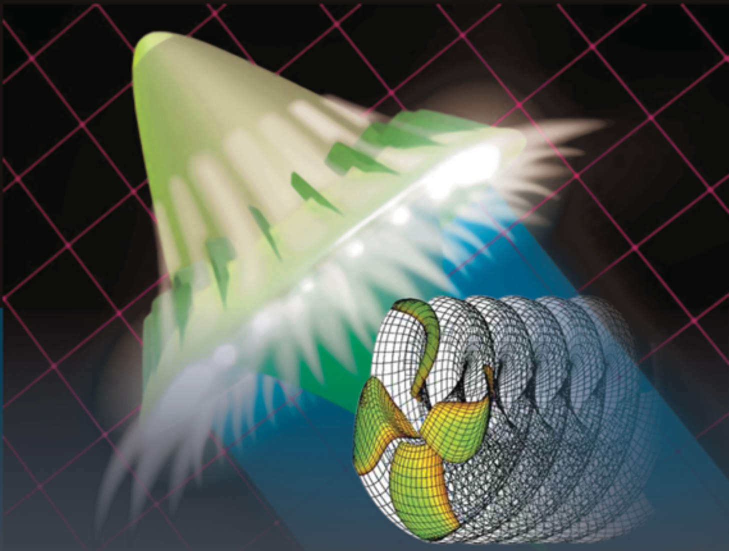

Aircraft and Automobile Propulsion

A Textbook

Himanshu Shekhar

Alpha Science

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Alpha Science International Ltd.
Oxford, U.K.

Aircraft and Automobile Propulsion
A Textbook
260 pgs. | 147 figs.

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ALPHA SCIENCE INTERNATIONAL LTD.
7200 The Quorum, Oxford Business Park North
Garsington Road, Oxford OX4 2JZ, U.K.

www.alphasci.com

ISBN 978-1-84265-778-2

E-ISBN 978-1-84265-975-5

Printed in India

Dedicated to

MY PARENTS

DR. KIRAN SHANKAR PRASAD (FATHER)
DR. KRISHNA PRASAD (MOTHER)

Preface

'Aircraft and Automobile Propulsion' is restricted to chemical propulsion of automobiles, where a working fluid is burnt to produce large amount of gases, which gives propulsive force for execution of motion. Propulsion is a multi-disciplinary science and the main theme of this book is basically chemical propulsion, which needs simultaneous understanding of chemistry of molecules, heat transfer processes, combustion mechanisms, mechanical engineering, mathematics and all gamut of science.

This book has three aspects in general. First is discussion on internal combustion (IC) engines. This part encompasses discussions on both theoretical cycles and operational engines simultaneously. Second aspect is introduction and deliberations on aircraft power plants, where specific requirements of air-worthiness are debated. Third aspect is exhaustive discussion on modes of heat transfer. Three modes of heat transfer namely conduction; convection and radiation are deliberated in detail in this book.

Overall, this book covers theoretical and practical aspects of internal combustion engines and engines for automobile propulsion. This book satisfie well the need of the students, teachers and institutes offering the courses in automobile propulsion. In addition, this book is very useful for graduation in mechanical engineering courses and other engineering streams also where propulsion, internal combustion engine, heat transfer and thermodynamics are essential part of curriculum. This book serves the requirements of multi-dimensional domain of students, teachers, librarians, researchers and industries.

Himanshu Shekhar

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CHAPTER 1

Cycles and Processes

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1.1 INTRODUCTION

In the internal combustion engines, the chemical energy of fuel is first converted into heat through combustion process and then this generated heat is converted into work. The working fluid for air standard cycle is air mixed with fuel, which may be petrol or diesel. Invariably, the air standard cycles are constituted by minimum four processes. It has heat rejection and heat addition processes. Other two processes are work consumed and work produced. Depending on nature of these processes, various cycles are prevalent. This chapter gives a brief description of various cycles and their control terms.

The chapter deals with the reciprocating engine where piston-cylinder arrangement is prevalent. The piston slides inside the cylinder executing compression stroke. At the end of compression heat addition takes place by combustion of fuel and useful work is extract during outward movement of piston executed by hot combustion gases. Once piston moves out of the cylinder, some part of useful work is stored in a flywheel, which releases work for the compression stroke of the piston. This chapter gives details of various air-standard cycles for reciprocating engines.

Objective

After studying this Chapter, you should be able to :

- Understand different air standard cycles
- Evaluate Otto, Diesel and Dual cycles
- Realize various types of efficiencies used in Internal Combustion Engines
- Know power output in terms of torque
- Calculate mean effective pressure and specific fuel consumption

1.2 THERMODYNAMIC PROCESSES AND CYCLES

Thermodynamics is a physical science and is based on observation of nature. It is made of Greek words *therme* meaning heat and *dynamis* means force; so it is study of energy conversion between heat and mechanical work. Then how it is different from heat transfer? If heat moves from one place to other, the thermodynamics only gives conditions responsible for movement of heat and at most how much heat can be transferred. However, heat transfer gives rate of transfer of heat also and ultimately it gives idea about time taken to heat an object. Heat transfer gives temperature at various points in the system, while thermodynamics only gives total amount of heat transferred and assumes entire system at a uniform temperature. **Thermodynamics gives quantity of heat while heat transfer gives rate of energy transfer.** There are certain other rate processes like mass transfer, momentum transfer, chemical kinetics etc.

In thermodynamics formulation of intuitive and primitive concepts has a major role. For example, if heat is given to a system, temperature will rise. Mathematical formulation of this intuitive idea is subject matter of thermodynamics. It involves several terms like energy, equilibrium, property, system, process, work, heat *etc.* All of them have a precise definition in thermodynamics. Before anything else, definition of thermodynamics is to be understood.

'Best way to study a subject is to understand it before you start'

Hotsopoulos and Keenan : Thermodynamics is the science of states and change of state of physical systems and the interaction between systems, which may accompany changes of states.

Callen : Thermodynamics is the study of the macroscopic consequence of myriads of atomic coordinates, which by virtue of the statistical averaging; do not appear explicitly in a macroscopic description of a system.

Epstein : Thermodynamics deals with systems, which, in addition to mechanical and electromagnetic parameters, are described by a specifically thermal one, namely the temperature or some equivalent of it. Thermodynamics is essentially a science about the conditions of equilibrium of systems and about the processes, which can go in states little different from the state of equilibrium.

Kestin : The science of thermodynamics is a branch of physics. It describes natural processes in which changes in temperature play an important part. Such processes involve the transformation of energy from one form to another. Consequently thermodynamics deals with the laws, which govern such transformation of energy.

Van Wylen and Sonntag : Thermodynamics is a science of energy and entropy.

In general, thermodynamics is (*i*) study of three 'E's namely energy, entropy and enthalpy (*ii*) study of heat and work interactions (*iii*) study of thermal effects on system (*iv*) study of energy, matter and the law governing their interactions. Whenever thermodynamic study is conducted, a region in space or a fixed mass or volume of material is considered. The constant volume or mass or control volume on which attention is focused for a given study is called '**system**'. Anything outside system is called '**surrounding**' and the demarcation of system is called '**boundary**'. Boundary is basically a hypothetical dividing line between system and surrounding. A diathermal boundary is one, which allows heat transfer, while adiabatic boundary means prevention of any heat transfer across the boundary.

Once system is defined, then it undergoes two types of interaction with surrounding. First type is called energy transfer. This energy transfer is also divided into two parts namely work transfer and heat transfer. **Heat transfer is basically all the energy transfers between the system and surrounding due to temperature difference.** Heat flow out of the system is negative heat transfer. **Work transfer** is all other forms of energy transfer and is defined in thermodynamics as **those energy transfers in which sole effect external to the system can be reduced to raising or lowering of weight.** Work transfer associated with raising of weight is taken positive as convention. This definition of work is different from work defined in mechanics. In mechanics, work is associated with displacement due applied force and is defined as product of force and displacement. However, in many energy conversion systems displacement is not easily available. Another form of interaction is called mass transfer. Mass transfer is associated with change in mass of the system and is always associated with the energy transfer. Mass transfer in absence of energy transfer is not possible.

Depending on these two forms of system-surrounding interactions, namely energy transfer and mass transfer, systems are classified in three different modules. First form is called **closed system**, in which mass transfer is prohibited and only energy transfer is permitted through boundary of the system. The mass of the system remains constant. Another form is called **open system**, which permits both mass and energy interactions. To study open system, generally closed volume concept is utilized, which is a fixed space, through which working fluids pass. Almost all the real systems are open system. Third type of system is called **isolated system**, which prevents interaction of both the types through their boundaries. Combination of both system and surrounding is an isolated system.

For each system, there are certain fixed properties on which attention is focused. Suppose in a class, each student possesses several unique characteristics like name, roll number, sex, religion, age, date of birth, father's name, height, marks obtained *etc.* Rather than specifying name, a number of these characteristics can help us in identification of a particular set of students. Male student with 5 feet 9 inch height born on 20 July may result in identification of a single student. There are different ways in which students in a class can be studied. It may be their sex-ratio, height profile; date of birth dispersion etc and for each study one particular property is considered. For a system, these set of characteristics are called '**property**'. Pressure is a property, volume is a property, and temperature is a property and so on.

Properties of a system may be classified as (*i*) **Intrinsic properties**, which are independent of mass of the system like pressure, temperature, internal energy, density *etc.* They are not additive. They are also called **thermostatic properties**. They arise due to mass and are evaluated by considering mass inside a system and without considering surrounding. They do not require any external datum point for their measurement. (*ii*) **Extrinsic properties** increase with increase in mass of the system like volume, energy *etc.* Ratio of two extrinsic properties is always intrinsic property. Properties acquire a definite value only at the state of equilibrium. The concept of thermodynamic equilibrium needs little explanation.

Thermodynamic equilibrium of a system is basically a combination of three forms of equilibrium – **mechanical equilibrium, thermal equilibrium and chemical equilibrium**. A system is said to be in mechanical equilibrium, if there is no unbalanced force or moment acting on the system. This ensures that system does not change position. Thermal equilibrium ensures that temperature of system and surrounding are same. There is no heat transfer across the diathermal boundary of the system and surrounding. There is no temperature change in the system. Existence of thermal equilibrium and no thermal interaction at equality of temperature is, in fact, derived from **zeroth law of thermodynamics**, which states – '**If a body is in thermal equilibrium separately with another two bodies, the other two bodies will also be in thermal equilibrium with each other**'. This law of thermodynamics can be expressed in several other forms. '**Energy can neither be created nor be destroyed**' is the statement of **first law of thermodynamics**. This indirectly indicates that energy can only change form. Another way to express this law is – '**Work cannot be executed without some form of equivalent amount of heat getting absorbed by the system**'. SI unit of work is joule (J), while for heat it is calories (cal) and 1 cal = 4.187 J. The conversion factor of heat to work is called **mechanical equivalent of heat**. Chemical equilibrium is another from of equilibrium, which ensures absence of any unaided spontaneous chemical reaction. The material, working fluid or content of the system should not undergo any chemical reaction of its own and bring any change in the composition of the system. Thermodynamic equilibrium is basically a hypothetical concept to understand and ensure that system properties are fixed and it does not change of its own. It also indirectly indicates that all the systems will lead to thermodynamic equilibrium, if left undisturbed. At thermodynamic equilibrium, properties of the system have a fixed value.

The distinguishing characteristics of the system are called properties of the system. Physical description of each system is characterized by certain quantities like pressure, volume, temperature *etc.* They are called properties of the system. These are all macroscopic in nature. Properties may be directly or indirectly observable characteristic of the system. Any combination of such characteristics is also a property. For example product of pressure and volume is property of a system. New properties can be defined in terms of already known properties. Such derived properties include enthalpy, Gibb's free energy, Helmholtz Function. Properties of a system are always called a state function or a point function and change of properties is dependent on end properties only.

For the system in thermodynamics, some of the unique properties of the systems need special attention. Properties of the system can be stated to exist in **conjugate pairs**. These pair in combination gives value of work. For example, in mechanics product of force and displacement gives work and they form a conjugate pair. Out of the elements of a conjugate pair, one is cause (force) and other is effect (displacement). For thermodynamics systems, one conjugate pair is pressure-volume, while another is temperature-entropy. Pressure is cause and it results in change in volume. Area under the curve on a pressure volume thermodynamic plane is equal to work interaction in the system. Similarly temperature is a cause, which results in change of entropy. As their product gives heat interaction during any change in the system, they form a conjugate pair. Here it must be noted that as per first law of thermodynamics, heat and work are mutually convertible. One interaction can be replaced totally by the other form of energy interaction. These four properties are important for understanding various energy transfer processes. So, each one is defined below.

Pressure (P) is derived from force directly. Force acting perpendicular to a surface is called thrust and thrust per unit area is called pressure. Force is a vector quantity, area is also a vector quantity, but pressure is a scalar quantity. It only indicates amplification of vector area to denote force acting on the surface. The scalar behaviour of pressure can be understood from the fact that hydraulic pressure at certain depth from free surface of water remains same and is independent of direction of considered plane at given depth. This makes it different from stress, which has identical units as pressure, but has tensor properties. Stress is force of resistance generated inside the body against an externally applied force, while pressure is constituted by the external forces directly. The application of pressure is seen in real life. A knife can cut vegetables with manual efforts. This is because, due to small area at the sharp edges of the knife, value of pressure becomes so high that vegetables get cut. However applying same force with a blunt knife or opposite edge of the knife require much higher force for execution of same operation. Similarly putting nail on wall, ease of movement for camels in desert, *etc.,* are some direct applications of pressure in real life.

Pressure is expressed in Pascal (Pa = N/m^2). Atmospheric pressure is measured by barometer. However, there are several other auxiliary units prevalent for indicating pressure. Their relations are given below.

1 atmospheric pressure = 1.0332 kg/cm^2 = 1.0332 ata = 1.01325 bar = 101.325 kPa = 1.01325 × 10^5 N/m^2 = 1.01325 × 10^5 Pa = 14.696 psi = 760 mm Hg = 10.26 m of water.

Pressure at a given point is called absolute pressure. This is used only when equation of state is to be used. In most of the real systems, it is change of pressure, which is important. So instead of absolute pressure, some relative variation of pressure is important. In real systems, value of pressure over atmospheric pressure is desirable. Such excess overpressure is called gauge pressure. In automobile tires, the pressure is expressed as 220 kPa/32 psi. These values are, in fact, gauge pressures and atmospheric pressure (101.325 kPa/14.696 psi) should be added to express the values correctly in absolute pressure. The absolute pressure in tires is 321.325 kPa/46.696 psi.

For pressure of gases, it needs a little more discussion. As per kinetic theory of gases, molecules of gases are in continuous motion inside a container. They collide amongst themselves and with the wall of the container in elastic manner and rebounds without any loss of energy. The net interactions of these molecules with wall of the container cause pressure.

▮▮ EXAMPLE 1.1

A manometer has two limbs. One limb is open to atmosphere and other is connected to a pipe. If mercury (density 13.6 g/cc) is filled in manometer and open limb is 300 mm above the height of mercury in other limb of the manometer, find pressure in the pipe.

SOLUTION

Gage pressure in the open limb of the pipe

= Pressure due to difference in mercury column

= Density of mercury × Height of mercury column

$= 13.6 \times 30 \text{ g/cm}^2 = 4.08/1000 \text{ kg/cm}^2$

$= 0.408 \times 101.325/1.0332$ kPa

= 40.0 kPa.

As limb connected to pipe has on lower side, pressure in pipe is higher than atmosphere. So, absolute pressure in the pipe = atmospheric pressure + gage pressure

= 101.325 kPa + 40 kPa = 141.325 kPa.

▮▮ EXAMPLE 1.2

In example 1.1, if height of mercury in open limb is 300 mm below the height of mercury in the other limb, find pressure in the pipe.

SOLUTION

Gage pressure remains same as 40.0 kPa as calculated in the previous solution. However, absolute pressure in the pipe will be lower than atmospheric pressure by the pressure equal to gage pressure.

So, absolute pressure in the pipe = atmospheric pressure – gage pressure = 101.325 kPa – 40 kPa = 61.325 kPa.

Volume (V) is another significant property for a system and is derived from the geometry of the system. Volume is how much three-dimensional space a substance (solid, liquid, gas, or plasma) or shape occupies or contains, often quantified numerically using the SI derived unit, the cubic meter. The volume of a container is generally understood to be the capacity of the container, *i.e.* the amount of fluid (gas or liquid) that the container could hold, rather than the amount of space the container itself displaces. Three dimensional mathematical shapes are also assigned volumes. Volumes of some simple shapes, such as regular, straight-edged, and circular shapes can be easily calculated using arithmetic formulas. The volumes of more complicated shapes can be calculated by integral calculus if a formula exists for the shape's boundary. One-dimensional figures (such as lines) and two-dimensional shapes (such as squares) are assigned zero volume in the three-dimensional space. The volume of a solid (whether regularly or irregularly shaped) can be determined by fluid displacement. Displacement of liquid can also be used to determine the volume of a gas. The combined volume of two substances is usually greater than the volume of one of the substances. However, sometimes one substance dissolves in the other. In that case, the combined volume is not obtained by addition of volumes of solvent and solute. In thermodynamics, volume is a fundamental parameter, and is a conjugate variable to pressure.

