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INTERNAL COMBUSTION ENGINES

THIRD EDITION

V GANESAN

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FOREWORD

Focussing on the need of a first level text book for the undergraduates, postgraduates and a professional reference book for practicing engineers, the author of this work Dr. V. Ganesan has brought forth this volume using his extensive teaching and research experience in the field of internal combustion engineering. It is a great pleasure to write a foreword to such a book which satisfies a long-felt requirement.

For selfish reasons alone, I wish that this book would have come out much earlier for the benefit of several teachers like me who have finished their innings a long time ago. For me, this would have been just the required text book for my young engineering students and engineers in the transportation and power fields. The style of the book reflects the teaching culture of premier engineering institutions like IITs, since a vast topic has to be covered in a comprehensive way in a limited time. Each chapter is presented with elegant simplicity requiring no special prerequisite knowledge of supporting subjects. Self-explanatory sketches, graphs, line schematics of processes and tables have been generously used to curtail long and wordy explanations. Numerous illustrated examples, exercises and problems at the end of each chapter serve as a good source material to practice the application of the basic principles presented in the text. SI system of units has been used throughout the book which is not so readily available in the currently-used books.

It is not a simple task to bring out a comprehensive book on an all-encompassing subject like internal combustion engines. Over a century has elapsed since the discovery of the diesel and gasoline engines. Excluding a few developments of rotary combustion engines, the IC engines has still retained its basic anatomy. As a descendent of the steam engine, it is still crystallized into a standard piston-in-cylinder mechanism, reciprocating first in order to rotate finally. The attendant kinematics requiring numerous moving parts are still posing dynamic problems of vibration, friction losses and mechanical noise. Empiricism has been the secret of its evolution in its yester years.

As our knowledge of engine processes has increased, these engines have continued to develop on a scientific basis. The present day engines have to satisfy the strict environmental constraints and fuel economy standards in addition to meeting the competitiveness of the world market. Today, the IC engine has synthesized the basic knowledge of many disciplines — thermodynamics, fluid flow, combustion, chemical kinetics and heat transfer as applied to a system with both spatial and temporal variations in a state of non-equilibrium. With the availability of sophisticated computers, art of multi-dimensional mathematical modelling and electronic instrumentation have added new refinements to the engine design. From my personal knowledge, Dr. Ganesan has himself made many original contributions in these intricate areas. It is a wonder for me how he has modestly kept out these details from the text as it is beyond the scope of this book. However, the reader is not denied the benefits of these investigations. Skillfully the overall findings and updated information have been summarized as is reflected in topics on combustion and flame propagation, engine heat transfer, scavenging processes and engine emissions – to name a few examples. Indeed, it must have been a difficult task to summarize the best of the wide ranging results of combustion engine research and compress them in an elegant simple way in this book. The author has also interacted with the curriculum development cell so that the contents of the book will cater to the needs of any standard accredited university.

I congratulate the author, Dr. V. Ganesan on bringing out this excellent book for the benefit of students in IC engines. While many a student will find it rewarding to follow this book for his class work, I also hope that it will motivate a few of them to specialize in some key areas and take up combustion engine research as a career. With great enthusiasm, I recommend this book to students and practicing engineers.

B. S. Murthy
Former Professor, IIT Madras

PREFACE TO THE THIRD EDITION

I am thankful to the teaching and student community for the overwhelming response for the second edition. I was receiving constant feed back from students and teachers. One of the requests was to add objective type questions, which I have included in all the chapters in this edition. This, I hope will help the students in testing their comprehension. The figures and text in all chapters have been thoroughly reviewed and the additions and deletions wherever necessary have been carried out in various chapters.

I am thankful to Prof. S. Sampath of Sri Venkateswara College of Engineering, Sriperumbudur, Chennai and Prof. A. Ramesh of Indian Institute of Technology Madras, who helped me in reviewing chapters and objective type questions.

This edition would not have been brought to this perfection but for the dedicated effort of Ms.Vijayashree, who is responsible right from the beginning in bringing out all the three editions. I am grateful to her for all the support in the preparation of the camera-ready copy.

I hope this edition will also receive the overwhelming support from the academia and practicing engineers. I will be thankful for any constructive criticism for improvements in future editions.

V GANESAN

PREFACE TO THE SECOND EDITION

Seeing the overwhelming response for the first edition and taking into account the recent developments taken place in the field of IC engines, it has been decided to bring out this second edition. Five new chapters, viz., Alternate Fuels, Electronic Injection Systems, Engine Emissions and Their Control, Engine Electronics and Supercharging have been added to make the book upto date. Variety of new problems have been added in various chapters and number of exercise problems have been increased. All the figures have been redrawn using computer.

The second edition would not have been possible but for the support of Ms.Vijayashree who did all the drawings and the text formatting with utmost sincerity and dedication. I am grateful to her for making this edition possible. I am thankful to all my research students who helped me in checking all the problems, text and figures. I hope and wish this second edition also will receive the overwhelming support from under-graduate, post-graduate and practising engineers. The author will be grateful for any constructive criticism for the improvement in future editions.

V GANESAN

PREFACE TO THE FIRST EDITION

Keeping in view the increasing importance of IC engines, various Universities are introducing courses on the subject as an intrinsic part of the thermal engineering curriculum. However, the lack of a suitable textbook has created difficulties in fully appreciating the basic principles and applications of IC engines. This book has been written to fulfill this need.

I have endeavoured to explain the various topics right from the fundamentals so that even a beginner can understand the exposition. Keeping this in view, chapters 1 to 15 are framed so that the book will be useful to both undergraduate and postgraduate students as well as to practising engineers.

In writing this book, I have kept in mind the tremendous amount of ground, which the student and the practising engineer of today is expected to cover. On this account, the work has been organized to form, it is hoped, a continuous logical narrative.

SI units have been consistently used throughout the book. The book includes a large number of typical worked out examples and several illustrative figures for an easier understanding of the subject. Exercises have also been provided in various chapters so that the inquisitive student may solve these problems and compare with the answers given.

Care has been taken to minimize the errors and typing mistakes. I would be obliged to the readers for finding out any such error and mistake, and would be grateful for any constructive criticism for the improvement of various topics in the book.

It would be impossible to refer in detail to the many persons who have been consulted in the compilation of this work. I am thankful to all my Punctilious and highly Devoted scholars, past and present, who have helped me in various ways in bringing out this book. I may be excused for not naming them individually.

I am particularly grateful for the help I have received from my colleagues at I.C. Engines Laboratory, Indian Institute of Technology, Madras and especially to Prof. P. Srinivasa Rao, who spared a great amount of his valuable time in going through the manuscript and discussing various points in each chapter. I wish to thank the Centre for Continuing Education of the Indian Institute of Technology, Madras for their help in the preparation of this book.

I would like to express my gratitude and offer my sincere thanks to Vijayashree, G. Venkatasubramanian and V. Satish Kumar, who were in-

strumental in composing and formatting the entire book. I will be failing in my duty if I do not acknowledge P. Chandrasekaran who has done all the drawings excellently.

Last but not the least my sincere thanks are due to my dear wife P. Rajalakshmi, my son G. Venkatasubramanian, and daughter G. Aparna who have borne with me for the last three and a half years and provided me with a quiet home where so much of this work was written.

V. GANESAN

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

e	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
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.

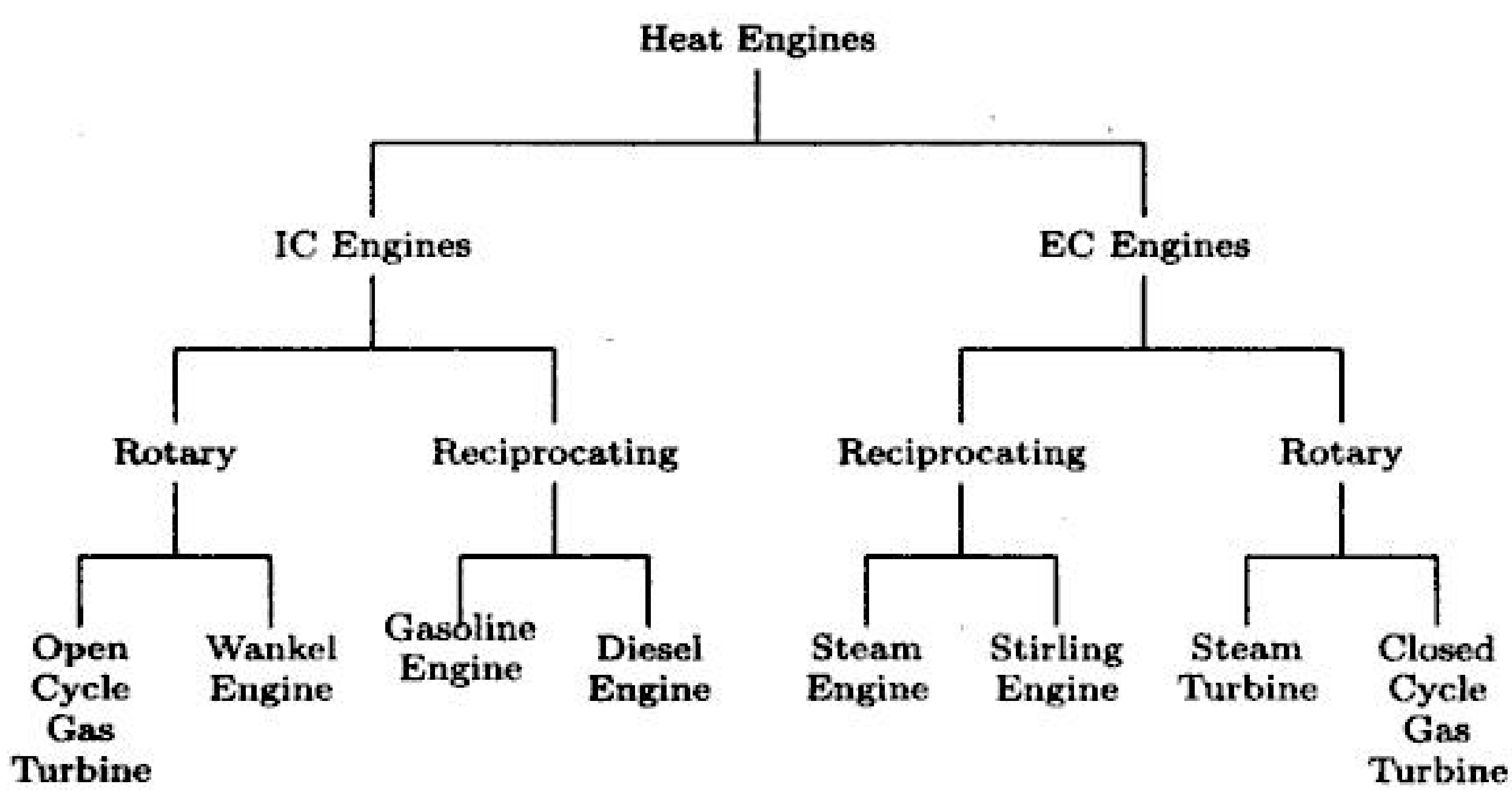


Fig. 1.1 Classification of Heat Engines

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.

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)

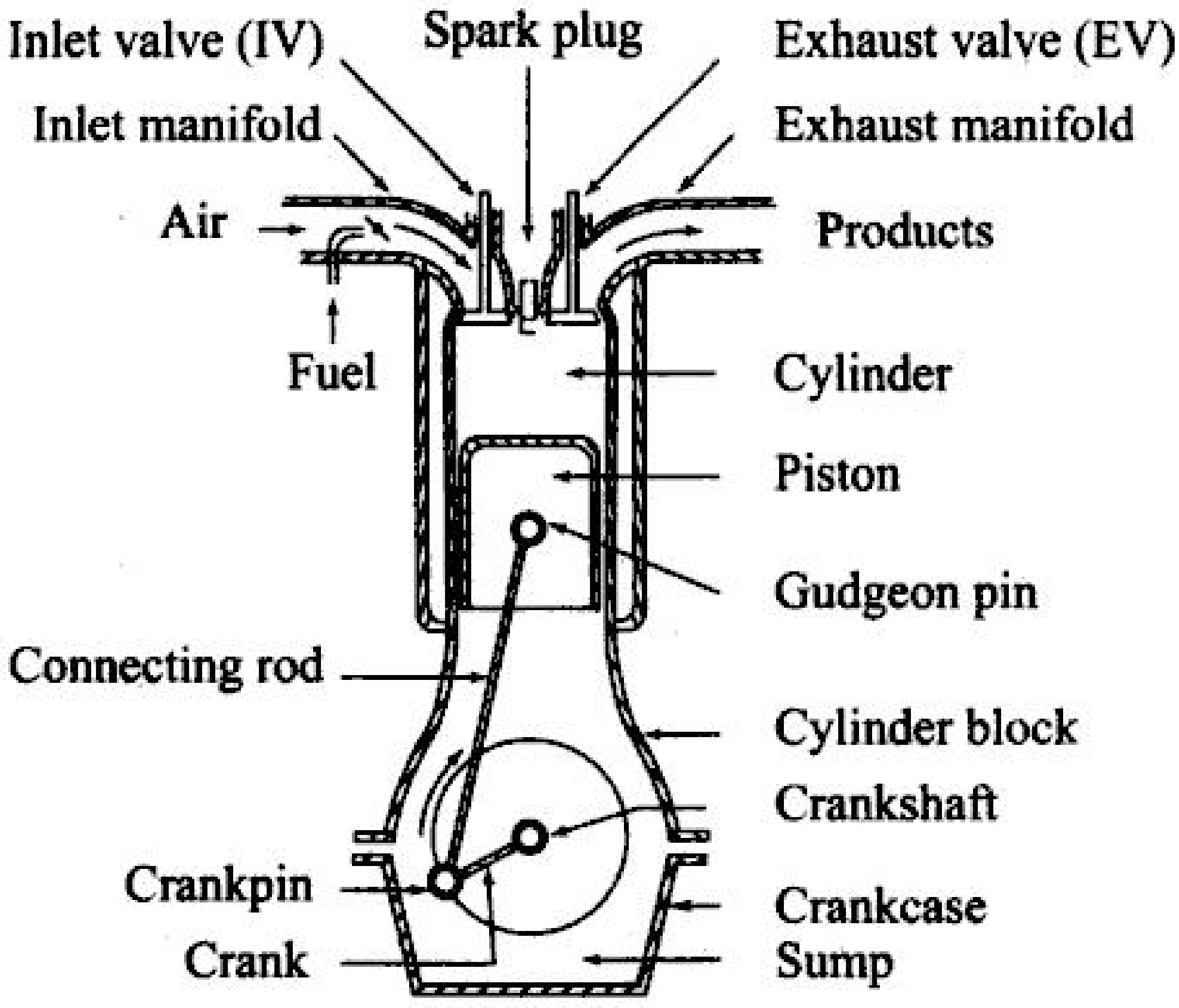


Fig. 1.2 Cross-section of a Spark-Ignition Engine

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.

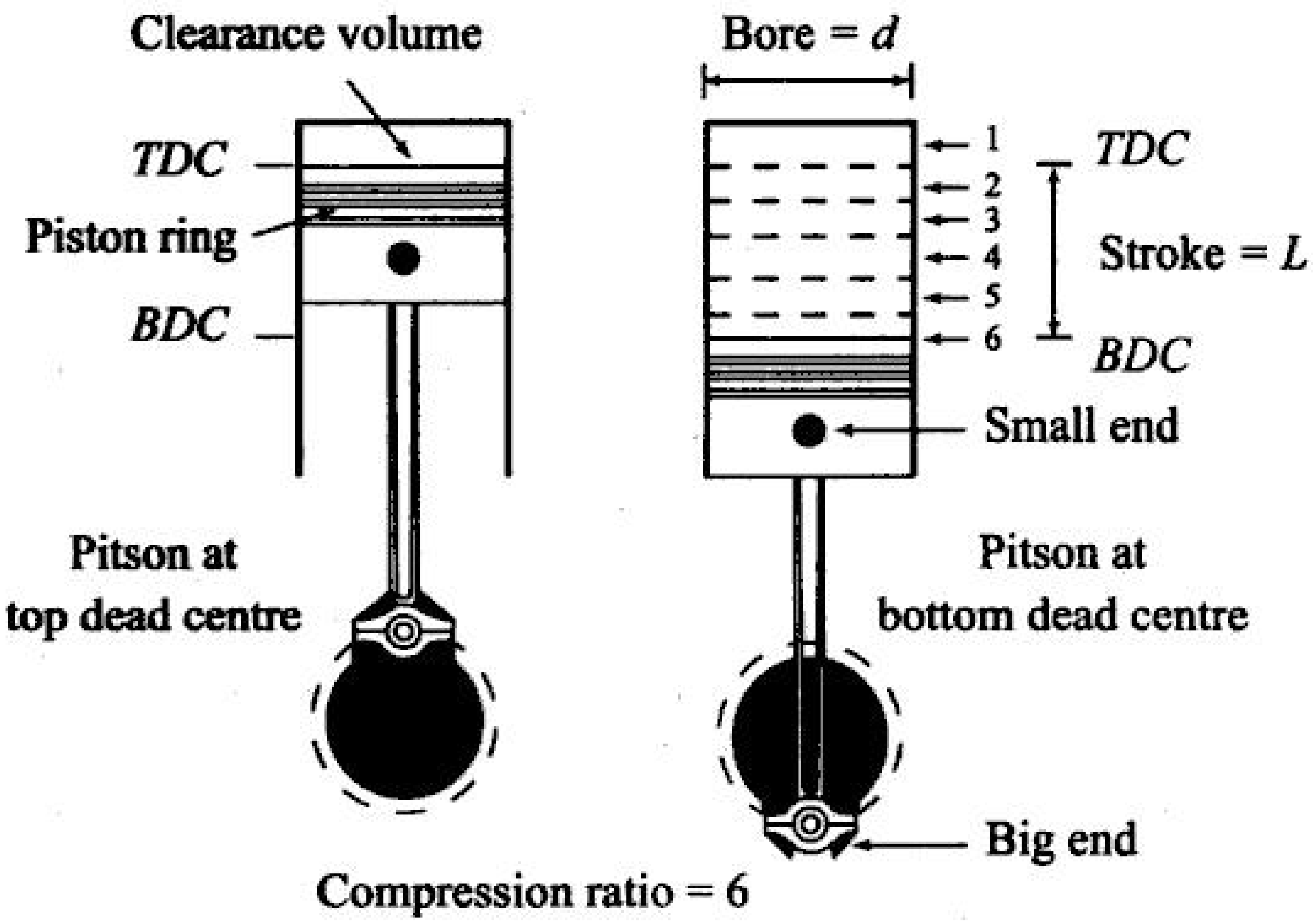


Fig. 1.3 Top and Bottom Dead Centres

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.

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

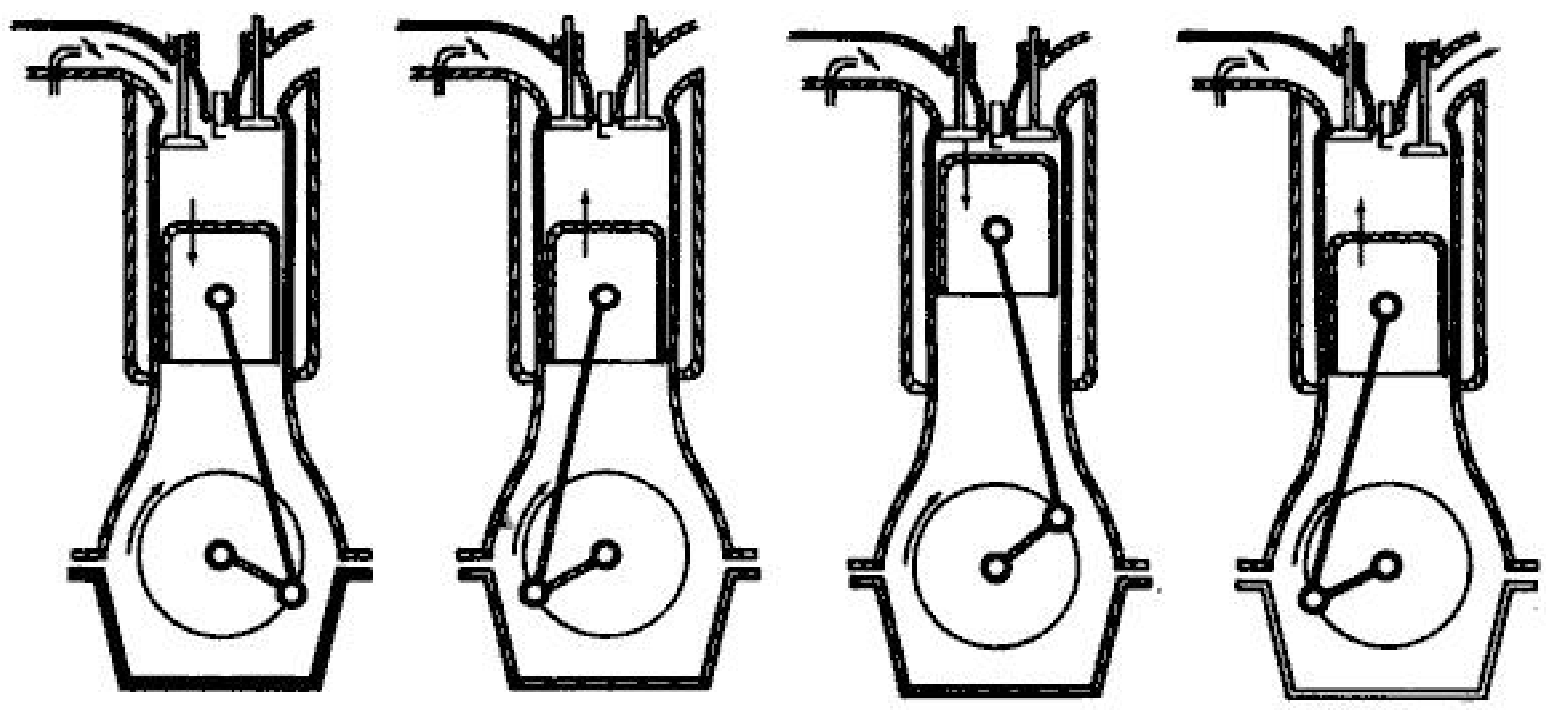
If an engine is to work successfully then it has to follow a cycle of operations in a sequential manner. The sequence is quite rigid and cannot be changed. In the following sections the working principle of both SI and CI engines is described. Even though both engines have much in common there are certain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

(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.

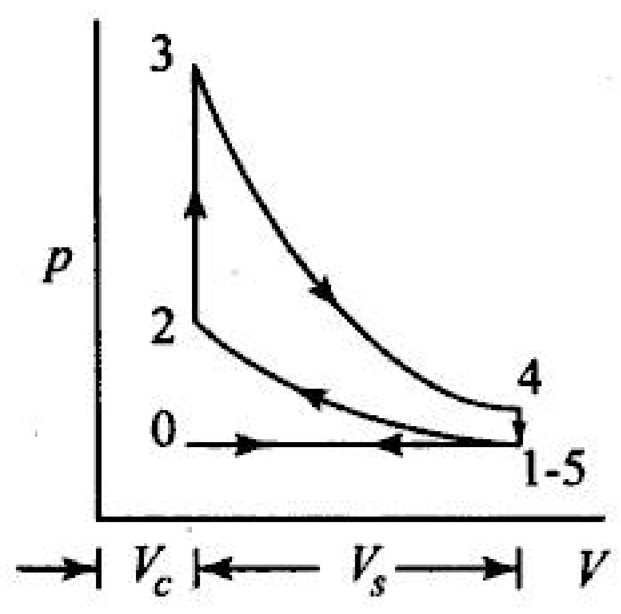


Fig. 1.5 Ideal p-V Diagram of a Four-Stroke SI Engine

(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 $2 \rightarrow 3$) should be as high as possible and the heat rejection (process $3 \rightarrow 4$) 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 as shown in Fig.1.6 is as follows:

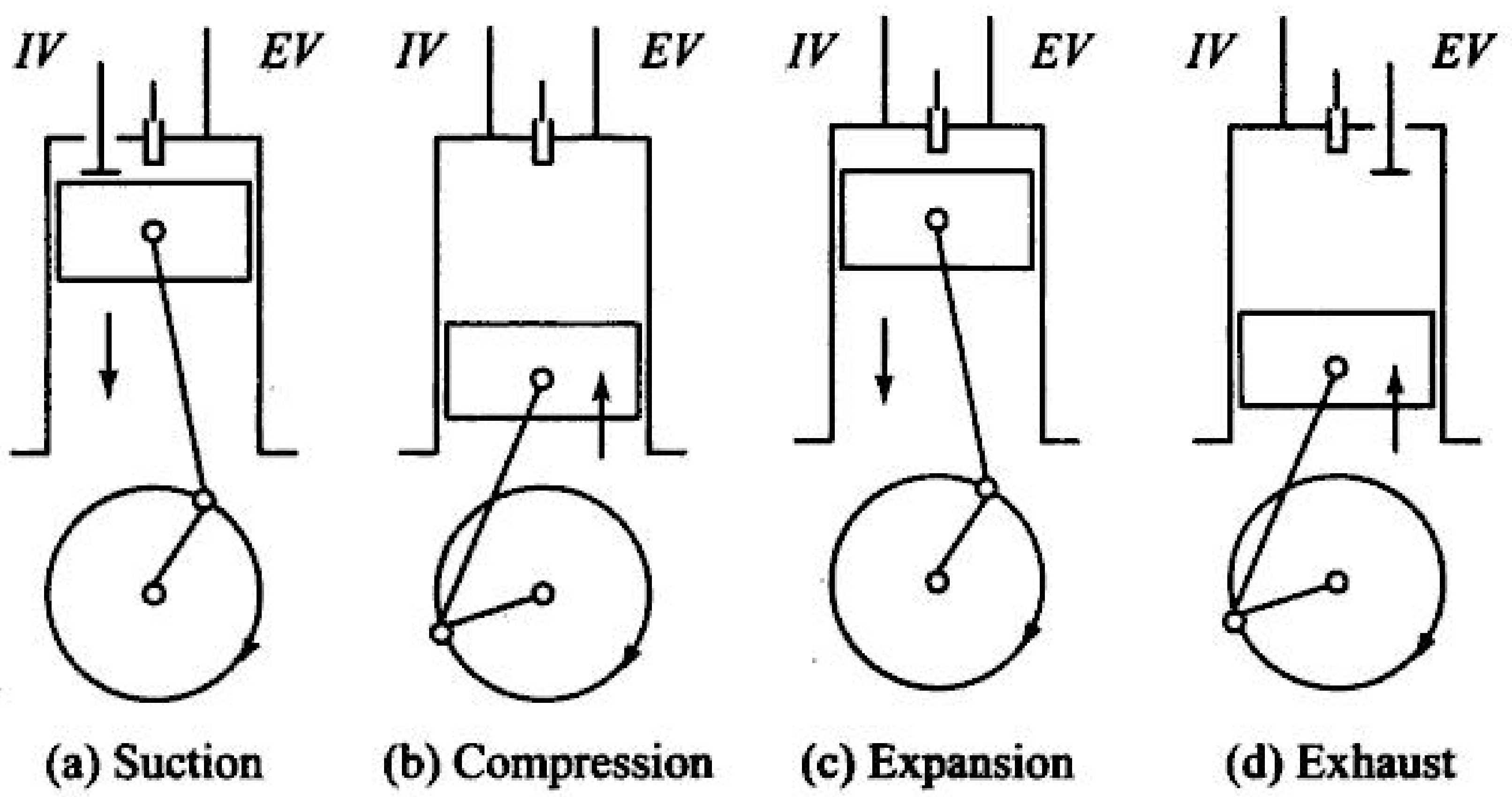


Fig. 1.6 Cycle of Operation of a CI Engine

(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).

(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.

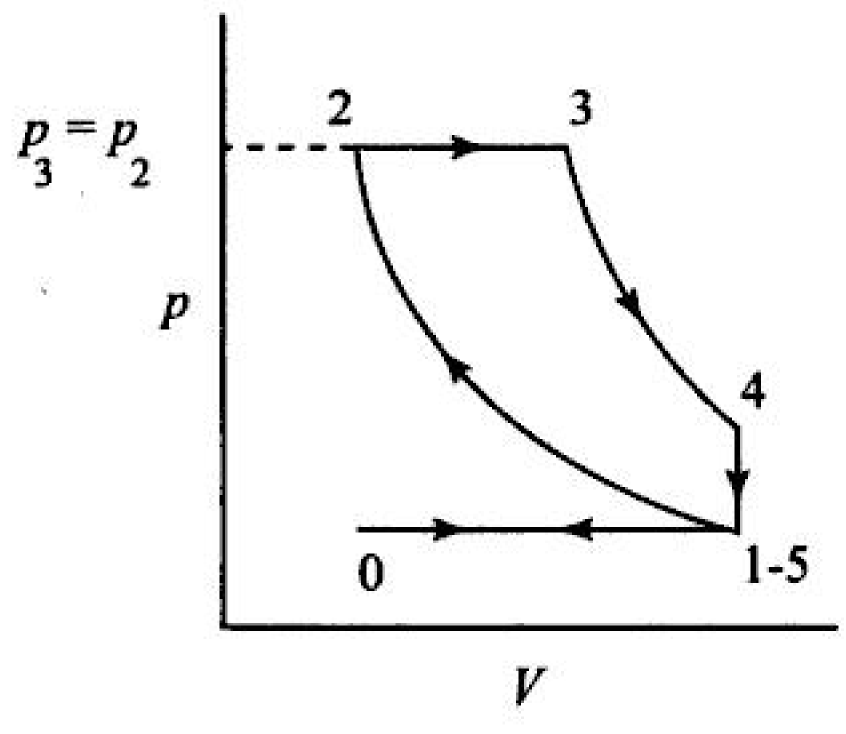


Fig. 1.7 Ideal p-V Diagram for a Four-Stroke CI Engine

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 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 process 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.

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 piston from

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.

