon soot particulate generation has been greatly reduced in modern CI engines by advanced design technology in fuel injectors and combustion chamber geometry. With greatly increased mixing efficiency and speeds, large regions of fuel- rich mixtures can be avoided when combustion starts. These are the regions where carbon soot is generated, and by reducing their volume, far less soot is generated. Increased mixing speeds are obtained by a combination of indirect injection, better combustion chamber geometry, better injector design and higher pressures, heated spray targets, and air-assisted injectors. Indirect injection into a secondary chamber that promotes high turbulence and swirl greatly speeds the air-fuel mixing process. Better nozzle design and higher injection pressures create finer fuel droplets which evaporate and mix quicker. Injection against a hot surface speeds evaporation, as do air-assisted injectors. Some modern, top-of-the-line CI automobile engines have reduced particulate generation enough that they meet stringent standards without the need for particulate traps.

15.16 REDUCING EMISSIONS BY CHEMICAL METHODS

Development work has been done on large stationary engines using cyanuric acid to reduce NO_x emissions. Cyanuric acid is a low-cost solid material that sublimes in the exhaust flow. The gas dissociates, producing isocyanide that reacts with NO_x to form N_2, H_2O, and CO_2. Operating temperature is about 500 °C. Up to 95% NO_x reduction can be achieved with no loss of engine performance. At present, this system is not practical for automobile engines because of its size, weight, and complexity.

Research is being done using zeolite molecular sieves to reduce NO_x emissions. These are materials that absorb selected molecular compounds and catalyze chemical reactions. Using both SI and CI engines, the efficiency of NO_x reduction is being determined over a range of operating variables, including air-fuel ratio, temperature, flow velocity, and zeolite structure. At present, durability is a serious limitation with this method.

Various chemical absorbers, molecular sieves, and traps are being tested to reduce HC emissions. HC is collected during engine startup time, when the catalytic converter is cold, and then later released back into the exhaust flow when the converter is hot. The converter then efficiently burns the HC to H_2O and CO_2. A 35% reduction of cold-start HC has been achieved.

H_2S emissions occur under rich operating conditions. Chemical systems are being developed that trap and store H_2S when an engine operates rich and then convert this to SO_2 when operation is lean and excess oxygen exists. The reaction equation is

$$H_2S + O_2 \quad \rightarrow \quad SO_2 + H_2 \tag{15.28}$$

15.16.1 Ammonia Injection Systems

Some large marine engines and stationary engines reduce NO_x emissions with an injection system that sprays NH_3 into the exhaust flow.

In the presence of a catalyst, the following reactions occur

$$4\ NH_3 + 4\ NO + O_2 \rightarrow 4\ N_2 + 6\ H_2O \qquad (15.29)$$

$$6\ NO_2 + 8\ NH_3 \rightarrow 7\ N_2 + 12\ H_2\ O \qquad (15.30)$$

Careful control must be adhered to, as NH3 itself is an undesirable emission.

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

15.17 EXHAUST GAS RECIRCULATION (EGR)

The most effective way of reducing NO_x emissions is to hold combustion chamber temperatures down. Although practical, this is a very unfortunate method in that it also reduces the thermal efficiency of the engine. Those who have undergone thermodynamics course are aware that to obtain maximum engine thermal efficiency it should be operated at the highest temperature possible.

Probably the simplest and practical method of reducing maximum flame temperature is to dilute the air-fuel mixture with a non-reacting parasite gas. This gas absorbs energy during combustion without contributing any energy input. The net result is a lower flame temperature. Any nonreacting gas would work as a diluent, as shown in Fig.15.11. Those gases with larger specific heats would absorb the most energy per unit mass and would therefore require the least amount; thus less CO_2 would be required than argon for the same maximum temperature. However, neither CO_2 nor argon is readily available for use in an engine. Air is available as a diluent but is not totally nonreacting. Adding air changes the air-fuel ratio and combustion characteristics. The one nonreacting gas that is available to use in an engine is exhaust gas, and this is used in all modern automobile and other medium-size and large engines.

Adding any non reacting neutral gas to the inlet air-fuel mixture reduces flame temperature and NO_x generation, Exhaust gas (EGR) is the one gas that is readily available for engine use.

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

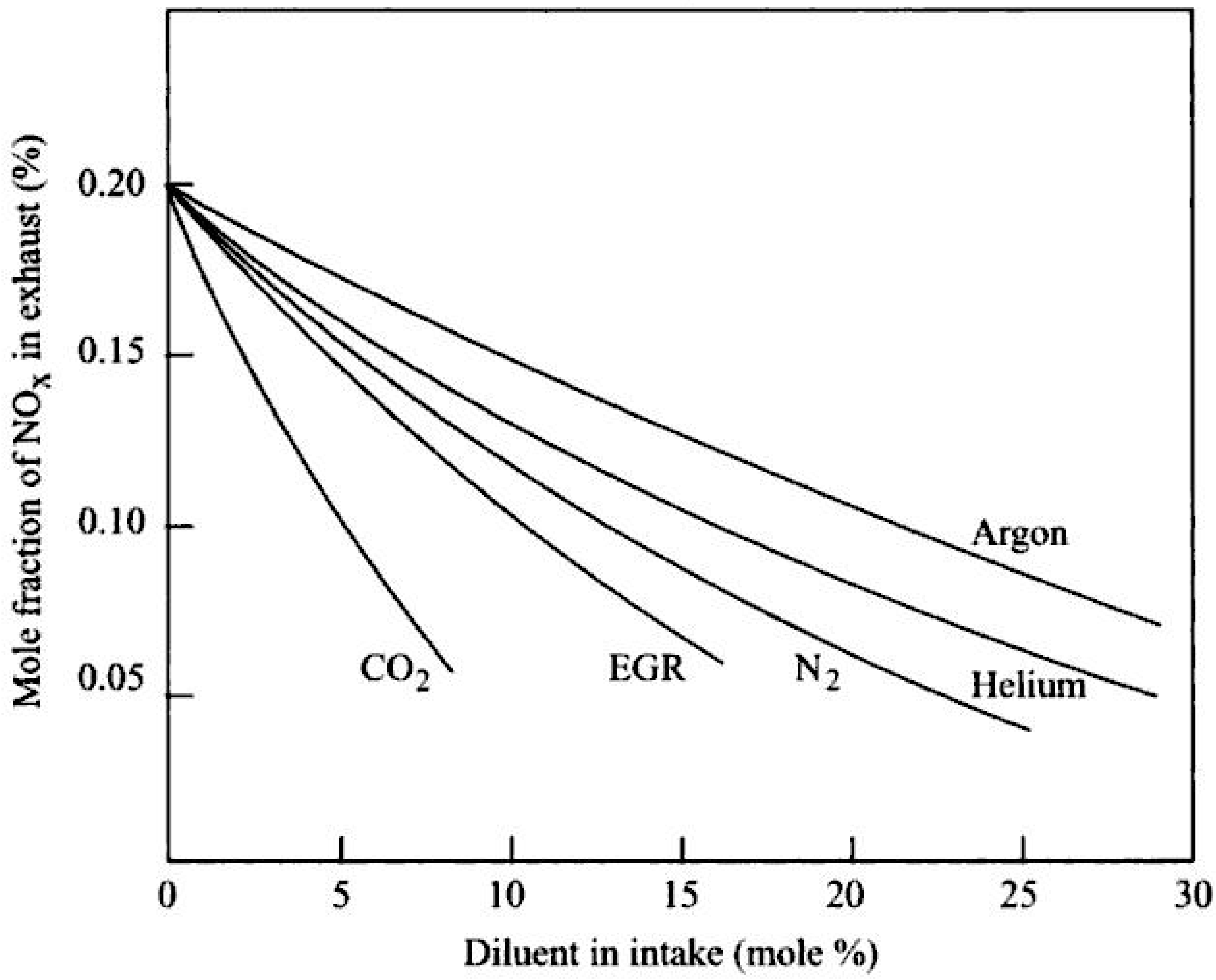


Fig. 15.11 NO_x Reduction using Non-combustible Diluent Gas to Intake Mixture

Management System. EGR is defined as a mass percent of the total intake flow

$$\mathrm{EGR} \quad = \quad \left(\frac{\dot{m}_{\mathrm{EGR}}}{\dot{m}_{cyl}}\right) \times 100 \tag{15.31}$$

where *cyl* is the total mass flow into the cylinders.

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

$$x_{ex} \quad = \quad \left(\frac{\mathrm{EGR}}{100}\right) \times (1 - x_r) + x_r \tag{15.32}$$

where x_r is the exhaust residual from previous cycle.

Not only does EGR reduce the maximum temperature in the combustion chamber, but it also lowers the overall combustion efficiency. Increase in EGR results in some cycle partial burns and, in the extreme, total misfires. Thus, by using EGR to reduce NO_x emissions, a costly price of increased HC emissions and lower thermal efficiency must be paid.

of the tank increases and since the tank is vented to atmosphere the vapour will flow out of the vent. This outflow of the vapour will increase if in addition to temperature rise of the liquified gasoline the vapour temperature is also increased.

The evaporation from the tank is affected by a large number of variables of which the ambient and fuel tank temperature, the mode of vehicle operation, the amount of fuel in the tank and the volatility of the fuel are important. Other significant factors are the capacity, design and location of the fuel tank with respect to the exhaust system and the flow pattern of the heated air underneath the vehicle.

Less the tank fill, greater is the evaporation loss. An approximate picture of the effect of tank fill and temperature are given in Table 15.2, This reflects the difference in the tank vapour space. Also when a vehicle is parked in a hot location the evaporation of the gasoline in the tank accelerates, so the loss due to evaporation are high.