EXAMPLE 1.3

Find volume of a figure made by a hemisphere placed over a cylinder of diameter of 30 mm and height of 50 mm. radius of hemi-sphere is same as that of cylinder and bottom end of the cylinder is flat.

SOLUTION

Given, Diameter, $D = 30$ mm,

Height of the cylinder, $H = 50$ mm.

$$\text{Volume of the cylinder} = \pi D^2 H/4 = \pi \times 302 \times 50/4$$

$$= 35342.9 \text{ mm}^3 = 35.3429 \text{ cc.}$$

$$\text{Volume of the hemi-sphere} = \pi D^3/12 = \pi \times 303/12$$

$$= 7068.58 \text{ mm}^3 = 7.06858 \text{ cc.}$$

$$\text{Total volume of the figure} = 35.3429 \text{ cc} + 7.06858 \text{ cc} = 42.4115 \text{ cc.}$$

EXAMPLE 1.4

Find work interaction in the process, which takes place in such a way that product of pressure and volume is constant. Pressure of the system changes from atmospheric pressure to 600 kPa and initial volume is 400 cc.

SOLUTION

Since pressure and volume are conjugate process, product of both can give work. For a given process, work interaction is given by area under the pressure volume curve. Here it is given that product of pressure and volume is the same. If first condition is expressed as subscript 1 and final condition by subscript 2, then

Given, $P_1 = 101.325$ kPa

$P_2 = 600$ kPa

$V_1 = 400$ cc

$PV =$ Constant

or, $P \times V = P_1 \times V_1 = P_2 \times V_2$

or, $V_2 = P_1 \times V_1/P_2 = 101.325 \times 400/600 = 67.55$ cc.

Work interaction

$$W = \int PdV = \int (P_1 \times V_1/V)\, dV$$
$$= (P_1 \times V_1) \int (1/V)\, dV = (P_1 \times V_1) \ln (V_2/V_1)$$
$$= (P_1 \times V_1) \ln (P_1/P_2)$$
$$= (P_2 \times V_2) \ln (P_1/P_2)$$
$$= (P_2 \times V_2) \ln (V_2/V_1)$$

(Any one of above 4 relations can be used for the calculation of work interaction)

$$= 101.325 \text{ kPa} \times 400 \text{ cc} \ln (101.325/600)$$
$$= 40530 \times -1.7786 \times 10^{-6} \text{ kJ}$$
$$= -0.0721 \text{ kJ.}$$

Since work interaction is negative, it indicates that work is done on the system. Another observation is calculation of volume is a redundant exercise and without calculating volume also calculations can be made for work interaction.

EXAMPLE 1.5

Solve example 1.4, if product of pressure and square of the volume is constant for the system.

SOLUTION

Given, $P_1 = 101.325$ kPa

$P_2 = 600$ kPa

$V_1 = 400$ cc

$PV^2 =$ Constant

or, $P \times V^2 = P_1 \times V_1^2 = P_2 \times V_2^2$

$$V_2 = V_1 \times \sqrt{P_1/P_2} = 400 \times \sqrt{101.325/600} = 164.38 \text{ cc.}$$

Work interaction,

$$\text{W} = \int P\, dV$$

$$= \int (P_1 V_1^2 / V^2)\, dV = (P_1 V_1^2) \int dV/V^2$$
$$= (P_1\ V_1^2)\ [(1/V_1) - (1/V_2)]$$
$$= 101.325 \times 400^2\ [(1/400) - (1/164.38)] \times 10^{-6}\ \text{kJ}$$
$$= -\ 0.0581\ \text{kJ.}$$

Since work interaction is negative, it indicates that work is done on the system. As far as absolute work done is concerned, this numerical value of work transfer is lower than that in the earlier case. This indicates that increasing power of volume in the equilibrium equation leads to reduction in numerical value of work. If product of pressure and cube of the volume is constant for the system, the numerical value of the work done will reduce further.

EXAMPLE 1.6

Solve example 1.4, if pressure is proportional to volume of the system.

SOLUTION

Given, $P_1 = 101.325$ kPa

$P_2 = 600$ kPa

$V_1 = 400$ cc

$P/V =$ Constant

or, $P/V = P_1/V_1 = P_2/V_2$

or, $V_2 = P_2 \times V_1/P_1 = 600 \times 400/101.325 = 2368.6$ cc.

Work interaction,

$$W = \int P dV = \int (P_1 V/V_1) dV$$
$$= (P_1/V_1) \int V dV$$
$$= (P_1/V_1)\ (V_2^2 - V_1^2)/2$$
$$= (P_2/V_2)\ (V_2^2 - V_1^2)/2$$

(Any one of above 2 relations can be used for the calculation of work interaction)

$$= (101.325/400) \times (2368.62 - 4002) \times 10^{-6}/2\ \text{kJ}$$
$$= 0.69\ \text{kJ.}$$

Since work interaction is positive, it indicates that work is done by the system.

Temperature (T) is a measure of degree of hotness or coldness of the body. In thermodynamics, heat flows from a body at higher temperature to a body at lower temperature and in fact second law of thermodynamics negates any reverse heat interaction of its own. As per Clausius statement of second law of thermodynamics – '**It is impossible to construct a device, which operating in a cycle will produce no effect other than transfer of heat from a body at low temperature to a body at high temperature**'. An equivalent statement of second law of thermodynamics is given by Kelvin Plank as – '**It is impossible for a system to produce work continuously by exchanging heat with a single heat reservoir, maintained at constant temperature**'. As per kinetic theory of gases, it is defined as interaction density. Higher temperature means more kinetic energy with gaseous molecules and subsequently less inter-collision time of the gaseous molecules. The temperature in SI unit is expressed in Kelvin. However there are several other units used to express temperature. Their relations are given below :

1 Kelvin = – 273.15°Celsius = – 459.67° Fahrenheit = 1° Rankin.

Various units are written with symbol derived from first alphabet of their names. Selection of scale for these units is arbitrary and for comparison, ice point and boiling point of water is considered as reference. For first two units, difference is 100 units, while for the later two the difference is 180 units. Their interrelations are given below :

$$T\ \text{K} = (T - 273.15)°\text{C} = [(T - 273.15) \times (180/100)] + 32°\text{F} = 180 \times T/100°\text{R}.$$

Temperature is measured by a device called thermometer. In thermometer, a property of the system is selected, which varies with temperature. Such properties are called thermometric properties. The properties can be volume, pressure, resistance, voltage or any other property of the substance, which varies with temperature. Mercury in glass thermometer, which is used to measure temperature in domestic application, uses length of mercury column as an indicator of temperature. Higher is temperature higher is length of mercury column. This is a volume based temperature measurement device. For thermometric properties certain requirements are set. First is linearity, which indicates that for a given change in temperature, the variation in thermometric property is fixed and is independent of temperature level. For example, if change in length for a change in temperature from 30°C to 70°C is 40 cm on a thermometer, then linearity demands that for any other temperature levels say from 200°C to 240°C, the change in length of column in thermometer should be 40 cm. linearity also demands that, for each degree Celsius rise in temperature 1 cm change in length of column must be seen at any temperature levels. Other requirements are reproducibility and consistency. The readings on thermometer and at least variation in thermometric property with temperature should be independent of place, atmospheric conditions, method of measurement, operator and other sundry factors affecting measured parameter.

EXAMPLE 1.7

Convert –40°C into degree Fahrenheit.

SOLUTION

Given temperature = –40°C.

Formulation: T°C = (180 × T/100) + 32°F

= (180 × –40/100) + 32°F = –72 + 32°F = –40°F

So at –40°, both Celsius and Fahrenheit scales give same temperature.

Using this identity, another conversion formulation, which is more uniform as compared to earlier formulation, can be developed between Celsius and Fahrenheit.

(T + 40)/100°C = (T + 40)/180°F.

Entropy (S) is a measure of quality of energy to do work. Higher quality means higher capability to do work. It is a conjugate property of temperature. Work reservoir has highest level of work output. Energy of work reservoir is of highest quality. The potential of a system to do work is also called availability. Total energy of a system is equal to sum of available energy and unavailable energy. Available energy is work, which is possible to derive from the system. Entropy is proportional to unavailable energy of the system. So, $dS = K \times dE_{UA}$. $= K \times (dE - dE_{AV})$. For an isolated system, there is no change in the total energy, $dE = 0$. As degradation of energy occurs, dE_{AV} must be negative thus making dS positive. This gives $dS_{\text{ISO}} > 0$ and is written as entropy of an isolated system always increases, it can never decrease.

For a work reservoir, entire energy is available as work. So total energy is same as available energy, thus making $dS = 0$ for work reservoir. A heat reservoir is characterized by constant temperature T and has infinite heat capacity. It has a fixed energy. $dE = dQ$ or $dS = k \times (dQ - dE_{AV})$. The constant of proportionality is equal to $1/T$ and $dS = dQ/T$ for a heat reservoir. When heat is added, entropy will increase or capacity of reservoir to do work increases.

Every system has entropy and it is a primitive concept. Entropy is an extensive property.

Work reservoir has fixed entropy. For other systems, it changes.

Entropy is a measure of unavailable energy.

Entropy of an isolated system is not conserved but increases monotonically.

If change in entropy is positive, it is an irreversible process. If it is zero, process is reversible. If change in entropy is negative, the process is impossible.

In thermodynamics, internal energy (U) also plays a major role. It is all forms of energy other than kinetic (KE) and potential (PE) energy. Total energy of a system, $E = U + PE + KE$. As per first law of thermodynamics, $dQ = dW + dE$. Here dW includes all types of quasi-static process including stretching, electric current flow through a resistor, magnetic interactions *etc.* Only PdV work is relevant for thermodynamic system. If there is no accounting for PE and KE, total energy can be replaced by internal energy. $dQ = dU + PdV$. For a constant pressure process, $dQ = d(U + PV)$. A function enthalpy is defined, $H = U + PV$, then $dQ = dH$. So, change in enthalpy is heat transfer in a quasi-static constant pressure process for a substance or system where PdV work is important. Enthalpy is a purely mathematical convenience and it is not limited to special process only. Enthalpy also represents condition for thermal equilibrium and in other processes, it still represents heat exchange. For ideal gases, internal energy and enthalpy depends on specific heat of the working fluid.

Although solids and liquids have only one heat capacity but gases exhibits two specific heats, mainly due to their high level of compressibility. The following are of special significance.

(*i*) Specific heat at constant volume (C_v)

(*ii*) Specific heat at constant pressure (C_p)

If we consider a unit mass of gas, these specific heats are called the principal specific heats of the gas. In this case, C_v denotes the specific heat of the gas at constant volume. It is defined as the amount of heat required to raise the temperature of 1 gram of the gas through 1°C at constant volume. The specific heat of the gas at constant pressure is denoted by C_p. It is defined as the amount of heat required to raise the temperature of 1 gram of the gas through 1°C at constant pressure. Instead of considering one gram of the gas, if we consider one mole of the gas, then specific heats are called gram molecular or molar specific heats of the gas.

C_p is greater than C_v. If a gas is heated at constant volume, the gas does no work against external pressure. In this case, the whole of the heat energy supplied to the gas is spent in raising the temperature of the gas.

If a gas is heated at constant pressure, its volume increases. In this case, heat energy is required for the following two purposes:

(*i*) To increase the volume of the gas against external pressure. (*ii*) To increase the temperature of 1 mole of gas through 1 K.

Thus, more heat energy is required to raise the temperature of 1 gram mole of gas through 1 K when it is heated at constant pressure than when it is heated at constant volume. The difference between C_p and C_v is the thermal equivalent of the work done by the gas in expanding against external pressure.

EXAMPLE 1.8

Find heat transaction constant volume process undergone by air, if temperature of air changes from 300K to 550 K. Carry out same calculation for constant pressure process. Specific heats at constant volume and constant pressure of air are 0.718 kJ/kg/K and 1.008 kJ/kg/K.

SOLUTION

Given that for the process volume is constant and specific heat at constant volume is needed for calculation. Heat transacted = C_v × (Change in temperature)

$$= 0.718 \times (550 - 300) \text{ kJ/kg} = 179.5 \text{ kJ/kg}.$$

Calculation is repeated for constant pressure process also. As pressure is constant, heat transaction needs utilization of specific heat at constant pressure.

$$\text{Heat transacted} = C_p \times (\text{Change in temperature})$$
$$= 1.008 \times (550 - 300) \text{ kJ/kg} = 252 \text{ kJ/kg}.$$

When properties of a system are completely defined, they are said to exist in a state. Properties are coordinates to define state of a system. They are state variables of the system. If properties of system change, they are said to have undergone a change of state. The succession of states passed through by the system during a change of state is called the path of change of state.

State postulate is one concept, which is helpful to develop functional relationship between intrinsic properties. Properties may be assigned an arbitrary numerical value, which will be referred for all subsequent calculations. To develop such relationship, it is needed to find out, how many intrinsic properties of a substance can be varied independently. Number of independent intrinsic properties required to fix the state of a substance is equal to one more than number of possible relevant quasi-static work modes. The equilibrium state of a simple homogenous substance is fixed by specifying the values of any two independent intrinsic properties.

If path is completely defined, then change of state is called a thermodynamic process *e.g.* isobaric process. In isobaric process, pressure is constant. There are several processes that are encountered in thermodynamics. If volume is constant during a process, it is called isochoric process. If temperature is constant, the process is called isothermal process and if entropy is constant, the process is called isentropic process. For an isentropic process, there is no heat transfer and process occurs in adiabatic manner. The governing relation is expressed as PV^γ = constant, where γ is isentropic exponent and is equal to ratio of specific heats of the gases. Value of isentropic exponent is generally 1.4 for air treated as ideal gases. For an ideal gas, combined gas law is given by the following relation.

Pressure × Volume = Mass × Gas constant × Temperature

$$P \times V = m \times R \times T.$$

Here gas constant has different values as per units of other properties of the system. **Universal gas constant can be 8.314 kJ/kg-mol/K or 82.04 atm litre/kg mol/K or 1.987 kcal/kg mol/K.**

Different processes have different representation on PV diagram. Isobaric is horizontal and isochoric is vertical. It is clear that, as 'n' rises, the curve shifts in clockwise direction. The expression of slope for any general curve represented by PV^n is given by $-nP/V$. Higher 'n' means higher numerical value of the slope or steeper curve. Isobaric process has, $n = 0$, slope is zero. Isothermal process has $n = 1$, adiabatic process has slope equal to 1.4 and isochoric process has n = infinity. So, curves are traced accordingly as figure 1.1. Sequence of slopes of different curves at any point is p-T-h-s-V in clockwise direction.

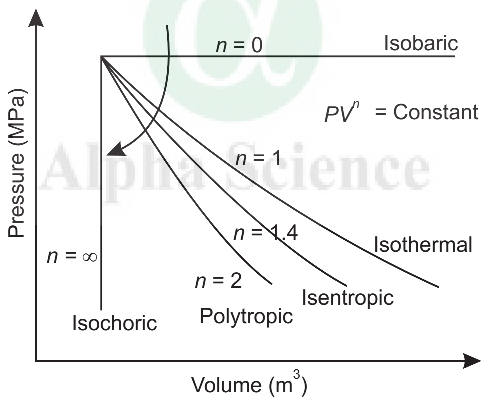


Fig. 1.1 : Variation of Different processes on P-V Plane

Area under P-V curve is work done in the process. Since work is a path function, it depends on the process undergone. Although it is expressed in terms of end condition, but complete description need specification about nature of the process. Different processes have different work expression.

Isobaric:	$W = P(V_2 - V_1)$
Isothermal:	$W = P_1 \times V_1 \times \ln (V_2/V_1)$
Polytropic:	$W = [P_1 \times V_1 - P_2 \times V_2] / (n - 1)$
Isochoric process:	$W = 0.$

Another thermodynamic plane is created by temperature and entropy, which forms a conjugate pair. Area under *T-S* diagram gives heat transfer is the process. Temperature should be in absolute scale. For a polytrophic process, first law of thermodynamics states that heat supplied to a system does two types of changes – increase in internal energy of the system and work output from the system. Heat supplied to the system must be equal to sum of increase in internal energy and work done by the system.

$$\begin{aligned} ds &= [du + p\,dv]/T = c_v dT/T + R\,dv/v;\ (\partial T/\partial s)v = T/c_v \\ &= [dh - v\,dp]/T = c_p dT/T - R\,dp/p;\ (\partial T\partial s)p = T/c_p. \end{aligned}$$

So many observations are possible from the basic fact that for an ideal gas $C_p > C_v$. In the example 1.7, it is clear that heat transfer in an isochoric process is less than the same in constant volume process. Isobaric and isochoric processes are plotted in figure 1.2.