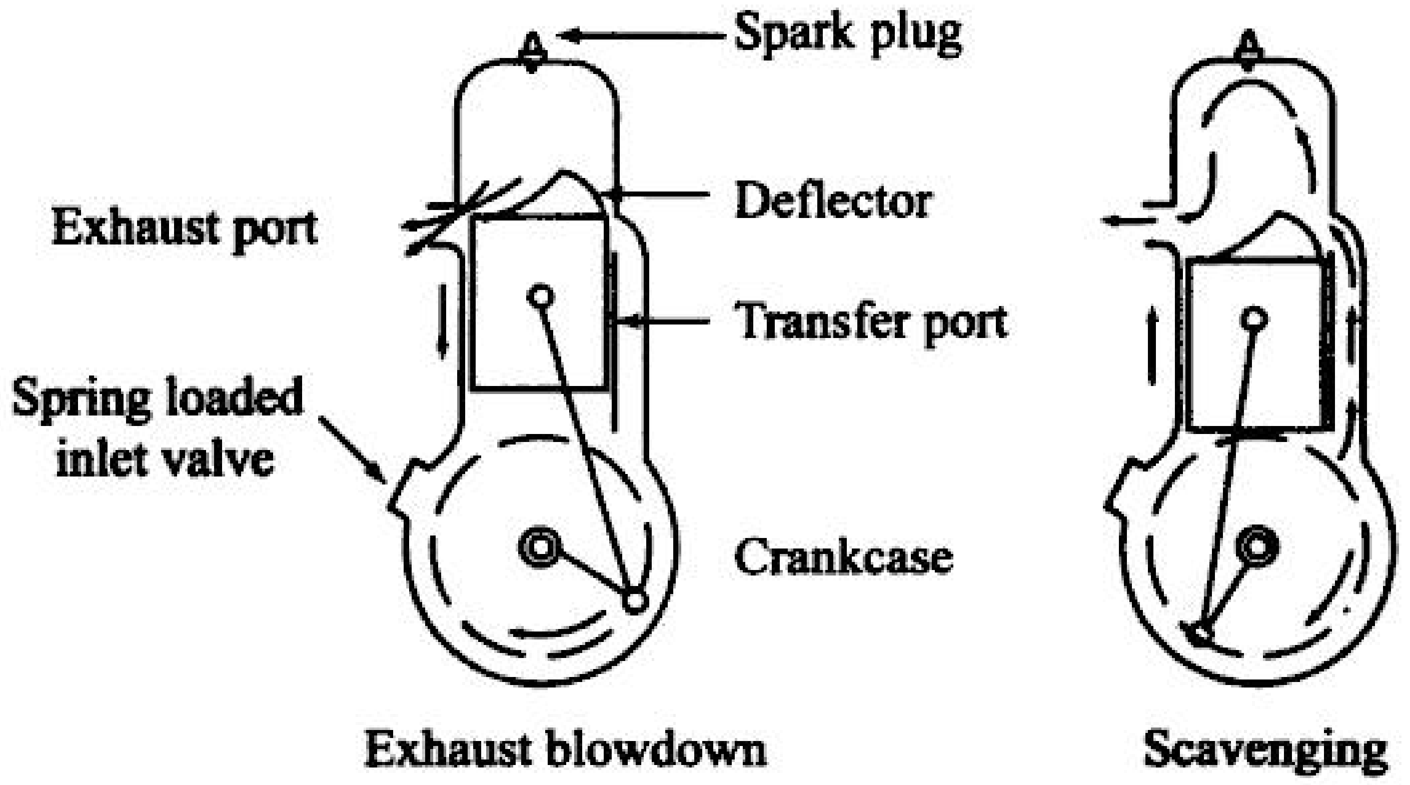


Fig. 1.8 Crankcase Scavenged Two-Stroke Engine

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

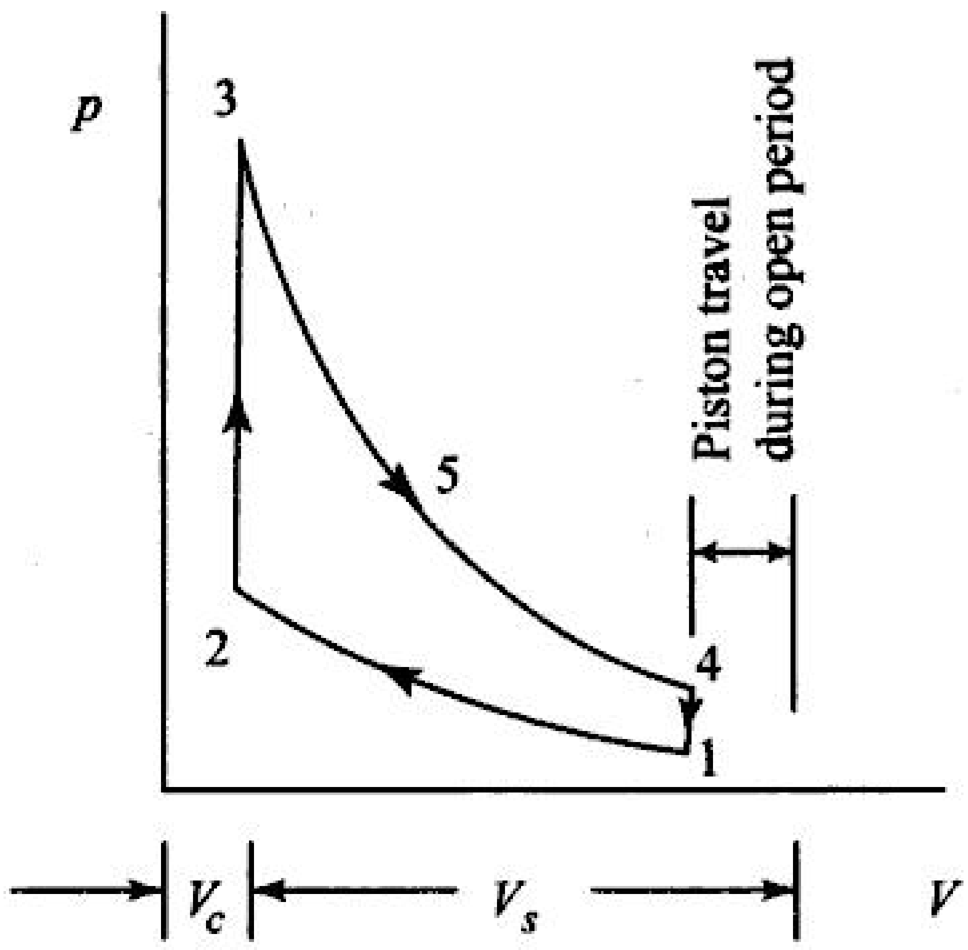


Fig. 1.9 Ideal Indicator Diagram of a Two-Stroke SI Engine

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

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.

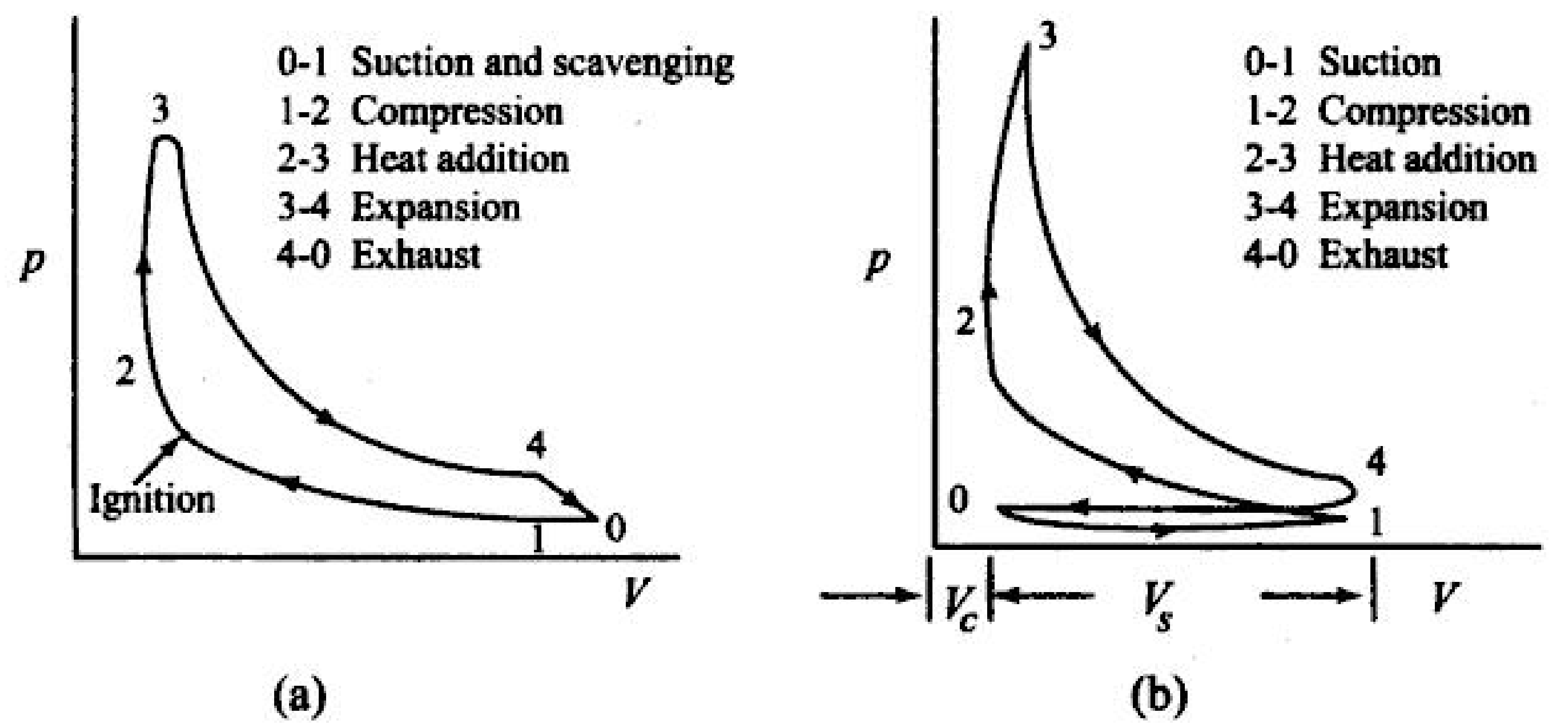


Fig. 1.10 Actual Indicator Diagrams of a Two-Stroke and Four-Stroke SI Engine

cylinder, type of ignition, type of cooling and the cylinder arrangement etc. Details are given in Fig.1.11.

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.

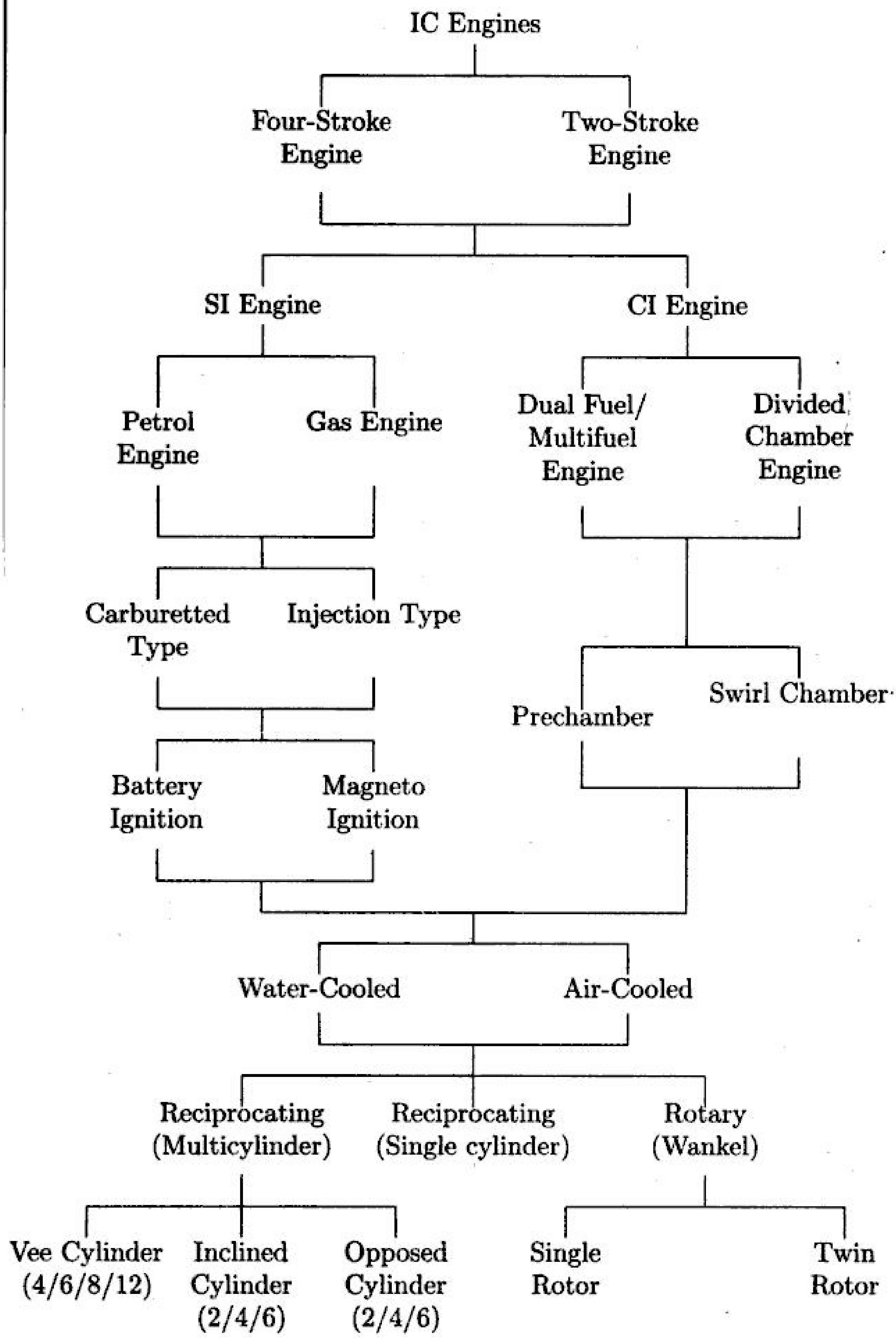


Fig. 1.11 Classification of Internal Combustion Engines

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 multi-cylinder 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.

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

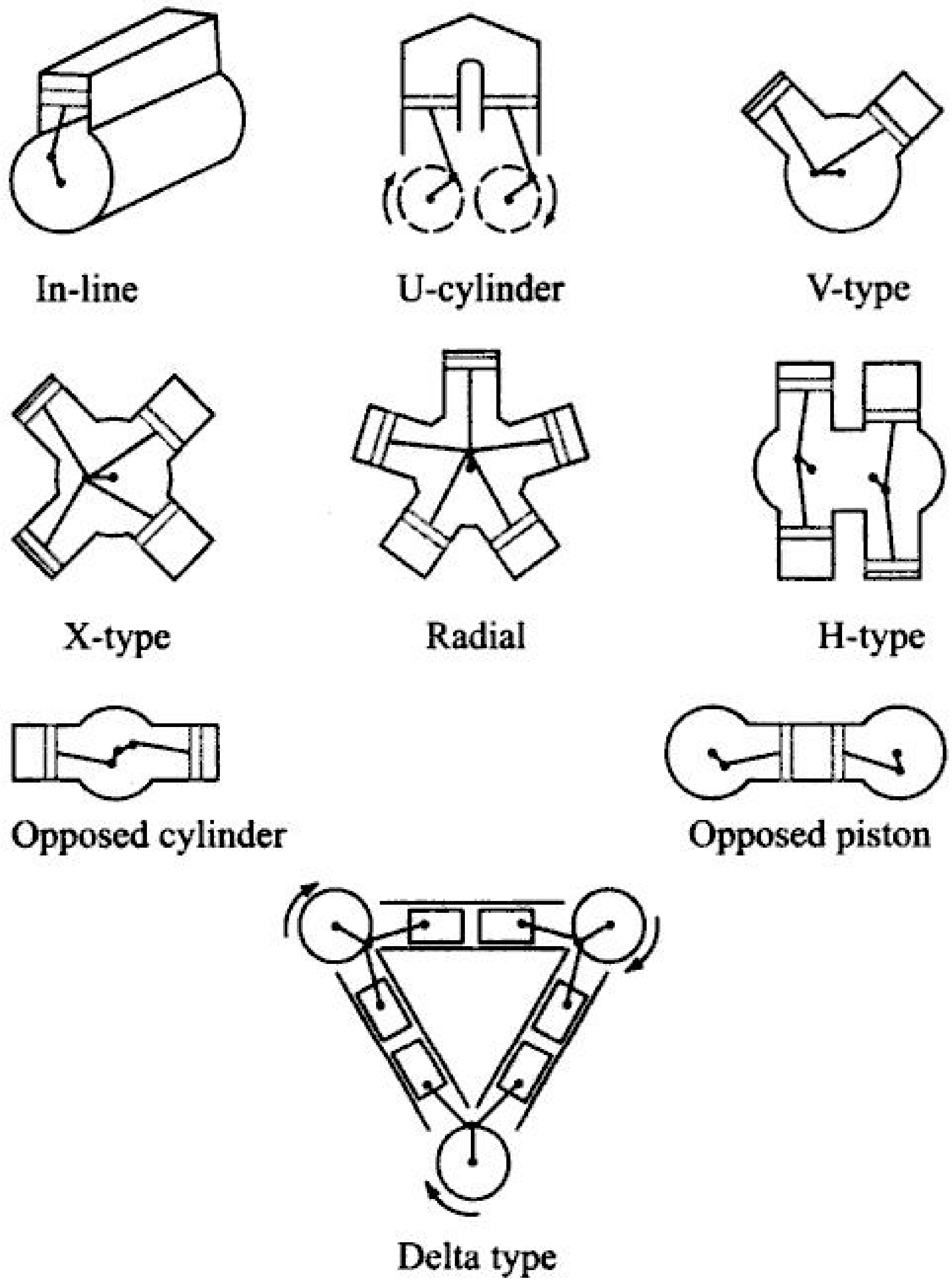


Fig. 1.12 Engine Classification by Cylinder Arrangements

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

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, Railways, Power,Marine	Steam Turbines	Power, Large Marine
Gas turbines	Power, Aircraft, Industrial, Marine	Close Cycle Gas Turbine	Power, Marine

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 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.

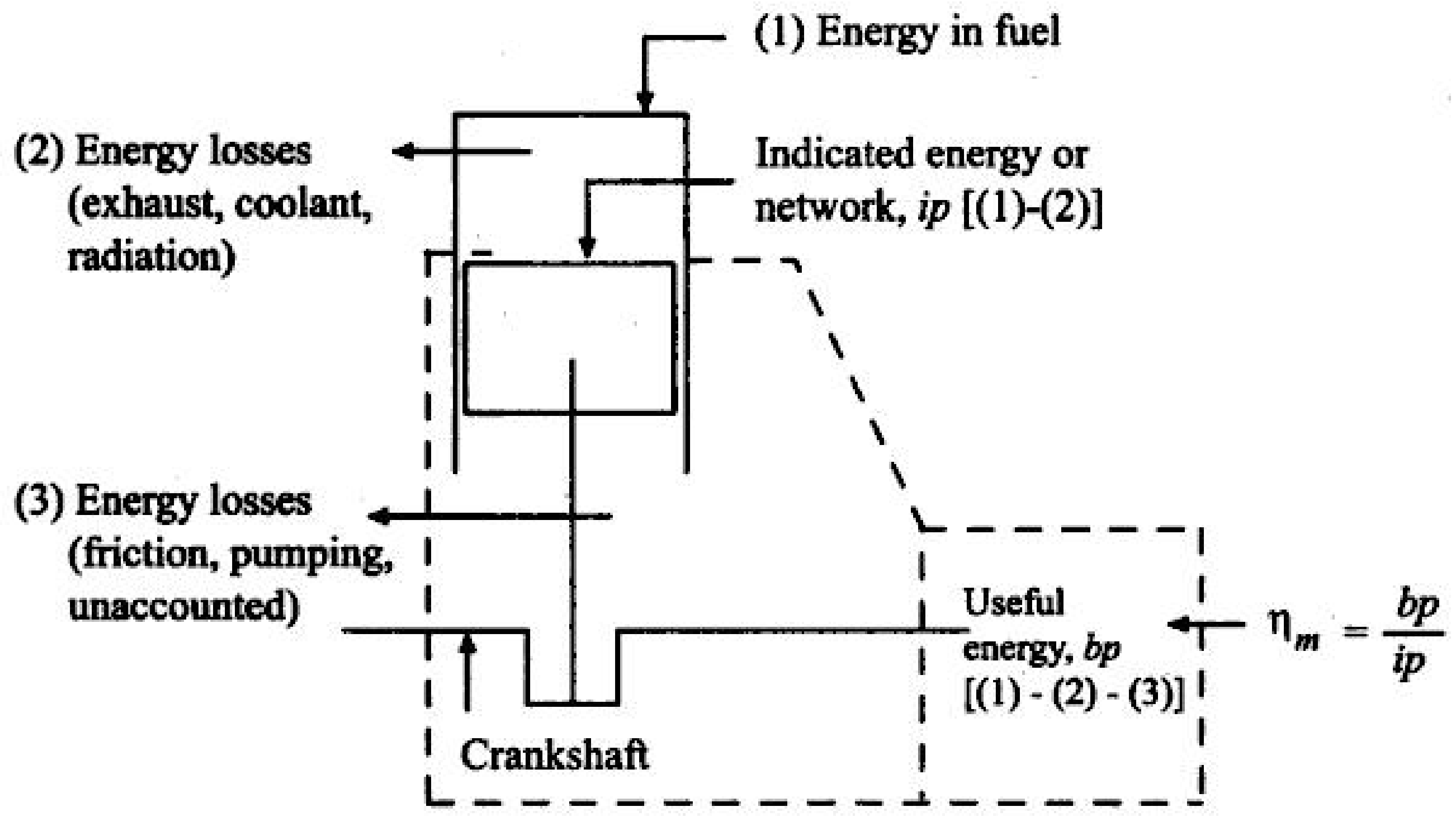


Fig. 1.13 Energy Flow through the Reciprocating Engine

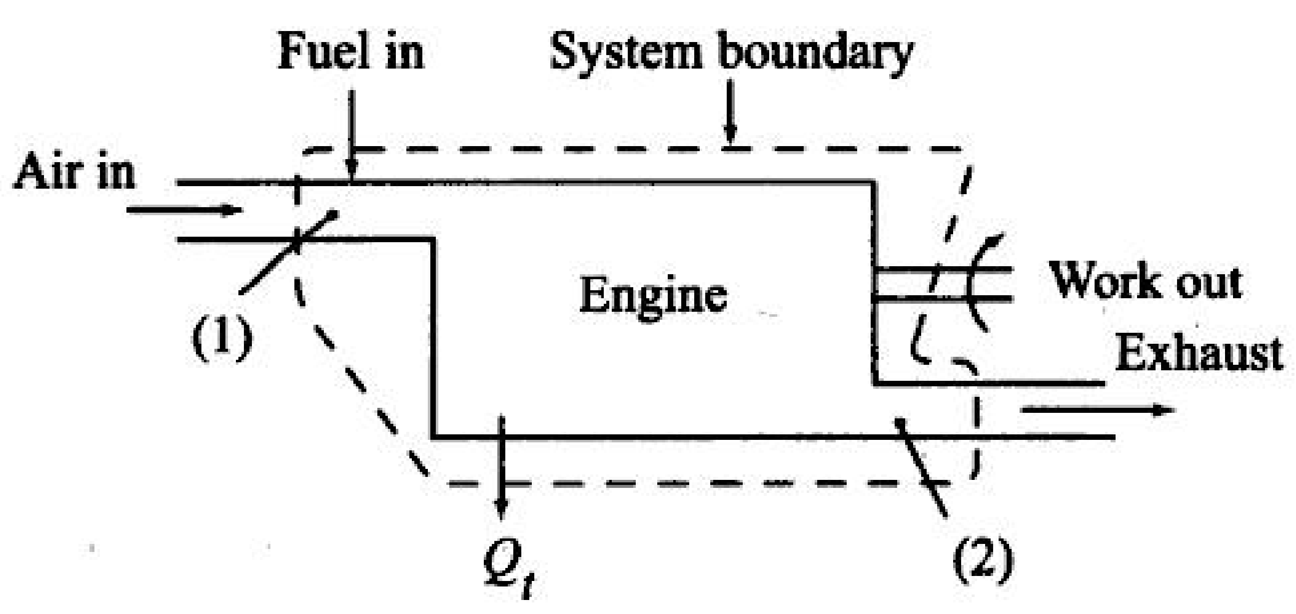


Fig. 1.14 Reciprocating Engine as an Open System

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	$(\overline{s}_p)$
(viii)	Specific power output	(P_s)
(ix)	Specific fuel consumption	(sfc)
(x)	Inlet-valve Mach Index	(Z)
(x)	Fuel-air or air-fuel ratio	$(F/A$ or $A/F)$
(xi)	Calorific value of the fuel	(CV)

Figure 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.

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

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

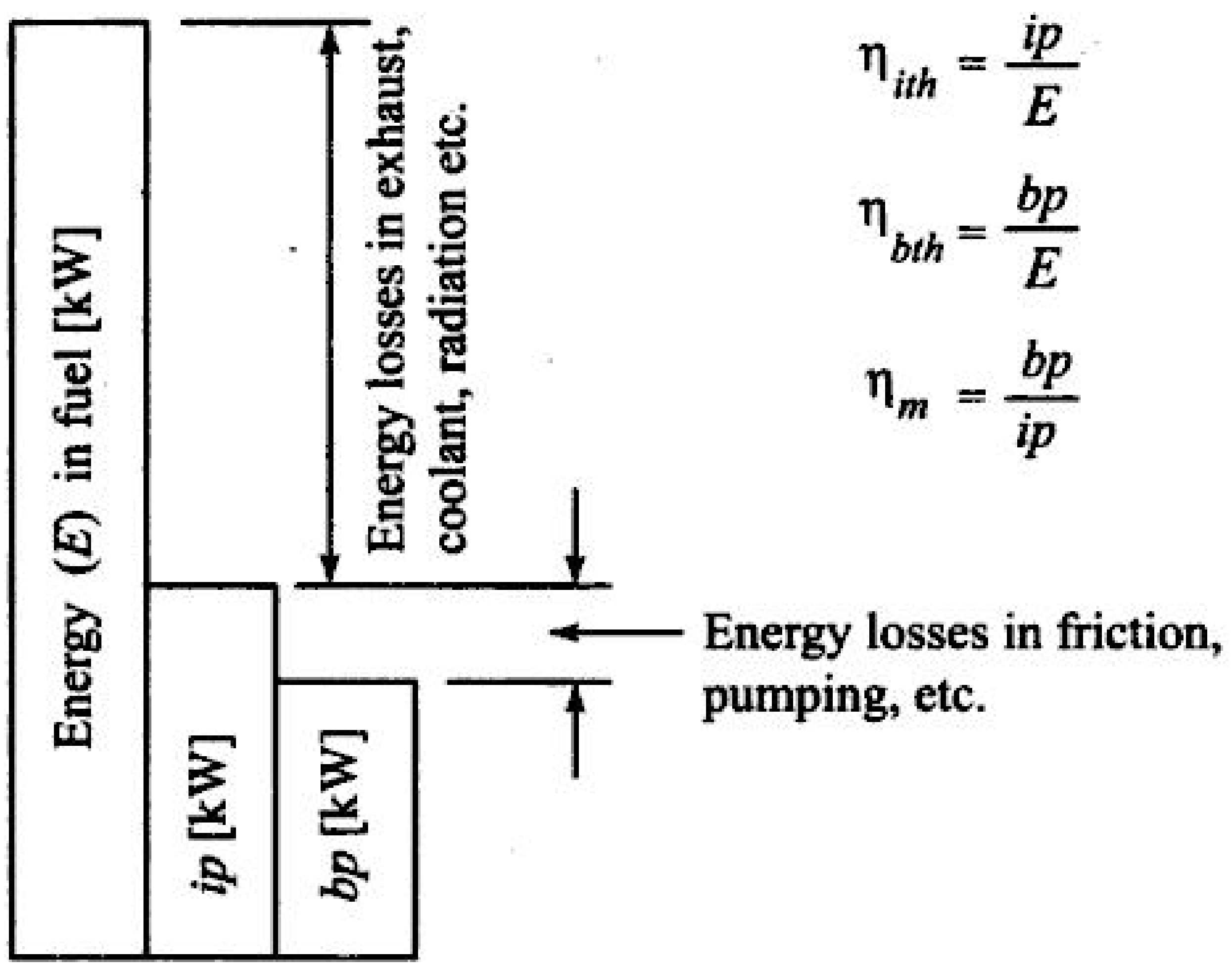


Fig. 1.15 Energy Distribution

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.

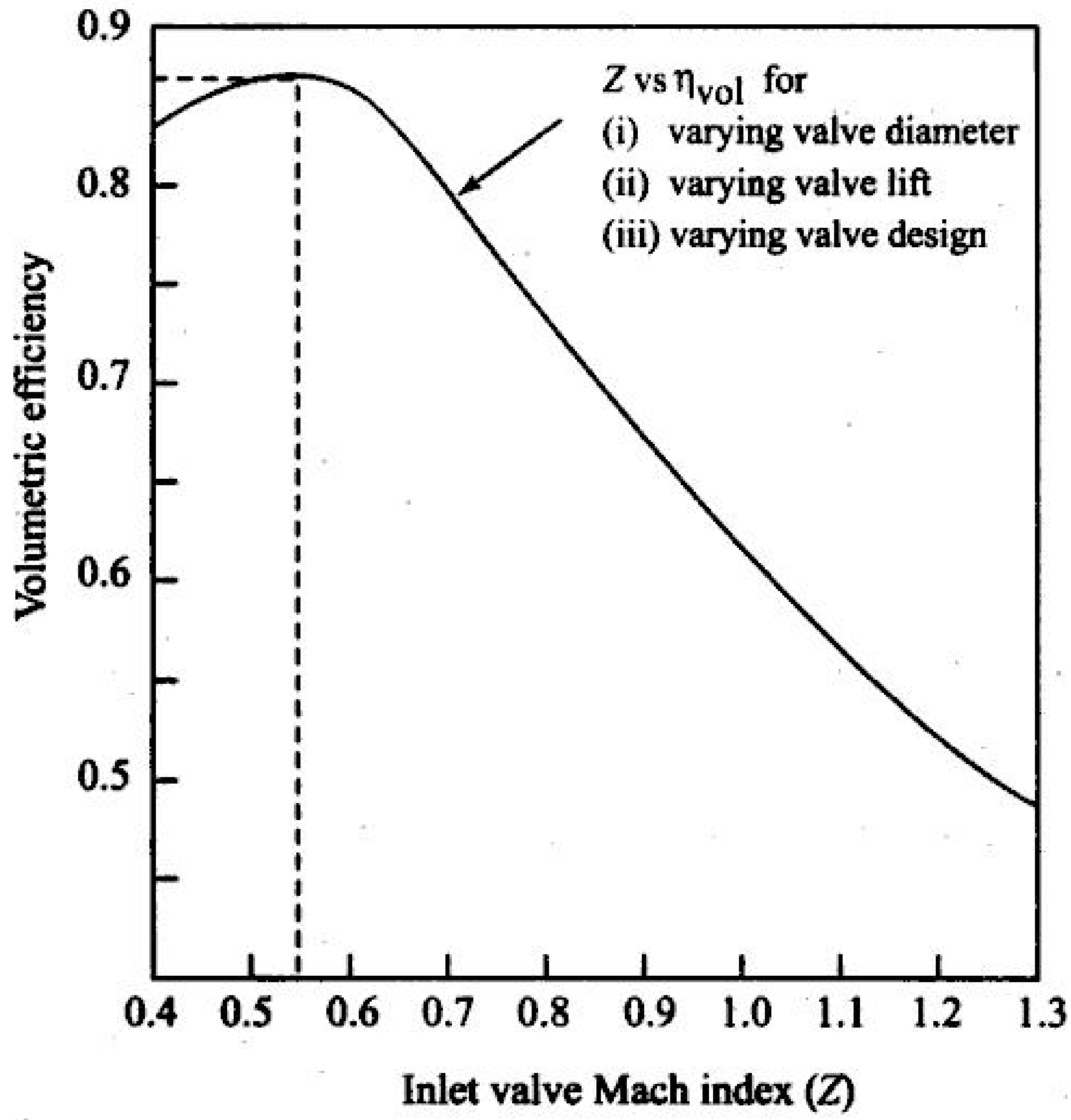


Fig. 1.16 Inlet-Valve Mach Index

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.