Table 15.2 Effect of Tank Fill on Evaporation Loss

Tank fill	Ambient temperature (°C)	Loss during operation (%)
Quarter	10	5.8
Half	15	1.1
Three-fourth	18	0.1
Full	22	0.0

The operational mode substantially affect the evaporation loss. When the tank temperature rises the loss increases. The vapour which goes out from a partially filled tank during vehicle operation called soak, is a mixture of air and hydrocarbon. After a prolonged high speed operation the HC per cent in the soak is as high as 60 per cent as compared to about 30 per cent after an overnight soak. The fuel composition also affects the tank losses. About 75 per cent of the HC loss from tank are C_4 and C_5 hydrocarbons.

Carburettor losses: The operation of an engine depends on the level of gasoline in the float chamber inside the carburettor. The engine stops running when the gasoline has been completely utilized. Heat produced by the engine causes evaporation of some quantity of gasoline from the float chamber. The evaporation of gasoline constitutes the main reason for the loss of gasoline from the carburettor.

The operation of the purge control valve is taken care of by the exhaust back pressure. Under idling conditions the fuel supply is cut off so that the level of HC can be reduced.

The ELCD completely controls all types of evaporative losses. However, the tolerance of the carburettor for supplying fuel-air ratio reduces to about 3 per cent only. This requires very accurate metering control.

15.19 MODERN EVAPORATIVE EMISSION CONTROL SYSTEM

A modern evaporative emission control system is illustrated in Fig.15.14. The fuel tank is fitted to the vapour-liquid separator which is in the form of a chamber on the fuel tank. Vapour from the fuel tank goes to the top of the separator where the liquid gasoline is separated and sent back to the fuel tank through the fuel return pipe. A vent valve or a vent hole is provided for the carburettor for the flow of fuel vapours. This vent hole is connected by a tube to the canister. Fuel vapours from the float chamber flow through the vent hole and the tube to the canister. The canister adsorbs the fuel vapours and stores them. *Adsorption* refers to the process of trapping of the gasoline vapours by the activated charcoal particles filled inside the canister. Vapour laden air from both the fuel tank and the carburettor passes through the canister. Hydrocarbons (HC) are left in the canister due to the process of adsorption, and air leaves from the canister into the atmosphere. When an engine is started, the inlet manifold sucks fresh air through the canister.

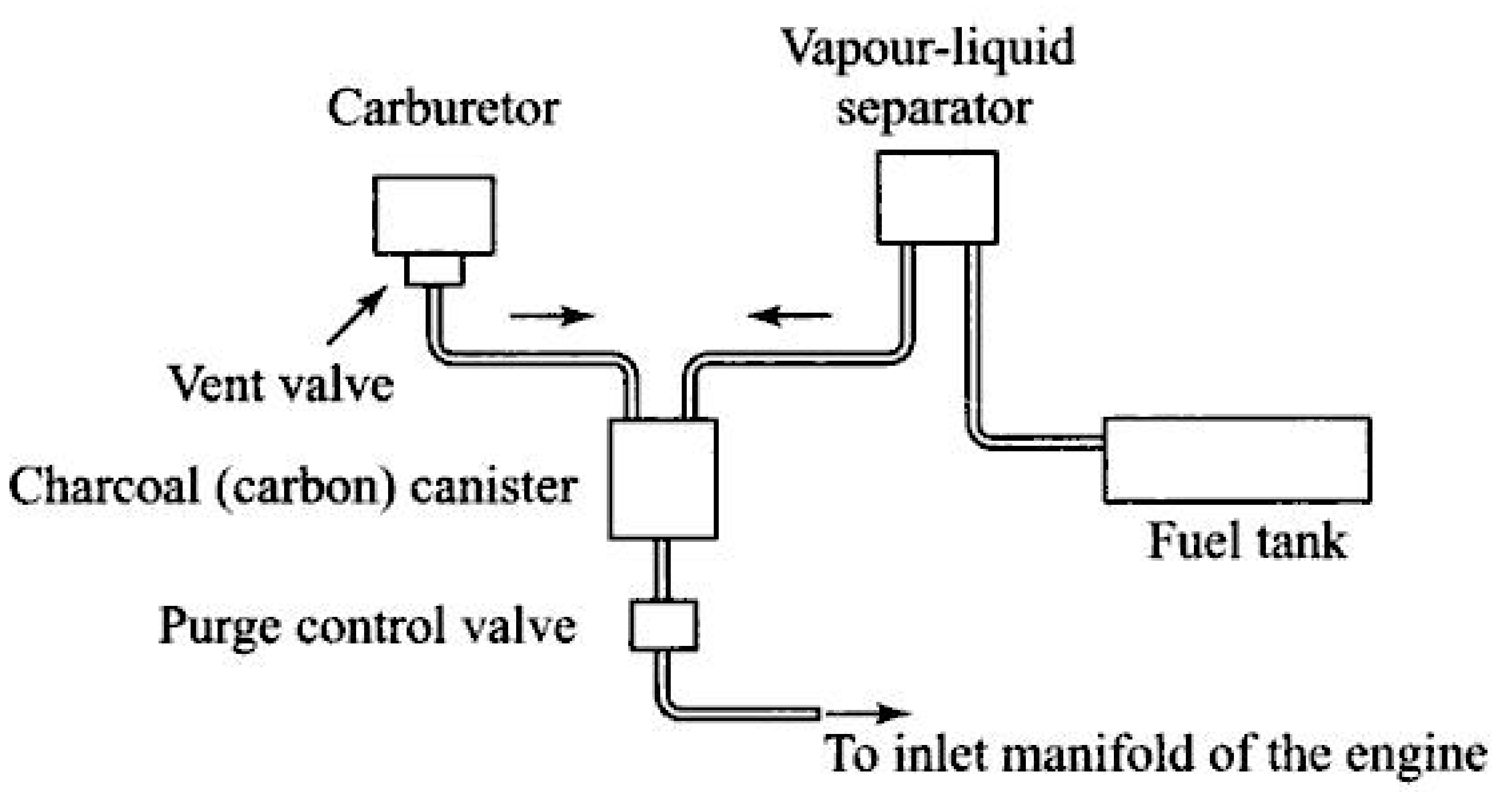


Fig. 15.14 Layout of Vapour Recovery System

The fresh air purges the gasoline vapours from the canister. 'Purging' is the process by which the gasoline vapours are removed from the charcoal particles inside the canister. The air carries the hydrocarbons (HC) through the purge control solenoid valve to the engine induction system. The purge control solenoid valve is controlled by the Electronic Control Module (ECM) of the Computer Command Control (CCC) system in modern automobiles.

15.19.1 Charcoal Canister

A charcoal canister used for trapping gasoline vapours is shown in Fig.15.15. This type of charcoal canister is used in the evaporative emission control system of a petrol engine. Fuel vapours from the float chamber of the carburetor enter into the canister though the left end passage. Fuel vapours from the fuel tank enter through the

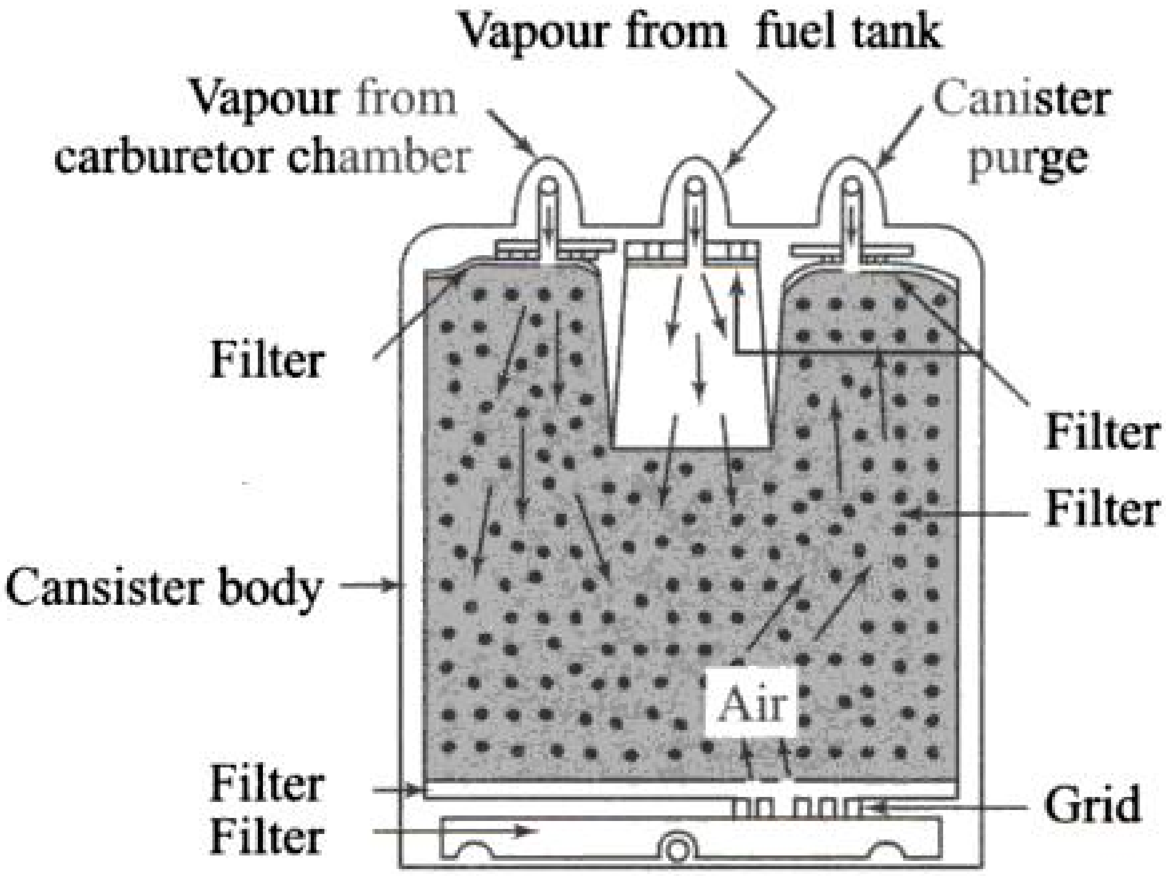


Fig. 15.15 Charcoal Canister

mid passage into the canister. The flow of these vapours is shown by the arrows pointing downwards. When the engine is not running, the fuel vapours flow in this manner . The fuel vapours are absorbed by the charcoal particles present in the canister. When the engine runs, air reaches the charcoal, canister due to the suction provided by the engine. This air carries away the hydrocarbons (HC) in the fuel vapours to the engine manifold. This purging action is shown at the right end of the charcoal canister by the arrows pointing upwards. As charcoal is a form of carbon, the charcoal canister is also called the carbon canister.