Slopes of isobaric as well as isochoric processes are positive on *T-S* diagram. So rise in temperature is associated with increase for entropy for both isobaric and isochoric processes.

Slope of constant volume line at a point on *T-S* plane is higher than that of constant pressure line. This indicates that for a given initial state, given higher temperature is attained with lower rise in entropy resulting in lower area under the curve on *T-S* plane, as compared to same for an isobaric process. So, heat transfer in isobaric process for same temperature rise is higher than same in isochoric process. This is confirmed by example 1.7 also.

From the first equation, if isothermal process ($dT = 0$) is considered, $ds = Rdv/v$, or $s_2 - s_1 = R \ln (v_2/v_1)$. As volume decreases $v_2 < v_1$, $s_2 < s_1$ and line shifts towards left or entropy of the system decreases.

From the second equation, if isothermal process ($dT = 0$) is considered, $ds = -Rdp/p$, or $s_2 - s_1 = -R \ln (p_2/p_1)$. As pressure decreases $p_2 < p_1$, $s_2 > s_1$ and line shifts towards right or entropy of the system increases.

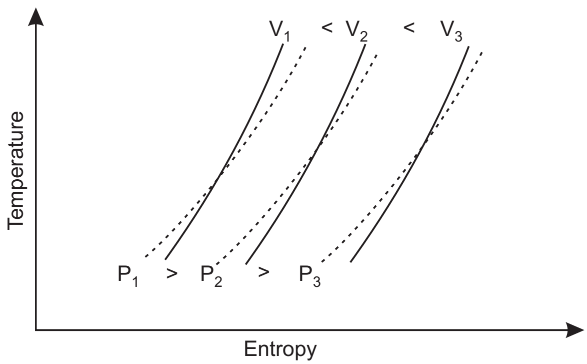


Fig. 1.2 : Variation of Different processes on *T-S* Plane

EXAMPLE 1.8

Find heat and work interactions in an isobaric process, in which initial volume and temperature of working fluid air is 300 cc and 300 K respectively. Constant pressure is 200 kPa and final volume is 200 cc. Mass of air is 0.5 g and specific heat at constant pressure is 1.008 kJ/kg/K.

SOLUTION

The process is carried out in isobaric manner, so pressure is constant. Air can be assumed to behave as ideal gas and gas law can be applicable. Initial state is represented by subscript 1 and final state by subscript 2. Since volume is reduced at constant pressure, it is a process of contraction at constant pressure, which is always associated with reduction in temperature. In other words, the process is a constant pressure cooling operation, where both temperature and volume reduces in proportion.

Given, $P_1 = 200$ kPa, $V_1 = 300$ cc, $T_1 = 300$ K

$P_2 = 200$ kPa, $V_2 = 200$ cc,

$\text{m cp} \times V/T = \text{Constant} \quad \text{or } T_2 = (P_2/P_1) \times (V_2/V_1) \times T_1 = 200 \text{ K}.$

$$\text{Work transfer, W} = \text{Pressure} \times \text{change in volume}$$

$$= 200 \times (200 - 300) \times 10^{-6} \text{ kJ} = -0.02 \text{ kJ}$$

Work is negative, so work is done on the system.

$$\text{Heat transfer, } Q = \text{m cp } (T_2 - T_1)$$

$$= 0.5 \times 10^{-3} \times 1.008 \times (200 - 300) = -0.0504 \text{ kJ}.$$

Heat transfer is negative. It indicates heat is taken out from the system. Numerically more heat is taken out of the system as compared to work done on the system, so internal energy of the system will decrease in this process. Change in internal energy of the system is $dU = Q - W = -0.0304$ kJ. Negative sign indicates reduction in internal energy of the system.

Quasi-static process is one important concept. In this case, system departs at every instance from initial equilibrium state only marginally or infinitesimally. In such cases, path followed by the system is basically succession of equilibrium states. Such processed are internally reversible. This is an ideal process. It is approximately realized when change occurs very slowly. All processes in real life are not quasi-static, because there is always some finite difference of P, T between system and surrounding.

A process is reversible if after it has been executed, it is possible to bring both system and surrounding involved in the interaction back to original state. It is possible to undo a reversible process in such a way that no trace of occurrence of that process exists. All real processes are irreversible.

Irreversibility occurs due to (*i*) lack of equilibrium (*ii*) free expansion (*iii*) dissipative effects (friction, heating of electric wire, fluid friction).

If final state after a succession of change is identical to the initial state of a thermodynamic system, it is called **thermodynamic cycle**. A simple cycle is described below :

Simple steam power system : This system takes care of the need of the society by converting thermal energy into mechanical or electrical energy. A simple system for the same contains four units – boiler, turbine, condenser and pump. The process starts with heating of water in the boiler, which generates steam at high pressure and temperature. External supply of heat is made to the system. The steam expands in the turbine and does useful work. Steam coming out of the turbine is at low temperature and pressure. It goes to condenser, where temperature is further lowered by extraction of heat. This is a heat sink or heat output device. After this, condensed water is pumped to increase its pressure. Water at higher pressure is sent to boiler, which heats it at constant high pressure to higher temperature. Thus cycle continues. Working media may vary also. Water, ammonia, organic fluid, mercury etc are used as working media in various systems.

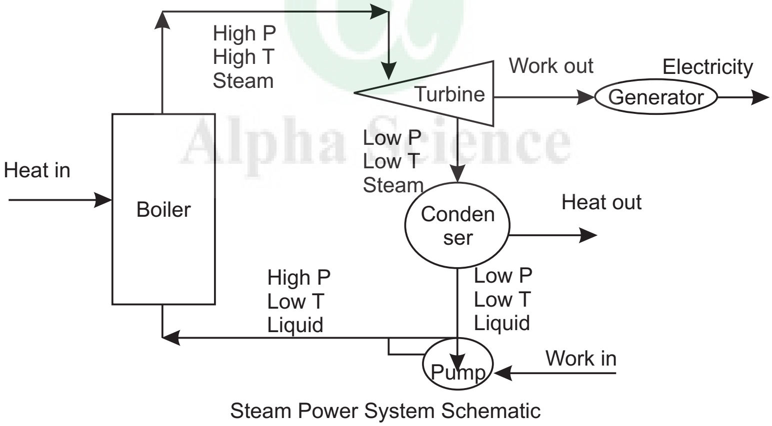


Fig. 1.3 : A Simple Thermodynamic Cycle

EXAMPLE 1.9

A simple thermodynamic cycle with air as working fluid has three processes, first is isobaric compression, followed by isochoric compression and followed by isothermal expansion to the initial state. Initial pressure, temperature and volume of the working fluid are 101.325 kPa, 300 K and 300 cc. Volume at the end of isobaric compression is 200 cc. Assume that air behaves as ideal gas with C_p *= 1.008 kJ/kg/K and* C_v *= 0.718 kJ/kg/K. Find work done in each process. Find net work done in the cycle.*

SOLUTION

Let subscripts 1, 2 and 3 indicates initial state, end of isobaric compression and end of isochoric compression, respectively.

Given, $P_1 = P_2 = 101.325$ kPa, $V_1 = 300$ cc, $T_1 = 300$ K, $V_2 = V_3 = 200$ cc

For an isothermal process, $P \times V$ = constant. So $P_1 V_1 = P_3 V_3$

So, $P_3 = P_1 V_1/V_3 = 101.325 \times 300/200 = 151.98$ kPa.

Work done in isobaric process 1-2, $W_{1\text{-}2} = P_1 (V_2 - V_1)$

$= -0.01013$ kJ

Work done in isochoric process 2-3, $W_{2\text{-}3} = 0$ kJ.

Work done in isothermal process 3-1, $W_{3\text{-}1} = P_1 V_1 \ln (V_1/V_3)$

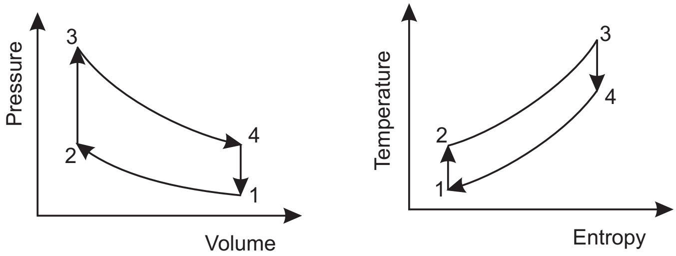
$= 0.0123$ kJ

Net work done by the working fluid in each cycle = –0.01013 + 0 + 0.0123 kJ = 0.002 kJ.

1.3 OTTO CYCLE

As described in figure 1.3, any cycle has four essential processes – heat addition, expansion work, heat rejection and compression work. During heat additional and heat rejection heat transactions takes place and invariably these processes are considered to occur as per different processes. Heat transactions results in temperature variation directly. However work interactions in real cycles are generally considered to occur in isentropic way. Temperature variations may occur during work interaction processes also.

Otto cycle was conceived in 1876 by a German engineer Nikolaus A. Otto. The cycle was based on 4 separate processes. Petrol engines work on Otto cycle. In the actual cycle, suction, compression, expansion and exhaust strokes are considered in a cycle and at the end of each cycle, fresh charge (fuel-air mixture) is taken inside the engine cylinder. So real systems never occur in a cyclic manner and working fluid does not undergo any cyclic process. But for analysis, the same can be idealized. Thermodynamic description of Otto cycle is given in figure 1.4. It has isentropic expansion and compression indicating work flow. Heat addition and heat rejection takes place in isentropic way. Process indicated by 1-2 is compression of air from pressure p_1 to pressure p_2. At the same time volume of air reduces from V_1 to V_2. No heat is added or rejected during the process. Volume of compressed air at the end of this stroke is called clearance volume (V_c). During next process indicated by 2-3, constant volume heat addition takes place. This process takes place by bringing chamber containing compressed air in contact with hot source reversibly. The pressure rises from p_2 to p_3. As there is no volume change, no work is done by the system during this process. During process 3-4, isentropic expansion of air takes place resulting in positive work output from the cycle. In this process, volume increases and pressure reduces. In this process work is done by air and there is no heat transfer. Finally, process 4-1 is constant volume heat rejection resulting in reduction of pressure. This process is executed by bringing air reversibly in contact with cold sink. This process is exactly opposite of process 2-3.



Process 1-2 : Isentropic Compression
Process 2-3 : Constant Volume Heat Addition
Process 3-4 : Isentropic Expansion
Process 4-1 : Constant Volume Heat Rejection

Fig. 1.4 : Otto Cycle on Thermodynamic Planes

Two important terms are used while describing this cycle. First term is called compression ratio defined by symbol 'r'. It is ratio of initial volume (V_1) to volume of air after compression ($V_2 = V_c$). Another term is called explosion ratio or pressure ratio and is denoted by 'α'. It is equal to ratio of pressures during constant volume heat addition process and is numerically given by p_3/p_2. Values of both 'r' and 'α' are always more than 1.

If specific heat of air at constant volume is given by C_v, and ratio of specific heats is given by γ, then

Heat added per unit mass of air during process 2-3

$$= C_v (T_3 - T_2)$$

Heat rejected per unit mass of air during process 4-1

$$= C_v (T_4 - T_1).$$

Net work done per unit mass of air during cycle

= (heat added – heat rejected) per unit mass of air

$$= C_v (T_3 - T_2) - C_v (T_4 - T_1)$$

For the isentropic processes 3-4 and 1-2,

$$T_3/T_4 = (V_4/V_3)^{\gamma-1} = (V_1/V_2)^{\gamma-1} = T_2/T_1$$

or, $$T_3/T_2 = T_4/T_1$$

Air standard efficiency for the cycle, η

$$= \text{Net work done/Heat Added}$$
$$= 1 - (T_4 - T_1)/(T_3 - T_2)$$
$$= 1 - [T_1(\{T_4/T_1\} - 1)]/[T_2(\{T_3/T_2\} - 1)]$$
$$= 1 - (T_1/T_2) = 1 - (V_1/V_2)^{1-\gamma} = 1 - \{1/r^{(\gamma-1)}\}.$$

Air standard efficiency of Otto cycle depends on compression ratio alone and is not dependent on temperature levels at which cycle is operating. Higher compression ratio gives higher air standard efficiency. However, compression ratio of petrol engines cannot be raised beyond certain levels due to abnormal combustion and is generally restricted to less than 8-10. At higher compression ratios, knocking or noisy combustion may take place, thus reducing the combustion efficiency of fuel in the system significantly. To enhance compression ratio without knocking some additives are used in fuel. However, selection of compression ratio depends on design of engine, fuel type, operating conditions and auxiliary heat sinks etc.

It is also observed that variation of air standard efficiency with compression ratio depends on value of compression ratio. At lower compression ratios, air standard efficiency changes more with a given change in compression than that at higher compression ratio. For ratio of specific heat as 1.4, changing compression ratio from 3 to 4, gives a variation of air standard efficiency from 35.56% to 42.56%. It means by changing compression ratio by 1, change in air standard efficiency is around 19.68%. For similar conditions, if compression ratio is changed from 9 to 10, air standard efficiency changes from 58.47% to 60.19%. This indicates that at higher compression ratio, same change of 1 unit in compression ratio brings a change of 2.94% in air standard efficiency. However in terms of percentage, at lower compression ratio ($r = 3$) 1 unit is equivalent to 33.33% rise in compression ratio and at higher compression ratio ($r = 9$), 1 unit rise in compression ratio is equivalent to 11.11% only. These are discrete calculations. For a continuous variation, it is observed that the percentage change in air standard efficiency is equal to a factor time's percentage change in compression ratio. The factor depends on both ratio of specific heats and compression ratio and is mathematically equal to $[(\gamma - 1)/\eta]$.

$$d\eta/\eta = [(\gamma - 1)/\eta] \times dr/r \text{ or, } d\eta/dr = [(\gamma - 1)/r\gamma].$$

For higher compression ratio value of air standard efficiency is higher and factor is lower. Same is depicted by the alternate expression also. So at higher compression ratio variation of air standard efficiency with compression ratio is less sensitive. Although for air as working fluid value of ratio of specific heats is constant and is equal to 1.4. However, effect of variation in air standard efficiency due to ratio of specific heats can also be ascertained. For monatomic gases like helium, argon, ratio of specific heats is 1.67 and for same compression ratio higher air standard efficiency is possible. For carbon dioxide and ethane, the values of ratio of specific heats are 1.3 and 1.2 respectively. For such systems lower air standard efficiency is obtained.

EXAMPLE 1.10

The bore and stroke of an engine working on Otto cycle are 20 cm and 30 cm respectively. The clearance volume is 0.001025 m^3. Calculate air standard efficiency.

SOLUTION

Refer figure 1.4

Given, Bore or diameter of cylinder, $D = 20$ cm

Stroke of piston, $L = 30$ cm

Clearance volume, $V_c = 0.001025$ m^3

Assume Ratio of specific heat for air, $\gamma = 1.4$

So, Swept volume, $V_s = (\pi/4)\, D^2$ L $= 0.00282$ m^3

Volume in the beginning of compression stroke,

$V_1 = V_s + V_c = 0.003852$ m^3

Volume at the end of compression stroke, $V_2 = V_c = 0.001025$ m^3

Compression ratio, $r = V_1/V_2 = 3.758$

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

$= 0.4111 = 41.11\%$.

EXAMPLE 1.11

Calculate compression ratio and air standard efficiency of an Otto cycle, if temperatures at the beginning and end of compression are 300 K and 500 K. Take ratio of specific heats as 1.4.

SOLUTION

Refer figure 1.4

Given,

Temperature at the beginning of compression, $T_1 = 300$ K Temperature at the end of compression, $T_2 = 500$ K

Air standard efficiency, $= 1 - T_1/T_2 = 40\%$.

For calculation of compression ratio, either efficiency can be used or isentropic compression process can be utilized.

From the calculated value, $0.4 = 1 - (1/r^{\gamma-1})$ or $r = 3.586$.

From isentropic expansion, $V_1/V_2 = (T_2/T_1)^{1/(\gamma-1)}$

$= (500/300)^{1/0.4} = 3.586$

EXAMPLE 1.12

In an ideal Otto cycle the air at the beginning of isentropic compression is at 1 kg/cm² and 15°C. The value of compression ratio is 8. If the heat added during the constant volume process is 250 kcal/kg, determine (a) the maximum temperature in the cycle, (b) the air standard efficiency, (c) the work done per kg of air, and (d) the heat rejected. Take C_v = 0.17 kcal/kg K and λ = 1.4.