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

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

$$d^3 = 343$$

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

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

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

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

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

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

$$ip - bp = 25$$

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

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

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

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

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

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

$$= \mathbf{7.5\ kW} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

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

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

$$= 0.773 = \mathbf{77.3\%} \qquad \overset{\text{Ans}}{\Longleftarrow}$$

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 10:1, 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 $= \dfrac{E_1}{N/120}$

$$= \frac{44.64 \times 120}{N} = \frac{5356.8}{N} \quad (2)$$

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

$$N \approx \mathbf{4500\ rpm} \quad \overset{\text{Ans}}{\Longleftarrow}$$

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

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

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

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

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

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

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

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

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

Volumetric efficiency =

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

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

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

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

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

$$= 7.96 \times 10^5\ \mathrm{N/m^2} = \mathbf{7.96\ bar} \qquad \xleftarrow{\text{Ans}}$$

$$\text{Mean piston speed} = \frac{2 \times 0.1 \times 2000}{60} = 6.67\ \mathrm{m/s} \qquad \xleftarrow{\text{Ans}}$$

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

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

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

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

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

$$\text{Actual volume of } H_2 \times CV = 1.52$$

$$\text{Actual volume of hydrogen taken in} = \frac{1.52 \times 10^6}{11000} = 138.18\ \mathrm{cc}$$

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

$$= 1105.44\ \mathrm{cc}$$

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

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

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

$$= \mathbf{1579.2\ cc} \qquad \xleftarrow{\text{Ans}}$$

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.

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

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

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

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

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

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

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

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.

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

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

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

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

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

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

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

$$fp = 31.5 \times 0.27 = \mathbf{8.5\ kW} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

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

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

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

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

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

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

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

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

Mechanical efficiency at half load

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

Mechanical efficiency at quarter load

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

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

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

Volume flow rate of fuel

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

Indicated thermal efficiency at full load

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

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, $d = L$.

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

$$= \frac{58.24 \times 60000}{\frac{\pi}{4}L^3 \times n \times 1} = 6 \times 10^5$$

$$L^3 n = 7.415 \quad (1)$$

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

$$\bar{s}_p = 12 = \frac{2LN}{60}$$

$$LN = 360 \quad (2)$$

Dividing (1) by (2), gives,

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

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

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

$$= 203 \text{ mm}$$

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

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

$$= \mathbf{1773.4 \text{ rpm}} \quad \overset{\text{Ans}}{\Longleftarrow}$$

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*	

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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

Multiple Choice Questions (choose the most appropriate answer)

1. Advantage of reciprocating IC engines over steam turbine is

   (a) mechanical simplicity
   (b) improved plant efficiency
   (c) lower average temperature
   (d) all of the above

2. The intake charge in a diesel engine consists of

   (a) air alone
   (b) air + lubricating oil
   (c) air + fuel
   (d) air + fuel + lubricating oil

3. Engines of different cylinder dimensions, power and speed can be compared on the basis of

   (a) maximum pressure
   (b) fuel consumption
   (c) mean effective pressure
   (d) unit power

4. Disadvantages of reciprocating IC engine are

   (a) vibration
   (b) use of fossil fuels
   (c) balancing problems
   (d) all of the above

5. Gudgeon pin forms the link between

   (a) piston and big end of connecting rod
   (b) piston and small end of connecting rod
   (c) connecting rod and crank
   (d) big end and small end

6. An IC engine gives an output of 3 kW when the input is 10,000 J/s. The thermal efficiency of the engine is

   (a) 33.3%
   (b) 30%
   (c) 60%
   (d) 66.6%

7. In a four-stroke IC engine cam shaft rotates at

   (a) same speed as crankshaft
   (b) twice the speed of crankshaft
   (c) half the speed of crankshaft
   (d) none of the above

8. Thermal efficiency of CI engine is higher than that of SI engine due to

   (a) fuel used
   (b) higher compression ratio
   (c) constant pressure heat addition
   (d) none of the above

9. SI engines are of

   (a) light weight
   (b) high speed
   (c) homogeneous change of fuel and oil
   (d) all of the above

10. Compression ratio in diesel engines is of the order of

    (a) 5–7
    (b) 7–10
    (c) 10–12
    (d) 14–20

11. In a reciprocating engine with a cylinder diameter of D and stroke of L, the cylinder volume is

    (a) $\frac{\pi}{4}D^2L\times$ clearance volume
    (b) $\frac{\pi}{4}D^2L-$ clearance volume
    (c) $\frac{\pi}{4}D^2L+$ clearance volume
    (d) $\frac{\pi}{4}D^2L\div$ clearance volume

12. Main advantage of a two-stroke engine over four-stroke engine is
    (a) more uniform torque on the crankshaft
    (b) more power output for the cylinder of same dimensions
    (c) absence of valves
    (d) all of the above

13. Engines used for ships are normally
    (a) four-stroke SI engines of very high power
    (b) two-stroke CI engines of very high power
    (c) four-stroke CI engines of high speed
    (d) two-stroke SI engines of high power

14. If L is the stroke and N is the rpm, mean piston speed of two-stroke engine is
    (a) LN
    (b) $\frac{LN}{2}$
    (c) $2LN$
    (d) none of the above

15. Equivalence ratio is
    (a) $\frac{\text{actual fuel} - \text{air ratio}}{\text{stoichiometric fuel} - \text{air ratio}}$
    (b) $\frac{\text{stoichiometric fuel} - \text{air ratio}}{\text{actual fuel} - \text{air ratio}}$
    (c) $\frac{\text{stoichiometric fuel} - \text{air ratio}}{\text{actual air} - \text{fuel ratio}}$

16. The volumetric efficiency of the SI engine is comparatively
    (a) lower than CI engine
    (b) higher than CI engine
    (c) will be same as CI engine
    (d) none of the above

17. The range of volumetric efficiency of a
    (a) 65 – 75%
    (b) 75 – 85%
    (c) 85 – 90%
    (d) 90 – 95%

18. Relative efficiency is the ratio of

    (a) $\frac{\text{actual thermal efficiency}}{\text{mechancial efficiency}}$

    (b) $\frac{\text{actual thermal efficiency}}{\text{air} - \text{standard efficiency}}$

    (c) $\frac{\text{air} - \text{standard efficiency}}{\text{actual thermal efficiency}}$

    (d) $\frac{\text{mechancial efficiency}}{\text{actual thermal efficiency}}$

19. Brake specific fuel consumption is defined as

    (a) fuel consumption per hour
    (b) fuel consumption per km
    (c) fuel consumption per bp
    (d) fuel consumption per brake power hour

20. Engine can be fired with

    (a) solid fuel
    (b) liquid fuel
    (c) gaseous fuel
    (d) any of the above fuels

Ans:					
	1. – (d)	2. – (a)	3. – (c)	4. – (d)	5. – (b)
	6. – (b)	7. – (c)	8. – (b)	9. – (d)	10. – (d)
	11. – (c)	12. – (d)	13. – (b)	14. – (c)	15. – (b)
	16. – (a)	17. – (c)	18. – (b)	19. – (d)	20. – (d)

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.

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

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

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

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.

F.P.S.	:	1 pound	=	1 slug $\times$ 32 ft/s^2
C.G.S.	:	1 dyne	=	1 gm $\times$ 1 m/s^2
M.K.S.	:	1 kgf	=	1 kg $\times$ 9.81 m/s^2
SI	:	1 Newton	=	1 kg $\times$ 1 m/s^2

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

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

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,

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

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

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

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

$$\frac{p_1}{p_t} = \frac{N_1}{N_t} \tag{2.7}$$

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.

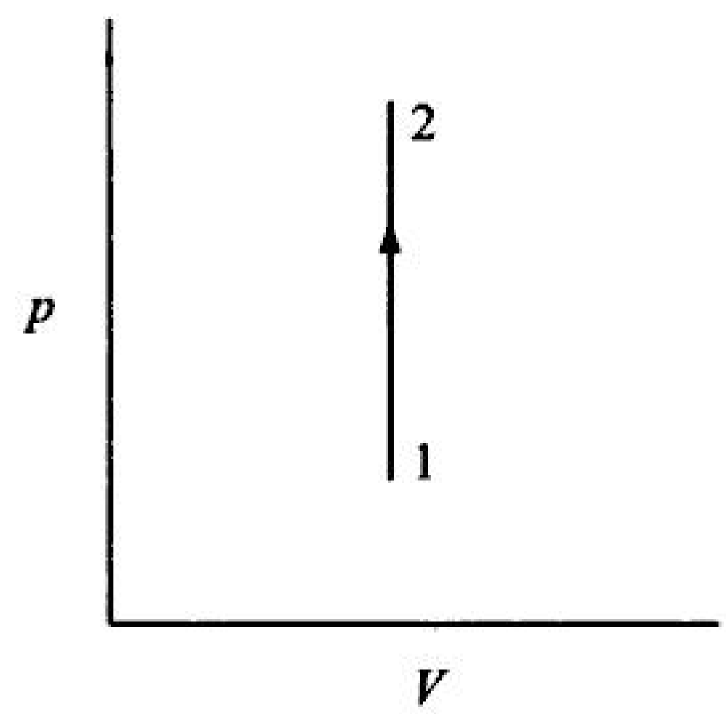

Fig. 2.1 Constant Volume Process

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.,

$$dU = \delta Q = mC_v dT = mC_v(T_2 - T_1) \quad (2.13)$$

For unit mass,

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

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

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

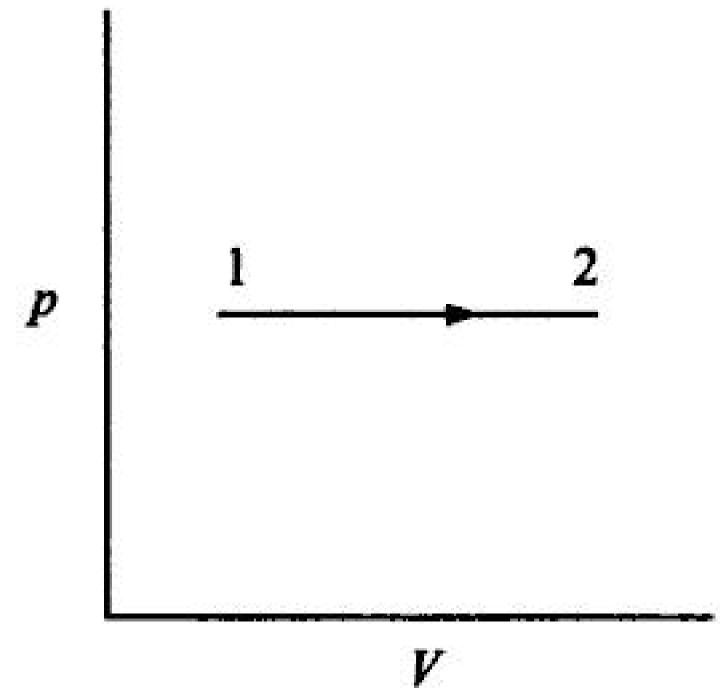

Fig. 2.2 Constant Pressure Process

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.

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

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

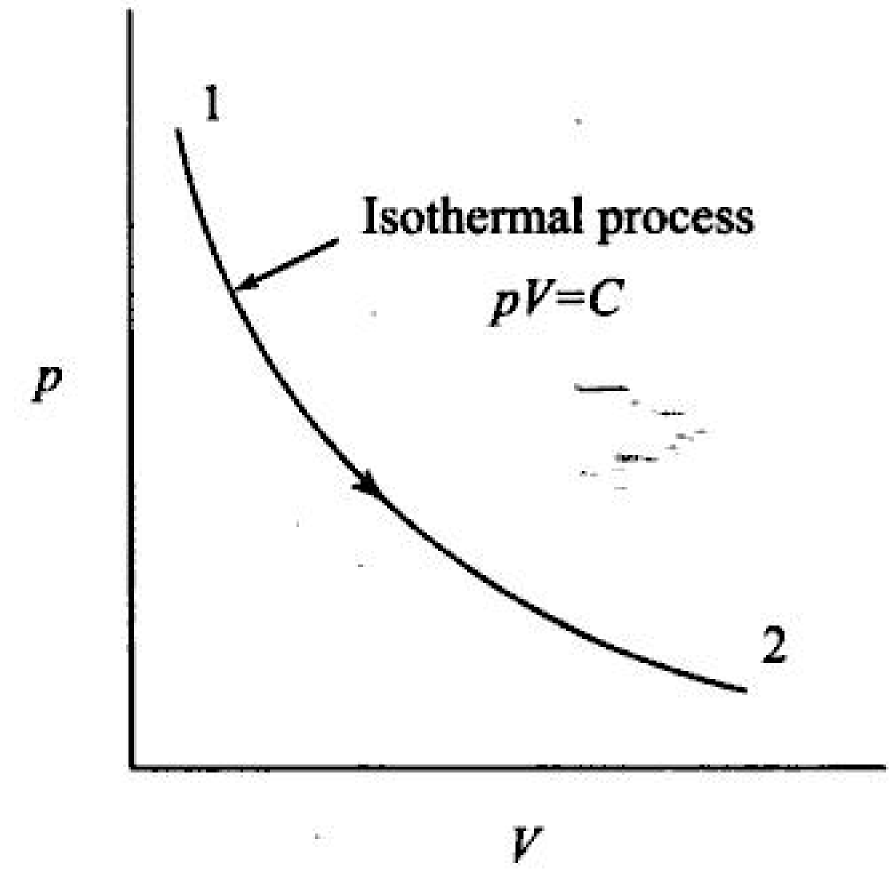

Fig. 2.3 Constant Temperature Process

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

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

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

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

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

when $C = p_1 V_1^\gamma = p_2 V_2^\gamma$,

$$W_{1-2} = \frac{p_2 V_2^\gamma V_2^{1-\gamma} - p_1 V_2^\gamma V_2^{1-\gamma}}{1-\gamma}$$

$$= \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$$

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

$$\begin{aligned} w_{1-2} &= \frac{R(T_1 - T_2)}{\gamma - 1} \\ &= C_v(T_1 - T_2) \quad = \quad -(u_2 - u_1) \\ \frac{p_1 V_1}{T_1} &= \frac{p_2 V_2}{T_2} \end{aligned}$$

therefore,

$$\begin{aligned} \frac{T_2}{T_1} &= \frac{p_2 V_2}{p_1 V_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} \left(\frac{V_2}{V_1}\right) \\ &= \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} \end{aligned} \tag{2.24}$$

$$\begin{aligned} \frac{T_2}{T_1} &= \frac{p_2 V_2}{p_1 V_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} \left(\frac{p_2}{p_1}\right) \\ &= \left(\frac{p_2}{p_1}\right)^{\left(1-\frac{1}{\gamma}\right)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \end{aligned} \tag{2.25}$$

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

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

$$p_1 V_1^n = p_2 V_2^n \tag{2.27}$$

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

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

and

$$w_{1-2} = \frac{p_1 V_1 - p_2 V_2}{n - 1} \tag{2.29}$$

2.7.6 Heat Transfer During Polytropic Process

Heat transfer per unit mass,

$$
\begin{aligned}
q_{1-2} &= (u_2 - u_1) + \int_1^2 pdV \qquad (2.30)\\
&= C_v(T_2 - T_1) + \frac{R(T_2 - T_1)}{1 - n}\\
&= \left(C_v + \frac{R}{1-n}\right)(T_2 - T_1)\\
&= \left(C_v + \frac{C_p - C_v}{1-n}\right)(T_2 - T_1)\\
&= \left(\frac{C_p - nC_v}{1-n}\right)(T_2 - T_1)\\
&= \left(\frac{C_v}{1-n}\right)\left(\frac{C_p}{C_v} - n\right)(T_2 - T_1)\\
&= \left(\frac{C_v}{1-n}\right)(\gamma - n)(T_2 - T_1)\\
&= \left(\frac{\gamma - n}{1-n}\right)C_v(T_2 - T_1) \qquad (2.31)
\end{aligned}
$$

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

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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	$p_1V_1^{n} = p_2V_2^{n}$ $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}$	$\left(\frac{\gamma-n}{1-n}\right)mC_v(T_2 - T_1)$	$\frac{p_1V_1 - p_2V_2}{n - 1}$	$\frac{n}{n-1}(p_2V_2 - p_1V_1)$	$mC_v(T_2 - T_1)$

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 :

$$\textit{Heat lost to cooling water and environment} = \frac{140 \times 60}{10} = 840 \text{ kJ/kg}$$

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

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

therefore,

$$\begin{aligned} w &= (U_1 - U_2) + (p_1V_1 - p_2V_2) + \frac{C_1^2 - C_2^2}{2} + q \\ &= -250 \times 10^3 + (1 \times 10^5 \times 0.5 - 8 \times 10^5 \times 0.2) \\ &\qquad + \frac{5^2 - 10^2}{2} - 140 \times 10^3 \\ &= -500037.5 \text{ J/kg} \approx -500 \text{ kJ/kg} \\ W &= mw = 10 \times (-500) \\ &= -5000 \text{ kJ/min} = -83.33 \text{ kJ/s} \end{aligned}$$

Power input = **83.33 N/m²kW** Ans

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,*

$$\frac{A_1 C_1}{v_1} = \frac{A_2 C_2}{v_2}$$

$$\frac{A_1}{A_2} = \frac{v_1 C_2}{v_2 C_1} = \frac{0.5 \times 10}{0.2 \times 5} = 5$$

$$\frac{D1}{D2} = \sqrt{5} = \mathbf{2.236}$$ Ans

2.3 A tank of volume 0.1 m^3 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; $V_1 = V = 0.1$ m^3
O_2 : $m_2 = 1.50$ kg; $t_2 = t = 20$ °C; $V_2 = V = 0.1$ m^3
CO_2 : $m_3 = 0.75$ kg; $t_3 = t = 20$ °C; $V_3 = V = 0.1$ m^3

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

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

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

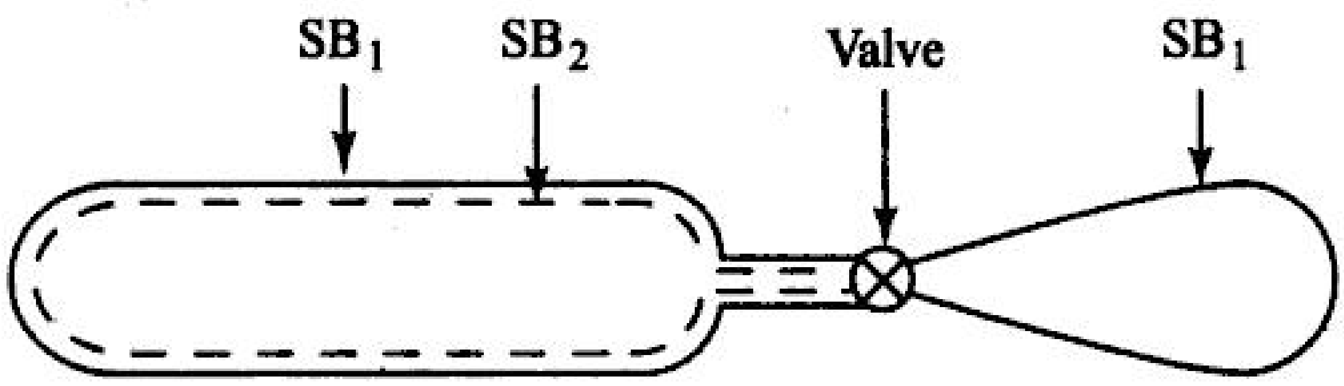
$$R = \frac{(m_{N_2} R_{N_2} + m_{O_2} R_{O_2} + m_{CO_2} R_{CO_2})}{m}$$

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

$$= \mathbf{274.98\ N/m^2 J/kg\ K}$$ 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,*

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

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

$$= \mathbf{-30390\ N/m^2 Nm}$$ Ans

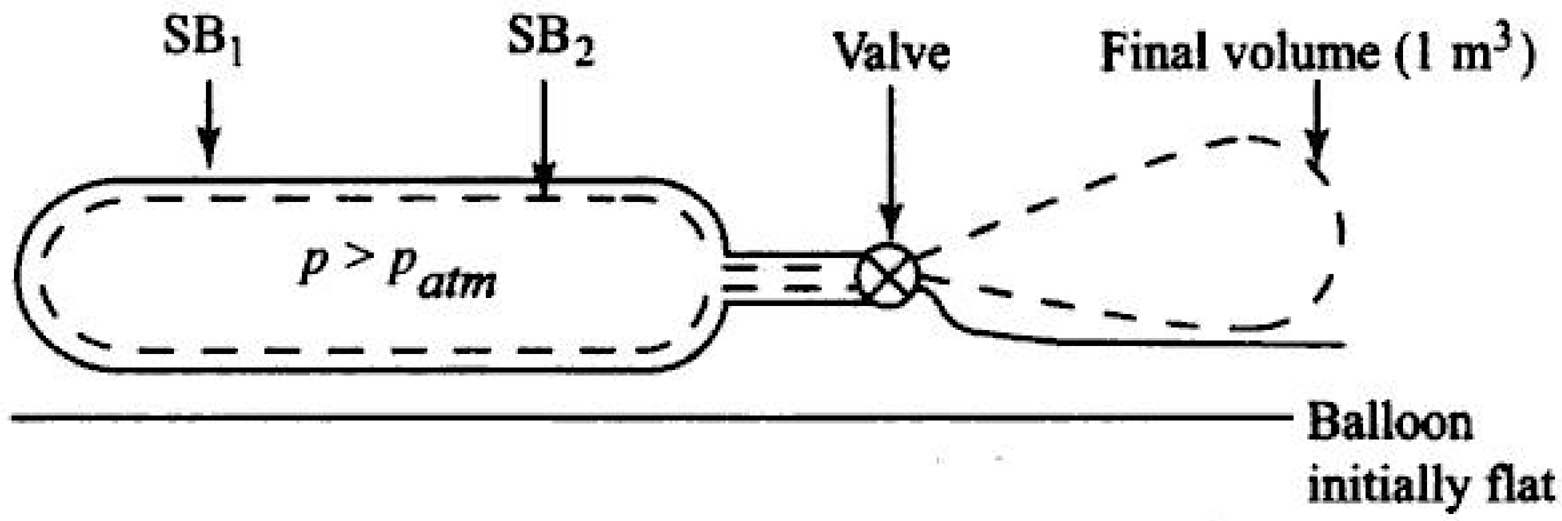
The work is negative as the boundary is contracting.

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,

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

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

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

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

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

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

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

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

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17. An isobaric process is a

    (a) constant pressure process
    (b) constant volume process
    (c) constant temperature process
    (d) constant entropy process

18. An isothermal process is a

    (a) constant pressure process
    (b) constant volume process
    (c) constant temperature process
    (d) constant entropy process

19. An isentropic process is a

    (a) constant pressure process
    (b) constant volume process
    (c) constant temperature process
    (d) constant entropy process

20. The statement that the entropy of a pure substance in complete thermodynamic equilibrium becomes zero at the absolute zero of temperature is known as

    (a) zeroth law of thermodynamics
    (b) first law of thermodynamics
    (c) second law of thermodynamics
    (d) third law of thermodynamics

Ans:					
	1. – (c)	2. – (b)	3. – (c)	4. – (a)	5. – (b)
	6. – (c)	7. – (c)	8. – (d)	9. – (c)	10. – (a)
	11. – (d)	12. – (a)	13. – (b)	14. – (c)	15. – (a)
	16. – (b)	17. – (a)	18. – (c)	19. – (d)	20. – (d)

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

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

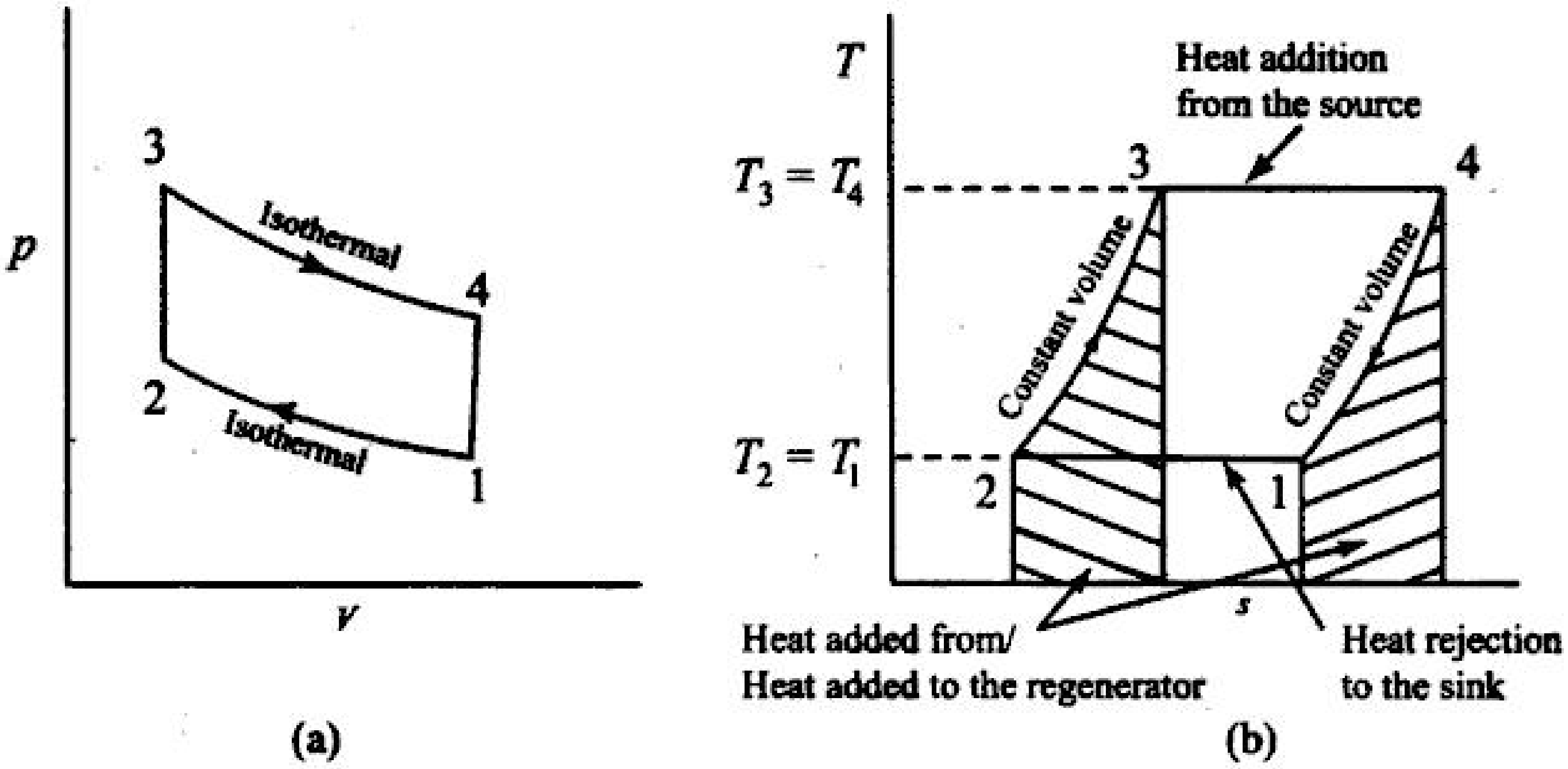


Fig. 3.3 Stirling Cycle

the amount of heat addition and rejection during constant volume processes is same. Hence, the efficiency of the cycle is given as

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

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

$$\eta_{Stirling} = \frac{T_3 - T_1}{T_3} \qquad (3.12)$$

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.

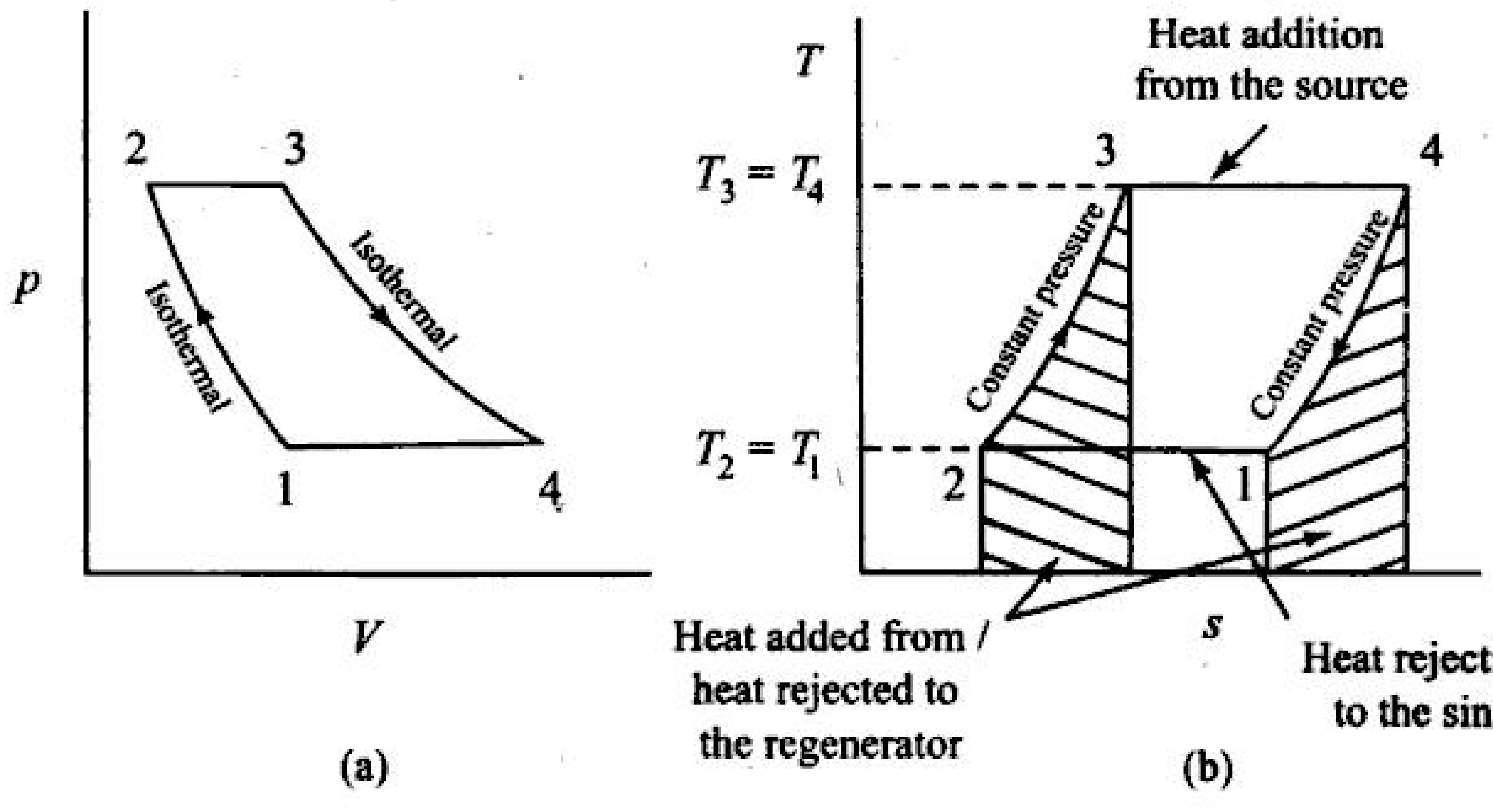


Fig. 3.4 Ericsson Cycle

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.