15.20 CRANKCASE BLOWBY

The blowby is the phenomenon of leakage past the piston and piston rings from the cylinder to the crankcase. The blowby HC emission are about 20 per cent of the total HC emission from the engine. This is increased to about 35 per cent if the rings are worn.

The blowby rate is greatly affected by the top land clearance and the position of the top ring because some of quenched gas is recycled in the combustion chamber and the ability of this to burn will depend on nearness to spark plug and the flame speed etc. and it will burn only when favourable conditions are there, otherwise it will go in the form of HC.

15.20.1 Blowby Control

The basic principle of all types or crankcase blowby control is recirculation of the vapours back to the intake air cleaner. There are a large number of different systems in use. Figure 15.16 shows typical closed or positive crankcase ventilation systems. In the PCV system the draft tube as shown in Fig.15.17 is eliminated and the blowby gases are rerouted back into the intake manifold or inlet of the carburettor. The blowby gases are consequently reintroduced into the combustion chamber where they are burned along with fresh incoming air and fuel. Since the blowby handling devices place the crankcase under a slight vacuum, they quickly became known as positive crankcase ventilation (PCV) systems.

15.20.2 Intake Manifold Return PCV System (Open Type)

Figure 15.16 (a) shows the intake manifold return PCV or open type system. It has a tube leading from crankcase or else the rocker arm cover through a flow control valve and into the intake manifold, usually, through an opening just below the carburettor. To provide proper ventilation of the interior of the engine, fresh air is usually drawn in through a rocker arm cover opposite that containing the PCV system. The closed PCV system Fig.15.16 (b) has a tube connected between the oil fill tube cap and the air cleaner, Both open and closed systems function in the same manner as long as the PCV valve remains unplugged. If the PCV valve plugs, using an open system, the blowby gases exhaust out of the oil fill tube cap and into the atmosphere. With PCV valve plugged it is no longer possible for fresh air crankcase ventilation to occur.

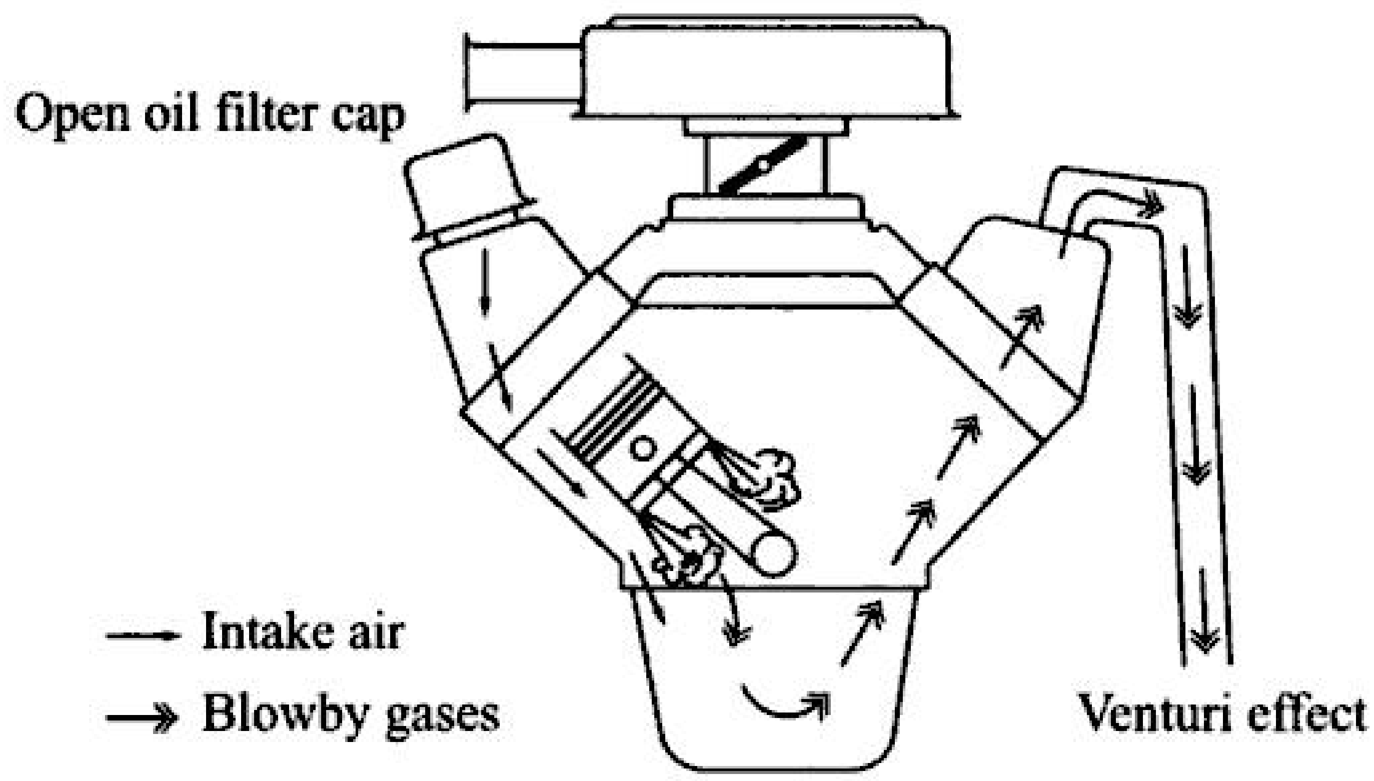


Fig. 15.17 Blowby Path without Positive Crankcase Ventilation

lift the lubricating oil. Also the carburettor has to be modified and adjusted to account for the charge coming from the crankcase in order to meter exact fuel-air ratio into the combustion chamber. The carburettor deposits and deposits on the blowby gas metering valve will significantly affect the performance of the carburettor. So high grade motor oil has to be used. In the closed ventilation system a provision is made for the blowby gases to escape to atmosphere in case of the metering valve failure.