SOLUTION

Refer figure 1.4

Given Pressure before compression stroke, p_1 = 1 kg/cm²

Temperature before compression stroke, T_1 = 15°C

= 288 K

Compression ratio, $r = 8$

Heat added, Q_1 = 250 kcal/kg

Specific heat at constant volume, C_v = 0.17 kcal/kg K

Ratio of specific heat, $\gamma = 1.4$

So, Air standard efficiency = $1 - (1/r^{\gamma-1})$ = 0.5647 = 56.47%. **Ans (*b*)**

Air standard efficiency = 1 – (heat rejected/heat added)

Heat rejected = {1 – (Air standard efficiency)} × heat added

= 108.8 kcal/kg **Ans (*d*)**

Work done per kg of air = heat added – heat rejected

= 250 kcal/kg – 108.8 kcal/kg

= 141.2 kcal/kg **Ans (*c*)**

Since compression stroke is isentropic, $p_2/p_1 = (V_1/V_2)^\gamma = r^\gamma$

So, pressure at the end of compression stroke, $p_2 = p_1 \times r^\gamma$ = 18.38 kg/cm²

Similarly, $T_2 = T_1 \times (V_1/V_2)^{\gamma-1}$ = 661.6 K

Heat added per stroke to the cycle, $Q_1 = C_v\,(T_3 - T_2)$

So, maximum temperature in the cycle, $T_3 = T_2 + Q_1/C_v$

= 2132.2 K

= 1859°C **Ans (*a*)**

EXAMPLE 1.13

In an ideal Otto cycle, the compression ratio is 6. The initial pressure and temperature of the air are 1 kg/cm² and 100°C. The maximum pressure in the cycle is 35 kg/cm². For 1 kg of air flow, calculate the values of the pressure, *volume and temperature at the four salient points of the cycle. What is ratio of heat supplied to heat rejected? Take C_v = 0.178 kcal/kg K and γ = 1.4.*

SOLUTION

Refer figure 1.4

Given, Compression ratio, $r = 6$

Pressure at the beginning of compression, p_1 = 1 kg/cm^2

Temperature at the end of temperature, T_1 = 100°C = 373 K

Maximum pressure in the cycle, p_3 = 35 kg/cm^2

Ratio of specific heats, $\gamma = 1.4$

Gas constant for air, R = 29.27 kgf-m/kg K

Volume at the beginning of compression, $V_1 = m \times R \times T/p_1$

$= 1.0918$ m^3.

Since 1-2 is an isentropic process with compression ratio of 6,

Volume at the end of compression, $V_2 = V_1/r = 0.18196$ m^3.

Pressure at the end of compression, $p_2 = p_1 \times r^\gamma = 12.28$ kg/cm^2.

Temperature at the end of compression, $T_2 = p_2V_2T_1/p_1V_1$

$= 763.4$ K.

Since 2-3 is a constant volume process,

Volume at the end of heat addition, $V_3 = V_2 = 0.18196$ m^3.

Temperature at the end heat addition, $T_3 = p_3\ T_2/p_2$

$= 2175.8$ K.

Volume at the end of expansion stroke, $V_4 = V_1 = 1.0918$ m^3.

Since 3-4 is an isentropic process

Pressure at the end of expansion stroke, $p_4 = p_3/r^\gamma$

$= 2.848$ kg/cm^2.

Since 4-1 is a constant volume process

Temperature at the end of expansion stroke, $T_4 = T_1 \times (p_4/p_1)$

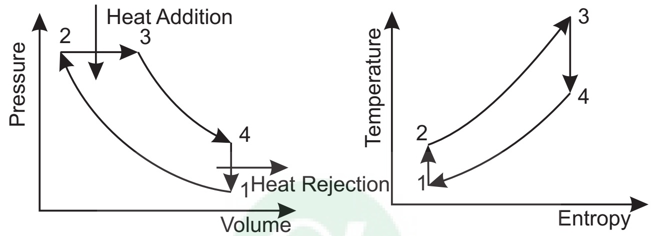
$= 1062.3$ K

Ratio of heat supplied to heat rejected $= (T_3 - T_2)/(T_4 - T_1)$

$= 2.049$.

1.4 DIESEL CYCLE

Original Diesel engine was invented by Rudolf Christian Karl Diesel (1958-1913), France in 1892. Diesel engines work on Diesel cycle. It differs from Otto cycle in only one respect. Heat addition is done at constant pressure instead of constant volume. In Otto cycle higher compression ratio gives higher air standard efficiency. Since in Otto cycle, both fuel and air mixture are introduced in the engine, obtaining higher air standard efficiency by introducing working fluid at higher compression ratio is restricted by abnormal combustion phenomena. At higher compression ratio, knocking may occur in the engine and air standard efficiency is restricted by this phenomena. In diesel cycle, only air is compressed rather than a mixture of air and fuel. The compression is carried to such an extent that auto ignition temperature of fuel is attained in the cylinder. Fuel is introduced in the engine in the compressed high temperature air. Fuel ignites instantaneously without any external spark plug or initiation device. This eliminates chances of knocking also.



Process 1-2 : Isentropic Compression
Process 2-3 : Constant Pressure Heat Addition
Process 3-4 : Isentropic Expansion
Process 4-1 : Constant Volume Heat Rejection

Fig. 1.5 : Diesel Cycle on Thermodynamic Planes

In practical systems, adding heat at constant volume is difficult. Thermodynamic description of Diesel cycle is given in figure 1.5. Process indicated by 1-2 is compression of air from pressure p_1 to pressure p_2. At the same time volume of air reduces from V_1 to V_2. No heat is added or rejected during the process. Volume of compressed air at the end of this stroke is called clearance volume (V_c). During next process indicated by 2-3, constant pressure heat addition takes place. In this process, the pressure is constant but volume rises from V_2 to V_3. During process 3-4, isentropic expansion of air takes place resulting in positive work output from the cycle. In this process, volume increases and pressure reduces. In this process work is done by air and there is no heat transfer. Finally, process 4-1 is constant volume heat rejection resulting in reduction of pressure.

In addition to various terms defined for Otto cycle, one more term is predominant in calculation for Diesel cycle. This term arises due to isobaric or constant pressure heating process, where volume changes from V_2 to V_3. Ratio of these two volumes is called cut-off ratio and is represented by symbol r_c (= V_3/V_2).

Heat addition in this case is given by the isobaric process 2-3.

Heat addition per cycle (process 2-3), $Q_1 = C_p\,(T_3 - T_2)$

Heat rejection per cycle (process 4-1), $Q_2 = C_v\,(T_4 - T_1)$

Net work done per unit mass of air during cycle

$$= \text{(heat added – heat rejected) per unit mass of air}$$

$$= C_p\,(T_3 - T_2) - C_v\,(T_4 - T_1)$$

For the isentropic processes 1-2,

$$T_1/T_2 = (V_2/V_1)^{\gamma-1} = (1/r)^{\gamma-1}$$

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

For isobaric process, 2-3, $T_3/T_2 = V_3/V_2 = r_c$

$$T_3 = T_1 \times (T_3/T_2) \times (T_2/T_1)$$

$$= T_1 \times r_c \times (r)^{\gamma-1}$$

For the isentropic processes 3-4,

$$T_4/T_3 = (V_3/V_4)^{\gamma-1} = (V_3/V_1)^{\gamma-1} = (r_c/r)^{\gamma-1}$$

$$T_4 = (T_4/T_3) \times T_3 = T_1 \times r_c^{\gamma}$$

Air standard efficiency for the cycle,

$$= \text{Net work Done/Heat Added}$$

$$= 1 - C_v\,(T_4 - T_1)/C_p\,(T_3 - T_2)$$

$$= 1 - (1/\gamma) \times [r_c^{\gamma} - 1] \,/\, [(r)^{\gamma-1} \times (r_c - 1)$$

Air standard efficiency of Diesel cycle is given by

$$\boldsymbol{\eta = 1 - \left\{\left(r_c^{\gamma} - 1\right)/\left[\gamma . r^{\gamma} - 1 . \left(r_c - 1\right)\right]\right\}.}$$

From the expression, it is clear that air standard efficiency of a diesel cycle depends on three parameters, namely compression ratio, cut-off ratio and ratio of specific heats. The effect of these parameters is depicted in figure 1.6. It is clear that as compression ratio increases air standard efficiency rises. This trend is similar to trend of air standard efficiency of an Otto cycle. As ratio of specific heat lowers, value of air standard efficiency also reduces. This trend is also similar to that for an Otto cycle. However, for diesel cycle, dependence on ratio of specific heat is on higher side. The additional parameter called cut-off ratio affects air standard efficiency of a diesel cycle significantly. As cut-off ratio increases, air standard efficiency reduces. So for higher air standard efficiency, higher value of compression ratio and ratio of specific heat is desirable, but value of cut-off ratio should be as small as possible. For a practical system, this indicates a quicker fuel injection.

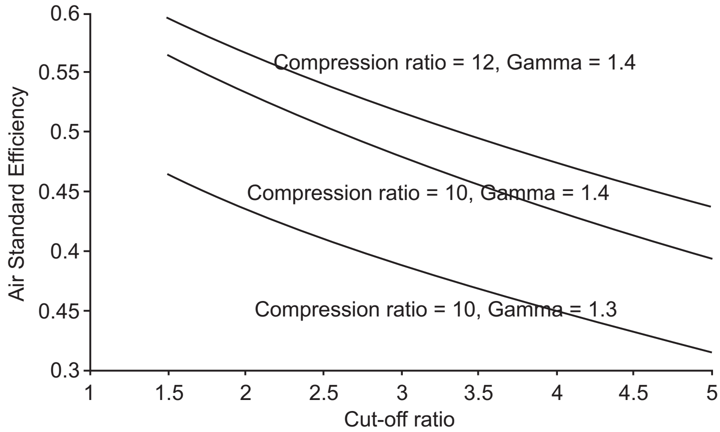


Fig. 1.6 : Variation of Air Standard Efficiency of Diesel Cycle

EXAMPLE 1.14

An air-standard Diesel cycle has a compression ratio of 14. The pressure at the beginning of the compression stroke is 1 kg/cm² and the temperature is 27°C. The maximum temperature is 2500°C. Determine the thermal efficiency and the mean effective pressure. Take C_p = 0.24 kcal/kg K, C_v = 0.17 kcal/kg K, γ = 1.4.

SOLUTION

Refer figure 1.5

Given, Compression ratio, $r = 14$

Pressure in the beginning of compression, $p_1 = 1$ kg/cm²

Temperature in the beginning of compression, $T_1 = 27°C$

$= 300$ K

Temperature at the end of heat addition, $T_3 = 2500°C = 2773$ K

For the isentropic process 1-2,

Pressure at the end of compression, $p_2 = p_1 \times r^{\gamma} = 40.23$ kg/cm².

Temperature at the end of compression, $T_2 = T_1 \times r^{\gamma-1} = 862.13$ K

Cut-off ratio, $r_c = V_3/V_2 = T_3/T_2 = 3.216$.

For the isentropic process 3-4

Temperature at the expansion, $T_4 = T_3 \times (V_3/V_4)^{\gamma-1}$

$= T_3 \times ([V_3/V_2] \times ([V_2/V_1])^{\gamma-1}$

$= T_3 \times (r_c/r)^{\gamma-1} = 1539.6$ K.

Heat added = $C_p\,(T_3 - T_2)$ = 458.6 kcal/kg.

Heat rejected = $C_v\,(T_4 - T_1)$ = 210.7 kcal/kg.

Thermal efficiency = 1 – (Heat rejected/Heat added)

= 0.5404 = 54.04%.

Gas constant for air, $R = C_p - C_v$ = 0.07 kcal/kg K

= 29.89 kgfm/kg K.

Volume at the beginning of compression, $V_1 = R \times T_1/p_1$

= 0.894 m^3/kg.

Stroke volume = $V_1 - V_2 = V_1\,(1 - 1/r)$ = 0.83 m^3/kg.

Mean effective pressure = net work per cycle/stroke volume

= (heat added – heat rejected)/stroke volume

= 12.75 kg/cm^2.

EXAMPLE 1.15

An ideal diesel engine has a diameter 10 cm and stroke 15 cm.the clearance volume 10% of the swept volume. Determine the compression ratio and the air standard efficiency of the engine, if cut-off takes place at 8% of the stroke. Take γ = 1.4.

SOLUTION

Refer figure 1.5

Given, $V_2 = 0.1 \times (V_1 - V_2)$.

or, Compression ratio, $r = (V_1/V_2) = 11$

Bore of the cylinder, d = 10 cm

Stroke of the engine, L = 15 cm

$V_3 - V_2 = 0.08 \times (V_1 - V_2)$ or cut-off ratio, $r_c = V_3/V_2 = 1.8$

Compression ratio, $r = (V_1/V_2) = 11$.

Air standard efficiency of Diesel cycle is given by

$$\eta = 1 - \{(r_c^{\gamma} - 1)/[\gamma . r^{\gamma-1} . (r_c - 1)]\} = 56.3\%.$$

EXAMPLE 1.16

An oil engine works on the ideal diesel cycle. The overall compression ratio is 16:1 and constant pressure energy addition ceases at 10% of the stroke. Intake conditions are 1 kg/cm^2 and 25°C. The engine uses 100 m^3 of air per hour. If γ = 1.4, determine (a) the maximum temperature and pressure in the cycle (b) the thermal efficiency of the engine.

SOLUTION

Refer figure 1.5

Given Compression ratio, $r = 16$

Pressure at the beginning of compression, $p_1 = 1$ kg/cm^2

Temperature at the beginning of compression, $T_1 = 25°C = 298$ K

Swept volume, $V_1 - V_2 = 100$ m^3

Volume change during constant pressure energy addition,

$V_3 - V_2 = 0.1\ (V_1 - V_2)$ or Cut-off ratio, $r_c = V_3/V_2 = 2.5$

Maximum pressure occurs at point '2' or point '3'. It can be calculated from isentropic compression formula with the help of pressure at '1'.

$$p_2 = p_3 = p_1 \times r^{\gamma} = 48.5 \text{ kg/cm}^2.$$

$$T_2 = T_1 \times r^{\gamma-1} = 903.3 \text{ K}$$

So, temperature at the end of constant pressure heat addition process can be calculated from temperature at the beginning of constant pressure heat addition.

$$T_3 = T_2 \times r_c = 2258.4 \text{ K}$$

Air standard efficiency of Diesel cycle is given by

$$\eta = 1 - \{(r_c^{\gamma} - 1) / [\gamma.\ r^{\gamma-1}.(r_c - 1)]\} = 59.1\%.$$

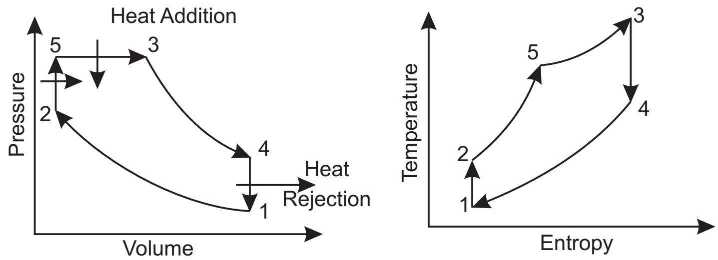
1.5 DUAL CYCLE

Diesel cycle efficiency increases as cut-off ratio, r_c decreases. So, it is desirable to keep the value of cut-off ratio, as small as possible to increase the efficiency. However, practically, it is very difficult to introduce fuel completely in a very short duration. So, fuel injection is partially done at constant pressure and partially at constant pressure. This is origin of dual cycle. This is possible in dual cycle by the combustion of fuel taking place partly at constant volume and partly at constant pressure. So it is also referred as mixed cycle. A typical dual cycle is represented on thermodynamic planes in figure 1.7.

For dual cycle, compression ratio, r is given by V_1/V_2. Pressure ratio, α during constant volume heat addition is given by p_5/p_2. Cut-off ratio (r_c) is given by ratio of volumes during constant pressure heat addition and is numerically given by V_3/V_5. In addition to already defined terms, one more term is added. This term is called expansion ratio and is represented by r_c. It is numerically calculated as V_4/V_3. Air standard efficiency of dual cycle is given by following expression.

Heat addition in this case is given by the isobaric process 2-3.