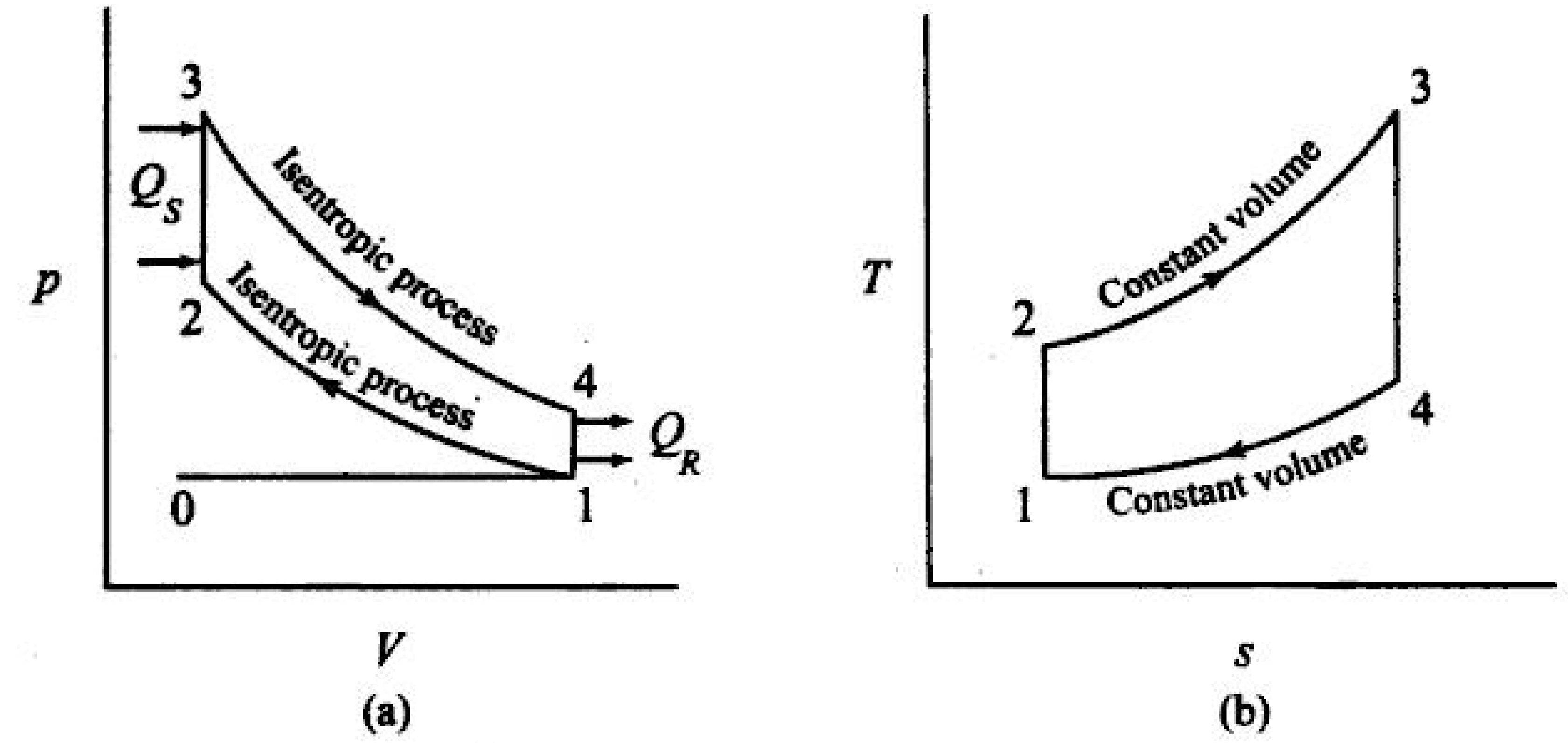


Fig. 3.5 Otto Cycle

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 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

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

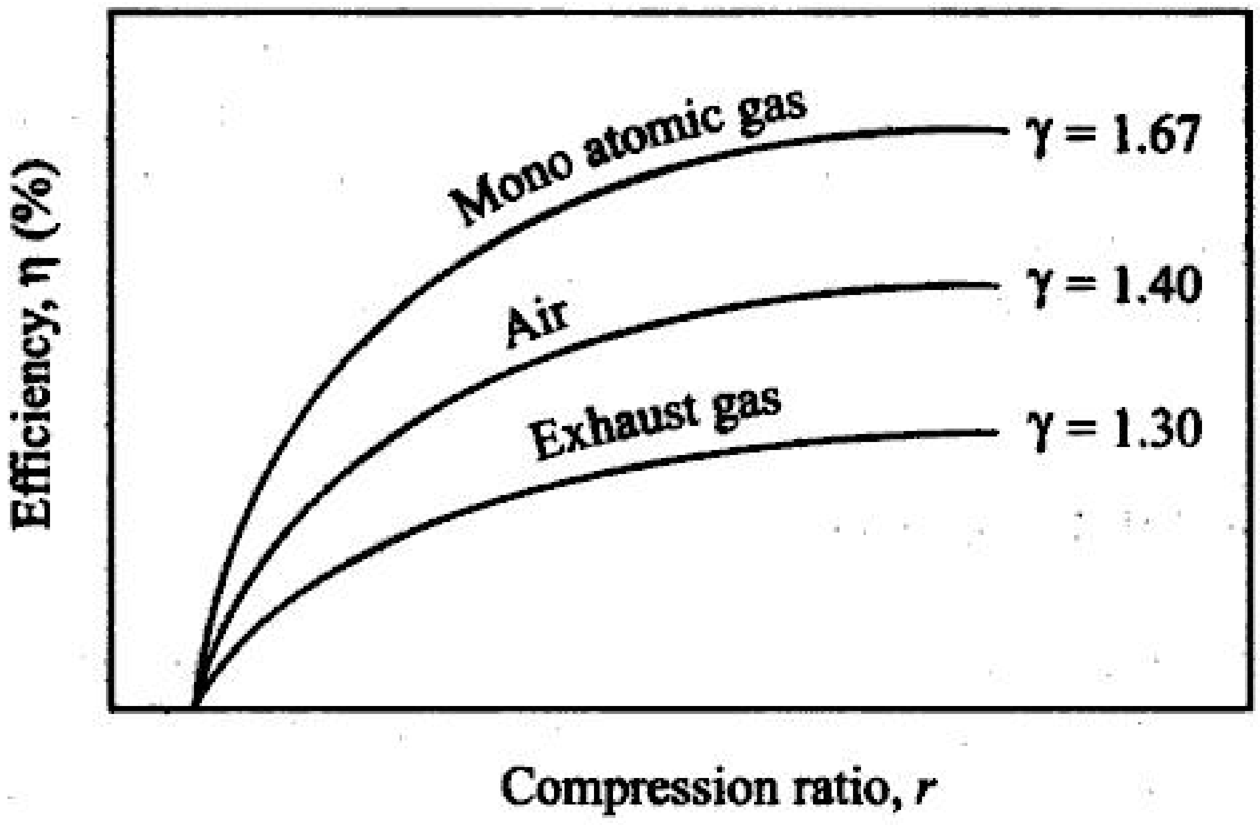


Fig. 3.6 Effect of r and γ on Efficiency for Otto Cycle

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

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

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

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

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

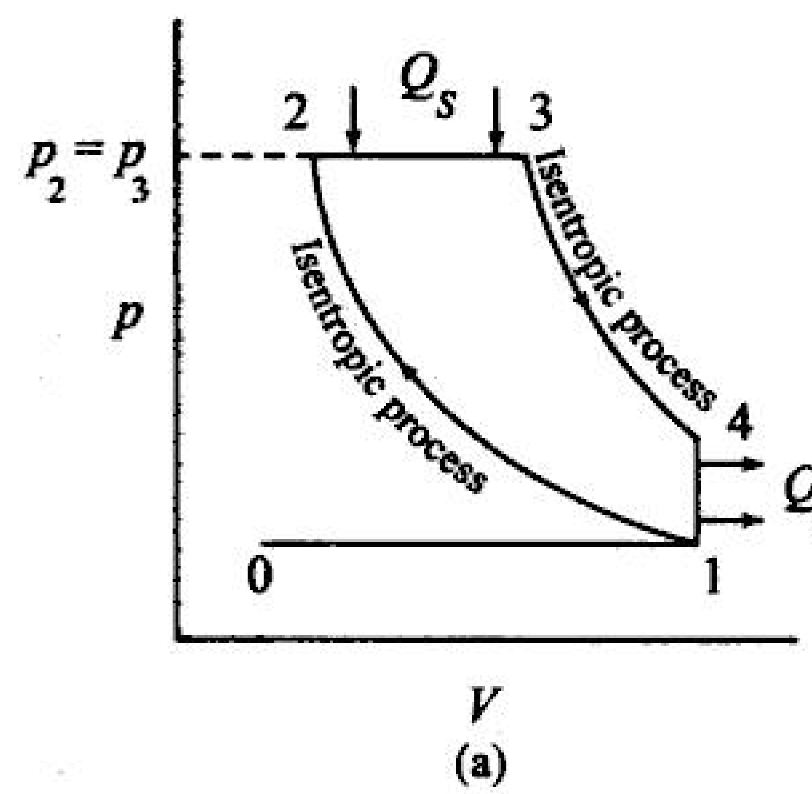
$$p_m = \frac{\frac{1}{\gamma-1}p_1V_1(r_p - 1)\left(r^{(\gamma-1)} - 1\right)}{V_2(r - 1)}$$

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

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.



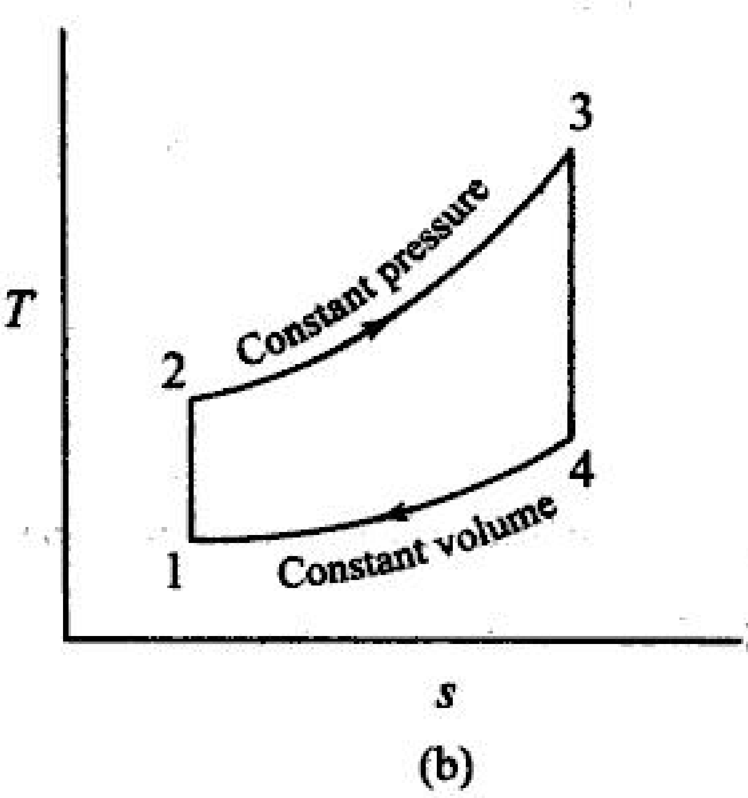


Fig. 3.7 Diesel Cycle

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.

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$$\text{Peak pressure}, p_3 = 3.65 \times 12.28 \times 10^5$$

$$= 44.82 \times 10^5 \text{ N/m}^2 = \mathbf{44.82\ bar} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Work output} = \text{Area of } p\text{-}V \text{ diagram}$$

$$= \text{Area under } (3 - 4) - \text{Area under } (2 - 1)$$

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

$$= \frac{mR}{\gamma - 1}[(T_3 - T_4) - (T_2 - T_1)]$$

$$R = C_p - C_v = 1.004 - 0.717$$

$$= 0.287 \text{ kJ/kg K}$$

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

$$= 6^{0.4} = 2.048$$

$$T_4 = \frac{T_3}{2.048} = \frac{2246.8}{2.048} = 1097.1 \text{ K}$$

$$\text{Work output/kg} = \frac{0.287}{0.4} \times [(2246.8 - 1097.1) - (615 - 300)]$$

$$= \mathbf{598.9\ kJ} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\eta_{Otto} = 1 - \frac{1}{r^{(\gamma-1)}} = 1 - \frac{1}{6^{0.4}} = 0.5116$$

$$= \mathbf{51.16\%} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

3.8 A spark-ignition engine working on ideal Otto cycle has the compression ratio 6. The initial pressure and temperature of air are 1 bar and 37 °C. The maximum pressure in the cycle is 30 bar. For unit mass flow, calculate (i) p, V and T at various salient points of the cycle and (ii) the ratio of heat supplied to the heat rejected. Assume $\gamma = 1.4$ and $R = 8.314$ kJ/kmol K.

Solution

Consider point 1,

$$n = \frac{m}{M} = \frac{1}{29}$$

$$V_1 = \frac{nRT_1}{p_1} = \frac{1 \times 8314 \times 310}{29 \times 10^5} = \mathbf{0.889\ m^3} \qquad \stackrel{\text{Ans}}{\Longleftarrow}$$

Solution

$$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{5.5^{0.4}} = 49.43\%$$

(Refer Fig.3.14)

$$\eta_{Atkinson} = 1 - \frac{\gamma(e-r)}{e^{\gamma} - r^{\gamma}}$$

$$r = 5.5$$

$$e = \frac{V_{4'}}{V_3} = \left(\frac{p_3}{p_{4'}}\right)^{\frac{1}{\gamma}} = \left(\frac{25}{1}\right)^{\frac{1}{1.4}} = 9.966$$

$$e^{\gamma} = 9.966^{1.4} = 25$$

$$r^{\gamma} = 5.5^{1.4} = 10.88$$

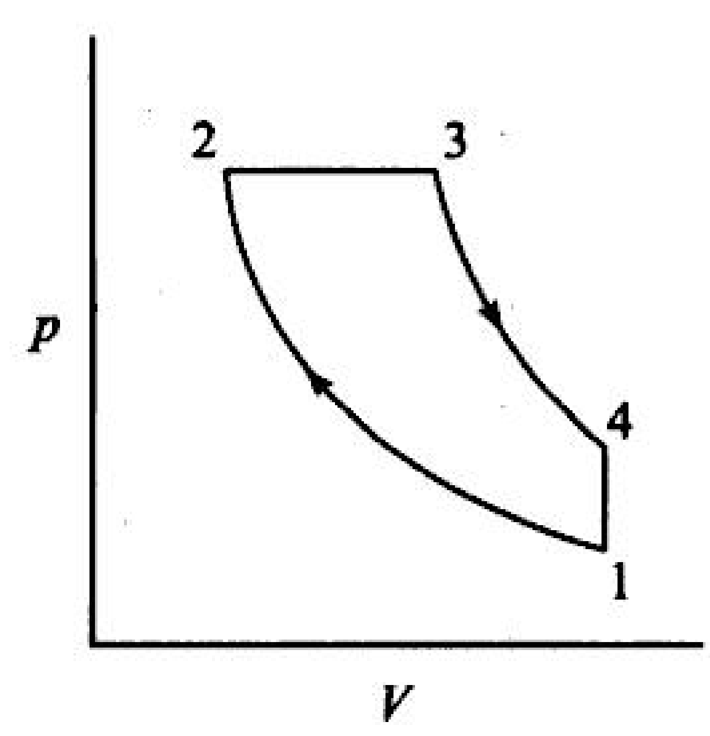
$$\eta_{Atkinson} = 1 - \frac{1.4 \times (9.966 - 5.5)}{25 - 10.88} = 55.72\%$$

$$\frac{\eta_{Atkinson}}{\eta_{Otto}} = \frac{55.72}{49.43} = \mathbf{1.127} \quad \xLeftarrow{\text{Ans}}$$

DIESEL CYCLE

3.11 A Diesel engine has a compression ratio of 20 and cut-off takes place at 5% of the stroke. Find the air-standard efficiency. Assume $\gamma = 1.4$.

Solution



$$r = \frac{V_1}{V_2} = 20$$

$$V_1 = 20V_2$$

$$V_s = 20V_2 - V_2 = 19V_2$$

$$V_3 = 0.05V_s + V_2 = 0.05 \times 19V_2 + V_2 = 1.95V_2$$

$$r_c = \frac{V_3}{V_2} = \frac{1.95V_2}{V_2} = 1.95$$

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

$$= 1 - \frac{1}{20^{0.4}} \times \left[\frac{1.95^{1.4} - 1}{1.4 \times (1.95 - 1)}\right] = 0.649$$

$$= \mathbf{64.9\%}$$ Ans

3.12 Determine the **ideal efficiency** of the diesel engine having a cylinder with bore 250 mm, stroke 375 mm and a clearance volume of 1500 cc, with fuel cut-off occurring at 5% of the stroke. Assume $\gamma = 1.4$ for air.

Solution

$$V_s = \frac{\pi}{4} d^2 L = \frac{\pi}{4} \times 25^2 \times 37.5$$

$$= 18407.8 \text{ cc}$$

$$r = 1 + \frac{V_s}{V_c} = 1 + \frac{18407.8}{1500} = 13.27$$

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

$$r_c = \frac{V_3}{V_2}$$

Cut-off volume $= V_3 - V_2 = 0.05V_s$

$$= 0.05 \times 12.27V_c$$

$$V_2 = V_c$$

$$V_3 = 1.6135V_c$$

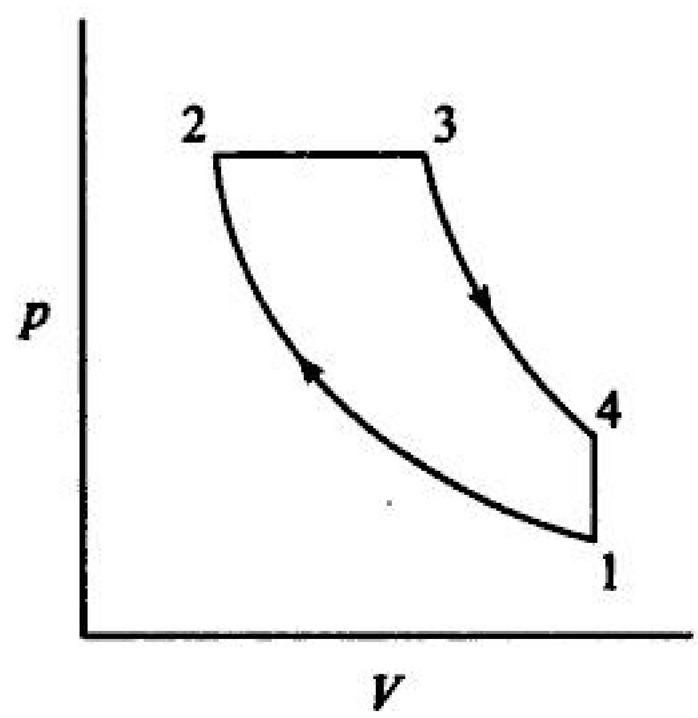
$$r_c = \frac{V_3}{V_2} = 1.6135$$

$$\eta = 1 - \frac{1}{13.27^{0.4}} \times \frac{1.6135^{1.4} - 1}{1.4 \times (1.6135 - 1)}$$

$$= 0.6052 = \mathbf{60.52\%}$$ Ans

3.13 In an engine working on Diesel cycle inlet pressure and temperature are 1 bar and 17 °C respectively. Pressure at the end of adiabatic compression is 35 bar. The ratio of expansion i.e. after constant pressure heat addition is 5. Calculate the heat addition, heat rejection and the efficiency of the cycle. Assume $\gamma = 1.4$, $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K.

Solution



Consider the process 1 – 2

$$\frac{V_1}{V_2} = r = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} = \left(\frac{35}{1}\right)^{\frac{1}{1.4}} = 12.674$$

$$\text{Cut-off ratio} = \frac{V_3}{V_2} = \frac{V_3}{V_1} \times \frac{V_1}{V_2}$$

$$= \frac{\text{Compression ratio}}{\text{Expansion ratio}} = \frac{12.674}{5} = 2.535$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{35}{1}\right)^{0.286} = 2.76$$

$$T_2 = 2.76 \times 290 = 801.7 \text{ K}$$

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

$$= \quad 1 - \frac{1}{1.4} \times \frac{1}{15^{0.4}} \times \left[\frac{\left(\frac{15}{10}\right)^{1.4} - 1}{\left(\frac{15}{10}\right) - 1} \right] \quad = \quad 0.63$$

$$= \quad \mathbf{63\%}$$ $\stackrel{\text{Ans}}{\Longleftarrow}$

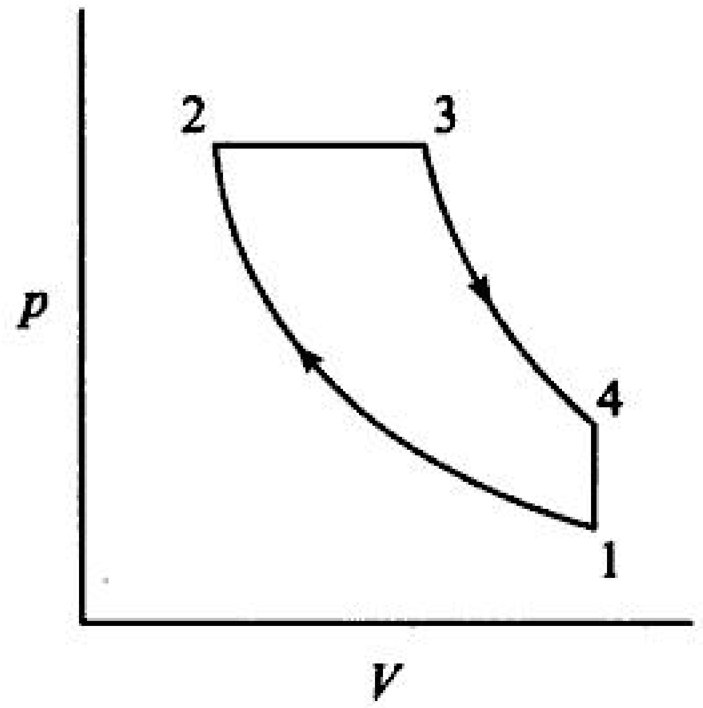
3.15 A Diesel engine works on Diesel cycle with a compression ratio of 15 and cut-off ratio of 1.75. Calculate the air-standard efficiency assuming $\gamma = 1.4$.

Solution

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

$$= \quad 1 - \frac{1}{15^{0.4}} \times \frac{1}{1.4} \times \left(\frac{1.75^{1.4} - 1}{1.75 - 1} \right) = 0.617 = \mathbf{61.7\%}$$ $\stackrel{\text{Ans}}{\Longleftarrow}$

3.16 A Diesel cycle operates at a pressure of 1 bar at the beginning of compression and the volume is compressed to $\frac{1}{16}$ of the initial volume. Heat is supplied until the volume is twice that of the clearance volume. Calculate the mean effective pressure of the cycle. Take $\gamma = 1.4$.



Solution

$$V_1 \quad = \quad 16V_2 \quad \text{and} \quad V_3 \quad = \quad 2V_2$$

Swept volume

$$= \quad V_1 - V_2 = (r-1)V_2 = 15\ V_2$$

$$V_2 \quad = \quad \frac{V_s}{15}$$

Consider the process 1 – 2

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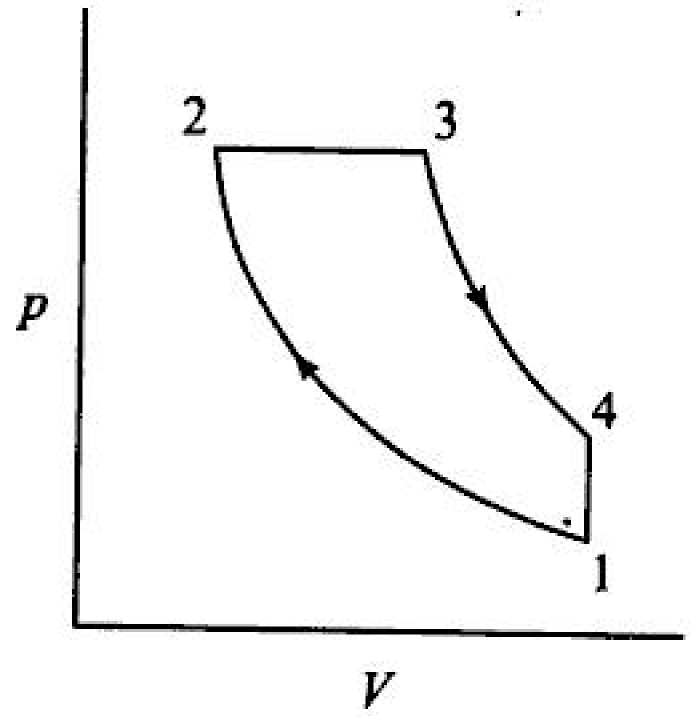


$$\frac{T_2}{T_1} = r^{(\gamma-1)} = 9^{0.4} = 2.408$$

$$T_2 = 308 \times 2.408 = 741.6 \text{ K} = \mathbf{468.6°C}$$ Ans

Consider the process 2 – 3,

$$p_3 = p_2 = 22.32 \times 10^5 \text{ N/m}^2 = \mathbf{22.32 \text{ bar}}$$ Ans

$$T_3 = 1773 \text{ K} = \mathbf{1500° \ C}$$ Ans

Consider the process 3 – 4,

$$\frac{T_3}{T_4} = r_e^{(\gamma-1)}$$

$$r_c = \frac{T_3}{T_2} = \frac{1773}{741.6} = 2.39$$

$$r_e = \frac{r}{r_c} = \frac{9}{2.391} = 3.764$$

$$\frac{T_3}{T_4} = 1.7$$

$$T_4 = \frac{T_3}{1.7} = \frac{1773}{1.7} = 1042.9 \text{ K}$$

$$= \mathbf{769.9° \ C}$$ Ans

$$\frac{p_3}{p_4} = r_e^{\gamma} = 3.764^{1.4} = 6.396$$

$$p_4 = \frac{p_3}{6.396} = \frac{22.32 \times 10^5}{6.396}$$

$$= 3.49 \times 10^5 \text{ N/m}^2 = \mathbf{3.49 \text{ bar}}$$ Ans

$$\eta_{Cycle} = \frac{\text{Work output}}{\text{Heat added}} = 1 - \frac{\text{Heat rejected}}{\text{Heat added}}$$

$$= 1 - \frac{q_{4-1}}{q_{2-3}}$$

$$q_{4-1} = C_v (T_4 - T_1)$$

$$= 0.717 \times (1042.9 - 308) = 526.9 \text{ kJ/kg}$$

$$q_{2-3} = C_p (T_3 - T_2)$$

$$= 1.004 \times (1773 - 741.6) = 1035.5 \text{ kJ/kg}$$

$$\eta_{Cycle} = 1 - \frac{526.9}{1035.5} = 0.4912 = \mathbf{49.12\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\text{Work output} = q_{2-3} - q_{4-1}$$

$$= 1035.5 - 526.9 = 508.6 \text{ kJ/kg}$$

$$\text{Power output} = \text{Work output} \times \dot{m}_a$$

$$\dot{m}_a = \frac{p_1 V_1}{RT_1} \times \frac{N}{2}$$

$$R = C_p - C_v = 0.287 \text{ kJ/kg K}$$

$$V_1 = V_s + V_c = \frac{9}{8} V_s$$

$$V_s = 6 \times \frac{\pi}{4} d^2\ L = 6 \times \frac{\pi}{4} \times 10^2 \times 12$$

$$= 5654.8 \text{ cc} = 5.65 \times 10^{-3} \text{ m}^3$$

$$V_1 = 5.65 \times 10^{-3} \times \frac{9}{8} = 6.36 \times 10^{-3} \text{ m}^3$$