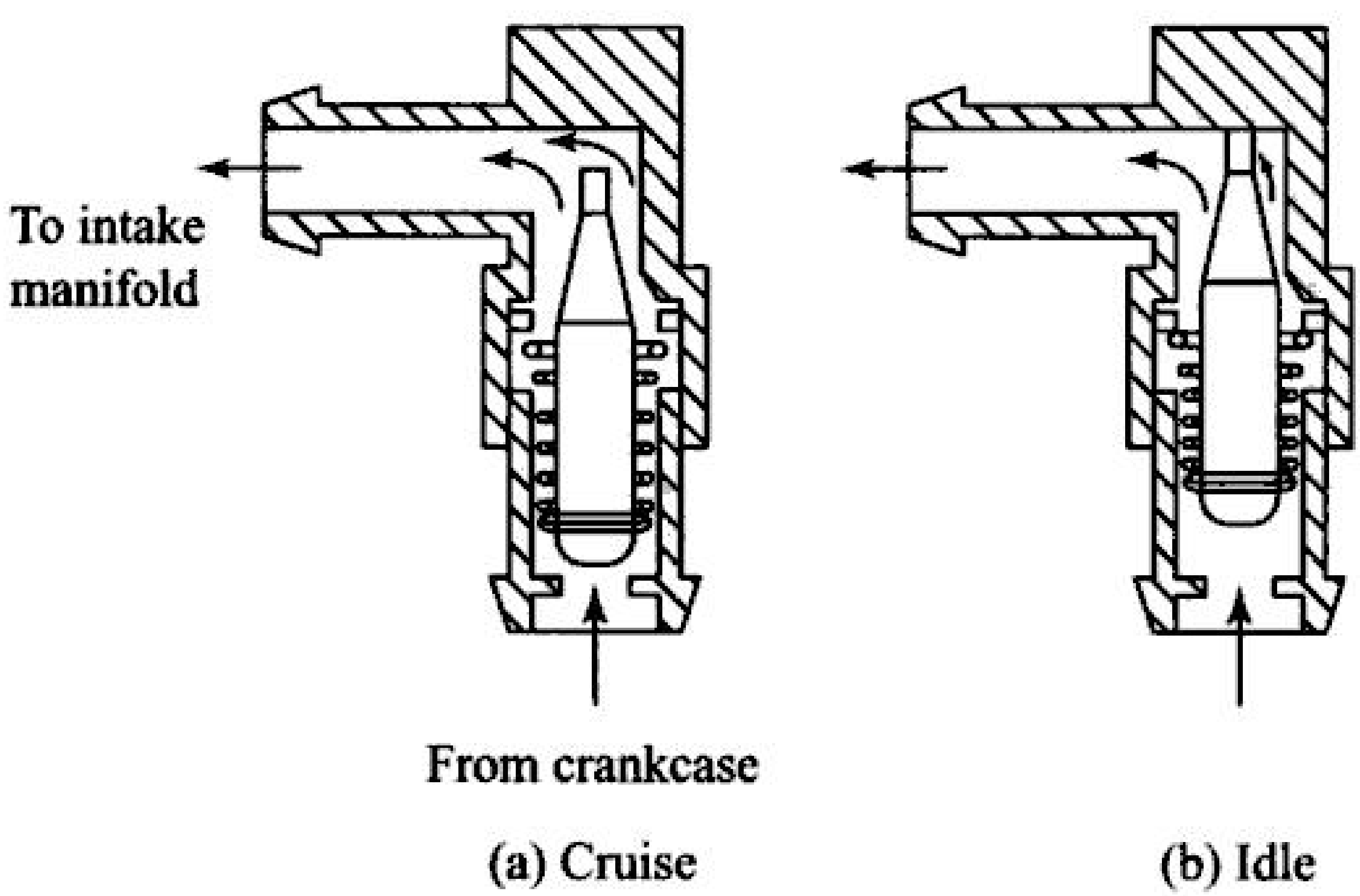


Fig. 15.18 Two Positions of PCV Valve

Review Questions

15.1 What are the problems created by exhaust emissions?

15.2 What causes the engine emissions?

15.3 Give a brief account of air pollution due to engines.

15.4 What are the major emissions that come out of engine exhaust?

15.5 Describe in detail the causes of hydrocarbon emissions from SI engines.

15.6 How knock emissions are caused and what are their effects on environment?

15.7 What are particulates? Describe in detail how particulate emissions are caused.

15.8 Give a brief account of other emissions from engines.

15.9 What is a thermal converter? How does it help to reduce emissions from engines?

15.10 What are catalytic converters? How are they helpful in reducing HC, CO and NO_x *emissions?*

15.11 Give a brief account of emissions from CI engines.

15.12 How can emissions be reduced using chemical methods?

15.13 What do you understand by the term EGR? Explain how EGR reduces NO_x *emission.*

15.14 Explain with a sketch the non-exhaust emission from a vehicle.

15.15 Explain with sketches how non-exhaust emission are controlled.

15.16 Explain with a neat sketch fuel system evaporation loss control device.

15.17 Give a layout of a vapour recovery system and explain.

15.18 With a neat sketch explain a charcoal canister for controlling non-exhaust emission.

15.19 What is crankcase blowby? How it is controlled?

15.20 Explain intake manifold open type PCV system.

16 MEASUREMENTS AND TESTING

16.1 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. It is beyond the scope of this book to discuss all the factors. In this chapter certain basic as well as important measurements and tests are considered.

(i) Friction power

(ii) Indicated power

(iii) Brake power

(iv) Fuel consumption

(v) Air flow

(vi) Speed

(vii) Exhaust and coolant temperature

(viii) Emissions

(ix) Noise

(x) Combustion phenomenon

The details of measurement of these parameters are discussed in the following sections.

16.2 FRICTION POWER

The difference between the indicated and the brake power of an engine is known as friction power. The internal losses in an engine are essentially of two kinds, viz., pumping losses and friction losses. During the inlet and exhaust stroke the gaseous pressure on the piston is greater on its forward side (on the under side during the inlet and on the upper side during the exhaust stroke), hence during both strokes the piston must be moved against a gaseous pressure, and this causes the so called pumping loss. The friction loss is made up of the friction between the piston and cylinder walls, piston rings and cylinder walls, and between the crankshaft and camshaft and their bearings, as well as by the loss incurred by driving the essential accessories, such as the water pump, ignition unit etc.

It should be the aim of the designer to have minimum loss of power in friction. Friction power is used for the evaluation of indicated power and mechanical efficiency. Following methods are used to find the friction power to estimate the performance of the engine.

(i) Willan's line method

(ii) Morse test

(iii) Motoring test

(iv) From the measurement of indicated and brake power

(v) Retardation test

16.2.1 Willan's Line Method

This method is also known as fuel rate extrapolation method. A graph connecting fuel consumption (y-axis) and brake power (x-axis) at constant speed is drawn and it is extrapolated on the negative axis of brake power. The intercept of the negative axis is taken as the friction power of the engine at that speed. The method of extrapolation

along the curve indicates the effect of part load efficiency of the engine. The pronounced change in the slope of this line near full load reflects the limiting influence of the air-fuel ratio and of the quality of combustion. Similarly, there may be slight curvature at light loads. This is perhaps due to the difficulty in injecting accurately and consistently very small quantities of fuel per cycle. Therefore, it is essential that great care should be taken in extrapolating the line and as many readings as possible should be taken at light loads to establish the true nature of the curve. The accuracy obtained in this method is reasonably good and compares favourably with other methods if extrapolation is carefully done.

16.2.2 Morse Test

The Morse test consists of obtaining indicated power of the engine without any elaborate equipment. The test consists of making inoperative, in turn, each cylinder of the engine and noting the reduction in brake power developed. With a gasoline engine each cylinder is rendered inoperative by *shorting* the spark plug of the cylinder; with a diesel engine by cutting off the supply of fuel to each cylinder. It is assumed that pumping and friction losses are the same when the cylinder is inoperative as well as during firing. This test is applicable only to multi cylinder engines. Referring to the Fig.16.2, the unshaded area of the indicator diagram is a measure of the gross power, *gp* developed by the engine, the dotted area being the pumping power, *pp*.

$$\text{Net indicated power per cylinder} = gp - pp$$

In this test the engine is first run at the required speed by adjusting the throttle in SI engine or the pump rack in CI engine and the output is measured. The throttle rack is locked in this position. Then, one cylinder is cut out by short circuiting the spark plug in the SI engine or by disconnecting the injector in the CI engine. Under this condition all the other cylinders will *motor* the cut out cylinder and the speed and output drop. The engine speed is brought to its original value by reducing the load. This will ensure that the frictional power is the same while the brake power of the engine will be with one cylinder less.

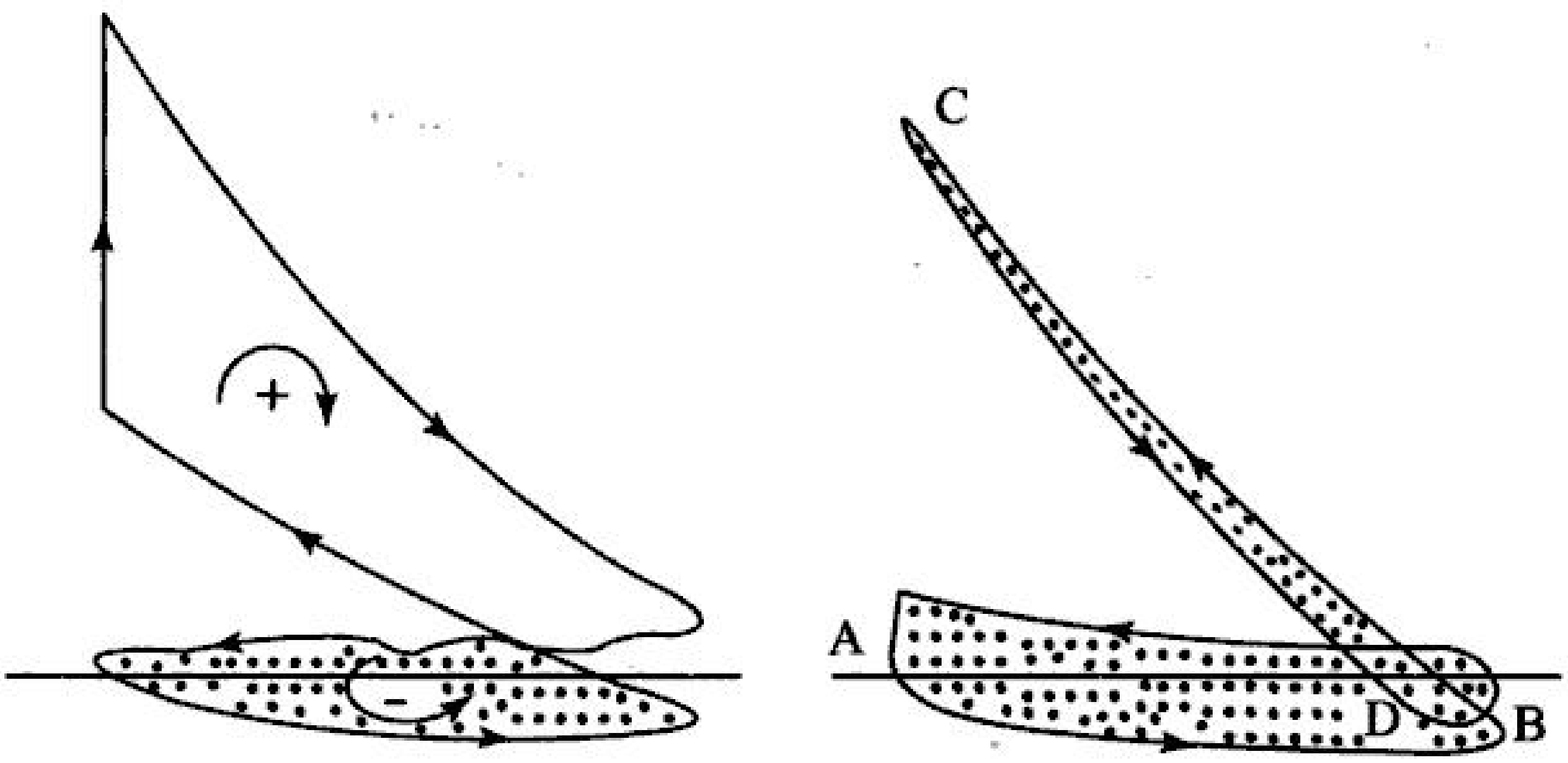
If there are k cylinders, then

$$ip_1 + ip_2 + ip_3 + ip_4 + \ldots + ip_k = \sum_{1}^{k} bp_k + fp_k \qquad (16.1)$$

16.3.1 Method using the Indicator Diagram

The device which measures the variation of the pressure in the cylinder over a part or full cycle is called an *indicator* and the plot of such information obtained is called an *indicator diagram*. Indicator diagram is the only intermediate record available in the account of total liberated energy before it is measured at the output shaft. Thus an indicator diagram gives a very good indication of the process of combustion and in the associated factors such as rate of pressure rise, ignition lag, etc. by its analysis. Also the losses occurring in the suction and exhaust strokes can be studied. It is very rare that an indicator diagram is taken to find indicated power only. It is almost invariably used to study engine combustion, knocking, tuning of inlet and exhaust manifolds, etc.