Heat addition per cycle (process 2-5-3), Q_1



Process 1-2 : Isentropic Compression

Process 2-5 : Constant Volume Heat Addition

Process 5-3 : Constant Pressure Heat Addition

Process 3-4 : Isentropic Expansion

Process 4-1 : Constant Volume Heat Rejection

Fig. 1.7 : Dual Cycle on Thermodynamic Planes

$= C_p (T_3 - T_5) + C_v (T_5 - T_2)$

Heat rejection per cycle (process 4-1), $Q_2 = C_v (T_4 - T_1)$

Net work done per unit mass of air during cycle

= (heat added – heat rejected) per unit mass of air

$= C_p (T_3 - T_5) + C_v (T_5 - T_2) - C_v (T_4 - T_1)$

For the isentropic processes 1-2,

$T_1/T_2 = (V_2/V_1)^{\gamma-1} = (1/r)^{\gamma-1}$

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

For isochoric process, 2-5, $T_5/T_2 = P_5/P_2 = \alpha$

$T_5 = T_1 \times (T_5/T_2) \times (T_2/T_1) = T_1 \times \alpha \times (r)^{\gamma-1}$

For isobaric process, 2-5, $T_3/T_5 = V_3/V_5 = r_c$

$T_3 = T_5 \times (T_3/T_5) = T_1 \times \alpha \times r_c \times (r)^{\gamma-1}$

For the isentropic processes 3-4,

$T_4/T_3 = (V_3/V_4)^{\gamma-1} = [(V_3/V_5)\ (V_5/V_2)\ (V_2/V_1)(V_1/V_4)]^{\gamma-1}$

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

$T_4 = (T_4/T_3) \times T_3 = T_1 \times \alpha \times r_c^{\gamma}$

Air standard efficiency for the cycle, η

= Net work Done/Heat Added

$= 1 - C_v (T_4 - T_1)/[C_p (T_3 - T_5) + C_v (T_5 - T_2)]$

$$\boldsymbol{\eta = 1 - [(\alpha . r_c^{\gamma} - 1) / [r(\gamma - 1).\{(\alpha - 1) + \gamma.\alpha.(r_c - 1)\}]]}$$

The dependence of air standard efficiency on already discussed parameters in the context of Otto and Diesel cycle has similar effects. Higher value of compression ratio and ratio of specific heats and lower value of cutoff ratio is desirable. However, the pressure ratio is one extra term added in the expression of air standard efficiency of a dual cycle. Variation of air standard efficiency with pressure ratio is plotted as figure 1.8. It is observed that higher value of pressure ratio gives higher air standard efficiency and rate of rise reduces at higher values of pressure ratio. Practically higher pressure ratio means more fuel addition at constant volume and it enhances mean temperature of heat addition also. This leads to higher value of air standard efficiency. However, the variation in air standard efficiency is very small numerically.

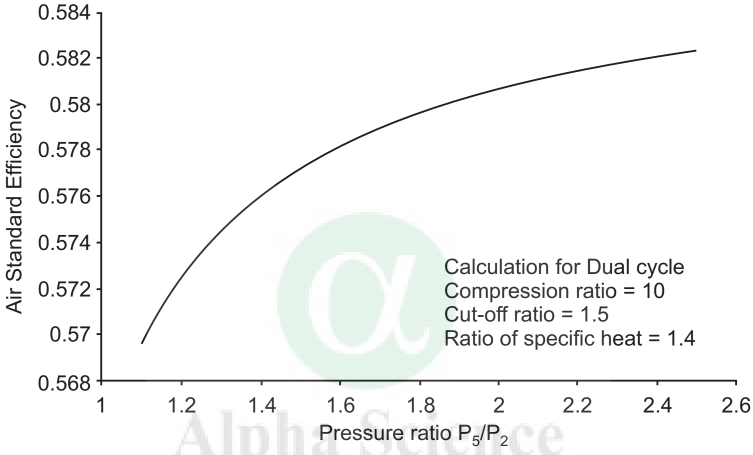


Fig. 1.8 : Variation of Air Standard Efficiency with Pressure Ratio of Heat Addition

If this expression is realized, efficiencies of both Otto and Diesel cycle can be obtained. To make dual cycle behave as Otto cycle, point '5' should coincide point '3'. This indicates that cut-off ratio, r_c is equal to 1. Similarly to get Diesel cycle from dual cycle, point '5' and point '2' should coincide. This means pressure ratio, α should be equal to 1.

EXAMPLE 1.17

A dual combustion cycle has an adiabatic compression volume ratio of 15:1. The conditions at the commencement of compression are 1 kg/cm^2, 25°C and 0.15 m^3. The maximum pressure of the cycle is 60 kg/cm^2 and the maximum temperature of the cycle is 1500°C. If $\gamma = 1.4$, calculate the pressure, volume and temperature at the corners of the cycle and the thermal efficiency of the cycle.

SOLUTION

Refer figure 1.7.

Given Compression ratio, $r = 15$

Initial conditions, $p_1 = 1$ kg/cm^2,

$V_1 = V_4 = 0.15$ m^3,

$T_1 = 25°C = 298$ K.

Maximum pressure, $p_3 = p_5 = 60$ kg/cm^2

Maximum temperature in the cycle, $T_3 = 1500°C = 1773$ K

Volume at the end of compression, $V_2 = V_5 = V_1/r = 0.01$ m^3.

Pressure at the end of compression, $p_2 = p_1 \times r^\gamma = 44.3$ kg/cm^2.

Temperature at the end of compression, $T_2 = T_1 \times r^{\gamma-1}$

$= 880.3$ K.

Temperature at the end of constant volume heat addition, $T_5 = T_2 \times p_5/p_2 = 1192.3$ K.

Volume at the end of constant pressure heat addition, $V_3 = V_5 \times T_3/T_5 = 0.01487$ m^3.

Pressure at the end of expansion, $p_4 = p_3\ (V_3/V_4)\gamma = 2.36$ kg/cm^2.

Temperature at the end of expansion, $T_4 = T_1 \times p_4/p_1$

$= 703.28$ K.

Thermal efficiency of the cycle = work output/Heat addition

= 1 – (heat rejected/heat added)

$= 1 - (C_v\,[T_4 - T_1]/\{C_v\,[T_5 - T_2] + C_p\,[T_3 - T_5]\}).$

$= 1 - ([T_4 - T_1]/\{[T_5 - T_2] + \gamma\,[T_3 - T_5]).$

= 74 %.

Pressure, volume and temperature at salient points of the dual cycle are tabulated below.

Point	*Pressure(kg/cm^2)*	*Volume(m^3)*	*Temperature (K)*
1	1	0.15	298
2	44.3	0.01	880.3
3	60	0.01487	1773
4	2.36	0.15	703.28
5	60	0.01	1192.3

■ ■ EXAMPLE 1.18

A dual combustion cycle has an adiabatic compression volume ratio of 15:1. The conditions at the commencement of compression are 1 kg/cm^2, 25°C and

0.15 m³. The maximum pressure of the cycle is 60 kg/cm² and heat transferred at constant pressure is same as heat transferred at constant volume. If $\gamma = 1.4$, calculate the pressure, volume and temperature at the cardinal points of the cycle and the thermal efficiency of the cycle.

SOLUTION

Refer figure 1.7.

Given Compression ratio, $r = 15$

Initial conditions, $p_1 = 1$ kg/cm²,

$V_1 = V_4 = 0.15$ m³,

$T_1 = 25°C = 298$ K.

Maximum pressure, $p_3 = p_5 = 60$ kg/cm²

$C_v (T_5 - T_2) = C_p (T_3 - T_5)$ or $T_3 = T_5 (1/\gamma + 1) - T_2/\gamma$

Volume at the end of compression, $V_2 = V_5 = V_1/r = 0.01$ m³.

Pressure at the end of compression, $p_2 = p_1 \times r^\gamma = 44.3$ kg/cm².

Temperature at the end of compression, $T_2 = T_1 \times r^{\gamma-1}$

$= 880.3$ K.

Temperature at the end of constant volume heat addition, $T_5 = T_2 \times p_5/p_2 = 1192.3$ K.

Maximum temperature, $T_3 = T_5 (1/\gamma + 1) - T_2/\gamma = 1415.15$ K

Volume at the end of constant pressure heat addition, $V_3 = V_5 \times T_3/T_5 = 0.01187$ m³.

Pressure at the end of expansion,

$$p_4 = p_3 (V_3/V_4)^\gamma$$
$$= 1.721 \text{ kg/cm}^2.$$

Temperature at the end of expansion,

$$T_4 = T_1 \times p_4/p_1$$
$$= 512.89 \text{ K}.$$

Thermal efficiency of the cycle = work output/Heat addition

$= 1 -$ (heat rejected/heat added)

$= 1 - (C_v [T_4 - T_1]/\{C_v [T_5 - T_2] + C_p [T_3 - T_5]\})$.

$= 1 - ([T_4 - T_1]/\{[T_5 - T_2] + \gamma [T_3 - T_5]\})$.

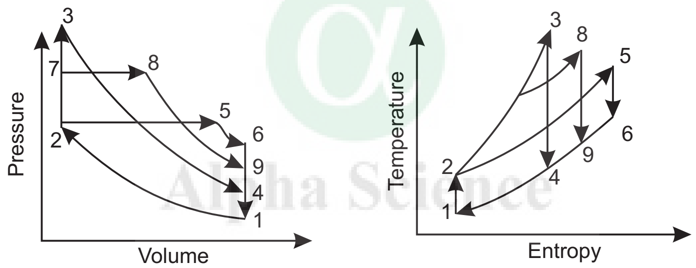
$= 65.56$ %.

Pressure, volume and temperature at salient points of the dual cycle are tabulated below.

Point	*Pressure(kg/cm^2)*	*Volume(m^3)*	*Temperature (K)*
1	1	0.15	298
2	44.3	0.01	880.3
3	60	0.01187	1415.15
4	1.721	0.15	512.89
5	60	0.01	1192.3

1.6 COMPARISON OF CYCLES

Otto cycle and Diesel cycles differ in the process of heat addition. In Otto cycle, heat addition occurs at constant volume, while in Diesel cycle, it occurs at constant pressure. Dual cycle lies in between both, as heat addition occurs partially at constant pressure and partially at constant volume. If compression ratio and heat addition is kept constant, all three cycles are depicted in figure 1.9.



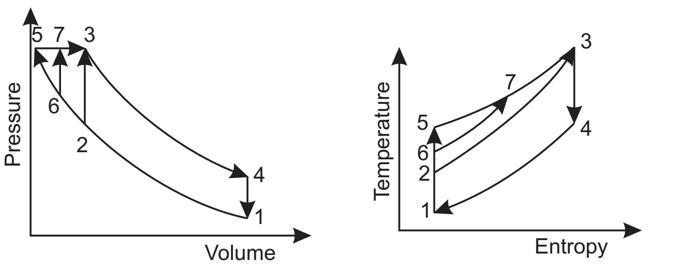
Cycle 1-2-3-4 : Otto Cycle
Cycle 1-2-5-6 : Diesel Cycle
Cycle 1-2-7-8-9 : Dual Cycle

1-2 : Isentropic Compression
2-3, 2-7 : Isobaric Heat Addition
2-5, 7-8 : Isochoric Heat Addition
3-4, 5-6, 8-9 : Isentropic Expansion
4-1, 6-1, 9-1 : Isochoric Heat Rejection

Fig. 1.9 : Comparisons of Cycles on Thermodynamic Planes

Same compression ratio is depicted by common isentropic compression depicted by process 1-2. Since heat addition is same in all the processes, comparison of Otto cycle and Diesel cycle can give a trend directly and performance of dual cycle will always lie in between the two. Heat addition in Otto cycle is given by expression $C_v\,(T_3 - T_2)$, where as heat addition in Diesel cycle is given by $C_p\,(T_5 - T_2)$. Both these quantities should be equal. Since for air as ideal gas, C_p is always greater than C_v, temperature difference in Otto cycle is higher than that in Diesel cycle. It clearly indicates that temperature after heat addition is higher in Otto cycle. Temperature after heat addition in dual cycle lies in between Otto and Diesel cycles. The same is depicted in temperature-entropy plot of figure 1.9. Since heat rejection in all the cases occur at constant volume, Otto cycle rejects least heat, while heat rejection in diesel cycle is the highest. For same heat input, higher heat rejection means lower work output from the cycle. Since efficiency is ratio of work output to heat addition in a cycle, Otto cycle is most efficient and Diesel cycle is least efficient for same compression ratio and same heat input. Actual comparison is dependent on control parameters and other cases are to be realized also. It must be noted that compression ratio of Otto cycle is always smaller than that of Diesel cycle. Since higher compression ratio in Otto cycle results in knocking, diesel cycle is designed to get higher compression ratio. Typically, Otto cycle operates around compression ratio of 4 to 8, where as Diesel cycle operating at compression ratio of 16 is also possible.

For comparison of other situations, if maximum temperature and maximum pressure realized in the cycle is kept same, then representation of various cycles on thermodynamic plane changes. The change is depicted in figure 1.10. Since peak pressure is maintained same, compression ratio of Otto cycle will be smaller than other two cycles and Diesel cycle will have highest compression ratio. Since peak temperature is also maintained same, expansion and subsequent heat rejection process is invariant for the type of cycles chosen. The given condition refers to same heat rejection. Close scrutiny of figure 1.10 reveals that area in the cycle depicted by Otto cycle is the least and the same for the Diesel cycle is highest. This indicates that for same heat rejection Otto cycle has lower work output. So Otto cycle will be least efficient.



Cycle 1-2-3-4 : Otto Cycle
Cycle 1-5-3-4 : Diesel Cycle
Cycle 1-6-7-3-4 : Dual Cycle

1-2, 1-6, 1-5 : Isentropic Compression
5-3, 7-3 : Isobaric Heat Addition
2-3, 6-7 : Isochoric Heat Addition
3-4 : Isentropic Expansion
4-1 : Isochoric Heat Rejection

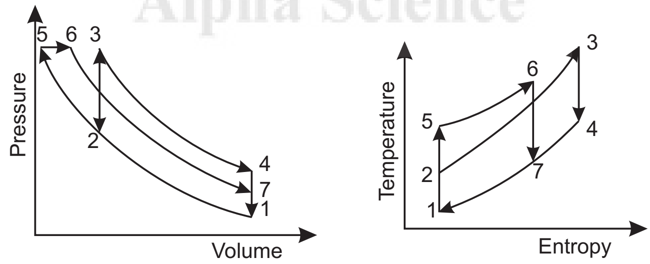
Fig. 1.10 : Comparisons for Same Maximum Temperature and Maximum Pressure

If W = work output, Q_1 = heat addition in the cycle and Q_2 = heat rejection in the cycle, then $Q_1 = W + Q_2$.

Air standard efficiency for any cycle = $W/Q_1 = W/(W + Q_2) = 1/(1 + Q_2/\mathrm{W})$.

If work output (W) is lower for same heat rejection (Q_2), as is the case with Otto cycle, (Q_2/W) will larger and ultimately air standard efficiency will be lower.

Another situation may be depicted by same maximum pressure and same work output. Since representing all three cycles on single thermodynamic plane becomes cluttered, only Otto cycle and diesel cycle are represented in figure 1.11. Performance of Dual cycle will lie in between performance of the two cycles. Since work output is same and maximum pressure is also restricted, area occupied by representation of both the cycle on thermodynamic plane should be the same. Otto cycle will have lower compression ratio so as to limit the maximum pressure. Contrary to this diesel cycle will have higher compression ratio. However, temperature realized in the Diesel cycle will be on lower side so as to maintain work output. It is clear from the given figure 1.11 that in case of diesel cycle, heat rejection is less. This leads to same conclusion as earlier that efficiency will be higher, if heat rejection is higher for same work output. For same maximum pressure and work output, diesel cycle is more efficient.



Cycle 1-2-3-4 : Otto Cycle
Cycle 1-5-6-7 : Diesel Cycle

1-2, 1-5 : Isentropic Compression
5-6 : Isobaric Heat Addition
2-3 : Isochoric Heat Addition
3-4, 6-7 : Isentropic Expansion
4-1, 7-1 : Isochoric Heat Rejection

Fig. 1.11 : Comparisons for Same Maximum Pressure and Work

Depending on various situations, efficiency of various cycles can be compared and the cycles behave differently for different restrictions. Practically Diesel cycle is more efficient than Otto cycle, because of higher compression ratio. Dual cycle is a compromise of efficiency to include isochoric heat addition also to the Diesel cycle.

EXAMPLE 1.19

Compare cycle efficiency of Diesel cycle and Dual cycle with compression ratio and cut-off ratio of 15 and 1.4 respectively. Pressure ratio for dual cycle is 1.2. Assume $\gamma = 1.4$.

SOLUTION

Given, Compression ratio, $r = 15$

Cut-off ratio, $r_c = 1.4$

Pressure ratio, $\alpha = 1.2$

Air standard efficiency of Diesel cycle is given by

$$\eta = 1 - \{(r_c^{\gamma} - 1)/[\gamma.r^{\gamma-1}.(r_c - 1)]\} = 63.63\%$$

Air standard efficiency of Dual cycle is given by

$$\eta = 1 - [(\alpha.r_c^{\gamma} - 1)/[r^{\gamma-1}.\{(\alpha - 1) + \gamma.\alpha.(r_c - 1)\}]] = 64.2\%.$$

Although efficiency of Otto cycle is not asked in this problem, but from the trend, it is clear that for same compression ratio (= 15) Otto cycle will have air standard efficiency higher than both the cycles. Since for Otto cycle it depends only on the compression ratio, air standard efficiency is = 1 – 1/150.4 = 66.15%.