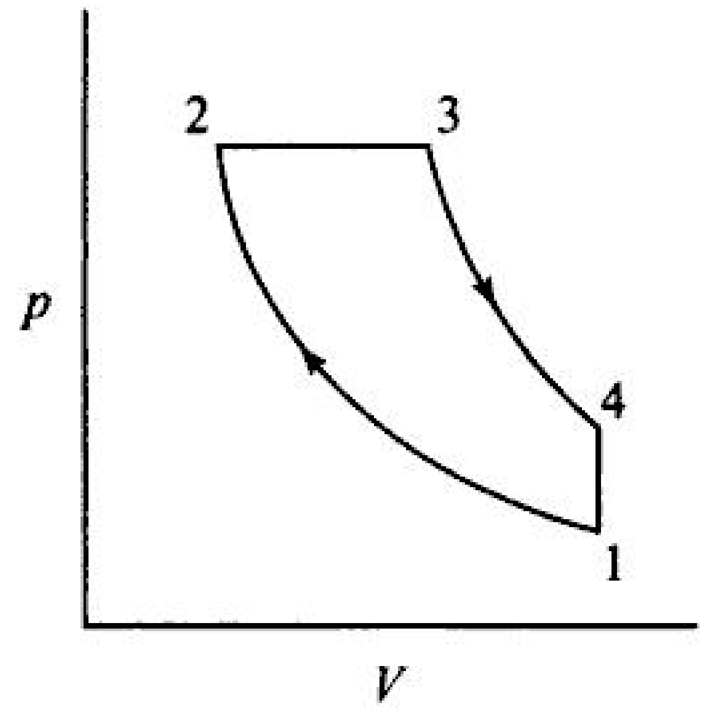
$$\dot{m}_a = \frac{1.03 \times 10^5 \times 6.36 \times 10^{-3} \times 30}{287 \times 308 \times 2} = 0.111 \text{ kg/s}$$

$$\text{Power output} = 508.6 \times 0.111 = \mathbf{56.45 \text{ kW}} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

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

$$\text{Work output} = p_m \times V_s = \text{Area } 1234$$



$$= \text{Area under } 2-3 + \text{Area under } 3-4 - \text{Area under } 2-1$$

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

$$r = \frac{V_1}{V_2} = 1 + \frac{V_s}{V_c} = 12$$

$$V_s = 11V_c \quad ; \quad V_2 = V_c$$

$$V_1 = V_4 = 12V_2 = 12V_c$$

$$V_3 = r_cV_2 = r_cV_c$$

$$\frac{p_2}{p_1} = r^\gamma = 12^{1.4} \doteq 32.42$$

$$p_2 = 32.42 \times 1.03 \times 10^5 = 33.39 \times 10^5 \text{ N/m}^2 = p_3$$

$$\frac{p_3}{p_4} = \left(\frac{r}{r_c}\right)^{1.4} = \frac{12^{1.4}}{r_c^{1.4}} = \frac{32.42}{r_c^{1.4}}$$

$$p_4 = \frac{33.39}{32.42} \times r_c^{1.4} \times 10^5 = 1.03r_c^{1.4} \times 10^5$$

$$\text{Area } 1234 = 33.39(r_cV_c - V_c) + \frac{33.39 \times r_cV_c - 1.03r_c^{1.4} \times 12V_c}{0.4} - \frac{33.39 \times V_c - 1.03 \times 12V_c}{0.4} \times 10^5$$

$$p_m \times V_s = 8 \times 11V_c \times 10^5$$

$$\text{Area } 1234 = p_m \times V_s$$

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$$\frac{T_2}{T_1} = r^{(\gamma-1)} = 10^{0.4} = 2.512$$

$$T_2 = 2.512 \times 300 = 753.6 \text{ K}$$

$$\frac{p_2}{p_1} = r^{\gamma} = 10^{1.4} = 25.12$$

$$p_2 = 25.12 \times 10^5 \text{ N/m}^2$$

Consider the process 3 – 4

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \frac{42}{25.12} = 1.672$$

$$T_3 = 1.672 \times 753.6 = 1260 \text{ K} = \mathbf{987° \ C} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$r_c = \frac{T_4}{T_3} = \frac{1773}{1260} = \mathbf{1.407} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Work done/kg} = \textit{Heat supplied - Heat rejected}$$

$$\textit{Heat supplied/kg} = C_v (T_3 - T_2) + C_p (T_4 - T_3)$$

$$= 0.717 \times (1260 - 753.6) + 1.004 \times (1773 - 1260)$$

$$= \mathbf{878.1 \ kJ} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

Consider the process 4 – 5

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

$$= \left(\frac{10}{1.407}\right)^{0.4} = 2.191$$

$$T_5 = \frac{T_4}{2.191} = \mathbf{809.2 \ K} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\textit{Heat rejected/kg} = C_v (T_5 - T_1)$$

$$= 0.717 \times (809.2 - 300) = 365.1 \text{ kJ}$$

$$\textit{Work output/kg} = 878.1 - 365.1 = \mathbf{513 \ kJ} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

$$\eta_{Dual} = \frac{\textit{Work output}}{\textit{Heat added}} = \frac{513}{878.1}$$

$$= 0.5842 = \mathbf{58.42\%} \quad \stackrel{\text{Ans}}{\Longleftarrow}$$

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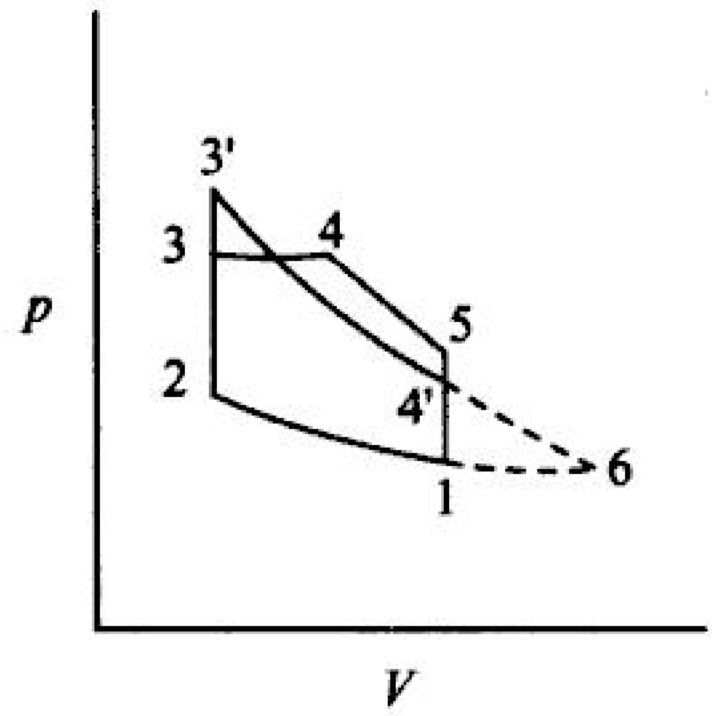
of charge would be obtained if it were possible to expand isentropically the exhaust gases to their original pressure of 1 bar. Assume that the charge has the same physical properties as that of air.

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

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = 310 \times 10^{0.4} = 778.7 \text{ K}$$

$$p_2 = p_1\left(\frac{V_1}{V_2}\right)^{\gamma} = 25.12 \text{ bar}$$

Consider the limited pressure cycle (123451)

$$T_3 = T_2\frac{p_3}{p_2} = 778.7 \times \frac{70}{25.12} = 2170 \text{ K}$$

Heat supplied at constant volume

$$= 0.717 \times (2170 - 778.7)$$

$$= 997.56 \text{ kJ/kg}$$

Heat supplied at constant pressure

$$= 2730 - 997.56 = 1732.44 \text{ kJ/kg}$$

Now,

3.17 Show that the efficiency of the Diesel cycle is lower than that of Otto cycle for the same compression ratio. Comment why the higher efficiency of the Otto cycle compared to Diesel cycle for the same compression ratio is only of a academic interest and not practical importance.

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

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:* $\eta_{th} = 65.25$ %

3.26 The pressure ratio during the compression in the cycle of an internal combustion engine with isochoric heat supply is equal to 18. Find the compression ratio, supplied and removed heat, work and efficiency, if during heat removal the temperature drops from 600 to 100 °C and the working fluid is air. Assume $\gamma = 1.4$ and $C_v = 0.717$ kJ/kg K.

Ans: (i) $r = 7.88$ (ii) $q_1 = 818.73$ kJ/kg
(iii) $q_2 = 358.5$ kJ/kg (iv) $w = 460.23$ kJ/kg
(v) $\eta_{th} = 56.21\%$

3.27 An oil engine working on the dual combustion cycle has a cylinder diameter of 20 cm and stroke of 40 cm. The compression ratio is 13.5 and the explosion ratio 1.42. Cut-off occurs at 5.1% of the stroke. Find the air-standard efficiency. Take $\gamma = 1.4$. *Ans:* 61.32 %

3.28 A compression-ignition engine working on Dual cycle takes in two-fifth of its total heat supply at constant volume and the remaining at constant pressure. Calculate :

(i) The pressure and temperature at the five cardinal points of the cycle.

(ii) The ideal thermal efficiency of the cycle.

Given : compression ratio = 13.1, Maximum pressure in the cycle = 45 bar, air intake at 1 bar and 15 °C, $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K.

Ans: (i) $p_1 = 1$ bar (ii) $p_2 = 36.3$ bar
(iii) $p_3 = 45$ bar (iv) $p_4 = 45$ bar
(v) $p_5 = 1.61$ bar (vi) $T_1 = 288$ K
(vii) $T_2 = 804.09$ K (viii) $T_3 = 996.88$ K
(ix) $T_4 = 1203.4$ K (x) $T_5 = 464.6$ K
(xi) $\eta_{th} = 63.36\%$

16. For the same peak pressure and heat input
    (a) $\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$
    (b) $\eta_{Otto} > \eta_{Diesel} > \eta_{Dual}$
    (c) $\eta_{Diesel} > \eta_{Dual} > \eta_{Otto}$
    (d) $\eta_{Diesel} > \eta_{Otto} > \eta_{Dual}$
17. For the same peak pressure and work output
    (a) $\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$
    (b) $\eta_{Otto} > \eta_{Diesel} > \eta_{Dual}$
    (c) $\eta_{Diesel} > \eta_{Otto} > \eta_{Dual}$
    (d) $\eta_{Diesel} > \eta_{Dual} > \eta_{Otto}$
18. Lenoir cycle is used in
    (a) SI engines
    (b) CI engines
    (c) pulse jet engines
    (d) gas turbines
19. A Brayton cycle consists of
    (a) two constant volume and two constant pressure processes
    (b) two constant volume and two isentropic processes
    (c) one constant pressure, one constant volume and two isentropic processes
    (d) none of the above
20. Brayton cycle is used in
    (a) Ramjet engines
    (b) gas turbines
    (c) pulse jet engines
    (d) CI engines
    (e) SI engines

Ans:	1. – (b)	2. – (a)	3. – (c)	4. – (b)	5. – (a)
	6. – (b)	7. – (b)	8. – (a)	9. – (b)	10. – (a)
	11. – (a)	12. – (c)	13. – (c)	14. – (a)	15. – (a)
	16. – (c)	17. – (d)	18. – (c)	19. – (c)	20. – (b)

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Above 1500 K the specific heat increases much more rapidly and may be expressed in the form

$$C_p = a_1 + k_1T + k_2T^2 \tag{4.3}$$

$$C_v = b_1 + k_1T + k_2T^2 \tag{4.4}$$

In Eqn.4.4 if the term T^2 is neglected it becomes same as Eqn.4.1. Many expressions are available even upto sixth order of T (i.e. T^6) for the calculation of C_p and C_v.

The physical explanation for increase in specific heat is that as the temperature is raised, larger fractions of the heat would be required to produce motion of the atoms within the molecules. Since temperature is the result of motion of the molecules, as a whole, the energy which goes into moving the atoms does not contribute to proportional temperature rise. Hence, more heat is required to raise the temperature of unit mass through one degree at higher levels. This heat by definition is the specific heat. For air, the values are

$C_p = 1.005$ kJ/kg K at 300 K $\quad C_v = 0.717$ kJ/kg K at 300 K

$C_p = 1.345$ kJ/kg K at 2000 K $\quad C_v = 1.057$ kJ/kg K at 2000 K

Since the difference between C_p and C_v is constant, the value of γ decreases with increase in temperature. Thus, if the variation of specific heats is taken into account during the compression stroke, the final temperature and pressure would be lower than if constant values of specific heat are used. This point is illustrated in Fig.4.1.

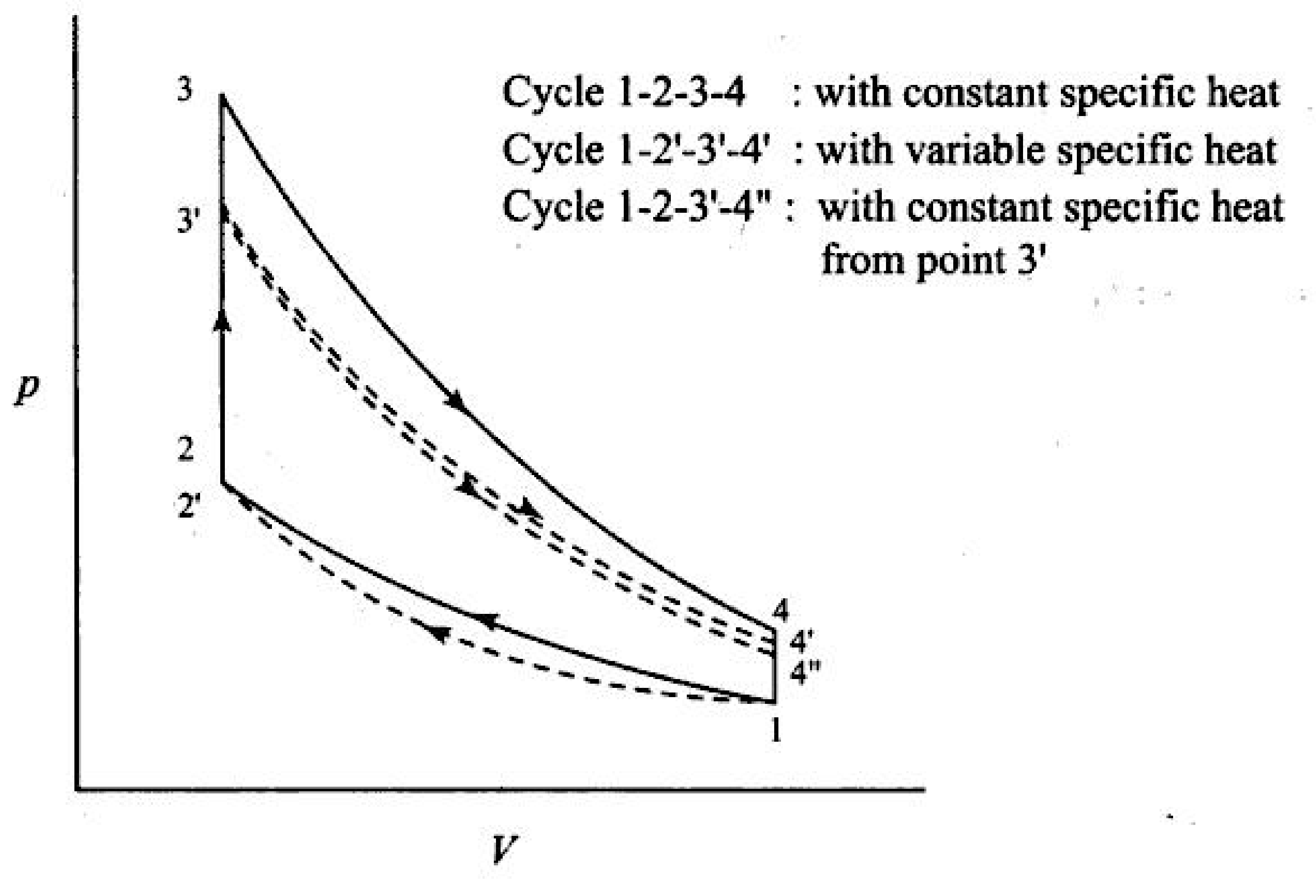

Fig. 4.1 Loss of Power due to Variation of Specific Heat

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.

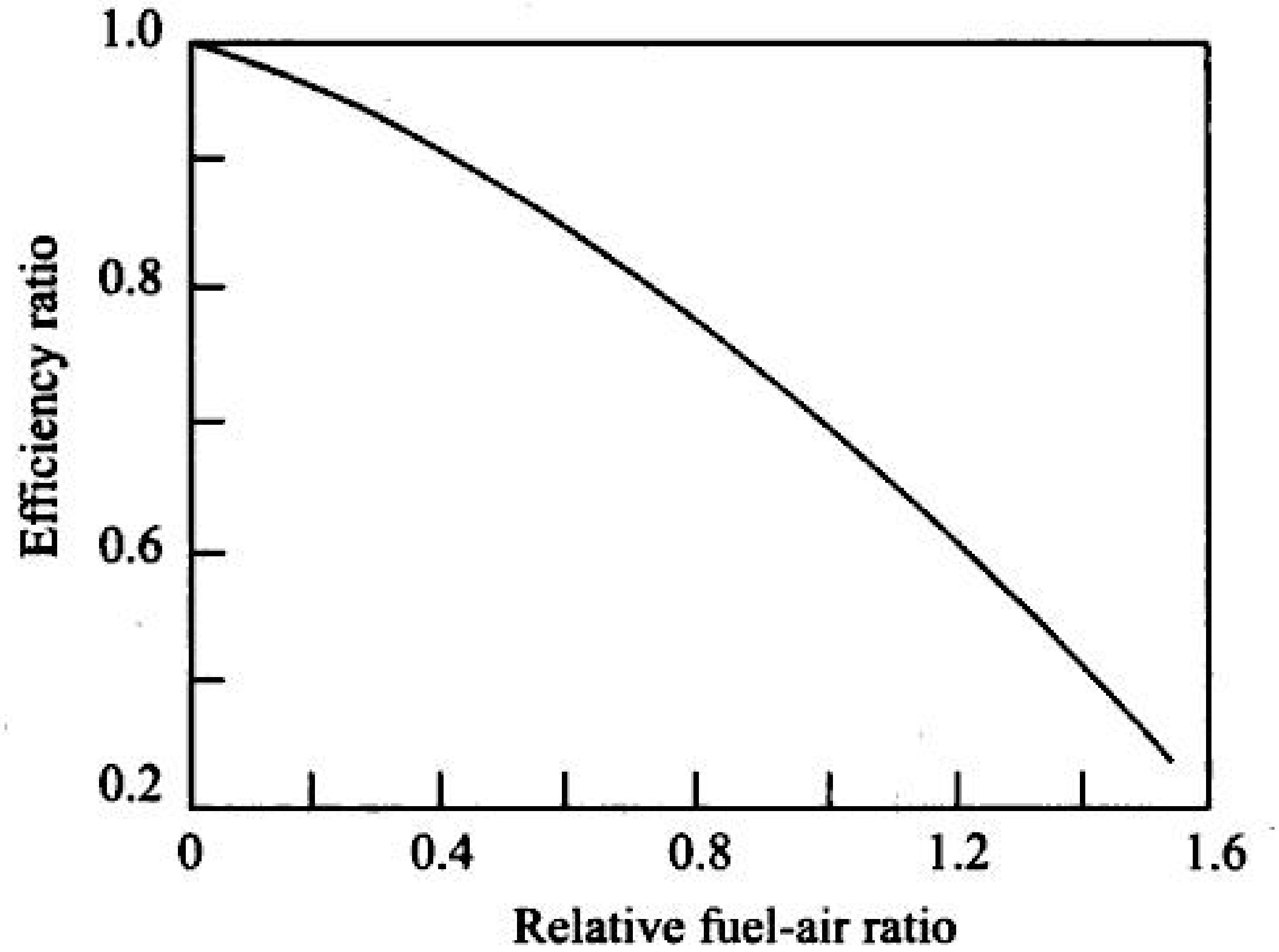

Fig. 4.5 Effect of Relative Fuel-Air ratio on Efficiency Ratio

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.

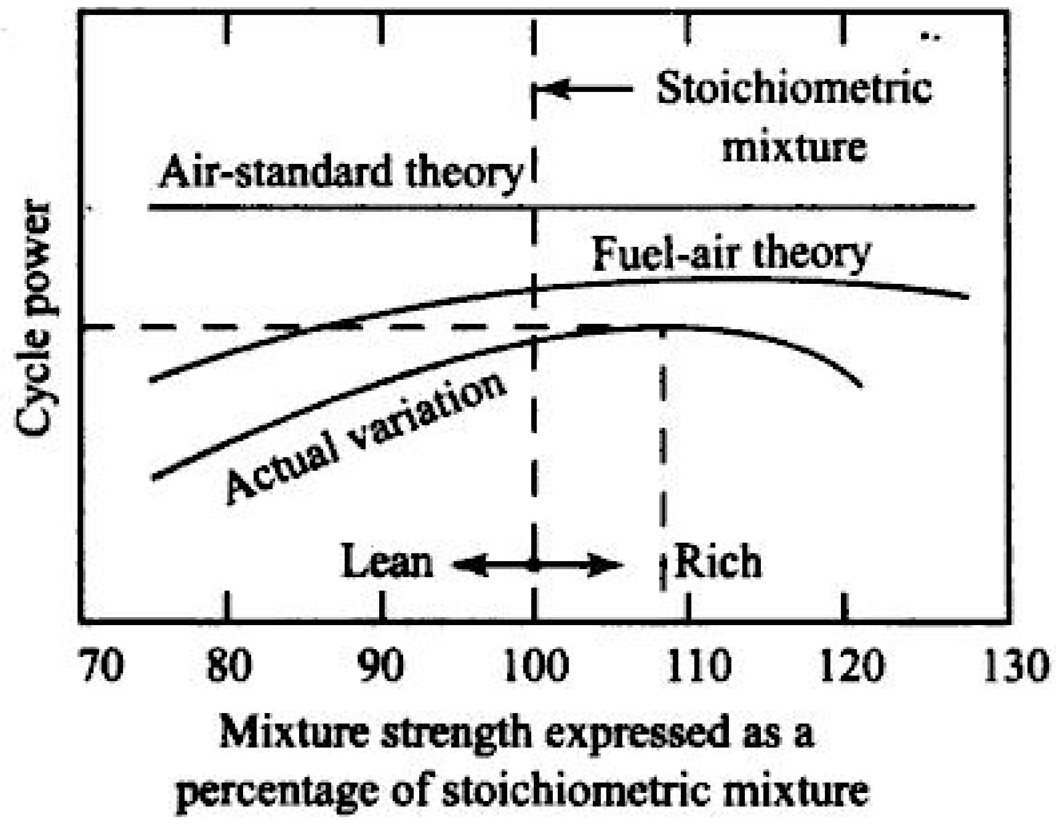

Fig. 4.11 Effect of Fuel-Air Ratio on Power

(iii) *Maximum temperature :* At a given compression ratio the temperature after combustion reaches a maximum when the mixture 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.

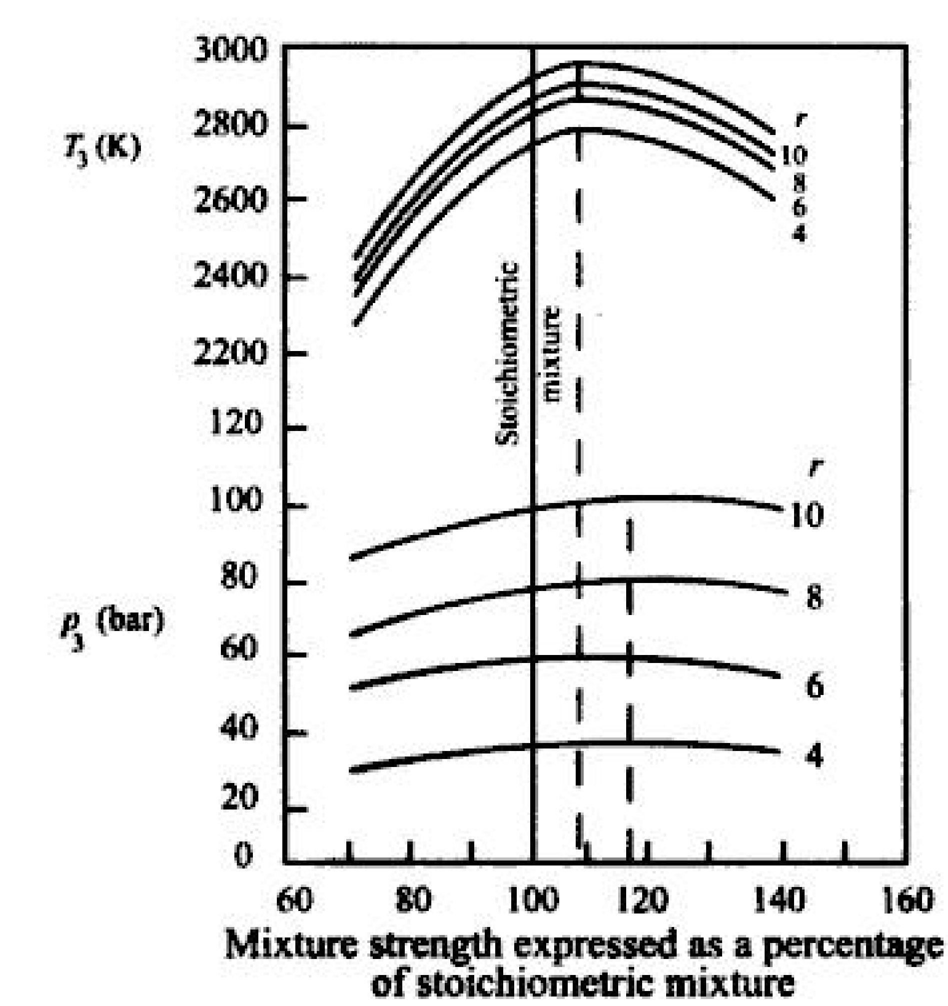

Fig. 4.12 Effect of Equivalence Ratio on T_3 and p_3

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,

$$-\frac{d\eta}{\eta} = \frac{\frac{R}{C_v^2}dC_v}{\frac{R}{C_v}+1} - \frac{\frac{R}{C_v^2}\left(r_c^{\left(\frac{R}{C_v}+1\right)}\right)\ln r_c dC_v}{r_c^{\left(\frac{R}{C_v}+1\right)}-1} + \frac{R}{C_v^2}\ln r dC_v$$

$$\frac{d\eta}{\eta} = -\frac{dC_v}{C_v}\frac{R}{C_v}\left(\frac{1-\eta}{\eta}\right) \times \left(\frac{1}{\frac{R}{C_v}+1} + \ln r - \frac{r_c^{\frac{R}{C_v}+1}\ln(r_c)}{r_c^{\frac{R}{C_v}+1}-1}\right)$$

$$\frac{d\eta}{\eta} = -\frac{dC_v}{C_v}\left(\frac{1-\eta}{\eta}\right)(\gamma-1)\left[\frac{1}{\gamma} + \ln r - \frac{r_c^{\gamma}\ln(r_c)}{r_c^{\gamma}-1}\right]$$

$$\gamma = 1.4$$

$$\frac{V_1}{V_2} = r = 20$$

$$V_1 = 20V_2$$

$$V_s = 20V_2 - V_2 = 19V_2$$

$$V_3 = 0.05V_s + V_2 = (0.05\times 19V_2) + V_2 = 1.95V_2$$

$$T_2 = T_1 r^{(\gamma-1)} = 373 \times 13^{0.4} = 1040.6 \text{ K}$$

For unit mass :
Consider the process 2-3,

$$Q_{2-3} = \frac{1}{2} \times 2000 = 1000 \text{ kJ}$$

$$Q_{2-3} = m\int_2^3 (0.709 + 0.000028T)dT$$

$$1000 = 0.709 \times (T_3 - 1040.6) + \frac{0.000028}{2} \times (T_3^2 - 1040.6^2)$$

$$T_3 = 2362.2 \text{ K}$$

$$p_3 = p_2\left(\frac{T_3}{T_2}\right) = 36.27 \times \left(\frac{2362.2}{1040.6}\right) \times 10^5$$

$$= \mathbf{82.34 \times 10^5\ N/m^2}$$ Ans

Consider the process 3-4,

$$Q_{3-4} = \frac{1}{2} \times 1000 = 500 \text{ kJ}$$

$$C_p = C_v + R = 0.996 + 0.000028\,T$$

$$Q_{3-4} = m\int_3^4 dT$$

$$500 = \int_3^4 (0.996 + 0.000028)dT$$

$$= 0.996 \times (T_4 - 2362.2) + \frac{0.000028}{2} \times (T_4^2 - 2362.2^2)$$

$$T_4 = 2830.04 \text{ K}$$

$$V_4 = V_3\left(\frac{T_4}{T_3}\right) = \frac{2830.04}{2362.2}V_3 = 1.198\,V_3$$

$$V_s = V_1 - V_3 = V_3(r-1) = 12\,V_3$$

$$\text{Cut-off \% of stroke} = \frac{V_4 - V_3}{V_s} \times 100 = \frac{V_4 - V_3}{12\,V_3} \times 100$$

$$= \frac{1.198 - 1}{12} \times 100 = \mathbf{1.65\%}$$ Ans

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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%

Multiple Choice Questions (choose the most appropriate answer)

1. The actual efficiency of a good engine is about

   (a) 100%
   (b) 85%
   (c) 50%
   (d) 25%

   of the estimated fuel-air cycle efficiency.

2. With dissociation peak temperature is obtained

   (a) at the stoichiometric air-fuel ratio
   (b) when the mixture is slightly lean
   (c) when the mixture is slightly rich
   (d) none of the above

3. With dissociation the exhaust gas temperature

   (a) decreases
   (b) increases
   (c) no effect
   (d) increases upto certain air-fuel ratio and then decreases

4. Fuel-air ratio affects maximum power output of the engine due to

   (a) higher specific heats
   (b) chemical equilibrium losses
   (c) both (a) and (b)
   (d) none of the above

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head into the water jacket or cooling fins. Some heat enters the piston head and flows through the piston rings into the cylinder wall or is carried away by the engine lubricating oil which splashes on the underside of the piston. The heat loss along with other losses is shown on the p-V diagram in Fig.5.8

Heat loss during combustion will naturally have the maximum effect on the cycle efficiency while heat loss just before the end of the expansion stroke can have very little effect because of its contribution to the useful work is very little. The heat lost during the combustion does not represent a complete loss because, even under ideal conditions assumed for air-standard cycle, only a part of this heat could be converted into work (equal to $Q \times \eta_{th}$) and the rest would be rejected during the exhaust stroke. About 15 per cent of the total heat is lost during combustion and expansion. Of this, however, much is lost so late in the cycle to have contributed to useful work. If all the heat loss is recovered only about 20% of it may appear as useful work. Figure 5.8 shows percentage of time loss, heat loss and exhaust loss in a Cooperative Fuel Research (CFR) engine. Losses are given as percentage of fuel-air cycle work. The effect of loss of heat during combustion is to reduce the maximum temperature and therefore, the specific heats are lower. It may be noted from the Fig.5.8 that of the various losses, heat loss factor contributes around 12%.