Pressure-volume, p-V and pressure-crank angle, p-θ, are the two types of indicator diagrams that can be obtained from an engine. Both these indicator diagrams are mutually convertible. An actual indicator diagram is shown in Fig.16.4(a) for a working cycle whereas Fig.16.4(b) is for a missed cycle. During a missed cycle of operation there is no power developed and therefore the entire area is shaded. The direction of the arrows show the path to be followed in the



(a) Working cycle (b) Missed cycle

Fig. 16.4 Actual Indicator Diagram of an Otto Engine

diagram. The sign of an area depends upon the direction in which it is traced and since the shaded area is traced in the reverse direction compared to the unshaded area, which has the opposite sign. The shaded area represents the work done in charging and discharging the cylinder. The elasticity of the column of exhaust gas results in

16.4.2 Rope Brake

The rope brake as shown in Fig.16.9 is another simple device for measuring *bp* of an engine. It consists of a number of turns of rope wound around the rotating drum attached to the output shaft. One side of the rope is connected to a spring balance and the other to a loading device. The power absorbed is due to friction between the rope and the drum. The drum therefore requires cooling.

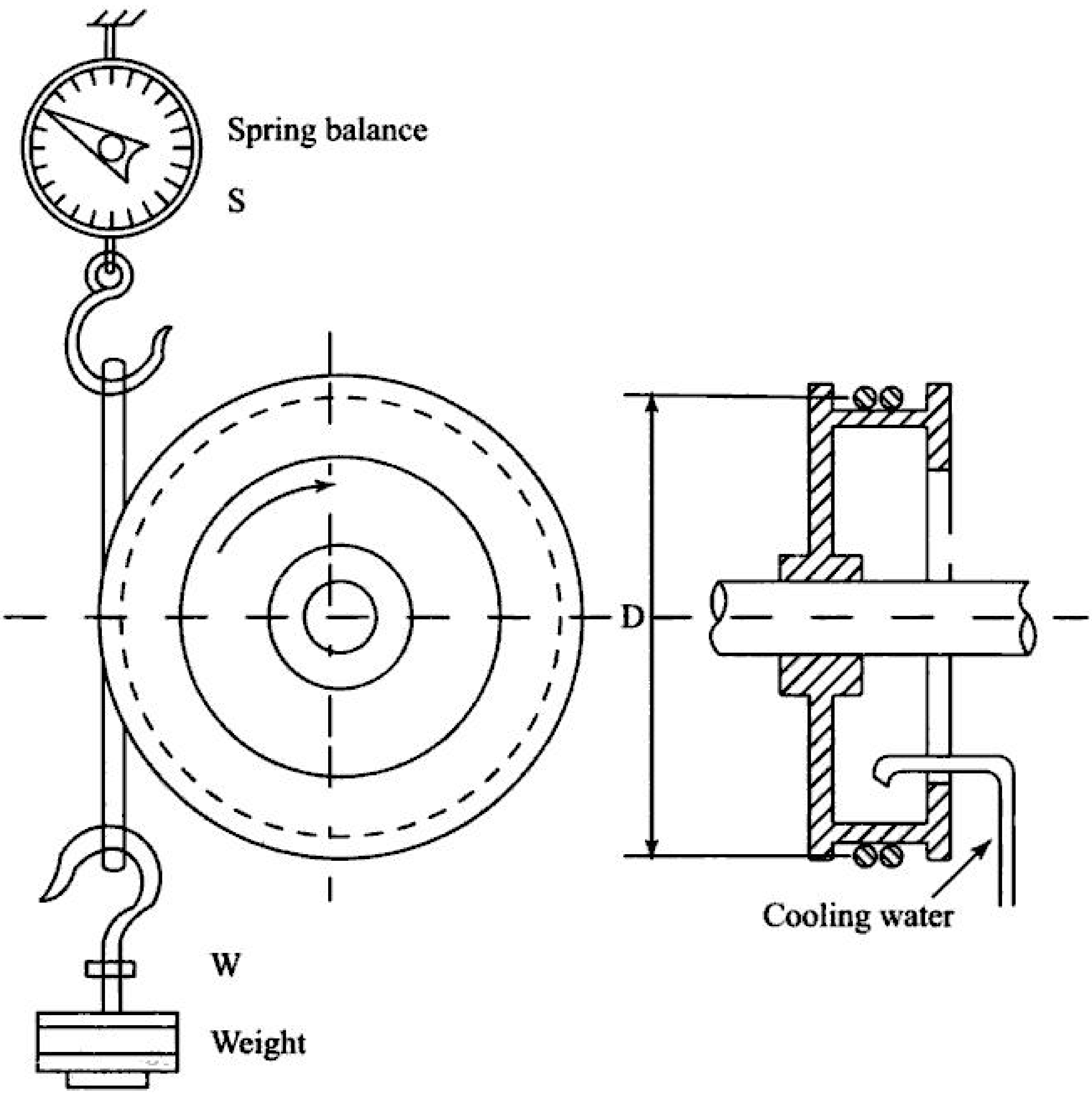


Fig. 16.9 Rope Brake

Rope brake is quite cheaper and can be easily fabricated but not very accurate because of changes in the friction coefficient of the rope with temperature.

The *bp* is given by

$$bp = \pi DN(W - S) \qquad (16.21)$$

(iv) The density of the fuel is dependent on temperature which can vary over a wide range (-10 °C to 70 °C) giving rise to a error in measurement.

(v) Some flowmeters which use a light beam, the measurement may be affected by the colour of the fuel.

(vi) The needle valve in the float bowl of the carburettor opens and closes periodically allowing fuel to surge into the float bowl. This may cause water hammer type effect making the turbine type flowmeter to continue to rotate even when fuel flow has stopped, thereby producing errors in flow measurements.

As already mentioned two basic type of fuel measurement methods are

(i) Volumetric type

(ii) Gravimetric type

16.5.1 Volumetric Type Flowmeter

The simplest method of measuring volumetric fuel consumption is using glass bulbs of known volume and having a mark on each side of the bulb. Time taken by the engine to consume this volume is measured by a stop watch. Volume divided by time will give the volumetric flow rate.

Burette Method: It consists of two spherical glass bulbs having 100 cc and 200 cc capacity respectively (Fig.16.14). They are connected by three way cocks so that one may feed the engine while the other is being filled. The glass bulbs are of different capacities so as to make the duration of the tests approximately constant irrespective of the engine load whilst the spherical form combines strength with a small variation of fuel head which is most important particularly in case of carburettor engines.

In order to avoid the error in sighting the fuel level against the mark on the burette photocells are used. Figure 16.14 shows such an arrangement in which the measurement is made automatic.

Automatic Burette Flowmeter: Figure 16.15 shows an automatic volumetric type fuel flow measuring system which is commercially available. It consists of a measuring volume (A) which has a photocell (B) and a light source (C) fitted in tubular housings. These housings are put opposite to each other at an angle such that a point of light is formed on the axis of the measuring volume as

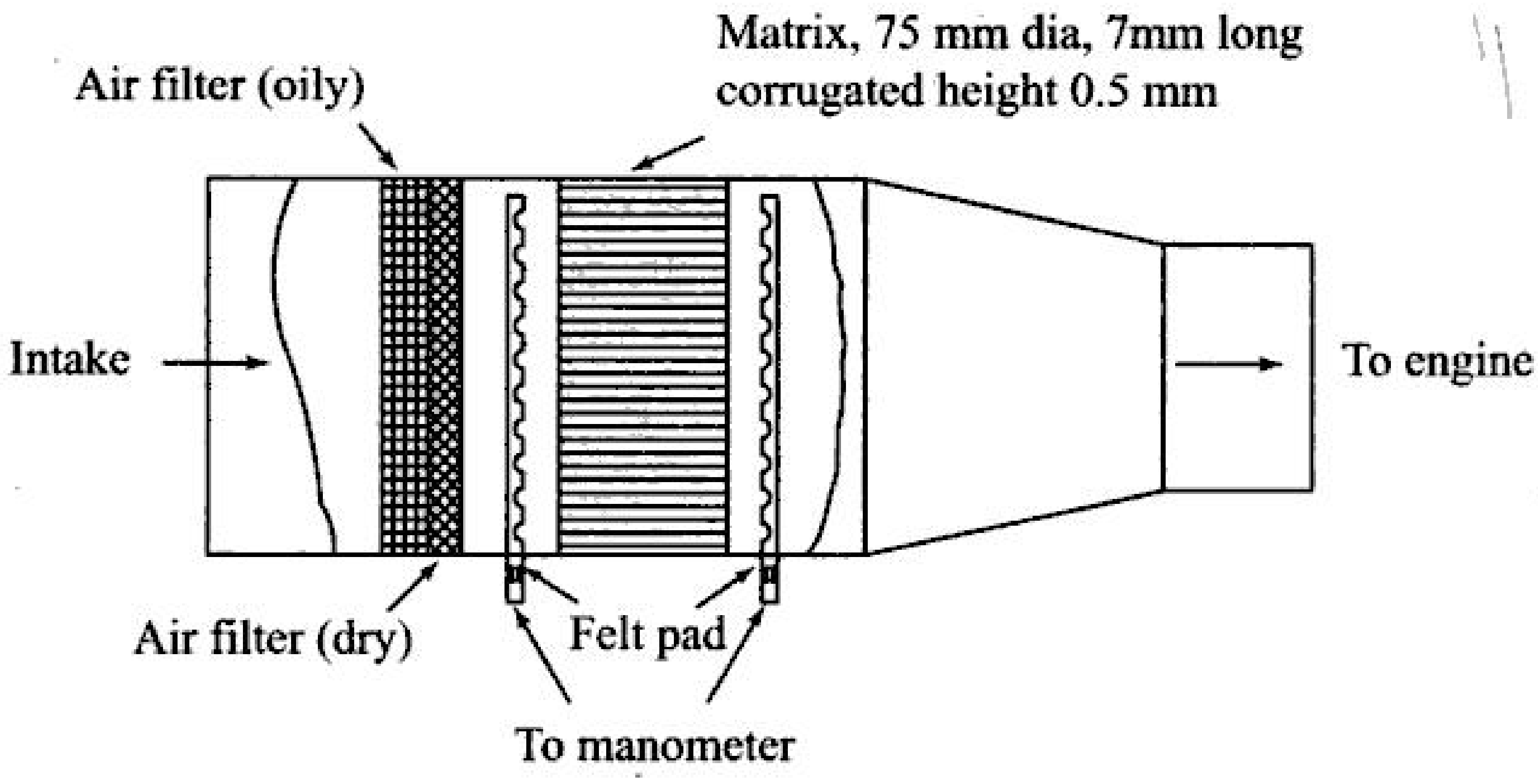


Fig. 16.20 Alcock Viscous-Flow Air Meter

The chief source of error in viscous meters arises from surface contamination of the small triangular passages. However, by ensuring good filtration at the entry to the meter, and not passing air through the meter unless readings are required, this trouble can be minimized. An advantage of viscous-flow meter is that larger range of flow can be measured without pressure head being too small. Nowadays positive displacement type of flowmeters are also used for the measurement of air consumption.