EXAMPLE 1.20

Compare air standard efficiency of Otto and Diesel cycle, if air is compressed from a pressure and temperature of 0.1 MPa and 300 K. Maximum pressure and maximum temperatures are restricted to 7 MPa and 1800 K. Assume $\gamma = 1.4$ and $C_v = 0.718$ kJ/kg/K.

SOLUTION

Refer figure 1.4 and figure 1.5.

Given Initial pressure, $p_1 = 0.1$ MPa

Initial temperature, $T_1 = 300$ K

Maximum temperature, $T_3 = 1800$ K

For the Otto cycle (figure 1.4)

Maximum pressure, $p_3 = 7$ MPa

$$T_2 = T_1\,(p_2/p_1)^{(\gamma-1/\gamma)} = (p_2/p_3) \times T_3$$

$$p_2 = 3.117 \text{ MPa}$$

$$T_2 = 801.49 \text{ K}.$$

Compression ratio, $r = (p_2/p_1)^{1/\gamma} = 11.667$

Air standard efficiency $= 1 - 1/r^{\gamma-1} = 62.57\%$

For the Diesel cycle (figure 1.5)

Maximum pressure, $p_2 = p_3 = 7$ MPa

$T_2 = T_1\ (p_2/p_1)^{(\gamma-1/\gamma)} = 1009.9$ K

Compression ratio, $r = (T_2/T_1)^{1/(\gamma - 1)} = 20.792$

Cut-off ratio, $r_c = (T_3/T_2) = 1.78235$

Air standard efficiency = $1 - \{(r_c^{\gamma}-1)/[\gamma.r^{\gamma-1}.(r_c - 1)]\} = 66.21\%$.

1.7 BRAKE THERMAL EFFICIENCY

The engine performance is indicated by efficiency. Thermal efficiency of an engine is important, since it determines how efficiently fuel is being used in the engine. All power in reciprocating engines is derived from chemical energy of the fuel. Calorific value of fuel indicates energy available per unit mass of the fuel. This is total chemical energy available in the form of heat with the engine. This whole energy cannot be utilized for driving the piston alone. Some part of energy is always lost to the exhaust, coolant and radiator as heat energy. The remaining energy is converted to power called indicated power (ip) derived from the engine. Sometimes power is expressed in horse power and is called indicated horsepower (ihp). This energy is passed through connecting rod to crankshaft. During this process, there are certain transmission losses due to friction and pumping. These losses are called friction power (fp). The remaining energy is useful mechanical energy termed as brake power (bp).

Ratio of indicated power to energy available from the fuel is termed as indicated thermal efficiency.

Indicated thermal efficiency = ip/energy supplied by the fuel.

Similarly brake thermal efficiency is defined as ratio of bp and energy available with the fuel.

Brake thermal efficiency = bp/energy supplied by the fuel.

1.8 MECHANICAL EFFICIENCY

Energy available with the fuel is in terms of heat. This heat is converted to work at piston and is expressed as indicated horse power (ihp). From this work available at the crankshaft is called brake horse power (bhp). The lost work in this transmission is called friction horse power (fhp = ihp – bhp). This part may be called delivered power. Mechanical efficiency of an engine is defined as ratio of bhp to ihp.

Mechanical efficiency = bhp/ihp = bhp/(bhp + fhp)
= 1 – (fhp/ihp).

It can also be expressed as ratio of brake thermal efficiency and indicated thermal efficiency.

1.9 OVERALL EFFICIENCY

Ratio of thermal efficiency of an actual cycle to that of the ideal cycle is called relative efficiency or efficiency ratio. It is an important criterion for the degree of development of the engine. At each stage of energy conversion, there are some losses in the engines. First of all, fuel conversion losses are there and fuel delivers less heat than ideally calculated based on mass of fuel their calorific values. This ratio is called fuel conversion efficiency. Overall efficiency takes into account all losses that takes place in an engine and is ratio of ideal heat possible by combustion of fuel to final work output. For best performance an engine should have high fuel conversion efficiency, high thermal efficiency, high piston speed and high volumetric efficiency.

1.10 VOLUMETRIC EFFICIENCY

Fuel exhibits combustion for limited value of fuel-air ratio. So quantity of fuel that can be introduced inside cylinder of a reciprocating engine is governed by the quantity of air intake possible during suction stroke. Volumetric efficiency indicates the breathing ability of the engine and is defined as the ratio of air actually inducted at ambient conditions to the swept volume of the engine. Swept volume is nothing but the product of cross-sectional area of cylinder and stroke length of the piston. Volumetric efficiency is calculated using mass or volume of air.

$$\text{Volumetric efficiency} = \frac{\text{Mass or volume of air actually indicated}}{\text{Mass or volume of air at intake temperature and pressure}}.$$

1.11 TORQUE AND MEAN EFFECTIVE PRESSURE

All reciprocating engines are designed to deliver power for rotation of the crack-shafts. In an engine, crack-shaft is powered at a rotational or angular speed. The rotational speed is generally expressed as rpm (revolution per minute). Torque derived from the engine is defined as power developed divided by angular speed of the engine. Torque is ability of the engine to do wok and power is the rate at which work is done.

Mean effective pressure (mep) is a measure of average pressure in the entire cycle. As explained earlier, work done in any cyclic process is given by area enclosed on pressure-volume plot for the cycle. In the cyclic process, pressure is generally variable. Mean effective pressure is defined as imaginary constant pressure which, if exerted on the piston during one stroke would develop the same network for the cycle as was obtained with variable pressure. It is a useful criterion for the comparison of relative size of reciprocating engines. It is also an indicator of how far actual engine efficiency will depart from the cycle efficiency.

Mean effective pressure,
mep = work done per cycle/stroke volume.

For Otto cycle, Mean effective pressure, mep

= Work done per cycle/stroke volume

$= (p_3V_3 - p_4V_4 - p_2V_2 + p_1V_1)/[(V_1 - V_2)(\gamma - 1)]$

$= p_1.r\ (\alpha - 1).(r^{\gamma-1} - 1)/[(\gamma - 1).(r - 1)]$.

1.12 SPECIFIC FUEL CONSUMPTIONS

The fuel consumptions are characteristics of an engine and are generally expressed in terms of specific fuel consumption (sfc). It is equal to mass flow rate of fuel per unit power developed by the engine. It is expressed in gram per horse power (hp) hour or kilo watt hour (kWhr). It is an important characteristic to compare performance of two engines or comparing the performance of the same engine at two loads. If brake power is considered, the parameter is called brake specific fuel consumption (bsfc). If indicated power is used it is called indicated specific fuel consumption (isfc).

Brake specific fuel consumption (bsfc) = fuel flow rate/ brake power
Indicated specific fuel consumption (bsfc) = fuel flow rate/ indicated power.

EXAMPLE 1.21

Find theoretical mean effective pressure and work of an ideal Otto cycle, which has a compression ratio of 6, clearance volume of 200 cc and pressure ratio during constant volume heat addition is 4. Assume initial pressure as 0.1 MPa and $\gamma = 1.4$.

SOLUTION

Refer figure 1.4

Given Compression ratio, $r = 6 = V_1/V_2$

Clearance volume, $V_2 = V_3$ =200 cc

Initial conditions, $p_1 = 0.1$ MPa,

Pressure ratio, $p_3/p_2 = 4$

Initial volume, $V_1 = V_4 = 6 \times 200$ cc = 1200 cc

Pressure at the end of compression, $p_2 = p_1 \times r^{\gamma} = 1.23$ MPa

Pressure at the end of heat addition, $p_3 = 4 \times p_2 = 4.91$ MPa

Pressure at the end of expansion, $p_4 = p_3/r^\gamma = 0.4$ MPa

Work done per cycle, $W = (p_1V_1 - p_2V_2 + p_3V_3 - p_4V_4)/(\gamma - 1)$

$= (0.1 \times 6 - 1.23 + 4.91 - 0.4 \times 6) \times 200/(1.4 - 1)$

$= 940$ J/cycle

Mean effective pressure $= W/(V_1 - V_2) = 940/(1000)$

$= 0.94$ MPa.

▮▮ EXAMPLE 1.22

Find mean effective pressure of an ideal Otto cycle for compression ratio of 6 with maximum and minimum pressure as 5 MPa and 0.1 MPa. Assume $\gamma = 1.4$.

SOLUTION

Given Compression ratio, $r = 6 = V_1/V_2$

Initial conditions, $p_1 = 0.1$ MPa.

Maximum Pressure, $p_3 = 5$ MPa.

Pressure at end of compression, $p_2 = 0.1 \times 61.4 = 1.23$ MPa

Pressure ratio, $\alpha = p_3/p_2 = 4.07$.

Mean effective pressure of an Otto cycle $= p_1.r\,(\alpha - 1).(r^{\gamma-1} - 1)/[(\gamma - 1).(r - 1)] = 0.9649$ MPa.

▮▮ EXAMPLE 1.23

For a dual combustion cycle, compression ratio is 12 and cut-off ratio is 1.615. Maximum pressure is 5.4 MPa. Temperature and pressure of air at inlet are 335 K and 0.1 MPa, then find (a) mean effective pressure (b) pressure and temperature at all the cardinal points (c) cycle efficiency. Assume $\gamma = 1.35$.

SOLUTION

Refer figure 1.7

Given Compression ratio, $r = 12 = V_1/V_2$

Cut-off ratio, $r_c = 1.615 = V_3/V_5$

Initial conditions, $p_1 = 0.1$ MPa, $T_1 = 335$ K.

Maximum pressure, $p_3 = p_5 = 5.4$ MPa

Ratio of specific heats, $\gamma = 1.35$

Pressure at the end of compression, $p_2 = p_1 \times r^\gamma = 2.86$ MPa.

Temperature at the end of compression, $T_2 = T_1 \times r^{\gamma-1}$

$= 799.4$ K.

Temperature at the end of constant volume heat addition, $T_5 = T_2 \times p_5/p_2 = 1509.3$ K.

Temperature at the end of constant pressure heat addition, $T_3 = T_5 \times r_c = 2437.6$ K.

Pressure at the end of expansion, $p_4 = p_3\,(V_3/V_4)^\gamma$

$$= p_3\,(r_c/r)^\gamma = 0.36 \text{ MPa.}$$

Temperature at the end of expansion, $T_4 = T_1\, x p_4/p_1 = 1206.6$ K.

Heat input, $Q_1 = C_v\,(T_5 - T_2) + C_p\,(T_3 - T_5) = 1963.1\, C_v$

Heat output, $Q_2 = C_v\,(T_4 - T_1) = 871.6\, C_v$

Work output, $W = Q_1 - Q_2 = 1091.5\, C_v$

Mean effective pressure = Work output/Swept volume

$$= W/(V_1 - V_2) = W/[V_1\,(1 - 1/r)]$$

$$= [1091.5/(1 - 1/12)] \times (C_v/V_1)$$

$$= 1190.73 \times [R/\{V_1\,(\gamma - 1)\}] = 3402.1 \times (R/V_1)$$

$$= 3402.1 \times (p_1/T_1) \text{ Use ideal gas eqn } p_1 \times V_1 = R \times T_1$$

$$= 1.0155 \text{ MPa.}$$

Pressure and temperature at the cardinal points are given in table below:

Points number	*1*	*2*	*3*	*4*	*5*
Pressure (MPa)	0.1	2.86	5.4	0.36	5.4
Temperature (K)	335	799.4	2437.6	1206.6	1509.3

Thermal efficiency of the cycle = Work output / Heat addition

$$= 1091.5/1963.1$$

$$= 55.6\%.$$

SUMMARY

In this chapter, air-standard cycles are introduced for reciprocating engines. Otto cycle, diesel cycle and dual cycle are explained and compared. Thermodynamic description as well as practical description of the engines is also described. Several terms are used while reference is made to these air- standard engines. Method to calculate air standard efficiency for all three engines is also described. Salient terms used for description of such engines like compression ratio, cut-off ratio, expansion ratio etc are also explained in the text. There are 5 types of efficiencies prevalent for such engines namely : (*i*) brake thermal efficiency (*ii*) indicated thermal efficiency (*iii*) mechanical efficiency (*iv*) efficiency ratio (*v*) Overall efficiency. All five are described and their calculation strategies are illustrated. Volumetric efficiency, torque, power, mean effective pressure and specific fuel consumption are described for completeness.

QUESTIONS

1. What is thermodynamics? How it is different from heat transfer?
2. What are various types of the systems?
3. How is thermodynamic equilibrium?
4. What is zeroth law of thermodynamics? Why it is called so? What is its utility?
5. What is first law of thermodynamics?
6. What is second law of thermodynamics?
7. Substantiate whether second law of thermodynamics is a conservative law or not?
8. Plot (*i*) isobaric (*ii*) isochoric (*iii*) isothermal and (*iv*) isentropic processes on (*a*) pressure-volume diagram and (*b*) temperature-entropy plane.
9. Explain operation of Otto cycle? Where are these cycles used?
10. Derive expression for air standard efficiency of Otto cycle?
11. What are limitations of Otto cycle?
12. What is Diesel cycle? How it superior to Otto cycle?
13. Derive expression for air standard efficiency of diesel cycle.
14. What are main problems with diesel cycle? How it is overcome with dual cycle?
15. Derive expression for air standard efficiency of dual cycle.
16. Compare air standard efficiency of Otto and diesel cycle for (*a*) same compression ratio and heat input (*b*) same heat input and maximum pressure (*c*) same maximum pressure and maximum temperature (*d*) same work output and maximum pressure.
17. What is friction power? How it is related to indicated power?
18. Derive expression for mean effective pressure of Otto cycle?
19. Write short notes:
    (*a*) Thermodynamic equilibrium
    (*b*) First law of thermodynamics
    (*c*) Thermodynamic processes
    (*d*) Conjugate properties
    (*e*) Temperature
    (*f*) Entropy
    (*g*) Thermal efficiencies
    (*h*) Mechanical efficiency
    (*i*) Brake specific fuel consumption.
    (*j*) Mean effective pressure
    (*k*) Air standard efficiency
    (*l*) Specific fuel consumption
    (*m*) Volumetric efficiency.

20. Compare the given terms

(*a*) Intrinsic and Extrinsic properties

(*b*) Closed and open system

(*c*) Stress and pressure

(*d*) Kelvin planks and Claussius statement of second law of thermodynamics

(*e*) Otto cycle and Diesel cycle

(*f*) Indicated power and brake power.

❖ ❖ ❖

CHAPTER

2

Aircraft Power Plants

STRUCTURE

- Introduction
- Objective
- Brief Description and Principles
- Jet Propulsion
- Propeller Propulsion
- Turboprop Propulsion
- Bypass Jet Propulsion
- Ramjet Propulsion
- Summary
- Questions

2.1 INTRODUCTION

This chapter gives a basic idea about engines used in aircrafts for propulsion. Discussion on various types of aircraft power plants, their merits and development requirements are discussed in this Chapter. This will give a basic technical understanding on one hand and equip with ready knowledge for implementation in design and efficiency aspects of aircraft propulsion.

Objective

After studying this chapter, you should be able to :

- Understand basic principles of aircraft propulsion,
- Know requirements and critical factors in aircraft propulsion,
- Assimilate development of various aircraft engines,
- Realize various types of engines used in aircrafts and missiles,
- Compare aircraft engines.

2.2 BRIEF DESCRIPTION AND PRINCIPLES

Aircraft propulsion is basically an air-breathing type of propulsion. It carries fuel and needs atmospheric air for oxygen. Fuel can be solid, liquid or gas, but for ease of combustion and easy handling, most of the time aviation fuels are liquid. Essential requirement of aircraft propulsion is to generate hot gases at high pressure by combustion of fuel in presence of atmospheric oxygen as oxidizer. Expansion of these hot combustion gases is utilized for extraction of work. Thrust produced by most of the systems is directly related to expansion of gases.

The main principle of propulsion is derived from Newton's laws of motion. In this system, certain mass is discharged at high velocity in the form of a jet and reaction of jet causes forward thrust for the propulsion of aircrafts. This system of propulsion is also called jet propulsion. In jet propulsion, ejection of mass at high velocity is very important and propulsive force is always generated in the direction opposite to the outgoing jet of mass. The ejected mass should possess certain properties. It should be a fluid and preferably a gas, which can be expanded through nozzle. The energy available with chemical bonds of the used fuel is extracted by combustion in the combustors, which raises temperature as well as pressure inside the propulsion units. Such thermal jet increases temperature of air sucked inside the propulsion units. This is principle utilized in air-breathing engines. However jet propulsion is not restricted to air-breathing engines alone. It can be implemented for non-air-breathing engines also.