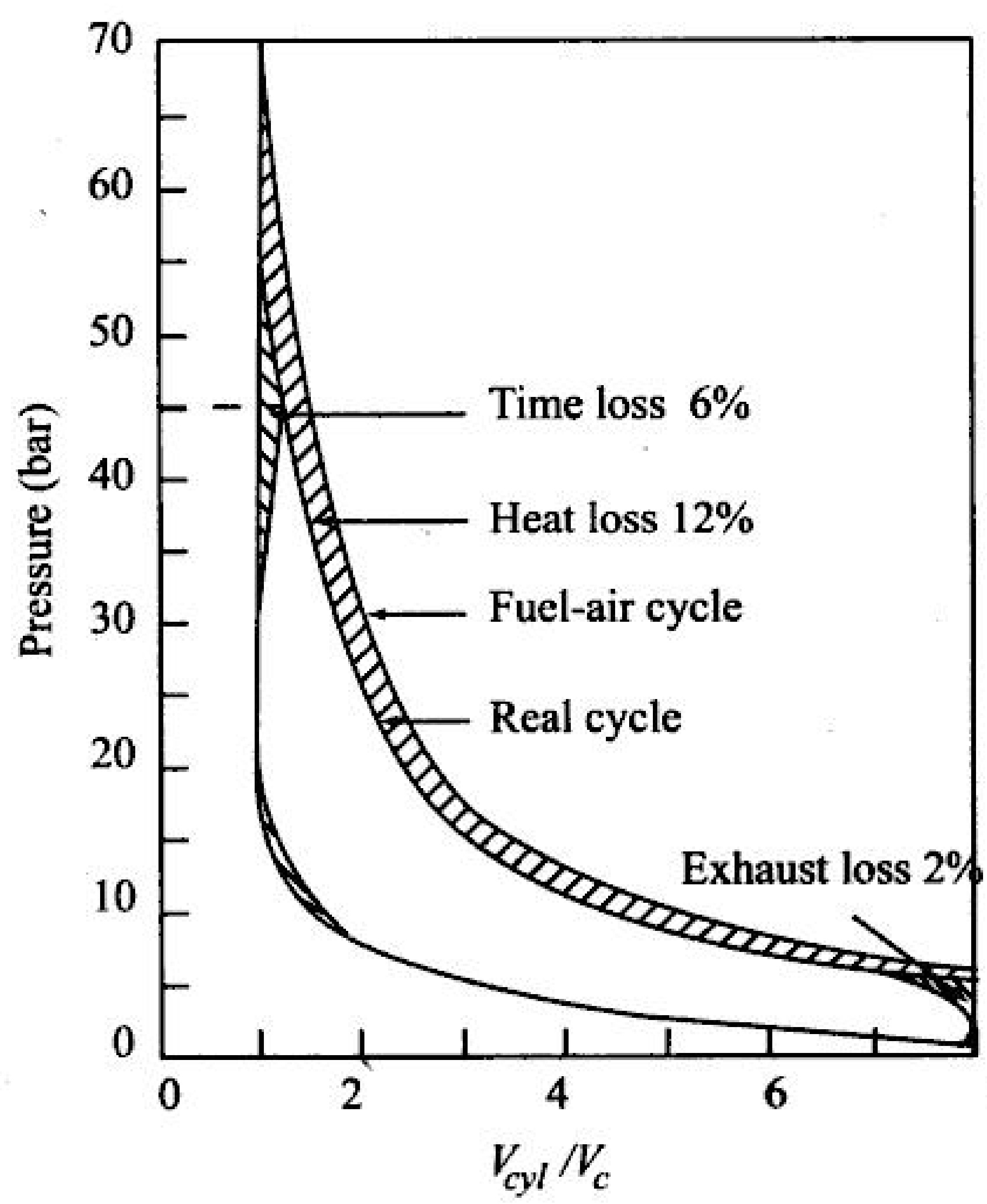

Fig. 5.8 Time Loss, Heat Loss and Exhaust Loss in Petrol Engines

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the flow may be chocked due to fluid friction. These losses can become greater than the benefit of the *ram*, and the charge per cylinder per cycle falls off.

The chosen intake valve setting for an engine operating over a range of speeds must necessarily be a compromise between the best setting for the low speed end of the range and the best setting for the high speed end.

The timing of the *exhaust valve* also affects the volumetric efficiency. The exhaust valve usually opens before the piston reaches BDC on the expansion stroke. This reduces the work done by the expanding gases during the power stroke, but decreases the work necessary to expel the burned products during the exhaust stroke, and results in an overall gain in output.

During the exhaust stroke, the piston forces the burned gases out at high velocity. If the closing of the exhaust valve is delayed beyond TDC, the inertia of the exhaust gases tends to scavenge the cylinder better by carrying out a greater mass of the gas left in the clearance volume, and results in increased volumetric efficiency. Consequently, the exhaust valve is often set to close a few degrees after TDC on the exhaust stroke, as indicated in Fig.5.10. It should be noted that it is quite possible for both the intake and exhaust valves to remain open, or partially open, at the same time. This is termed the *valve overlap*. This overlap, of course, must not be excessive enough to allow the burned gases to be sucked into the intake manifold, or the fresh charge to escape through the exhaust valve.

The reasons for the necessity of valve overlap and valve timings other than at TDC or BDC, has been explained above, taking into consideration only the dynamic effects of gas flow. One must realize, however, that the presence of a mechanical problem in actuating the valves has an influence in the timing of the valves.

The valve cannot be lifted instantaneously to a desired height, but must be opened gradually due to the problem of acceleration involved. If the sudden change in acceleration from positive to negative values are encountered in design of a cam. The cam follower may lose the contact with the cam and then be forced back to close contact by the valve spring, resulting in a blow against the cam. This type of action must be avoided and, hence, cam contours are so designed as to produce gradual and smooth changes in directional acceleration. As a result, the opening of the valve must commence ahead of the time at which it is fully opened. The same reasoning applies for the closing time. It can be seen, therefore, that the timing of valves depends on dynamic and mechanical considerations.

Both the intake and exhaust valves are usually timed to give the most satisfactory results for the average operating conditions of the particular engine, and the settings are determined on the prototype of the actual engine.

6.2.2 Gaseous Fuels

Gaseous fuels are ideal and pose very few problems in using them in internal combustion engines. Being gaseous, they mix more homogeneously with air and eliminate the distribution and starting problems that are encountered with liquid fuels. Even though the gaseous fuels are the most ideal for internal combustion engines, storage and handling problems restrict their use in automobiles. Consequently, they are commonly used for stationary power plants located near the source of availability of the fuel. Some of the gaseous fuels can be liquefied under pressure for reducing the storage volume but this arrangement is very expensive as well as risky. Because of the energy crisis in the recent years considerable research efforts are being made to improve the design and performance of gas engines which became obsolete when liquid fuels began to be used.

6.2.3 Liquid Fuels

In most of the modern internal combustion engines, liquid fuels which are the derivatives of liquid petroleum are being used. The three principal commercial types of liquid fuels are benzyl, alcohol and petroleum products. However, petroleum products form the main fuels for internal combustion engines as on today.

6.3 CHEMICAL STRUCTURE OF PETROLEUM

Petroleum as obtained from the oil wells, is predominantly a mixture of many hydrocarbons with differing molecular structure. It also contains small amounts of sulphur, oxygen, nitrogen and impurities such as water and sand. The carbon and hydrogen atoms may be linked in different ways in a hydrocarbon molecule and this linking influences the chemical and physical properties of different hydrocarbon groups. Most petroleum fuels tend to exhibit the characteristics of that type of hydrocarbon which forms a major constituent of the fuel.

The carbon and hydrogen combine in different proportions and molecular structures to form a variety of hydrocarbons. The carbon to hydrogen ratio which is one of the important parameters and their nature of bonding determine the energy characteristics of the hydrocarbon fuels. Depending upon the number of carbon and hydrogen atoms the petroleum products are classified into different groups.

The differences in physical and chemical properties between the different types of hydrocarbon depend on their chemical composition and affect mainly the combustion processes and hence, the proportion of fuel and air required in the engine. The basic families of hydrocarbons, their general formulae and their molecular arrangement are shown in Table 6.1.

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7. For SI engines fuels most preferred are

   (a) aromatics
   (b) paraffins
   (c) olefins
   (d) napthenes

8. For CI engine fuels most preferred are

   (a) napthenes
   (b) paraffins
   (c) olefins
   (d) aromatics

9. Octane number of iso-octane is

   (a) 0
   (b) 30
   (c) 60
   (d) 100

10. Ignition quality of diesel fuel is indicated by its

    (a) octane number
    (b) cetane number
    (c) flash point
    (d) fire point

Ans:					
	1. – (b)	2. – (c)	3. – (a)	4. – (a)	5. – (c)
	6. – (b)	7. – (a)	8. – (b)	9. – (d)	10. – (b)

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Table 7.2 Emission Levels and Comparison between CNG-driven Vehicles and Petrol-driven Vehicles

	Emission levels		
Pollutants	Emission norms 1977	Petrol with catalytic converter	CNG with catalytic converter
CO (g/km)	5.60	0.92	0.05
HC (g/km)	–	0.36	0.24
NO_x (g/km)	1.92	0.25	0.93

7.8.5 Liquefied Petroleum Gas (LPG)

Propane and butane are obtained from oil and gas wells. They are also the products of the petroleum refining process. For automobile engines, two types of LPG are used. One is propane and the other is butane. Sometimes, a mixture of propane and butane is used as liquid petroleum gas in automobile engines. Liquid petroleum gases serve as fuel in place of petrol. They are used widely in buses, cars and trucks. Liquid petroleum gases are compressed and cooled to form liquid. This liquid is kept in pressure tanks which are sealed. Table 7.3 gives the comparison of petrol with LPG.

7.8.6 Advantages and Disadvantages of LPG

Liquefied petroleum gas has higher potential as an alternate fuel for IC engines. The advantages and disadvantages of using LPG are

Advantages :

(i) LPG contains less carbon than petrol. LPG powered vehicle produces 50 per cent less carbon monoxide per kilometre, though only slightly less nitrogen compounds. Therefore emission is much reduced by the use of LPG.

(ii) LPG mixes with air at all temperatures.

(iii) In multi-cylinder engines a uniform mixture can be supplied to all cylinders.

(iv) Since the fuel is in the form of vapour, there is no crankcase dilution.

(v) Automobile engines can use propane, if they have high compression ratios (10:1).

(vi) LPG has high antiknock characteristics.

(vii) Its heat energy is about 80 per cent of gasoline, but its high octane value compensates the thermal efficiency of the engine.

(viii) Running on LPG translates into a cost saving of about 50%.

(ix) The engine may have a 50 per cent longer life.

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(c) equal
(d) none of the above

4. Small amount of gasoline is often added to alcohol to
   (a) reduce the emission
   (b) to increase the power output
   (c) to increase the efficiency
   (d) to improve cold weather starting

5. Methanol by itself is not a good CI engine fuel because
   (a) its octane number is high
   (b) its cetane number is low
   (c) both (a) and (b)
   (d) none of the above

6. Anti-knock characteristics of alcohol when compared to gasoline is
   (a) higher
   (b) lower
   (c) equal
   (d) none of the above

7. Alcohols alone cannot be used in CI engines as
   (a) their self ignition temperature is high
   (b) latent heat of vaporization is low
   (c) both (a) and (b)
   (d) none of the above

8. Advantage of hydrogen as an IC engine fuel
   (a) high volumetric efficiency
   (b) low fuel cost
   (c) No HC and CO emissions
   (d) relatively safe

9. Disadvantage of hydrogen as a fuel in IC engine
   (a) storage is easy
   (b) low NO_x emissions
   (c) detonating tendency
   (d) easy handling

to provide for the various special requirements like starting, idling, variable load and speed operation and acceleration will be included. Figure 8.7 shows the details of a simple carburetor.

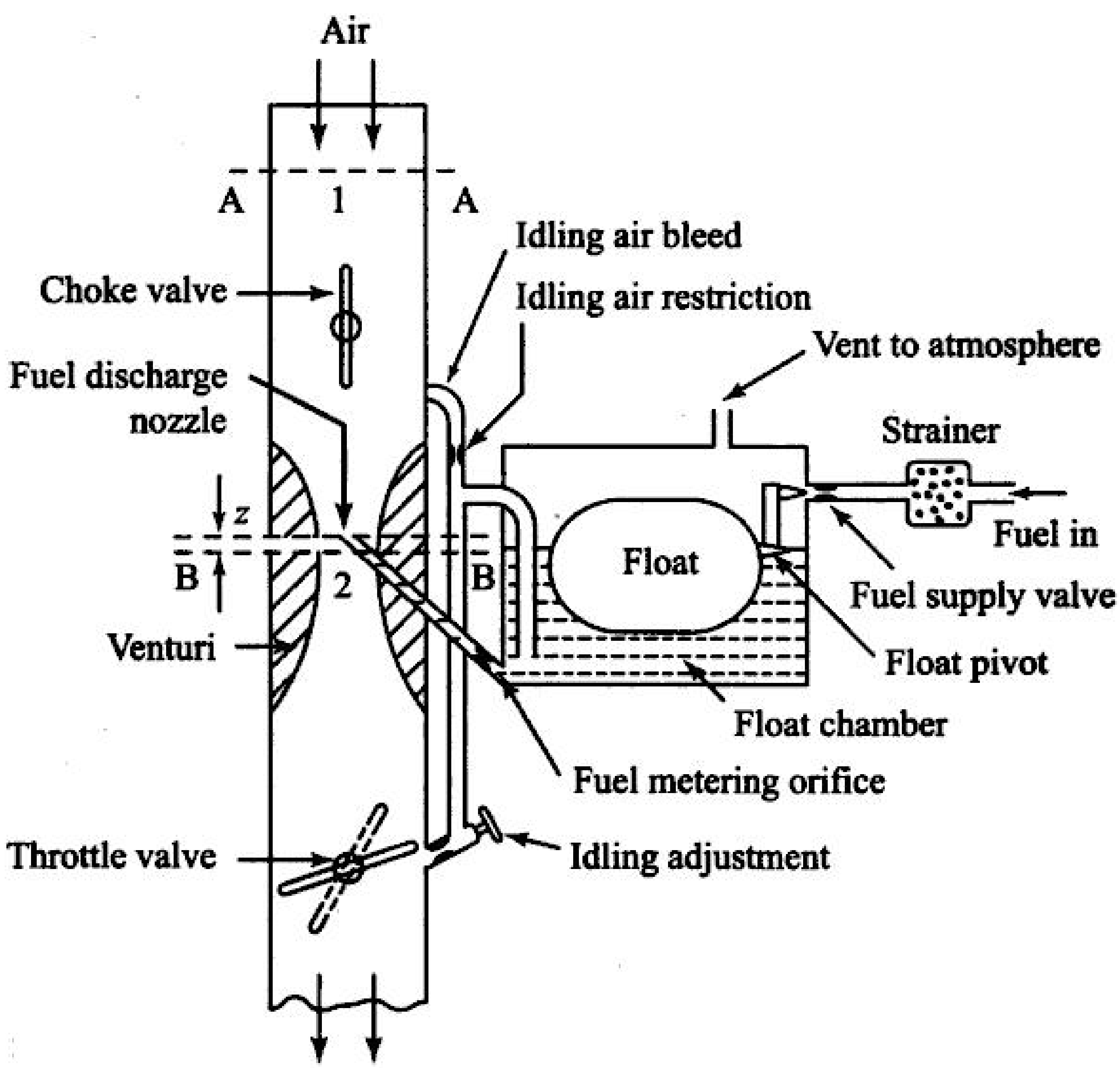

Fig. 8.7 The Simple Carburetor

The simple carburetor mainly consists of a float chamber, fuel discharge nozzle and a metering orifice, a venturi, a throttle valve and a choke. The float and a needle valve system maintains a constant level of gasoline in the float chamber. If the amount of fuel in the float chamber falls below the designed level, the float goes down, thereby opening the fuel supply valve and admitting fuel. When the designed level has been reached, the float closes the fuel supply valve thus stopping additional fuel flow from the supply system. Float chamber is vented either to the atmosphere or to the upstream side of the venturi.

During suction stroke air is drawn through the venturi. As already described, venturi is a tube of decreasing cross-section with a minimum area at the throat. Venturi tube is also known as the choke tube and is so shaped that it offers minimum resistance to the air flow. As the air passes through the venturi the velocity increases reaching a maximum at the venturi throat. Correspondingly, the pressure decreases reaching a

$$A/F \text{ ratio} = \frac{\dot{m}_{a_{actual}}}{\dot{m}_{f_{actual}}}$$

$$\frac{A}{F} = 0.1562\frac{C_{da}}{C_{df}}\frac{A_2}{A_f}\frac{p_1\phi}{\sqrt{2T_1\rho_f(p_1-p_2-gz\rho_f)}} \tag{8.19}$$

8.9.1 Air–Fuel Ratio Neglecting Compressibility of Air

When air is considered as incompressible, Bernoulli's theorem is applicable to air flow also. Hence, assuming $U_1 \approx 0$

$$\frac{p_1}{\rho_a} - \frac{p_2}{\rho_a} = \frac{C_2^2}{2} \tag{8.20}$$

$$C_2 = \sqrt{2\left[\frac{p_1-p_2}{\rho_a}\right]} \tag{8.21}$$

$$\dot{m}_a = A_2C_2\rho_a = A_2\sqrt{2\rho_a(p_1-p_2)} \tag{8.22}$$

$$\dot{m}_{a_{actual}} = C_{da}A_2\sqrt{2\rho_a(p_1-p_2)} \tag{8.23}$$

$$A/F \text{ ratio} = \frac{\dot{m}_a}{\dot{m}_f}$$

$$= \frac{C_{da}}{C_{df}}\frac{A_2}{A_f}\sqrt{\frac{\rho_a(p_1-p_2)}{\rho_f(p_1-p_2-gz\rho_f)}} \tag{8.24}$$

If $z = 0$

$$\frac{\dot{m}_a}{\dot{m}_f} = \frac{C_{da}}{C_{df}}\frac{A_2}{A_f}\sqrt{\frac{\rho_a}{\rho_f}} \tag{8.25}$$

8.9.2 Air–Fuel Ratio Provided by a Simple Carburetor

(i) It is clear from expression for $\dot{m}_f$ (Eq.8.17) that if (p_1-p_2) is less than $gz\rho_f$ there is no fuel flow and this can happen at very low air flow. As the air flow increases, (p_1-p_2) increases and when $(p_1-p_2) > gz\rho_f$ the fuel flow begins and increases with increase in the differential pressure.

(ii) At high air flows where (p_1-p_2) is large compared to $gz\rho_f$ the fraction $gz\rho_f/(p_1-p_2)$ becomes negligible and the air-fuel ratio approaches

$$\frac{C_{d_a}}{C_{d_f}}\frac{A_2}{A_f}\sqrt{\frac{\rho_a}{\rho_f}}$$

(iii) A decrease in the density of air reduces the value of air-fuel ratio (i.e., mixture becomes richer). It happens at

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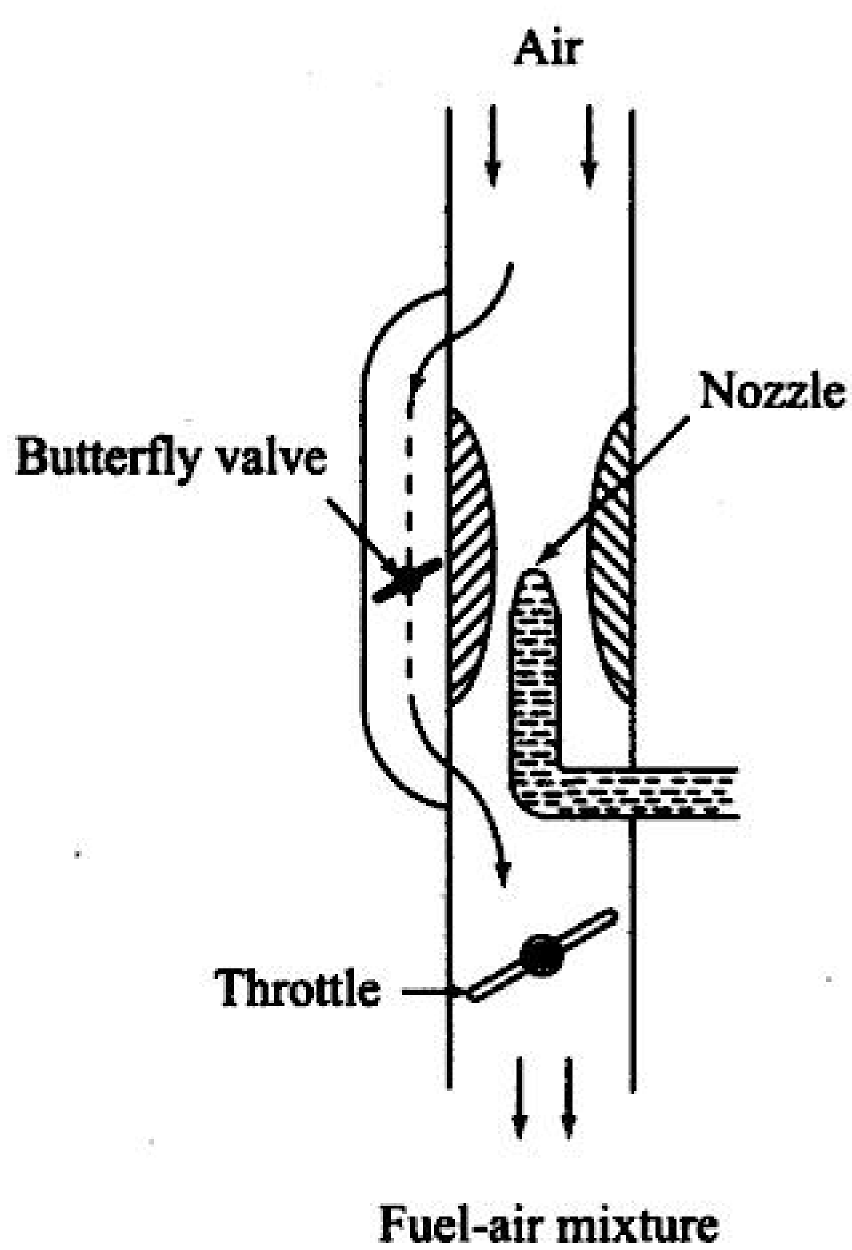

Fig. 8.18 Auxiliary Port

8.12.1 Anti-dieseling System

An SI engine sometimes continuous to run for a very small period even after the ignition is switched off. This phenomenon is called dieseling (after running or run-on). Dieseling may take place due to one or more of the following:

(i) Engine idling speed set to high.

(ii) Increase in compression ratio due to carbon deposits.

(iii) Inadequate or low octane rating.

(iv) Engine overheating.

(v) Too high spark plug heat range.

(vi) Incorrect adjustment of idle fuel-air mixture (usually toulene).

(vii) Sticking of throttle.

(viii) Requirement of tune up of engine.

(ix) Oil entry into the cylinder.

Some modern automobiles use antidieseling system to prevent dieseling system. This system has a solenoid valve operated idling circuit. With

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7. Injection system in which the pump and the injector nozzle is combined in one housing is known as

   (a) common rail system
   (b) distributor system
   (c) unit injector system
   (d) individual pump and nozzle system

8. Main advantage of pintaux nozzle is

   (a) better cold starting performance
   (b) ability to distribute the fuel
   (c) good penetration
   (d) good atomization

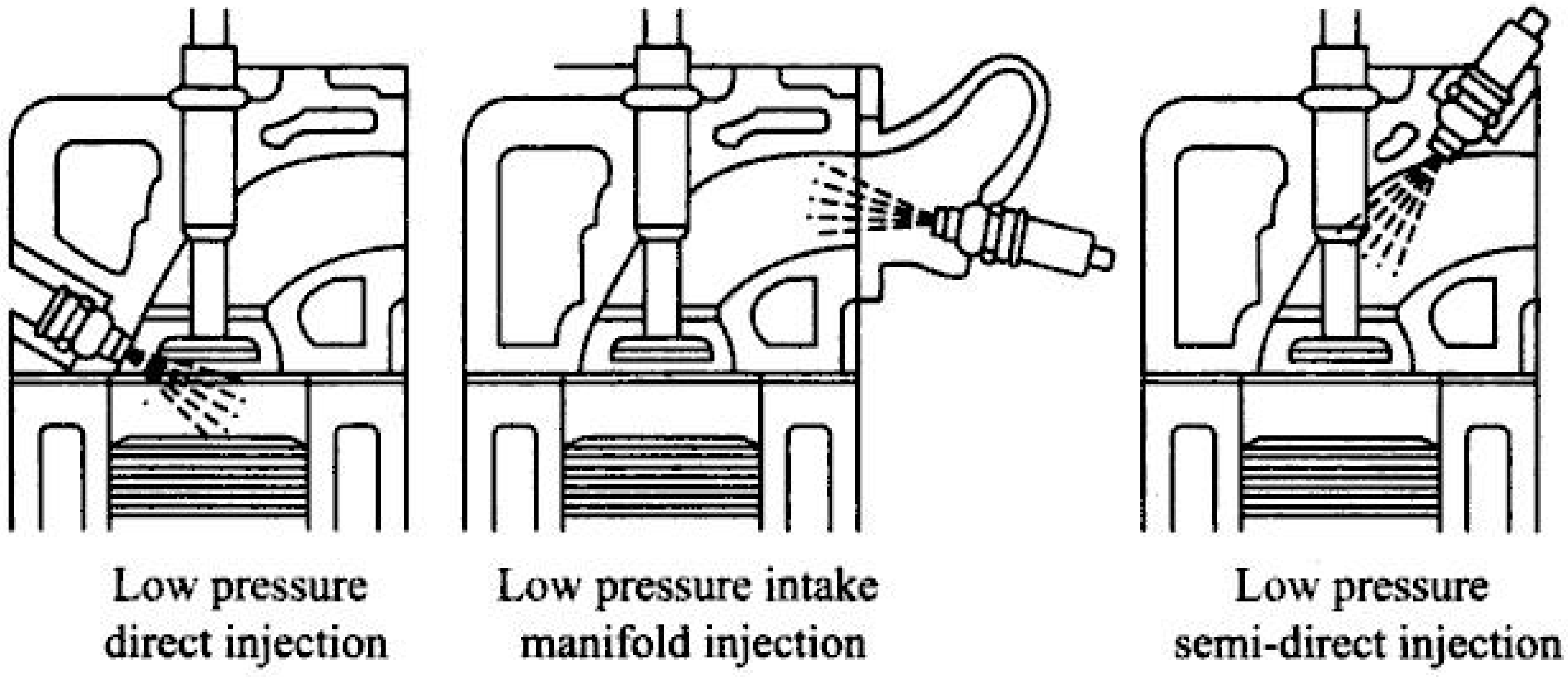
9. The most accurate gasoline injection system is

   (a) direct injection
   (b) port injection
   (c) throttle body injection
   (d) manifold injection

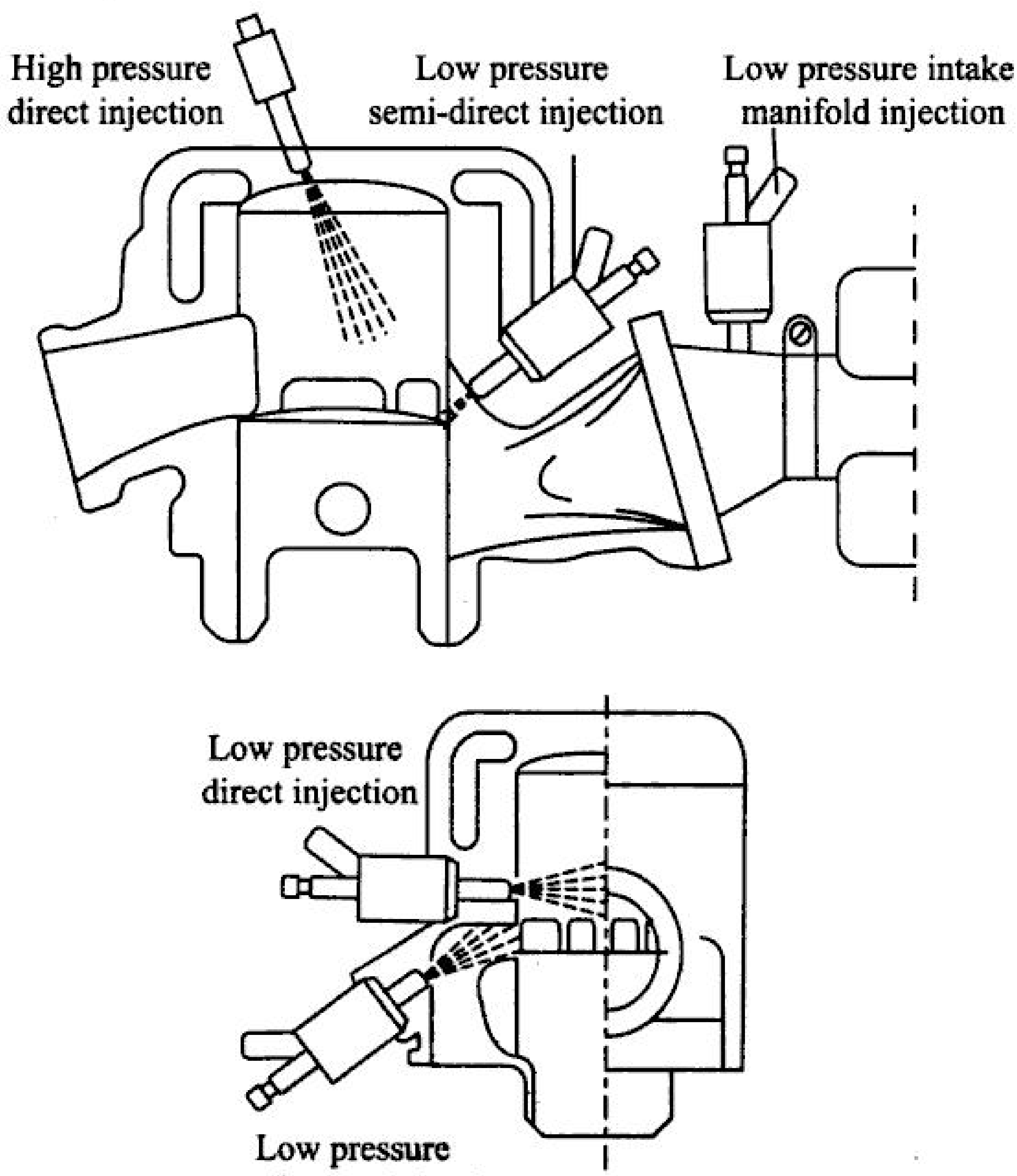
10. Advantage of fuel injection in SI engine is

   (a) low initial cost
   (b) low maintenance requirements
   (c) increased volumetric efficiency
   (d) none of the above

Ans: 1. – (b) 2. – (d) 3. – (b) 4. – (d) 5. – (b)
6. – (c) 7. – (c) 8. – (a) 9. – (b) 10. – (c)