16.7 SPEED

Speed measurement is an art. Speed of the engine is widely used in the computation of power, design and development. Measurement of speed is accomplished by instruments like mechanical counters and timers, mechanical tachometers, stroboscope, electric counters, tachometers, electric generators, electronic pulse counters etc. The best method of measuring speed is to count the number of revolution in a given time. This gives an accurate measurement of speed. Many engines are fitted with such revolution counters. A mechanical or electrical tachometer can also be used for measuring speed. Both these types are affected by temperature variation and are not very accurate. For accurate and continuous measurements of speed a magnetic pick-up placed near a toothed wheel coupled to the engine shaft can be used. The magnetic pick-up will produce a pulse for every revolution and a pulse counter will accurately measure the speed.

the same as the gas temperature. Suppose if T_b is set above the suction temperature, then at the beginning of the compression stroke as the temperature raises there will be some absorption of the infrared by the cylinder gas and the emission level will be less than from the black body.

The detector will record a decrease in intensity. Further increase in temperature of the cylinder gas will produce a further decrease in recorded intensity until a minimum is reached. As the stroke continues the intensity will now increase until at C the rate of absorption of the infrared waves from the black body source equals the rate of emission from the cylinder gases. At this point the cylinder temperature equals the black body temperature T_b. The crank angle at which this occurs can be measured in the normal manner. As series of black body temperatures T_b is taken and the procedure is repeated, a temperature-crank angle diagram can be obtained. Cleanliness of the quartz windows, window radiation, absorption and homogeneity of the cylinder temperature, cycle-to-cycle variations are factors upon which the accuracy of the method depends.

Another technique is the spectrographic method which is not only used for measuring the local gas temperature but also for studying the chemical reactions in the cylinder. Basis of this method depends on the measurement of the light intensities associated with reacting species at defined wavelengths at which the intensities are at a maximum. One method which has been successfully used to study the formation of oxides of nitrogen during combustion is described. Figures 16.32(a) and 16.32(b) show the details.

A number of quartz windows, W are fitted in the cylinder head. The light source is of 2×2 mm size fitted in the combustion chamber. The core of light emitted from the cylinder is split into four separate beams in mirror M_2, each of which is brought to a separate focus by mirrors M_3 and M_4. Three beams (2, 3 and 4) are focused onto a photo-multiplier with an interference filter to pass wave lengths 0.38 μm and 0.61 μm with a band pass of 100°A and a wave length of 0.75 μm at a band pass of 300°A. The fourth channel is monitored with a monochromator and photomultiplier. The light intensities I at the wave length 0.38 μm, 0.68 μm and 0.61 μm are recorded on a oscillograph relative to the crank angle as shown in Fig.16.32(c). The 0.38 μm wavelength corresponds to the reaction

$$CO + O \rightarrow CO_2 + h\nu \tag{16.23}$$

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17.3.7 Charge Efficiency

The charge efficiency shows how well the piston displacement of a four-stroke engine is utilized. Various factors affecting charge efficiency are:

(i) the compression ratio.

(ii) the amount of heat picked up during passage of the charge through intake manifold.

(iii) the valve timing of the engine.

(iv) the resistance offered to air-fuel charge during its passage through induction manifold.

17.3.8 Combustion Efficiency

Combustion efficiency is the ratio of heat liberated to the theoretical heat in the fuel. The amount of heat liberated is less than the theoretical value because of incomplete combustion either due to dissociation or due to lack of available oxygen. Combustion efficiency in a well adjusted engine varies from 92% to 97%.

17.4 ENGINE PERFORMANCE CHARACTERISTICS

Engine performance characteristics are a convenient graphical presentation of an engine performance. They are constructed from the data obtained during actual test runs of the engine and are particularly useful in comparing the performance of one engine with that of another. In this section some of the important performance characteristics of the SI engines are discussed.

It is to be noted that there is a certain speed, within the speed range of a particular engine, at which the charge inducted per cylinder per cycle will be the maximum. At this point, the maximum force can therefore be exerted on the piston. For all practical purposes, the torque, or engine capacity to do work, will also be maximum at this point. Thus, *there is a particular engine speed at which the charge per cylinder per cycle is a maximum, and at approximately this same speed, the torque of the engine will be a maximum.*

As the speed of the engine is increased above this speed the quantity of the indicated charge will decrease. However, the power output of the engine increases with speed due to more number of cycles are

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driven turbines. In this case, the exhaust gas from engine cylinders drives a turbine which is connected to the engine crankshaft, thus increasing engine output. Engines having this type of power booster are known as *turbocompound engines.*

Many of the parameters entering into the performance of four-stroke CI engine are similar to those already analyzed for SI engines. Hence, the performance characteristics of CI engines are not discussed separately.

17.7 HEAT BALANCE

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 turbocompounding. The two main parts of the heat not available for work are the heat carried away by the exhaust gases and the cooling medium. Figure 17.8 illustrates the same for spark-ignition engines. A typical heat balance for compression-ignition engines is illustrated in Fig.17.9.

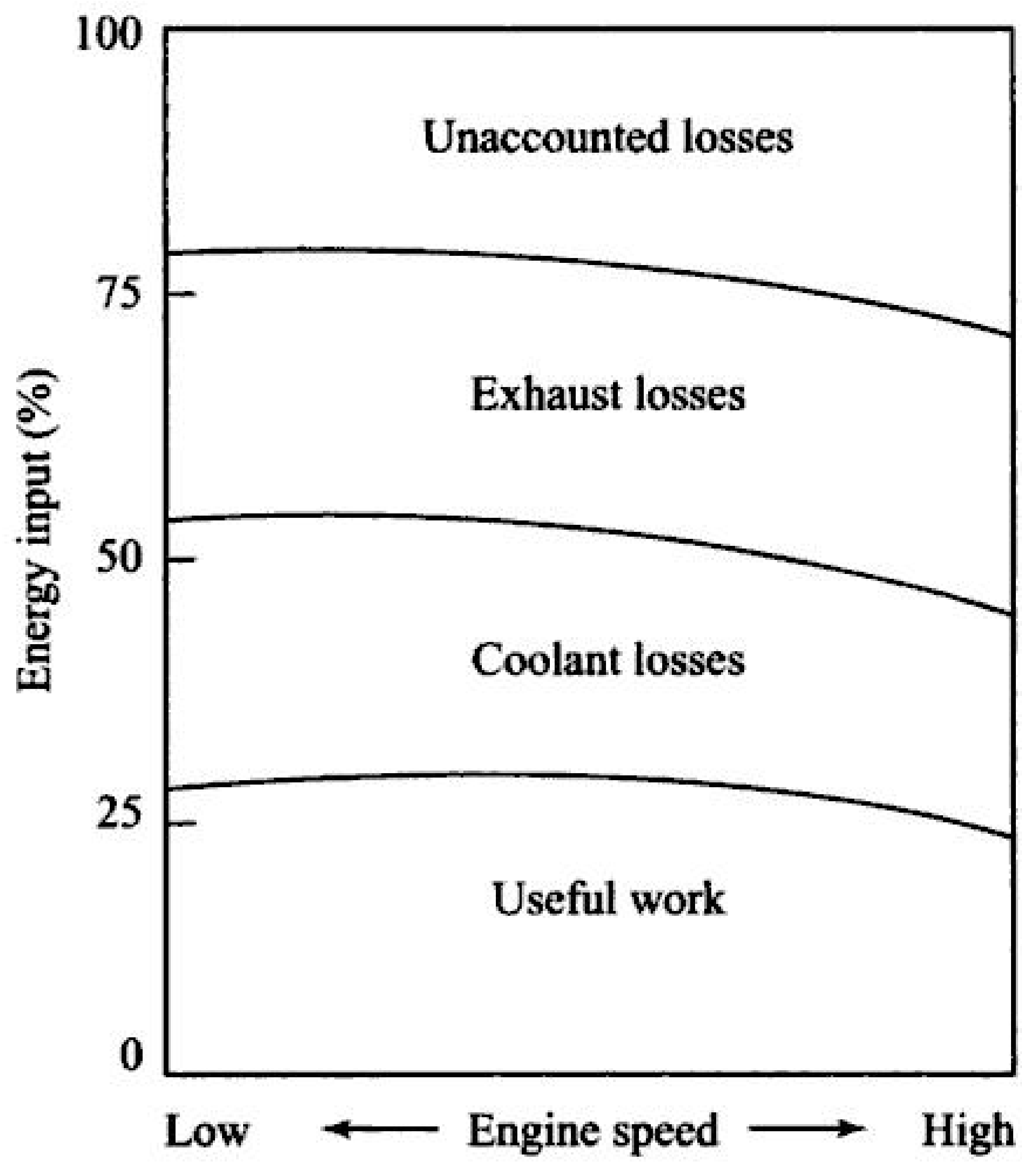

Fig. 17.8 Heat Balance Diagram for a Typical SI Engine

$$\eta_{ith} = \frac{\eta_{bth}}{\eta_m} \times 100 = \frac{0.214}{0.80} \times 100$$

$$= \mathbf{26.75\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$imep = \frac{\frac{bp}{\eta_m} \times 60000}{LAnK}$$

$$= \frac{\frac{31.39}{0.8} \times 60000}{0.12 \times \frac{\pi}{4}0.1^2 \times \frac{1600}{2} \times 4} = 7.8 \times 10^5 \text{ Pa}$$

$$= \mathbf{7.8\ bar} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$bsfc = \frac{\dot{m}_f}{bp} = \frac{0.2 \times 60}{31.39}$$

$$= \mathbf{0.382\ kg/kW\ h} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