The non-air-breathing jet propulsion engines are utilized in rocket propulsion, where external supply of oxidizer (air) is not needed and fuel contains molecules, where both fuel and oxidizers segments are present. On combustion, breakage of chemical bond generates high temperature high pressure gases, which expands through rearward positioned nozzle, resulting in forward thrust as a reaction. There is no need of any external air supply for such engines. Performance of such engines is independent of outside atmosphere. Change in altitude and forward speed of the complete unit does not affect performance of rocket engines. Such systems can operate in vacuum also.

In all aircraft engines, gas turbines are used for propulsion. In the system, air is sucked in at high velocity. The velocity of air is reduced in the diffuser, situated in the front section of the engine. This results in rise in pressure by the compression of gases. Fuel is injected in the compressed air for combustion. After combustion, generated hot combustion gases are passed through gas turbine. Gases gain velocity and expand through turbines to give power primarily to run the compressor. Spare energy of expansion gives propulsive power. Final expansion of the gases through the nozzle is responsible for acceleration of flow and generation of propulsive thrust. Each of these subcomponents has different work and heat transfer operations and energy transformation results in various possible changes in the system.

When aircraft moves, the first section or component that feels outside atmosphere is called diffuser. This is placed behind intake ducts of the air breathing engines, employed in aircrafts. Air is sucked into the diffuser because of motion of aircraft at high velocity. Since high velocity may result in poor combustion and flame stability related issues, velocity of incoming stream of air is reduced in diffuser. This leads to rise in pressure of the incoming air. Indirectly diffuser converts kinetic energy of incoming air into pressure. This resembles compressor to some extent and the operation in a diffuser is called ram compression. Ideally process inside diffuser can be considered isentropic.

After diffuser air velocity reduces and pressure is slightly higher than ambient outside atmospheric pressure. This air is passed through next component of the aircraft engine called compressor. Compressor invariably draws power from the turbine part of the engine component. The process through compressor is also isentropic and incoming air is compressed to required pressure ratio. Temperature rise of air is always accompanied with compression. After compression high pressure high temperature air is available in the engine, which enters combustor or combustion chamber. Fuel in atomized droplet form is injected in the combustor. Fuel catches fire immediately on introduction in the high temperature environment. Atmospheric air is consumed and chemical energy of the fuel is released during this operation. Combustion results in large volume of gases and ultimately, pressure rise also occurs. Highest temperature as well as pressure is realized after combustion chamber in any engine. Combustion is generally assumed to occur at constant pressure inside combustor of an air-breathing engine of aircraft. In combustor, constant pressure heat addition takes place from thermodynamic point of view.

After combustor, turbine is next component in which high temperature, high pressure, and high volume combustion product is fed. In turbine, the incoming gases expand and work is produced. Some part of work is utilized to drive compressor and remaining for propulsion of the aircraft. Process in turbine is also isentropic in ideal situation and it results in reduction of pressure and temperature. Downstream, nozzle is placed, which expands the incoming gases further. The process in a nozzle is opposite to diffuser. In nozzles, gases expand and velocity of stream increases at the cost of pressure. This is also accompanied with reduction in temperature of the gases. The high velocity exhaust gases produce thrust required for the forward movement of the aircraft. State of air, while passing through various turbojet components, is depicted in figure 2.1.

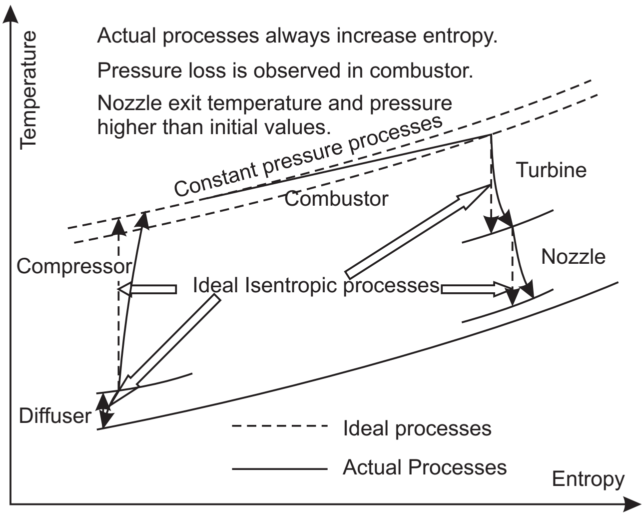


Fig. 2.1 : Actual Processes in a Simple Turbojet Engine

As depicted in figure 2.1, compression of air in diffuser is ideally isentropic and it results in increase of temperature as well as pressure. Constant pressure curved lines are shown with positive slopes on temperature entropy plane. However, actual flow of incoming air through diffuser is always associated with some losses and as a result of this the outgoing air from diffuser end up with higher entropy and temperature than expected. This is depicted by solid line. The outgoing air from diffuser enters compressors, where it should ideally be compressed in isentropic manner as shown by dotted line above. However, in actual scenario, again there is some rise in temperature and entropy of the air. Fuel is injected in combustor and is burnt. The air becomes loaded with combustion products and there is variation in properties of air slightly from that of air. Ratio of specific heats, specific heats at constant volume and pressure change for the working fluid. Although process through combustor is ideally considered to occur at constant pressure, but again some pressure losses are expected in combustor and outgoing gases from combustor are always fed to turbine at lower pressure than ideally expected. Expansion in a turbojet occurs in two phase. Both are assumed to occur in isentropic manner ideally but are always practically non-ideal. First gases from combustor outlet are expanded in turbine, which generates enough power to run the compressor. Turbine power is not responsible for motion of air crafts. It is expansion of gases through nozzle that provides necessary thrust due to momentum exchange for the propulsion of the aircrafts. Higher the velocity of combustion gases through the nozzle, higher will be propulsive thrust. Flow of working fluid through each component is non-ideal. The ratio of ideal to actual temperature ratio is depicted by efficiency of that particular component.

For flight-worthy configurations, several terms are introduced in course of time as performance indicators of aircraft engines. Specific fuel consumption, compactness, engine mass, life of components, modularity of elements, pollution levels etc are time and again placed as governing parameters for new ideas and designs. From performance point of view, several terms are of prime importance.

First is air standard efficiency, which has same role as played by thermal efficiency in standard Otto-cycle. This is dependent on pressure ratio (Pr), alone. Since the cycle resembles Otto cycle, air standard efficiency is also having same expression as that of Otto cycle. However, in Otto cycle, efficiency is expressed in terms of compression ratio, which is ratio of volumes before and after compression. But in aircraft engines, it is pressure ratio (not volume ratio), which is accounted. So, efficiency is not having same expression as thermal efficiency of an Otto cycle. Air standard efficiency of aircraft engine is given by $\{1 - 1/[P_r^{(\gamma-1/\gamma)}]\}$.

Thrust is another parameter, which is a measure of performance of a jet engine. It has two components – one is momentum thrust and other is pressure thrust. Momentum thrust is generated by change in momentum of air and fuel. It is given by mass flow rate multiplied by velocity of flow. At the inlet condition, only air enters with velocity of aircraft. However, at the exit plane both fuel and air contributes in the mass flow rate calculation. If fuel air ratio is given by 'f', then mass flow rate at exit is $(1+f)$ times mass flow rate at the inlet. In general, air is gas and fuel is liquid. For most simplified calculations, mass of fuel at the exit side of the aircraft engine can be neglected. However, initial velocity of incoming air, u_i attains a very high value at the nozzle outlet, usually given by u_e. Sometimes inlet velocity (u_i) can be neglected in light of very high value of exit velocity ($u_e >> u_i$).

$$\textbf{Momentum thrust} = M_a\,[(1 + f).u_e - u_i\,].$$

Pressure thrust is generated due to pressure mismatch at the outlet conditions. It is given by exit area multiplied by pressure difference.

$$\textbf{Pressure thrust} = A_e\,(P_e - P_i).$$

Inlet pressure can be taken equal to atmospheric pressure in normal situations. If flow is subsonic contribution of pressure thrust is almost negligible due to low change in pressure value. Additionally, even if there is some difference in the pressure values, exit cross-sectional area is so small that compared to momentum thrust value of pressure thrust is negligible. So for all practical purposes, pressure thrust is not considered for aircraft propulsion. However, for rocket propulsion, it has some significant value. For simplification effective jet velocity (U_j) is proposed for aircraft engines.

Thrust, $F = M_a [(1 + f).U_j - u_i]$,
where $U_j = u_e + A_e (P_e - P_i)/M_a$.

Product of thrust and velocity of the aircraft gives thrust power of the aircraft engine.

Power, $P = F.u = M_a [(1 + f).U_j - u_i].\ u_i$
$\simeq M_a.\ [U_j - u_i].\ u_i$.

Ratio of speeds of outgoing jet and aircraft is another important parameter. This is called speed ratio. The speed of aircraft is velocity of incoming air given by u_i. The outgoing jet can be assumed to have an effective velocity of U_j. So flight to jet speed ratio is given by ω (= u_i/U_j). When speed ratio is 1, thrust produced is zero because incoming jet and outgoing jet attains same velocity. For normal flight conditions, effective jet velocity must be greater than flight speed.

Maximum value of thrust power is achieved if $U_j = 2.u_i$. This means effective speed ratio, ω = ½. Maximum value of this thrust power is given by $M_a.u_1^2$.

However change in energy of incoming and outgoing jet of fluid gives propulsive power. It includes useful power given by thrust and wasted part of available power. Wasted part of available power is the residual kinetic energy of the gas discharged from the nozzle. The difference of propulsive power and thrust power is called leaving losses, which forms significant part of available power.

Propulsive power, $PP = M_a [U_j^2 - u_i^2]/2$.

Propulsive efficiency is another important efficiency term which accounts for conversion of jet power to useful thrust. It is ratio of thrust power and propulsive power.

Propulsive efficiency, $PE = 2.[U_j - u_i].u_i/[U_j^2 - u_i^2]$
$= 2.u_i./.[U_j + u_i]$.
$= 2.\omega/[1 + \omega]$.

At the start of the engine, aircraft is stationary and incoming jet velocity (u_i) is zero. Under such conditions (ω = 0), thrust power is also zero. Propulsive efficiency is zero. If maximum thrust is achieved in an aircraft (ω = ½), then propulsive efficiency is 66.66%. If effective velocity of jet is equal to incoming stream speed of air, expression for thrust power and propulsive power, both becomes zero. Under limiting condition, propulsive efficiency is calculated for (ω = 1). Maximum efficiency of 100% is achieved under such circumstances. Specific thrust is thrust produced per unit mass flow of gases or air. It has more significance for rocket engines.

Specific thrust is the highest at zero flight speed. It decreases with rise in speed of the flight. Thrust specific fuel consumption (TSFC) is another term which is ratio of rate of fuel flow and produced thrust. Infact overall efficiency is another term which correlates chemical energy of fuel with the thrust power. Chemical energy of the fuel is given by calorific value (CV) of the fuel, which is energy produced per unit weight of the fuel. Thrust power is given by $F.u_i$. Overall efficiency is equal to thrust power divided by chemical energy of the fuel.

Overall efficiency = $F.u_i./M_f.$(CV)
= u_i./(TSFC) . (CV).

EXAMPLE 2.1

Air flow through a turbojet engine is 50 kg/s and it propels an aircraft at a speed of 300 m/s. Nozzle is 95% efficient and isentropic enthalpy drop in the nozzle is 200 kJ/kg. If air fuel ratio is 80 and combustion efficiency is 95%, find fuel consumption, propulsive power, thrust power, thrust specific fuel consumption, propulsive efficiency, thermal efficiency, and overall efficiency. Calorific value of the fuel is 40000 kJ/kg.

SOLUTION

Mass flow rate of air, m_a = 50 kg/s

Mass flow rate of the fuel, m_f = 500/85 = 0.588 kg/s

Mass flow rate of gases through the nozzle, m = 50.588 kg/s

Enthalpy drop in the nozzle results in gain of velocity of the working fluid.

$$U_j/2000 = 200 \times 0.95$$

Flow velocity through the nozzle, U_j = 616.4 m/s

Thrust produced, $F = m.U_j - m_a.u_i$

$= 50.588 \times 616.4 - 50 \times 300 = 16.17$ kN

Thrust specific fuel consumption = m_f/F = 0.0364 kg/kN/s

Thrust power, $P = F \times u_i$ = 4854.733 kW.

Propulsive power, $PP = ½\,[m \times U_j^2 - m_a \times u_1^2]$ = 7360.4 kW.

Propulsive efficiency = Thrust power/Propulsive power

= 4854.733/7360.4 = 65.95%.

Heat supplied to the system = 0.95 × 0.588 × 40000 kJ/s

= 22344 kW

Thermal efficiency = Propulsive power/Heat supplied

= 7360.4/22344 = 32.94%

Overall efficiency = Thrust power/Heat supplied

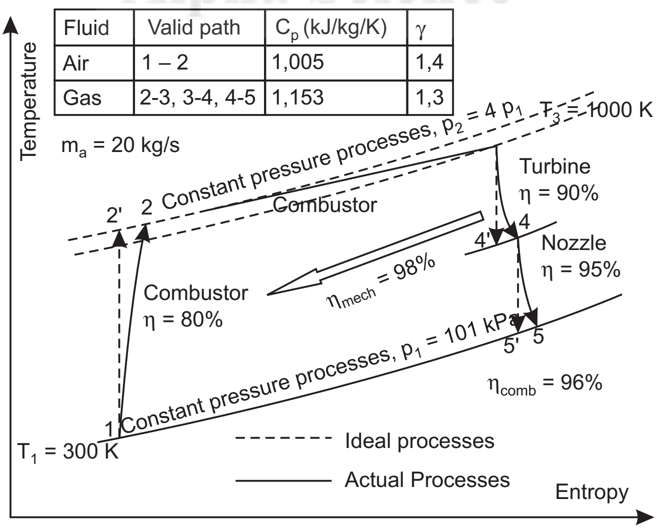
= 4854.733/22344 = 21.72%.

▮▮ EXAMPLE 2.2

For a simple jet propulsion engine, efficiencies of various sub- systems are given: Compressor = 80%, Turbine = 90%, Combustion = 96%, Mechanical efficiency = 98%, Nozzle = 95%. Mass flow rate of air is 20 kg/s and engine realizes a maximum temperature of 1000 K. Air at inlet condition has following properties: Temperature = 300 K, Pressure = 101 kPa, Ratio of specific heats for air = 1.4, Specific heat at constant pressure for air = 1.005 kJ/kg.K. Ratio of specific heats for combustion gases = 1.3, Specific heat at constant pressure for combustion gas = 1.153 kJ/kg/K. If overall pressure ratio is 4:1, then find (i) Pressure ratio in the nozzle, (ii) Effective jet velocity, (iii) Thrust and (iv) Specific thrust. Neglect mass of fuel. Take suitable assumptions.

SOLUTION

The given data is valid for a simple jet propulsion engine consisting of compressor, combustor, turbine and nozzles. Ideally, processes inside compressor, turbine and nozzle are isentropic and that in combustor is isobaric. Since efficiencies are given, all the calculations are to be done for isentropic change of properties and then efficiency terms is used to alter the given outlet conditions. There is no diffuser used in the system. The system is depicted in figure below.



Calculations are made in progressive manner from inlet condition using relations available.

For the isentropic process, 1-2′, take $\gamma = 1.4$,

$$T_2' = T_1 \times (p_2/p_1)^{\gamma-1/\gamma} = 300 \text{ K} \times 4^{1/3.5} = 445.8 \text{ K}$$

For the actual process in compressor, 1-2, take $\eta = 0.80$

$$T_2 = T_1 + (T_2' - T_1)/\eta = 482.3 \text{ K}$$

Pressure at the end of compressor, $p_2 = 4 \times 101$ kPa = 404 kPa.

Compressor work = $C_p\ (T_2 - T_1)$

$$= 1.005 \times (482.3 - 300) \text{ kJ/kg} = 183.16 \text{ kJ/kg}$$

Turbine work = compressor work/η_{mech}

$$= 183.16/0.98 = 186.9 \text{ kJ/kg}$$

Turbine work = $C_p\ (T_3 - T_4)$

$$= 1.153 \times (1500 - T_4) = 186.9 \text{ kJ/kg}$$

Temperature at exit of turbine, $T_4 = 1337.9$ K

For the isentropic process in turbine 3-4, take $\eta = 0.90$

$$T_4' = T_3 - (T_3 - T_4)/\eta = 1319.9 \text{ K}$$

Pressure at the end of turbine, $p_4 = p_3 \times (T_4' / T_3)^{\gamma/(\gamma-1)}$

$$= 404 \text{ kPa } (1319.9/1500)^{1.3/0.3} = 232.1 \text{ kPa.}$$

Pressure ratio in the nozzle = p_4/p_5 = 232.1/101 = 2.298

For isentropic process in nozzle, 4 – 5′, take $\gamma = 1.3$,

$$T_5' = T_4\ (p_5/p_4)^{\gamma-1/\gamma}$$

$$= 1337.9 \text{ K} \times (101/232.1)^{0.3/1.3} = 1104.2 \text{ K}$$

For the actual process 4-5, take $\eta = 0.95$

$$T_5 = T_4 + \eta\ (T_5' - T_4) = 1115.9 \text{ K.}$$

Since kinetic energy in nozzle is gained at the cost of reduction in temperature, initial velocity of gas in nozzle can be assumed to be zero and outlet velocity can be calculated using relation $C_5^2/2000 = C_p\ (T_4 - T_5) = 1.153 \times (1337.9 - 1115.9)$.