(a) Gasoline injection in four-stroke engines



(b) Fuel injection in two-stroke engines

Fig. 10.2 Different Methods of Fuel Injection

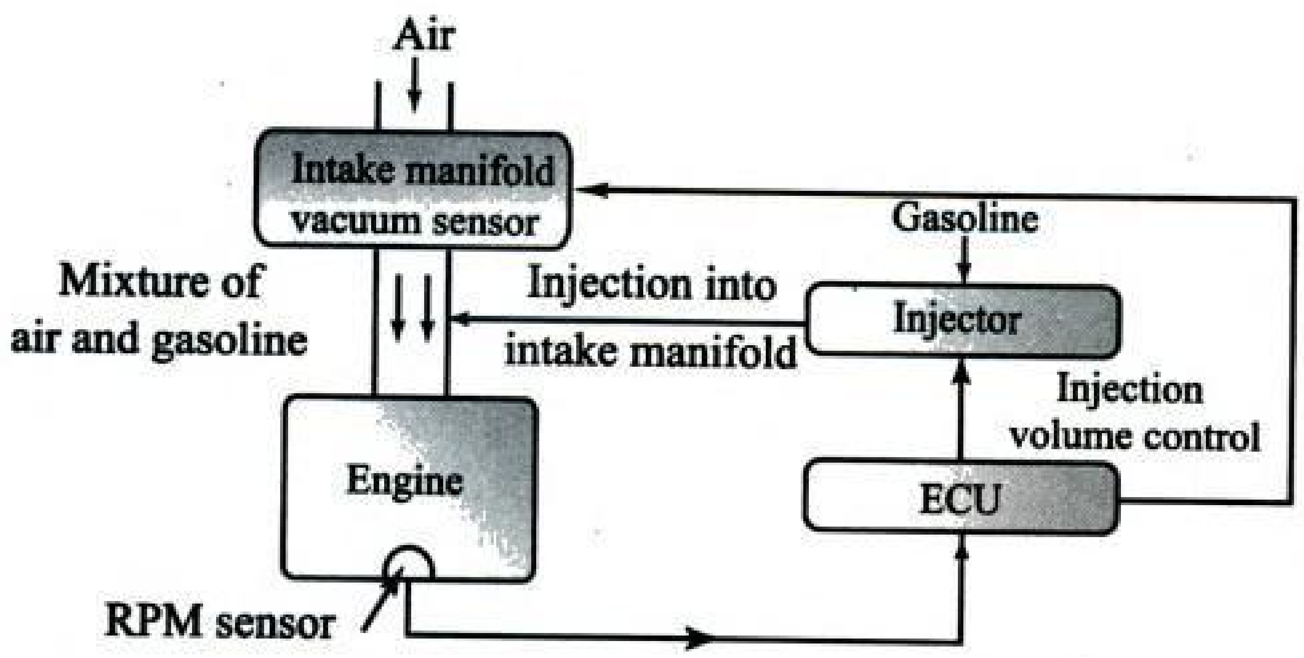


Fig. 10.6 D-MPFI Gasoline Injection System

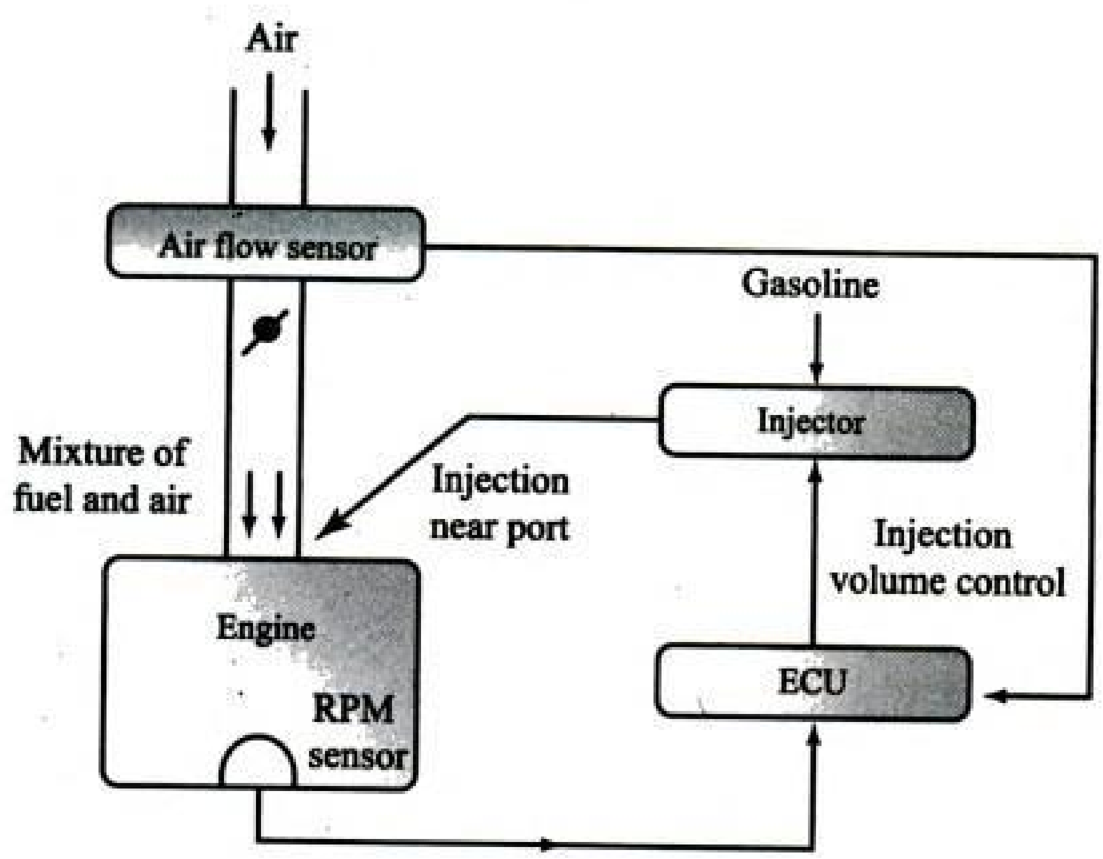


Fig. 10.7 L-MPFI Gasoline Injection System

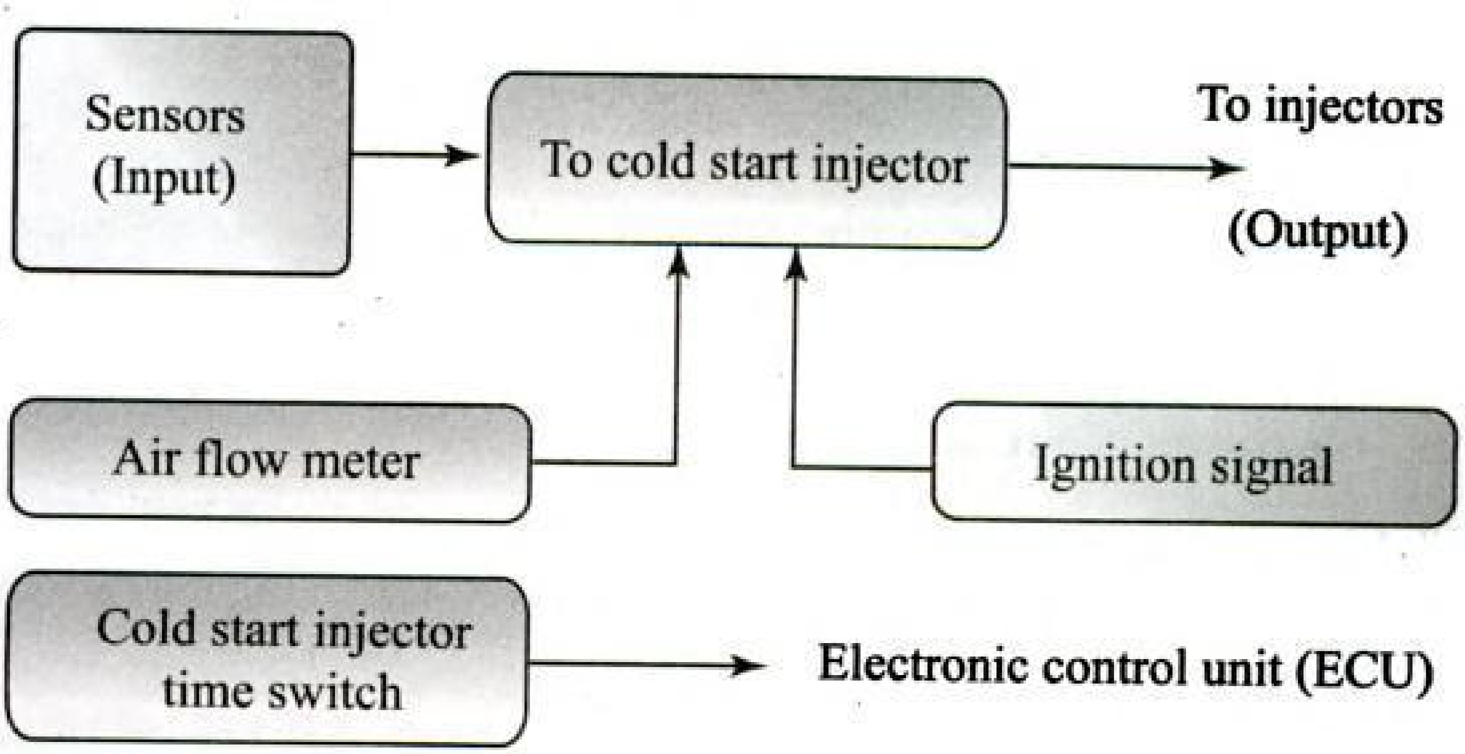


Fig. 10.8 MPFI-Electronic Control System

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The injection timing for one cylinder of this four cylinder engine is described below.

In one cylinder, the piston moves up from BDC (Bottom Dead Centre) to TDC (Top Dead Centre) during the exhaust stroke. Just before the piston reaches TDC during this exhaust stroke, injection of the fuel takes place into the inlet manifold of this cylinder at about 60° crankangle before TDC. This injected fuel mixes with the air in the air intake chamber. Thus the air-fuel mixture is obtained.

At the beginning of the suction stroke, intake valve opens and the air-fuel mixture is sucked into the cylinder during the suction stroke.

According to the firing order, the injection of the fuel takes place inside the inlet manifolds of the other three cylinders at various timings.

In this four cylinder engine, the ECU calculates the appropriate injection timing for each cylinder and the air fuel-mixture is made available at each suction stroke.

In order to meet the operating conditions, the injection valve is kept open for a longer time by ECU. For example, if the vehicle is accelerating, the injection valve will be opened for longer time, in order to supply additional fuel to the engine.

10.8 GROUP GASOLINE INJECTION SYSTEM

In an engine having group gasoline injection system, the injectors are not activated individually, but are activated in groups. In a four-cylinder engine also there are two groups, each group having two injectors. In a six-cylinder engine, there are two groups, each group having 3 injectors.

Figure 10.12 shows a block diagram with sensors and the Electronic Control Unit (ECU), for a group injection system. Sensors for detecting pressure in the manifold, engine speed in rpm, throttle position, intake manifold air temperature and the coolant temperature send information to the ECU. With this information, the ECU computes the amount of gasoline that the engine needs. The ECU then sends signals to the injectors and other parts of the system. The timing of the injectors is decided by the engine-speed sensor.

The injectors are divided into two groups. Based on the signals from the speed sensor, the ECU activates one group of injectors. Subsequently, the ECU activates the other group of injectors. For example, the injector grouping for a six-cylinder engine is shown in Fig.10.13. Injectors for cylinders 1, 3 and 5 open at the same time and inject gasoline into the intake manifold. After these injectors close, the injectors for the cylinders 2, 4 and 6 open and inject gasoline.

Figure 10.14 shows a port injection using the electronic group fuel-injection system for an eight-cylinder engine. Eight injectors are connected to a fuel system and are divided into two groups, each group having four injectors. Each group of injectors is alternately turned on by the ECU. When the crankshaft makes two revolutions, the injectors are turned on once. Thus it is seen that the modern engines are controlled more and

quantity to be momentarily increased. The turbocharger boost pressure can be used to detect the mass flow rate of air, which can be used to decide the fuel injection quantity. Alternatively a hot wire sensor can measure the mass flow rate of air. Various electronically controlled injection systems are discussed below in some detail.

10.10.1 Electronically Controlled Unit Injectors

The schematic layout of the entire system is indicated in Fig.10.15. Unit injectors can be combination of high-pressure pumps and injectors in one unit. They do not have high-pressure lines and hence the injection lag is low. The main high-pressure pump is situated above the injector. Fuel is fed into the high-pressure pump by a supply gear pump at low pressure. The plunger of the high-pressure pump is pushed down at the appropriate time by a cam, and rocker mechanism. A simplified cross section of the unit injector and phases of injection are shown in Fig.10.16. The fuel pushed down by the injector just bypasses the injection nozzle till the solenoid controlled spill valve closes the spill port. The closure of the spill port initiates the injection process. The injection stops when the solenoid valve opens the spill port. The timing and duration of the square pulse given to the solenoid can thus control the fuel timing and injection quantity. The solenoid can also be opened and closed more than once to have a pilot injection spray followed by the main spray. The pressure of injection is however controlled by the rate of displacement of the fuel and the size of the hole in the nozzle. The ECU generates the pulses to operate the solenoid controlled spill valve.

10.10.2 Electronically Controlled Injection Pumps (Inline and Distributor Type)

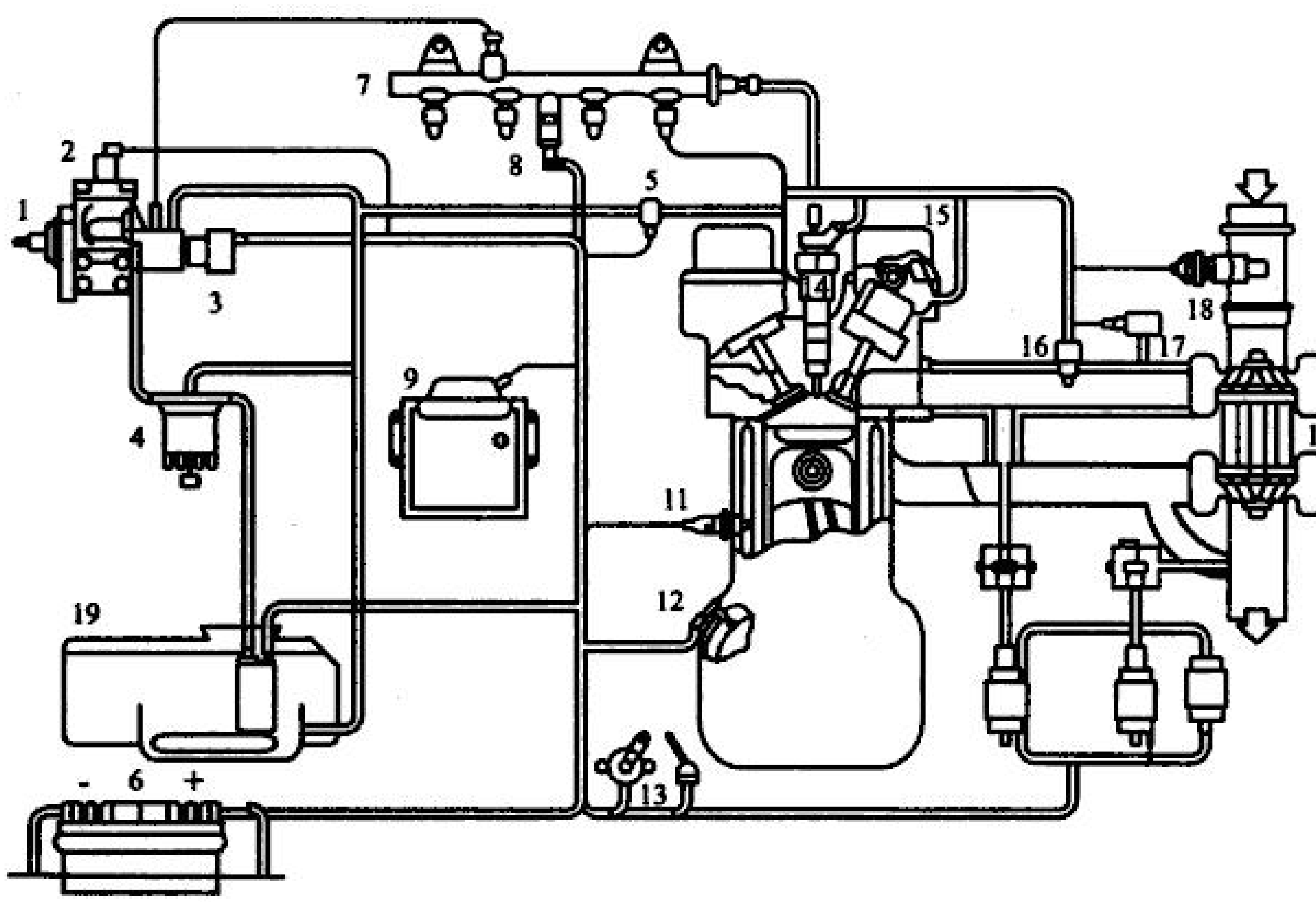
Diesel engines use inline and distributor pumps. The start of injection in the conventional inline element is determined by the instant when the top of the plunger covers the bypass or the spill ports. The end of delivery occurs when the helical slot or groove on the plunger uncovers these ports. The start of delivery is fixed but the end of delivery depends on the amount of fuel to be delivered.

In the case of the electronically controlled system there will be a control sleeve which can be moved up and down by an actuator which is controlled by the ECU (Electronic Control Unit). The ECU determines the amount of fuel to be injected depending on the throttle position, engine speed, and other parameters. Once this is obtained the control sleeve is positioned so that the required quantity of fuel can be injected. The timing of injection is still done mechanically.

Distributor pumps use control sleeves for metering the injected quantity. Thus they can be easily be made to work with an electronically controlled solenoid actuator. The principle of operation is similar to the one explained above.

head through lines. The injector is solenoid operated. It received pulses from the ECU to open the same.

The engine directly drives the pump of the common rail system. It is generally of the multi-cylinder radial piston type. The generated pressure is independent of the injection process unlike conventional injection systems. The rail pressure pump is generally much smaller that conventional pumps and also is subjected to lesser pressure pulsations. The injection occurs when the solenoid is energized. The quantity of fuel injected is directly dependent on the duration of the pulse when the injection pressure is constant. Sensors on the crankshaft indicate its position and speed and so the timing of injection and its frequency can be controlled. A typical layout of the common rail fuel injection system is indicated in Fig.10.18. Fuel



1	High pressure pump	7	High pressure accumulator	13	Accelerator pedal sensor
2	Element shutoff valve	8	Rail pressure sensor	14	Injector
3	Pressure control valve	9	ECU	15	Camshaft speed sensor
4	Fuel filter	10	Turbocharger	16	Intake air temperature sensor
5	Fuel temperature sensor	11	Coolant temperature sensor	17	Boost pressure sensor
6	Battery	12	Crankshaft speed	18	Air mass meter

19 Fuel tank with prefilter and presupply pump

Fig. 10.18 Sensors of a Common Rail Injection System, together with Various System Components

from the tank is lifted by a low pressure pump and passed through a filter. The pump is generally run by an electric motor independent of the engine speed. The main pumping element can be a conventional gear pump or of the roller cell type. The roller cell pump has a rotor with radial slots. These

11
IGNITION

11.1 INTRODUCTION

In spark-ignition engines, as compression ratio is lower, and the self-ignition temperature of gasoline is higher, for igniting the mixture for the initiation of combustion an ignition system is a must.

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

By implication, ignition is merely a prerequisite for combustion. Therefore, the study of ignition is a must to understand the phenomenon of combustion so that a criterion may be established to decide whether ignition has occurred. Although the ignition process is intimately connected with the initiation of combustion, it is not associated with the gross behaviour of combustion. Instead, it is a local small-scale phenomenon that takes place within a small zone in the combustion chamber.

In terms of its simplest definition, ignition has no degree, intensively or extensively. Either the combustion of the medium is initiated or it is not. Therefore, it is reasonable to consider ignition from the standpoint of the beginning of the combustion process that it initiates.

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

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11.11 MAGNETO IGNITION SYSTEM

Magneto is a special type of ignition system with its own electric generator to provide the necessary energy for the system. It is mounted on the engine and replaces all the components of the coil ignition system except the spark plug. A magneto when rotated by the engine is capable of producing a very high voltage and does not need a battery as a source of external energy.

A schematic diagram of a high tension magneto ignition system is shown in Fig.11.9. The high tension magneto incorporates the windings to generate the primary voltage as well as to step up the voltage and thus does not require a separate coil to boost up the voltage required to operate the spark plug.

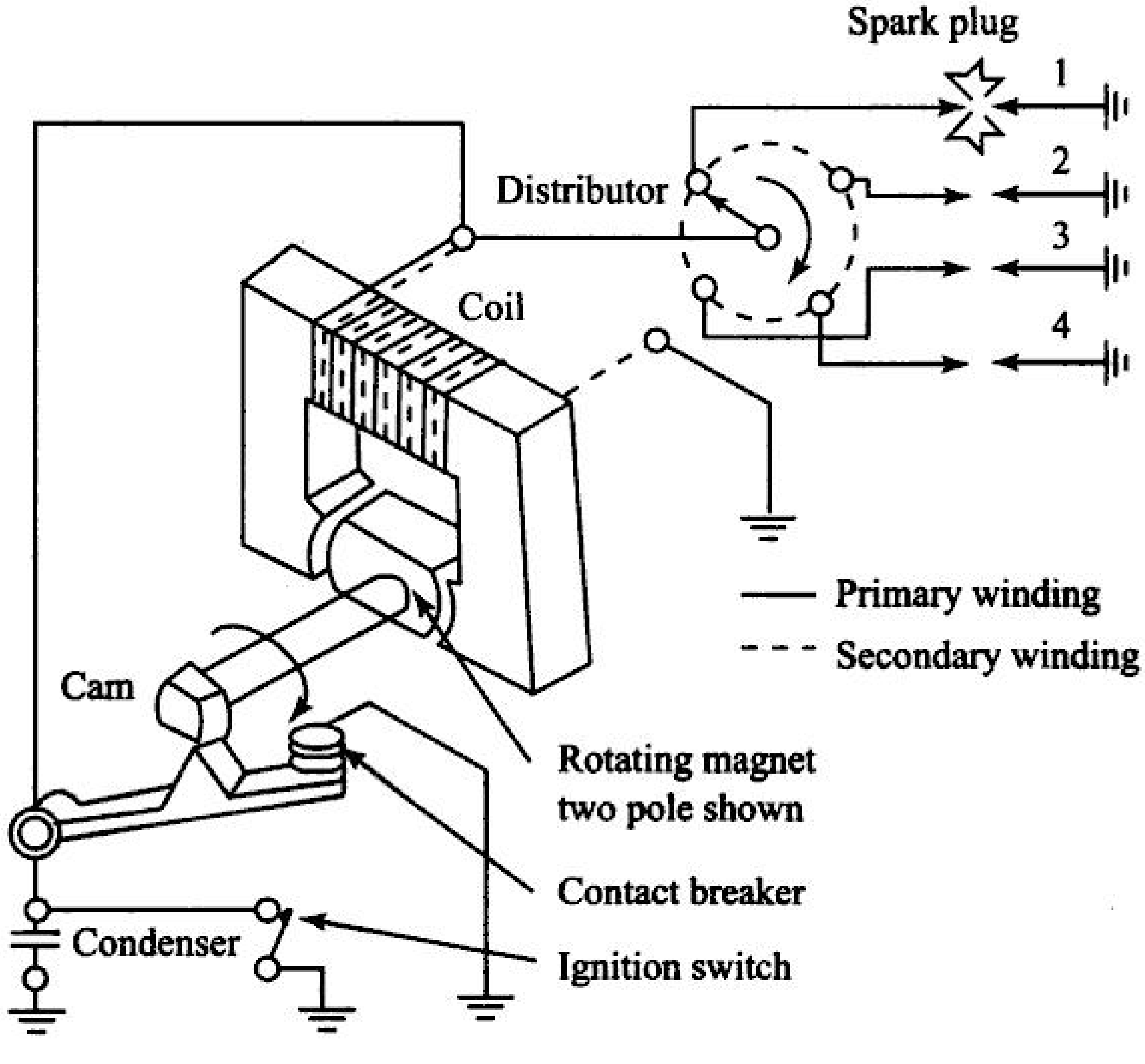

Fig. 11.9 High Tension Magneto Ignition System

Magneto can be either *rotating armature type* or *rotating magnet type*. In the first type, the armature consisting of the primary and 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.

plugs as in the conventional breaker system. The control module contains timing circuit which later closes the primary circuit so that the build up of the primary circuit current can occur for the next cycle. There are many types of pulse generators that could trigger the electronic circuit of the ignition system.

A magnetic pulse generator where a gear shaped iron rotor driven by the distributor shaft rotates past the pole of a stationary magnetic pickup, is generally used. The number of teeth on the rotor is equal to the number of cylinders. The magnetic field is provided by a permanent magnet. As each rotor tooth passes the magnet pole it first increases and then decreases the magnetic field strength ψ linked with the pickup coil wound on the magnet, producing a voltage signal proportional to $\frac{d\psi}{dt}$. In response to this the electronic module switches off the primary circuit coil current to produce the spark as the rotor tooth passes through alignment and the pick up coil voltage abruptly reverses and passes through zero. The increasing portion of the voltage waveform, after this voltage reversal, is used by the electronic module to establish the point at which the primary coil current is switched on for the next ignition pulse.

11.12.2 Capacitive Discharge Ignition (CDI) System

The details of capacitive discharge ignition system are shown in Fig.11.12. In this system, a capacitor rather than an induction coil is used to store the ignition energy. The capacitance and charging voltage of the capacitor determine the amount of stored energy. Ignition transformer steps up the primary voltage generated at the time of spark by the discharge of the capacitor through the thyristor to the high voltage required at the spark plug.

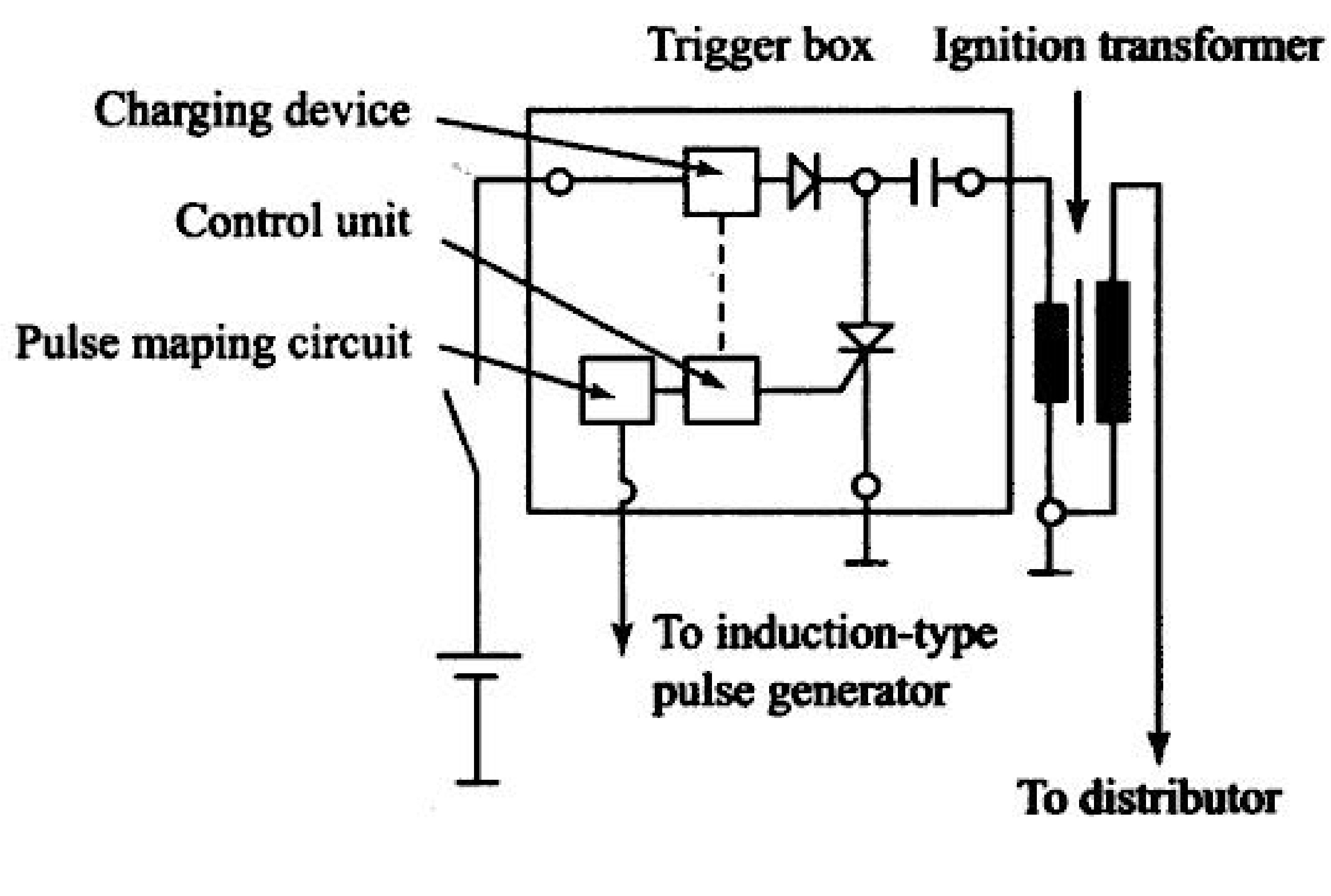

Fig. 11.12 Schematic of Capacitive Discharge Ignition (CDI) System

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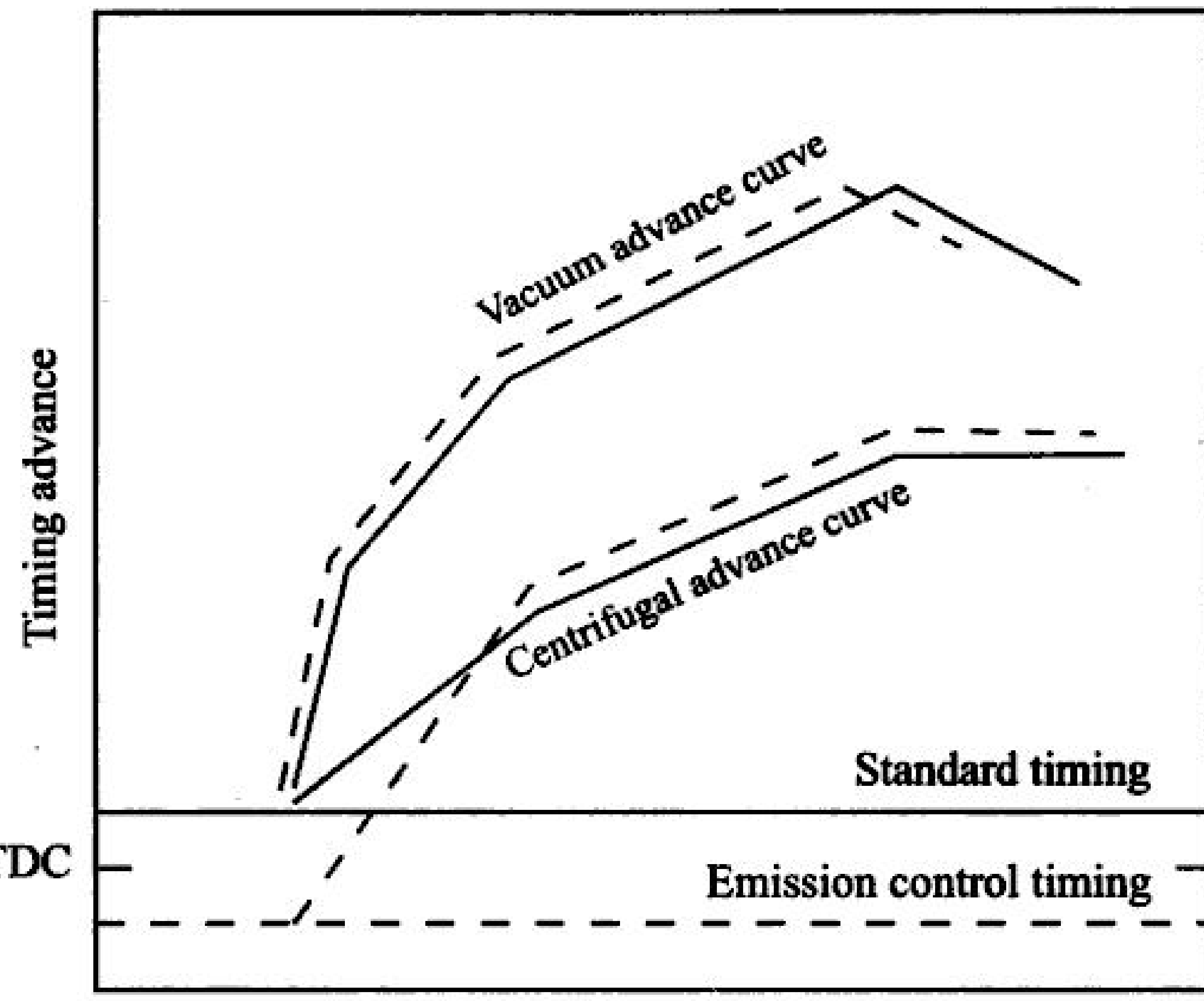