17.14 The air flow to a four cylinder, four-stroke oil engine is measured by means of a 5 cm diameter orifice having a coefficient of discharge of 0.6. During a test on the engine the following data were recorded : bore = 10 cm; stroke = 12 cm; speed = 1200 rpm; brake torque = 120 Nm; fuel consumption = 5 kg/h; calorific value of fuel = 42 MJ/kg; pressure drop across orifice is 4.6 cm of water; ambient temperature and pressure are 17 °C and 1 bar respectively. Calculate (i) the thermal efficiency on brake power basis; (ii) the brake mean effective pressure and (iii) the volumetric efficiency based on free air condition.

Solution

$$bp = \frac{2\pi NT}{60000} = \frac{2 \times \pi \times 1200 \times 120}{60000}$$

$$= 15.08 \text{ kW}$$

$$\eta_{bth} = \frac{15.08 \times 60}{\frac{5}{60} \times 42000} \times 100 = \mathbf{25.85\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$p_{im} = \frac{bp \times 60000}{LAnK}$$

$$= \frac{15.08 \times 60000}{0.12 \times \frac{\pi}{4} \times 0.1^2 \times \frac{1200}{2} \times 4}$$

$$= 4 \times 10^5 \text{ Pa} = \mathbf{4\ bar} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

NTP, find (i) the charge of air per working cycle as measured at NTP and (ii) the volumetric efficiency.

Solution

Assume 760 mm of Hg = 1 bar

$$\text{Gas pressure} = 1 + \frac{100}{13.6} \times \frac{1}{76}$$

$$= 1.097 \text{ bar}$$

$$\text{Volume of coal gas at NTP} = 0.3 \times \frac{1.097}{1} \times \frac{273}{290}$$

$$= 0.31 \text{ m}^3/\text{min}$$

$$\text{Vol. of coal gas used/explosion} = \frac{0.31}{100}$$

$$= 0.0031 \text{ m}^3 \text{ at NTP}$$

$$\text{Extra air missed/cycle} = 0.0031 \text{ m}^3 \text{ at NTP}$$

$$\text{Volume of air taken at NTP} = \frac{mRT}{p}$$

$$= \frac{3 \times 287 \times 273}{1 \times 10^5}$$

$$= 2.35 \text{ m}^3/\text{min}$$

The engine is running at 240 rpm and therefore there must be 120 firing cycles per minute. However, there are only 100 cycles per minute. Hence, there are 20 missed cycles. The 2.35 m^3 of air per minute at NTP must be made up of 120 normal air charges, V, together with 20 missed cycles each equivalent to 0.0031 m^3 at NTP

$$20 \times 0.0031 + 120V = 2.35$$

$$V = 0.019 \text{ m}^3$$

$$\text{Total volume of charge at NTP} = 0.019 + 0.0031$$

$$= \mathbf{0.022 \text{ m}^3} \quad \xLeftarrow{\text{Ans}}$$

$$\text{Displacement volume} = \frac{\pi}{4} \times 0.25^2 \times 0.5$$

$$= 0.0245 \text{ m}^3$$

Mean specific heat of steam	=	2.1 kJ/kg K
Sensible heat of water at room temp.	=	62 kJ/kg
Latent heat of steam	=	2250 kJ/kg

Find *ip*, *bp* and draw up a heat balance sheet for the test in kJ/min and in percentage.

Solution

$$ip = \frac{p_{im}LAn}{60000}$$

$$= \frac{3 \times 10^5 \times 0.28 \times \frac{\pi}{4} \times 0.22^2 \times 350}{60000}$$

$$= \mathbf{18.63\ kW} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$bp = \frac{W\pi Nd}{60000}$$

$$= \frac{65 \times 9.81 \times \pi \times 350 \times 1}{60000}$$

$$= \mathbf{11.68\ kW} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Heat supplied/min} = \frac{4 \times 43000}{60}$$

$$= \mathbf{2866.7\ kJ} = \mathbf{100\%}\textit{(let)} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat equivalent of bp

$$= 11.68 \times 60$$

$$= \mathbf{700.8\ kJ/min} = \mathbf{24.4\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat lost to cooling water

$$= \mathbf{696.7\ kJ/min} = \mathbf{24.3\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

1 kg of H_2 *produces 9 kg of* H_2O. *Therefore,*

H_2O *produced per kg of fuel burnt*

$$= 9 \times \%H_2 \times \textit{mass of fuel/min}$$

$$= 9 \times 0.15 \times \frac{4}{60} = 0.09\ \text{kg/min}$$

Total heat carried away by the exhaust gases

$$= 2006.4 + 176.4$$

$$= \mathbf{2182.8\ kJ/min} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Total heat accounted} = 1620 + 1755.6 + 2182.8$$

$$= 5558.4\ \mathbf{kJ/min} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Unaccounted loss} = 5733.3 - 5558.4$$

$$= \mathbf{174.9\ kJ/min} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Indicated thermal efficiency} = \frac{ip \times 60}{\textit{Heat supplied/min}} \times 100$$

$$= \frac{33 \times 60}{5733.3} \times 100$$

$$= \mathbf{34.5\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Brake thermal efficiency} = \frac{27 \times 60}{5733.3} \times 100$$

$$= \mathbf{28.3\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Mechanical efficiency} = \frac{bp}{ip} \times 100 = \frac{27}{33} \times 100$$

$$= \mathbf{81.8\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat input (per minute)	(kJ)	Heat expenditure (per minute)	(kJ)
Heat supplied by fuel	5733.3	1. Heat equivalent of *bp*	1620
		2. Heat lost to cooling medium	1755.6
		3. Heat lost in exhaust	2182.8
		4. Unaccounted losses	174.9
		Total	**5558.4**

17.35 A gasoline engine has a stroke volume of 0.0015 m^3 and a compression ratio of 6. At the end of the compression stroke, the pressure is 8 bar and temperature 350°C. Ignition is set so that the pressure rises along a straight line during combustion and attains its highest value of 25 bar after the piston has travelled 1/30 of the stroke. The charge consists of a gasoline-air mixture in proportion by mass 1 to 16. Take R = 287 J/kg K,

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Heat equivalent of bp

$$= \quad 13.56 \times 60 = 813.6 \text{ kJ/min} = \mathbf{21\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat lost to cooling water

$$= \quad \frac{440}{60} \times 4.18 \times 36 \quad = \quad 1103.5 \text{ kJ/min}$$

$$= \quad \mathbf{28.5\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat carried away by dry exhaust gas

$$= \quad \frac{5.4}{60} \times (30 + 1 - 9 \times 0.15) \times 1 \times (350 - 17)$$

$$= \quad 888.6 \text{ kJ/min} = \mathbf{23\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat carried away by the steam in the exhaust gas

$$= \quad 9 \times 0.15 \times \frac{5.4}{60} \times (3180 - 4.18 \times 17)$$

$$= \quad 377.7 \text{ kJ/min} == \mathbf{9.8\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Unaccounted loss (by difference)

$$= \quad 3870 - (813.6 + 1103.5 + 888.6 + 377.7)$$

$$= \quad 686.6 \text{ kJ/min} = \mathbf{17.7\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Heat input (per minute)	%	Heat expenditure (per minute)	%
Heat supplied by fuel	100	1. Heat equivalent to *hp*	21.0
		2. Heat lost to cooling water	28.5
		3. Heat carried away by dry exhaust	23.0
		4. Heat lost in steam	9.8
		5. Unaccounted losses	17.7
		Total	**100**

17.40 A gas engine working on the constant-volume cycle gave the following results during a one-hour test run. Cylinder diameter 24 cm; stroke 48 cm; torque 770 Nm; average speed 220 rpm; average explosion per minute 77; *mep* 7.5 bar; volume of gas used 12 m^3 at 17 °C and 770 mm of mercury pressure; lower calorific value of gas 21 MJ/m^3 at NTP; inlet and outlet temperature of cooling water are 25 °C and 60 °C respectively;

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18.3 POSITION DISPLACEMENT AND SPEED SENSING

Position displacement and speed sensing are very important in the engine management system. For such sensing inductive, hall effect, potentiometric, electro optical, differential transformer and strain gauge sensors are extensively used for these applications particularly in automobiles and engine laboratories. The details and working principle of various sensors are discussed in the following sections.

18.3.1 Inductive Transducers

A typical inductive transducer is shown in Fig.18.3. As can be seen, there is a permanent magnet. When the toothed piece moves it changes the permeance of the magnetic circuit and this changes the magnetic flux. Thus a voltage gets developed when the flux field varies as the toothed piece moves. The voltage output alternates about the mean on the plus and minus sides. It can be further sent to an electronic circuit to shape it into a square pulse. The toothed piece can be the teeth on the crank shaft of the engine as shown in Fig. 18.1, so that the frequency of the pulses can give the speed of the shaft.