Fig. 11.18 Typical Distributor Advance Curve for Lower HC and CO Exhaust Emission

Review Questions

11.1 What is meant by ignition? What is the interrelation between ignition and combustion?

11.2 What are various types of ignition system that are commonly used?

11.3 Explain the basic energy requirements for spark ignition.

11.4 What is capacitance spark and how is it produced?

11.5 With a neat sketch explain an induction coil.

11.6 How does gas movement affect spark-ignition?

11.7 Comment on the spark energy and its duration in the initiation of combustion.

11.8 What are the important requirements of the high voltage ignition source for the spark-ignition process?

11.9 What are the two conventional types of ignition systems that are normally used in automobiles?

11.10 Mention the various important qualities of a good ignition system.

11.11 With a neat sketch explain the battery ignition system.

11.12 Explain TCI ignition system with a sketch.

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 is caused by heat transfer and diffusion of burning fuel molecules from the combustion zone to the adjacent layers of unburnt mixture. The flame front is a narrow zone separating the fresh mixture from the combustion products. The velocity with which the flame front moves, with respect to the unburned

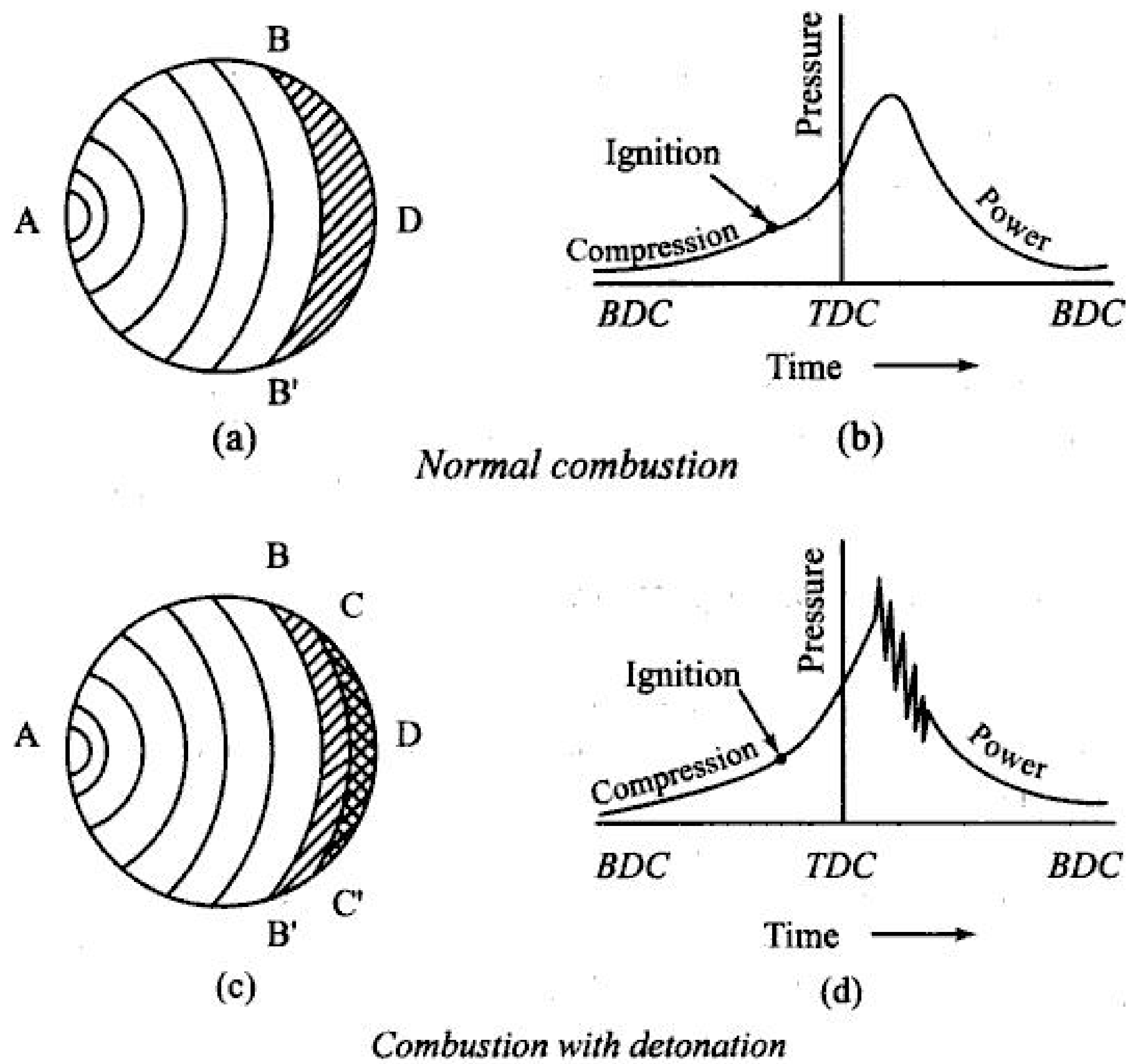

Fig. 12.6 Normal and Abnormal Combustion

oxidation may take place in the end charge leading to further increase in temperature. In spite of these factors if the temperature of the end charge had not reached its self-ignition temperature, the charge would not autoignite and the flame will advance further and consume the charge BB′D. This is the normal combustion process which is illustrated by means of the pressure-time diagram, Fig.12.6(b).

However, if the end charge BB′D reaches its autoignition temperature and remains for some length of time equal to the time of preflame reactions the charge will autoignite, leading to knocking combustion. In Fig.12.6(c), it is assumed that when flame has reached the position BB′, the charge ahead of it has reached critical autoignition temperature. During the preflame reaction period if the flame front could move from BB′ to only CC′ then the charge ahead of CC′ would autoignite.

Because of the autoignition, another flame front starts traveling in the opposite direction to the main flame front. When the two flame fronts collide, a severe pressure pulse is generated. The gas in the chamber is subjected to compression and rarefaction along the pressure pulse until pressure equilibrium is restored. This disturbance can force the walls of the combustion chambers to vibrate at the same frequency as the gas. Gas vibration frequency in automobile engines is of the order of 5000 cps. The pressure-time trace of such a situation is shown in Fig.12.6(d).

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Table 12.3 gives a comparative statement of various characteristics that reduce knocking in spark-ignition engines and compression-ignition engines.

Table 12.3 Characteristics Tending to Reduce Detonation or Knock

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

12.18 COMBUSTION CHAMBERS FOR CI ENGINES

The most important function of the CI engine combustion chamber is to provide proper mixing of fuel and air in a short time. In order to achieve this, an organized air movement called the air swirl is provided to produce high relative velocity between the fuel droplets 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.

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13.1.2 Pumping Loss

In case of the four-stroke engines, a considerable amount of energy is spent during intake and exhaust processes. The pumping loss is the net power spent by the engine (piston) on the working medium (gases) during intake and exhaust strokes. In the case of two-stroke engines this is negligible since the incoming fresh mixture is used to scavenge the exhaust gases.

13.1.3 Power Loss to Drive Components to Charge and Scavenge

In certain types of four-stroke engines the intake charge is supplied at a higher pressure than the naturally aspirated engines. For this purpose a mechanically driven compressor or a turbine driven compressor is used. Accordingly the engine is called the supercharged or turbocharged engine. In case of a supercharged engine, the engine itself supplies power to drive the compressor whereas in a turbocharged engine, the turbine is driven by the exhaust gases of the engine. These devices take away a part of the engine output. This loss is considered as negative frictional loss. In case of two-stroke engines with a scavenging pump, the power to drive the pump is supplied by the engine.

13.1.4 Power Loss to Drive the Auxiliaries

A good percentage of the generated power output is spent to drive auxiliaries such as water pump, lubricating oil pump, fuel pump, cooling fan, generator etc. This is considered a loss because the presence of each of these components reduces the net output of the engine.

13.2 MECHANICAL EFFICIENCY

The various losses described above can be clubbed into one heading, viz., the mechanical losses. The mechanical losses can be written in terms of mean effective pressure, that is frictional torque divided by engine displacement volume per unit time. Therefore, frictional mean effective pressure, $fmep$, can be expressed as

$$fmep = mmep + pmep + amep + cmep$$

where $mmep$: mean effective pressure required to overcome mechanical friction

$pmep$: Mean effective pressure required for charging and scavenging

$amep$: mean effective pressure required to drive the auxiliary components

$cmep$: mean effective pressure required to drive the compressor or scavenging pump

Consider the case of two parallel plates filled with viscous oil in between them, of which one is stationary and other is in motion with a constant velocity in the direction as shown in Fig.13.2. It is assumed that

(i) the width of the plates in the direction perpendicular to the motion is large so that the flow of lubricant in this direction is negligibly small

(ii) the fluid is incompressible

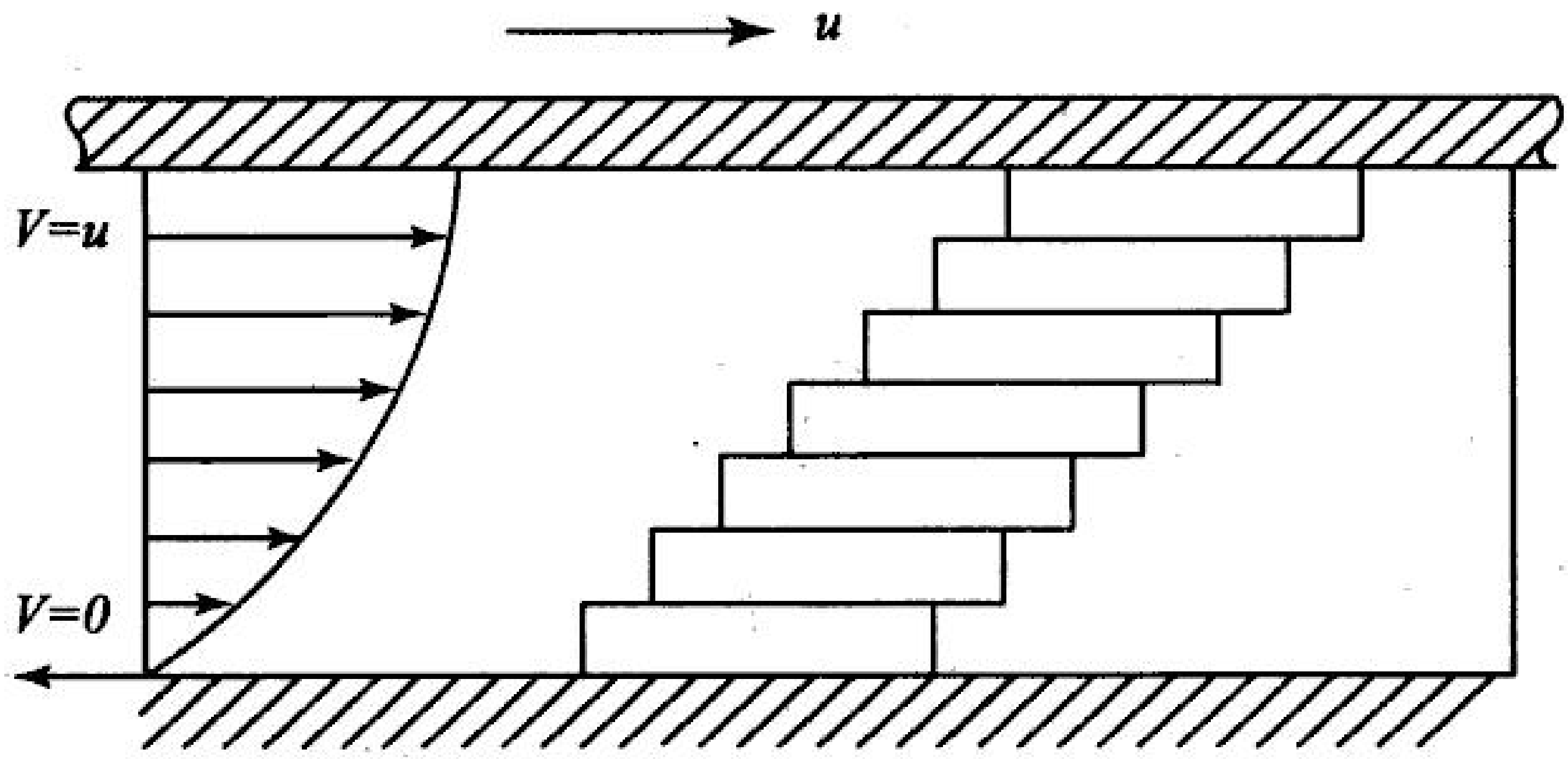

Fig. 13.2 Mechanism of Lubrication in Parallel Surfaces

Let us imagine the film as composed of a series of horizontal layers and the force, F causing layers to deform or slide one over another, just like a deck of cards. The first layer clinging to the moving surface will move with the plate because of the adhesive force between the plate and the oil layer while the next layer is moving by at slower pace. The subsequent layers below keep moving at gradually reducing velocities. The layer clinging to the surface of the stationary plate will have zero velocity. The reason is that each layer of the oil is subjected to a shearing stress and the force required to overcome this stress is the fluid friction. Thus the velocity profile across the oil film varies from zero at the stationary plate to the velocity of the plate at the moving surface, as shown in Fig.13.2.

In the above example, the fluid or internal friction arose because of the resistance of the lubricant to shearing stress. A measure of the resistance to shear is a property called *dynamic viscosity or coefficient of viscosity.*

Now let us consider that the above mentioned plates are non-parallel and the upper one (A′B′) is moving while the lower one (AB) remains stationary. The cross section of the fluid at the leading edge is less than that of the trailing edge. This arrangement is shown in Fig.13.3. The discussion with regard to Fig.13.2 indicates that the velocity distributions across the oil film at the edges are expected to have the shape shown by dotted line in Fig.13.3.

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13.12 ADDITIVES FOR LUBRICANTS

The lubricating oil should possess all the above properties for the satisfactory engine performance. The modern lubricants for heavy duty engines are highly refined which otherwise may produce sludge or suffer a progressive increase in viscosity. For these reasons the lubricants are seasoned by the addition of certain oil-soluble organic compounds containing inorganic elements such as phosphorus, sulphur, amine derivatives. Metals are added to the mineral based lubricating oil to exhibit the desired properties. Thus oil soluble organic compounds added to the present day lubricants to impart one or more of the following characteristics.

(i) anti-oxidant and anticorrosive agents

(ii) detergent-dispersant

(iii) extreme pressure additives

(iv) pour point depressors

(v) viscosity index improvers

(vi) antifoam agent

(vii) oiliness and film-strength agents

13.12.1 Anti-oxidants and Anticorrosive Agents

Oxidation of the lubricating oil is slow at temperatures below 90 °C but increase at an exponential rate when high temperatures are encountered. Oxidation is undesirable, not only because sludge and varnish are formed but also because of the formation of acids which may be corrosive. Thus the additive has the dual purpose of preserving both the lubricant and the components of the engine. To accomplish these purposes, the additive must nullify the action of metals in catalyzing oxidation; copper is especially active as an oxidation catalyst of hydrocarbon. The additives may be alkaline to 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

carried away by lubricating oil and heat lost by radiation amounts to 5 per cent of the total heat supplied. Unless the engine is adequately cooled engine seizure will result. In this chapter the details of engine heat transfer, heat rejection and cooling are considered.

14.2 VARIATION OF GAS TEMPERATURE

There is an appreciable variation in the temperature of the gases inside the engine cylinder during different processes of the cycle. Temperature inside the engine cylinder is almost the lowest at the end of the suction stroke. During combustion there is a rapid rise in temperature to a peak value which again drops during the expansion. This variation of gas temperature is illustrated in Fig.14.1 for various processes in the cycle.

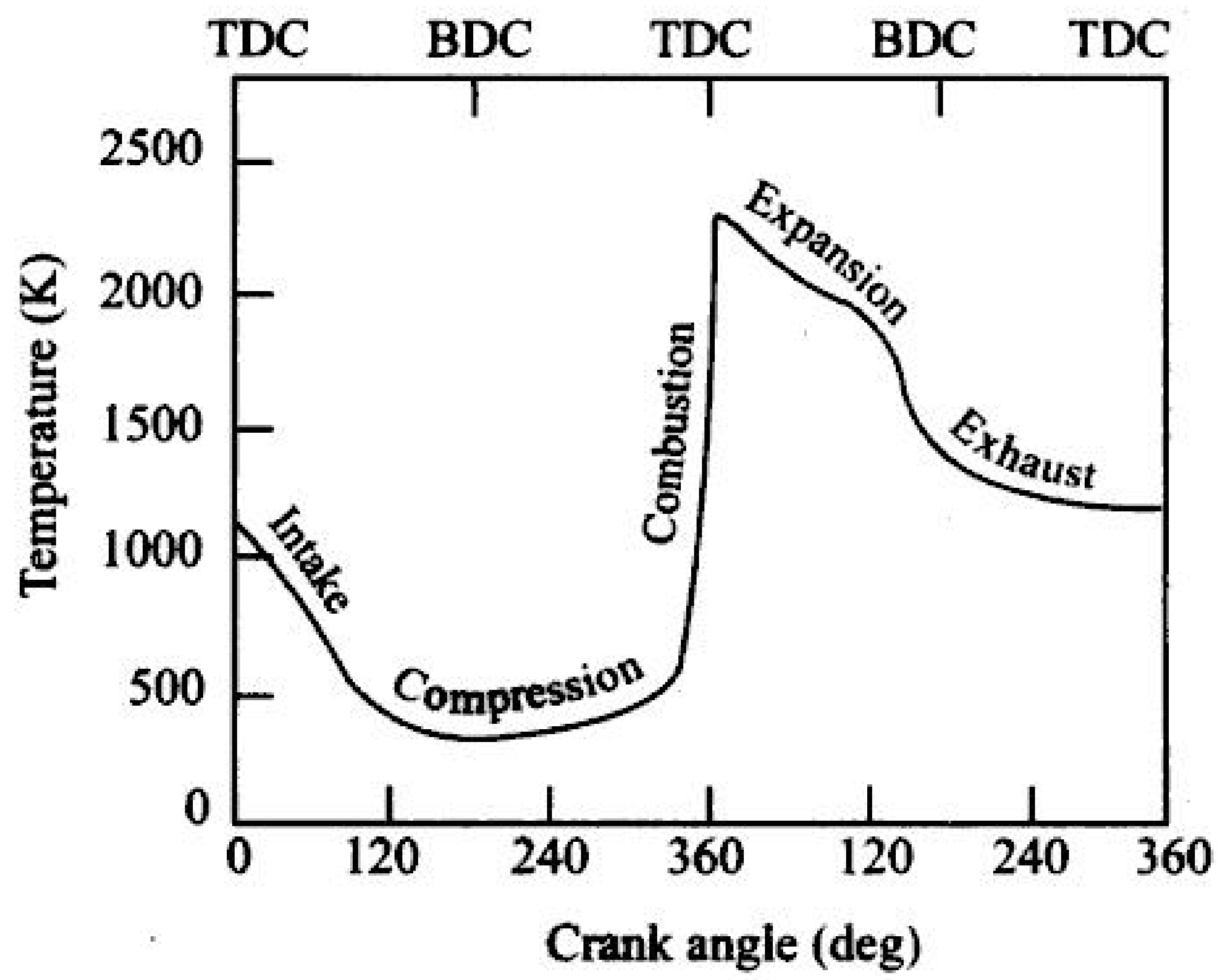

Fig. 14.1 Gas Temperature Variation during a Cycle

14.3 PISTON TEMPERATURE DISTRIBUTION

The piston crown is exposed to very high combustion temperatures. Figure 14.2 gives the typical values of temperature at different parts of a cast iron piston. It may be noted that the maximum temperature occurs at the centre of the crown and decreases with increasing distance from the centre. The temperature is the lowest at the bottom of the skirt. Poor design may result in the thermal overloading of the piston at the centre of the crown. The temperature difference between piston outer edge and the centre of the crown is responsible for the flow of heat to the ring belt through the path offered by metal section of the crown. It is, therefore, necessary to increase the thickness of the crown from the centre to the

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Motoring method, however, gives reasonably good results and is very suitable for finding the losses imparted by various engine components. This insight on the losses caused by various components and other parameters is obtained by progressive stripping off of the engine. First the full engine is motored, then the test is conducted under progressive dismantling conditions keeping water and oil circulation intact. Then the cylinder head can be removed to evaluate by difference, the compression loss. In this manner, piston rings, pistons, etc. can be removed and evaluated for their effect on overall friction.

16.2.4 From the Measurement of Indicated and Brake Power

This is an ideal method by which fp is obtained by computing the difference between indicated power obtained from an indicator diagram and brake power obtained by a dynamometer. This method is mostly used only in research laboratories as it is necessary to have elaborate equipment to obtain accurate indicator diagrams at high speeds.

16.2.5 Retardation Test

This test involves the method of retarding the engine by cutting the fuel supply. The engine is made to run at no load and rated speed taking into all usual precautions. When the engine is running under steady operating conditions the supply of fuel is cut-off and simultaneously the time of fall in speeds by say 20%, 40%, 60%, 80% of the rated speed is recorded. The tests are repeated once again with 50% load on the engine. The values are usually tabulated in an appropriate table. A graph connecting time for fall in speed (x-axis) and speed (y-axis) at no load as well as 50% load conditions is drawn as shown in Fig.16.3.

From the graph the time required to fall through the same range (say 100 rpm) in both, no load and load conditions are found. Let t_2 and t_3 be the time of fall at no load and load conditions respectively. The frictional torque and hence frictional power are calculated as shown below.

Moment of inertia of the rotating parts is constant throughout the test.

$$\text{Torque} = \text{Moment of Inertia} \times \text{Angular Acceleration}$$

Let ω be the angular velocity and $\frac{d\omega}{dt}$ be the angular acceleration.

$$T = I\frac{d\omega}{dt} \tag{16.5}$$

$$I = MK^2 \tag{16.6}$$

therefore,

$$T = MK^2\frac{d\omega}{dt} \tag{16.7}$$

$$d\omega = \frac{T}{MK^2}dt \tag{16.8}$$

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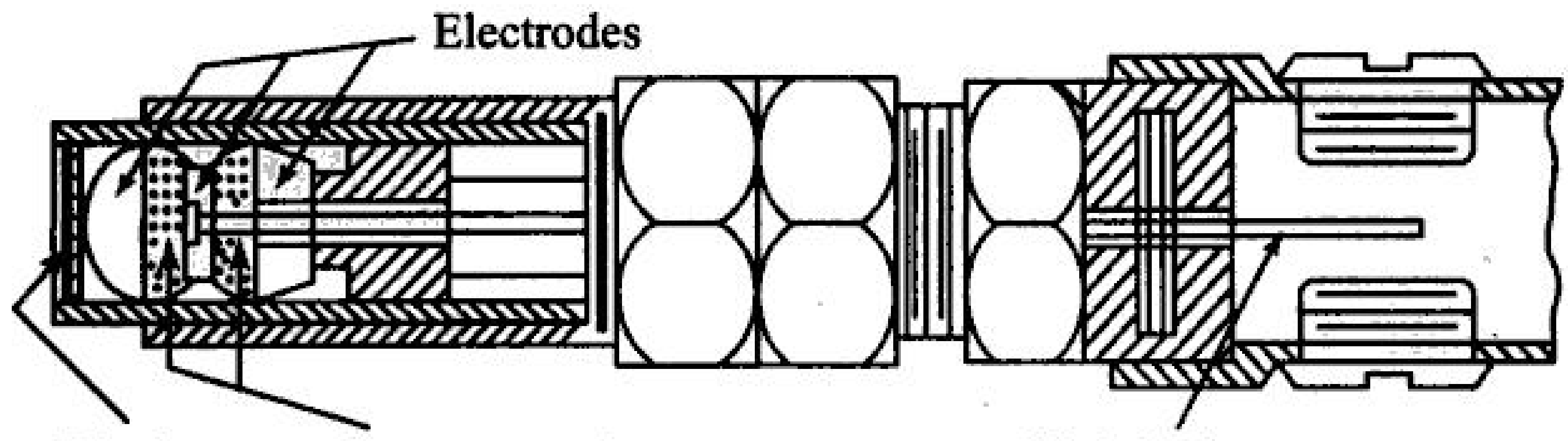

Fig. 16.5 Pressure Transducer

to augment the signal so that it can be displayed on the oscilloscope or recorded in a recorder.

The pick-up used for measuring the cylinder pressure must have a linear output and a good frequency response. Its temperature behaviour should also be satisfactory, i.e., the effect of heat should not affect its performance. Also it must have a low acceleration sensitivity. Of all the pressure transducers currently available the piezo electric quartz transducers are most satisfactory for all normal uses in internal combustion engine measurements as far as the sensitivity is concerned. The quartz transducer, which has a natural frequency greater than 50 kHz, has a sensitivity of about one-tenth of the other types of pressure transducers in which inductive, capacitive or strain-gauge principles are applied. The temperature effect is about 0.005 per cent per degree centigrade change in temperature of the crystal.

The use of electrical pressure pickups avoid almost all the difficulties of a mechanical indicator and gives inertia free operation. However, the greatest problem when using these transducers is the difficulty in calibrating them. A notable exception is strain gauge type transducer which can be satisfactorily calibrated.

Usually the device used to display the p-θ diagram is the storage type cathode ray oscilloscope, CRO. The CRO provides almost inertia free recording and displaying of the pressure signal. The principle of CRO is that a cathode ray can be deflected by the variation of an electric current. It can be of electromagnetic or electrostatic type. Usually the pressure signal from the pressure transducer and the time signal from the time-base pick-up are applied to the two beams of dual beam oscilloscope. This produces a p-θ or p-t diagram on the screen of the oscilloscope which can be observed or photographed. A typical time base unit consists of a permanent magnet with coil and a V-shaped pole piece and a rotating disc having slots in which a magnetic material is fixed. When the disc rotates and a slot passes the permanent magnet it generates a voltage according to its depth and produces a peak on the oscilloscope screen. Usually slots are 1° apart with deeper slots at 10° intervals and still deeper at 90° interval so that a complete degree timing diagram is produced. The disc is so adjusted on the engine shaft that when the deepest slot is against the magnet poles it shows the

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(i) By using experimental results obtained from engine tests.

(ii) By analytical calculation based on theoretical data.

Engine performance is really a relative term. It is represented by typical characteristic curves which are functions of engine operating parameters. The term performance usually means how well an engine is doing its job in relation to the input energy or how effectively it provides useful energy in relation to some other comparable engines.

Some of the important parameters are speed, inlet pressure and temperature, output, air-fuel ratio etc. The useful range of all these parameters is limited by various factors, like mechanical stresses, knocking, over-heating etc. Due to this, there is a practical limit of maximum power and efficiency obtainable from an engine. The performance of an engine is judged from the point of view of the two main factors, viz., engine power and engine efficiency. Besides the overall efficiency, various other efficiencies are encountered when dealing with the theory, design and operation of engines. These factors are discussed in more detail in the following two sections.

17.2 ENGINE POWER

In general, as indicated in section 1.7, the energy flow through the engine is expressed in three distinct terms. They are indicated power, *ip*, friction power *fp* and brake power, *bp*. Indicated power can be computed from the measurement of forces in the cylinder and brake power may be computed from the measurement of forces at the crankshaft of the engine. The friction power can be estimated by motoring the engine or other methods discussed in Chapter 16. It can also be calculated as the difference between the *ip* and *bp* if these two are known, then,

$$ip = bp + fp \tag{17.1}$$

$$fp = ip - bp \tag{17.2}$$

In the following sections, the usually employed formulae for the computation of power are discussed.

17.2.1 Indicated Mean Effective Pressure (p_{im})

It has been stated in section 17.2 that *ip* can be computed from the measurement of forces developed in the cylinder, viz., the pressure of the expanding gases.

As already described, in chapters dealing with cycles, the pressure in the cylinder varies throughout the cycle and the variation can be expressed with respect to volume or crank angle rotation to obtain p-V or p-θ diagrams respectively. However, such a continuous variation does not readily lend itself to simple mathematical analysis in the computation of *ip*. If an average pressure for one cycle can be used, then the computations becomes far less difficult.

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