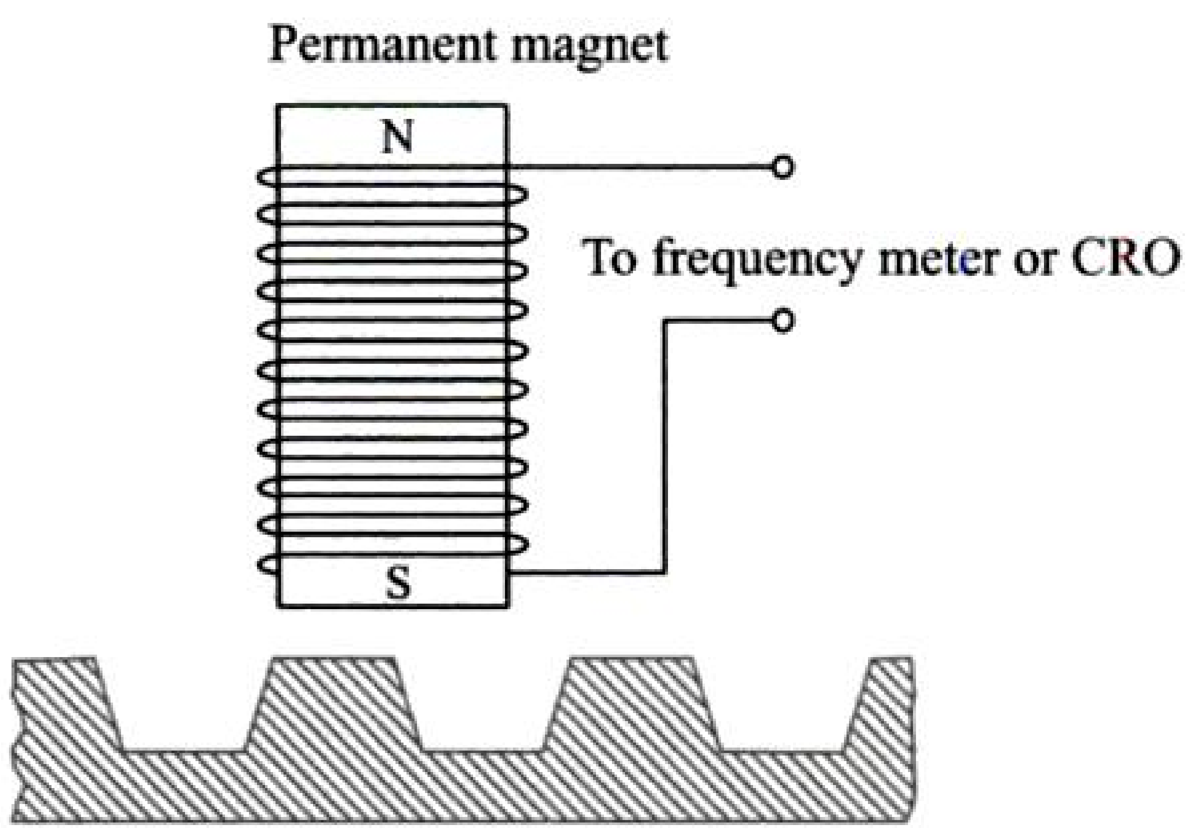

Fig. 18.3 Variable Reluctance Pickup

These transducers are often used for this application. Their output becomes stronger as the variation in the flux rate becomes faster. Thus they are used only for dynamic applications. They can also be used to determine the position of the cam shaft if at some point on the shaft a tooth is removed as shown in Fig. 18.1. In that case the

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18.7 EXHAUST OXYGEN SENSOR

Catalytic converters can operate efficiently only in a narrow band of air fuel ratios. In order to maintain the excess air ratio, λ, with in a narrow range, a closed loop control of the air fuel ratio is needed.

A schematic of the lambda (λ) sensor is indicated in Fig.18.24. The outer section of the ceramic body is in contact with the exhaust whilst the inner section is in contact with ambient air. The ceramic is made of zirconium dioxide. The surfaces are coated with porous platinum which acts as electrodes. A porous protective ceramic coating is also applied on the outer surface.

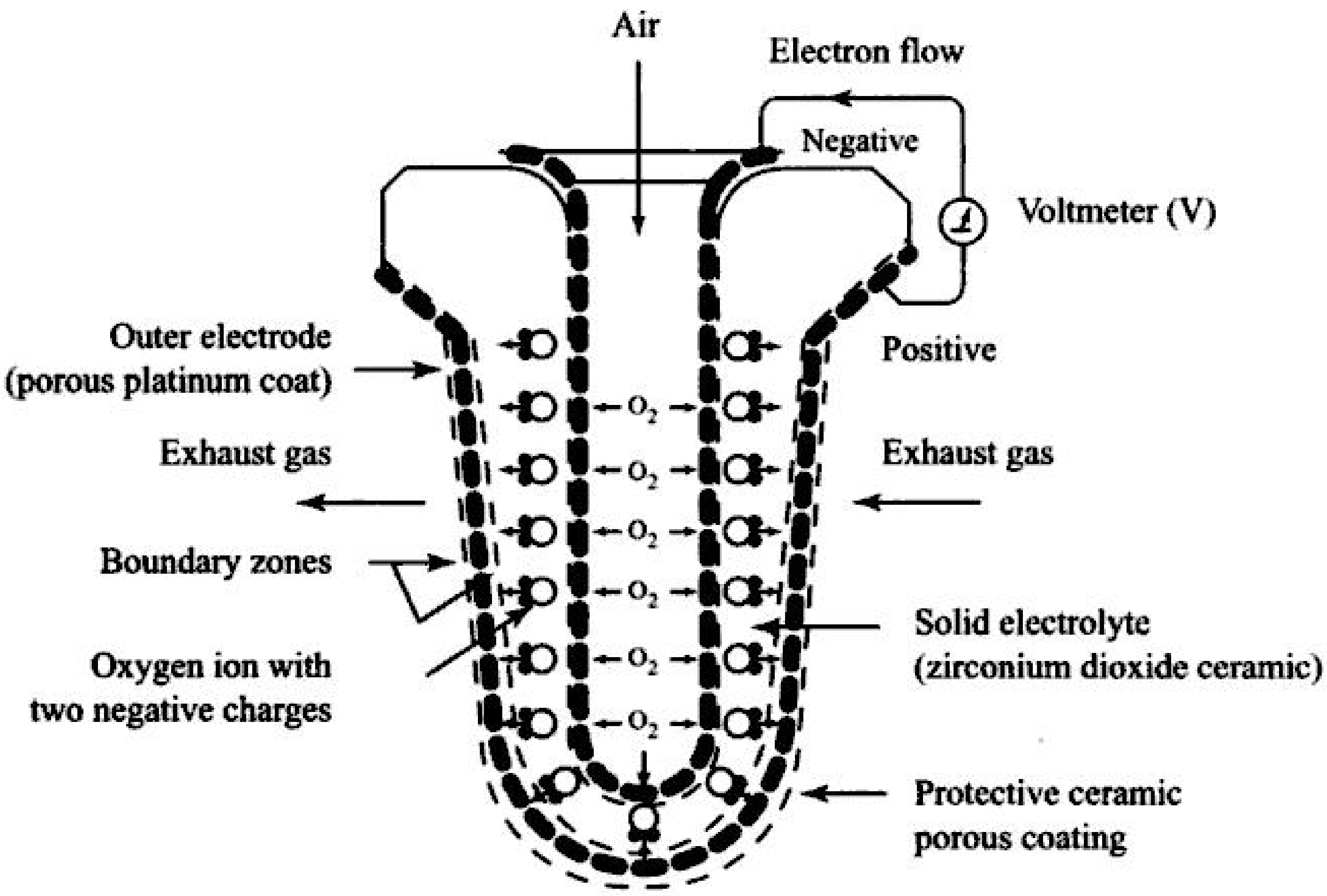

Fig. 18.24 Oxygen Level Sensor

The lambda sensor measures the oxygen concentration in the exhaust gases. When the oxygen concentration on both sides of the cell are different there is a voltage output. The voltage characteristics of this sensor are indicated in Fig.18.25. The voltages given are for a sensor operating at about 600 °C. The temperature of the sensor affects the conductivity of the ions. Thus the curve depends strongly on the sensor temperature. The response also depends on the sensor temperature. At the ideal operating temperature of about 600 °C, the response time is less than 50 ms. Hence, the Lambda sensor and the closed loop control are active only after the sensor reaches

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by supplying air or air-fuel mixture at a pressure which is higher than the atmospheric pressure. This will increase the density, thereby the mass of air or air-fuel mixture inducted for the same swept volume. This in turn will increase the power output of the engine.

This method of supplying air or fuel-air mixture higher than the pressure at which the engine naturally aspirates, by means of a boosting device is called the supercharging. The device which boosts the pressure is called supercharger.

19.2 SUPERCHARGING

Supercharging of internal combustion engines is in practice for a long time as a method for improving engine power output. Entering the millennium, a new trend is appearing. The trend points to small displacement engines in order to meet emission legislation on fuel consumption and emission control. The consumers, however, still demands the same performance they are used to.

A good way to meet these needs is to have supercharging which may be called forced induction. As already stated, the purpose of supercharging an engine is to raise the density of the air charge, before it enters the cylinders. Thus, the increased mass of air will be inducted which will then be compressed in each cylinder. This makes more oxygen available for combustion than the conventional method of drawing the fresh air charge into the cylinder (naturally aspirated). Consequently, more air and fuel per cycle will be forced into the cylinder, and this can be effectively burnt during the combustion process to raise the engine power output to a higher value than would otherwise be possible.

The points to be noted in supercharging an engine summarized as:

(i) Supercharging increases the power output of the engine. It does not increase the fuel consumption, per brake kW hour.

(ii) Certain percentage of power is consumed in compressing the air. This power has to be taken from the engine itself. This will lead to some power loss. However, it is seen that the net power output will be more than the power output of an engine of the same capacity, without supercharging.

(iii) The engine should be designed to withstand the higher forces due to supercharging.

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