Effective jet velocity, C_5 = 715.5 m/s

Take combustion efficiency as 96%,

Specific thrust = $\eta \times C_5$ = 686.9 N.s/kg

Thrust produced = 686.9 × 20 N = 13737.9 N.

It is worth to have a quick look at the development of aircraft engines for fighter aircrafts. Although aero-engines evolved from piston engine, in early days, the development of aircrafts for battlefield application was concentrated mainly on the carrying and firing a gun in air-borne position. Major development of aircraft propulsion engine took place due to two World Wars.

During First World War, aircrafts used to have a rotor, called propeller mounted in the rear portion of the aircraft behind the pilot, so that pilot can aim and fire from their guns. Such structures were called pusher aircrafts, but they have higher drag and were relegated to secondary positions due to tractor aircrafts. Tractor aircrafts used front mounted propeller and firing line of the guns are to be located away from the propeller area. This is accomplished either by providing angular firing, or by rearward firing or by above the rotor firing positions. The synchronization of aiming and firing from a moving platform was very difficult and head-on attack on incoming enemy became difficult with such configurations. Later on technological evolution resulted in development of a synchronized gear, which can be timed in such a way that firing and rotor locations are distinct and rotor does not come in line during firings. During World War I, aircrafts were dominantly fitted with rotary engines.

However, in course of time, during period in-between two world wars, stationary radial engines captured propulsion domain of aircrafts. This resulted in increase of engine power five folds to a level as high as around 650 kW. Power of radial engine was raised to around 1500 kW by the end of Second World War. Although radial engines have higher drag, but they were preferred for Naval applications due to less stringent separate cooling requirements. However, parallel development of inline engines also took place. These engines had better power-to-weight ratio and were sleek. They were preferred by land based forces. Inline engines matched in power to their radial counterparts in same time frame. For a short duration rocket powered aircraft engines were also devised and used, but by 1944, first turbojet engine was designed and built. Turbojets ruled since then in the area of military aircrafts. Major advantages include less engine weight, far greater reliability, little cooling problem and adaptable for safer les flammable fuel.

The turbojets were classified in five generation as per time frame. First generation were prevalent during 1940s to 1950s and were first replacements of piston driven aircraft engines. To achieve higher and faster flights, the piston engines have a limited maximum speed, governed by achievement of sonic speed by the tip of the blades of the rotor. These limitations gave birth to the slower turbojets of first generation. Although they were faster, they were not based on much matured technology. As a result engine used to be delicate and massive. This results in slower adjustments of power and lack of maturity limited induction of first generation turbojet aircraft engines significantly. One of the major drawbacks of first generation aircrafts were their subsonic speed and the same was addressed comfortably by second generation aircraft engines. They covered an era of 1950s to 1960s and using advanced technologies like after burners, sound barrier is broken by these aircrafts.

Further developments for third generation (1960s-1970s) aircrafts were enhancement in maneuverability and ground attack capabilities by equipping them with advanced weapons. Use of turbofan with after burner evolved during this era. Fourth generation aircraft evolved during 1970s to 1990s and gave aircraft a multi-role capabilities. One advanced technologies perfected during this period for aircrafts were stealth technologies. Fifth generation is current vintage evolved after 2005.

Earlier engines were centrifugal flow engine with a thrust output of around 4-5 kN. This figure is presently around 200 kN with axial flow turbofans with after burner. At present, aircraft engines use multi-stage compressors and turbines and have supersonic speeds. In facts, various concepts of aircraft propulsion were used in modern aircraft engines to make them more efficient in different speed zones. With afterburners, if speed of a jet engine is increased, efficiency rises. Actually, such engines operates as turbojets at low speeds and as ramjets as higher speeds. At lower speeds, energy is derived from combustion of the fuel before expansion in the turbine and in true sense; turbojet is mainly responsible for the entire power generation. At higher speeds, compression takes place in the shock-waves and bypass flow of incoming air becomes predominant. In this case, no fuel is consumed before turbine after compression. Entire combustion of fuel takes place in after burner alone. In fact at very high speeds of the order of 2-3 Mach, the sock compression is sufficient to raise the temperature of incoming air to sufficient high level and fuel consumption becomes zero. Compressor–turbine combination does not deliver any power and bypass jet of air in ramjet mode delivers entire power.

Advanced version of turbojets is utilized in modern jet fighters, while helicopters invariably base their propulsion on turbo shaft power plants. Bigger military transporters use high bypass ratio turbofan or turboprop engines. The current trend is harnessing more power by lighter, smaller and simpler aircraft engines. Reliability, resistance to battle damage, easier maintenance, fuel efficiency, environment friendly exhaust are some additional features always inculcated in the new design of aircraft engines. In subsequent parts, thermodynamics andoperational characteristics of these aircraft engines are described in details.

Consideration of materials for construction of gas turbines in aero-engine has been a prime concern since beginning. In early engine designs, coating of turbine blades are contemplated to be a remedial measure for realized high temperature and offset effects of design shortfalls. Earlier engines are designed on the concept of ‘built and test’ but later, this philosophy is replaced by incorporating modeling techniques, optimization of material, manufacturing process, reduced cycle time and predictive techniques in the design of the aero engines. Fifty years back, steel dominated the construction materials, which is replaced with two major alloy classes – nickel based super alloys and titanium alloys. Steel is continued to be used in bearings and shafts where high strength and hardened surfaces are needed. Aluminum alloy have virtually disappeared from the aero-engine domains. In addition to material development, increasing reliance is being placed on coatings and surface treatment to achieve reliability and performance retention targets of the engines.

The development of any product has to undergo 4 stage process including product planning, full concept definition, product development and in-service management. After product planning a preliminary launch of the product occurs, this is followed by full launch at the end of concept development. After product development, product delivery phase starts. But this model is more of a business oriented development cycle. Turbine blades are critical components of an aero-engine and it should have provision for improved cooling and application of thermal barrier coating. This may enhance engine thrust, reduce weight, reduce fuel consumption and of course improves life. The expected benefits in terms of fuel burn, payload and operating cost are derived from such systems.

In addition to turbine, compressors also use blades and these blades are subjected to impact of outside atmospheric air. Compared to cargo engines or high flying aircraft engines, military engines, while operating in desert condition has very hostile atmosphere and blades are subjected to erosion by sand particles. Coating in such cases will have effective erosion resistance coating. For such erosive atmosphere, coating based on conventional titanium nitride coatings is now replaced by multilayer ceramic/ceramic or metal/ceramic combinations for improved wear resistance. For high and intermediate pressure turbines, nickel based alloys have dominated. Invariably such blades are subjected to severe condition of temperature and stress during service and sometimes the temperature may be higher than melting point of the alloys being employed. These blades must have demanding mechanical properties and environmental stability of the blade system. Creep and fatigue strength also play a major role from mechanical properties point of view and from environmental side, corrosion and oxidation are always taken into consideration.

In early 1960s, premature blade failure due to environmental degradation is observed and surface coating was considered fit to offset such problems. Simple, cheap aluminide coatings improved blade life significantly. However temperature remains a critical issue in such developments and further quest for higher performance raised the severity of atmospheric degradation, making aluminide coating insufficient for further use. So, platinum-aluminide coating is evolved. The latest trend is application of thermal barrier coating on blades, which prevents high temperature to percolate deep inside blades. This can improve operating temperature by at least 100 K. the thermal barrier coating may be Yttria-stabilized zirconia and is generally applied by electron beam physical vapour deposition. This process results in development of columnar grain microstructure necessary to resist thermal and mechanical strains. The requirement of durability of coating and above all bonding with base metals are major concerns.

Above all turbine entry temperature is the controlling factor, which is raised by cooling provisions and selection of advanced materials of construction. From the material of construction point of view, all engines are based on single crystal alloys in most demanding high pressure turbine stages and the enhancement in turbine entry temperature (TET) is achieved by research in material science. Different materials used in construction of turbine blades have different peak temperature capabilities. Turbine entry temperature was generally of the order of 950 K in the beginning. Wrought iron works in the range of around 1000 K and is generally uncooled. Provision of cooling can be incorporated in cast alloys and it can withstand a temperature of around 1100 K. With thermal barrier coating, turbine entry temperature can be raised to as high as 1500 K. For a service life of 2500 hours, uncoated blades may show severe hot corrosion, while aluminized coated blades does not show any sign even if subjected to marine atmosphere, where severe corrosion is expected.

2.3 TURBOJET PROPULSION

Turbojet, as a concept was prevalent in early days but realization of operational engines started in 1930s in Italy, Germany, UK and other European countries. It has been in use for propulsion of aircrafts and missiles. Turbojet utilizes combined effects of turbines and high speed fluid jet for realization of desired power. As it is an air-breathing engine, it has a simple or complex air intake system. Design of intake depends on the flight speed and the geometry of aircrafts. The incoming air is compressed in an axial or a centrifugal compressor. This raises pressure of incoming air at the cost of its velocity. After compression, pressure and temperature both rises and liquid fuel is injected in compressed air stream. This resembles heat addition process. After heat addition, it is passed through turbine to extract power. A part of turbine power is used for driving compressor and remaining part of energy is extracted during expansion of fluid stream through nozzle. In nozzle high pressure fluid stream is converted to high velocity fluid jet to produce thrust. Turbojet has five components namely intake, compression, combustion, turbine and nozzle. Except nozzle, all other parts are used for gas generation while nozzle helps in generation of thrust. The components of a turbojet engine are depicted in figure 2.2.

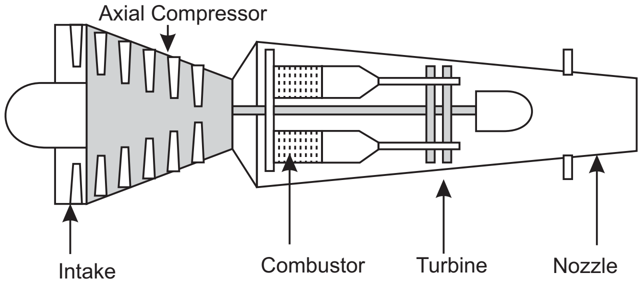


Fig. 2.2 : A Simple Turbojet with Axial Compressor

After air intake, compressor becomes an important component of turbojets. For centrifugal compressors, pressure ratio may vary up to 6-8 in single stage. For axial compressors, several stages are used. Each stage has a combination of stator and rotor. In the rotor, fluid is accelerated while in stator, the kinetic momentum is recovered in the form of advanced pressure. Pressure ratio in single stage of axial compressor may vary from 1.4 to 1.6.

From thermodynamic point of view, the process occurring inside a turbojet engine can be described as figure 2.3. Air at local ambient condition and zero velocity is available as depicted by point '0' on the temperature entropy plot. First, air is sucked-in through intake at certain velocity. This process is depicted by '0-1'. Intake portion does no change in condition of air from temperature point of view and is depicted by process '1-2'. Next portion depicts ram-compression and in some cases, a diffuser also. During movement of air through compressor, velocity reduces and pressure rises. This is simultaneously associated with rise in temperature of gas streams. The process is isentropic and is depicted by process '2-3'.

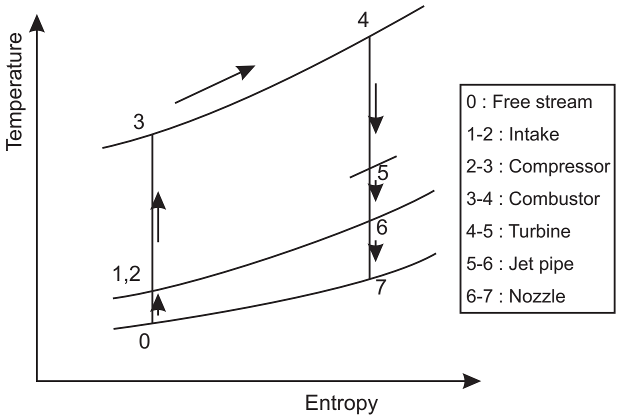


Fig. 2.3 : Representation of Turbojet Engine on Thermodynamic Plane

During motion of air through combustor, first fuel is injected in the beginning and burns during the process. It may be considered as constant process heat addition and is depicted by process ‘3-4’. Here rise is temperature is expected and this process is always associated with highest temperature of the cycle. Such high temperature gases are expanded through turbine, where positive work is extracted. This process is isentropic and is depicted by process ‘4-5’. If afterburner is used, process is depicted by ‘5-6’ else turbine process continues till point ‘6’ on thermodynamic plane. Expansion of compressed gases takes place in nozzle and pressure is converted to kinetic energy. This flow is also isentropic and is depicted by process ‘6-7’.

In turbojet engines, compressor pressure ratio is an important parameter is equal to ratio at the end and beginning of the compression process ($\sim p_3/p_2$). Another relevant parameter is turbine pressure ratio, which is decided by the power required by the compressor. It is equal to ratio of pressure after passage through turbine to pressure at the entry of the turbine (p_5/p_4).

The expression for thrust from turbojet engine is given below.
$F = m_a [V_7 (1 + f) - V_1] + A_7 (p_7 - p_0)$, where m = mass of air, V = volume, A = area of cross section, p = pressure, subscripts denote condition at respective points on thermodynamic plane.

There are several advantages of turbojet engines. It is used mostly in larger planes. It has simple construction and diffuser action is predominant. It has lower weight and it occupies lower space also. It is suitable for long distance flight at higher speed and altitude. Flow of air is also continuous and there is no extra air flow line needed for operation. It has compression ratio varying from 5 to 18 and flight speeds of 260 m/s. However, it needs a longer runway to take off and is not an economically viable option for short distance flights. At lower speeds, the thrust reduces significantly, thereby reducing propulsive efficiency. At low altitude and lower speed, thrust specific fuel consumption is much higher than contemporary engines.

EXAMPLE 2.3

A turbojet aircraft flies with a velocity of 300 m/s at an altitude where pressure and temperature are 30 kPa and –50°C. The compressor has a pressure ratio of 10 and air enters at a speed of 50 kg/s. Temperature at turbine inlet is 1500 K. Find temperature and pressure of gases at compressor, turbine and nozzle exit, nozzle pressure ratio, velocity of gas at nozzle exit, jet speed ratio and propulsive efficiency.

SOLUTION

The process is named as below:

1-2: ram compression of incoming air in the diffuser

2-3: compression of air in compressor

3-4: constant pressure heat addition

4-5: isentropic expansion of gases in the turbine

5-6: isentropic expansion of gases in the nozzle

Given $T_1 = -\ 50°C = 223.16$ K, $p_1 = 30$ kPa.

$u_i = 300$ m/s, $p_3/p_2 = 10$, $T_4 = 1500$ K.

$m_a = 50$ kg/s

Assume All processes are isentropic.

- Neglect mass increase of working fluid due to fuel.
- Property of combustion gases resembles that of air.
- Specific heat at constant pressure, $C_p = 1.005$ kJ/kg/K
- Ratio of specific heats, $\gamma = 1.4$
- Turbine generates power to run compressor only.
- No pressure drop in operation, $p_3 = p_4$ and $p_1 = p_6$
- Velocity of jet at nozzle entrance is negligible.

Temperature at compressor inlet, $T_2 = T_1 + u_1^2/2\ C_p$

$= 277.94$ K.

Pressure at compressor inlet, $p_2 = p_1 \times (T_2/T_1)^{\gamma/(\gamma-1)}$

$= 55.48$ kPa

Pressure at compressor exit, $p_3 = 10 \times p_2 = 554.8$ kPa

Pressure at turbine entry, $p_4 = p_3 = 554.8$ kPa

Temperature at compressor exit, $T_3 = T_2 \times (p_3/p_2)^{\gamma-1/\gamma}$

$= 277.94 \text{ K} \times 101/3.5 = 536.6$ K

Temperature drop in turbine = Temperature drop in compressor.

or, $T_4 - T_5 = T_3 - T_2$,

Temperature at turbine exit, $T_5 = T_4 - T_3 + T_2$

$= 1500 - 536.6 + 277.94 = 1214.3$ K

Pressure at turbine exit, $p_5 = p_4 \times (T_5/T_4)^{\gamma/(\gamma-1)} = 286$ kPa

Pressure at nozzle exit, $p_6 = p_1 = 30$ kPa

Nozzle pressure ratio = $p_5/p_6 = 286/30 = 9.533$

Temperature at nozzle exit, $T_6 = T_5 \times (p_6/p_5)^{\gamma-1/\gamma} = 637.6$ K

Velocity of jet at nozzle exit,

$V_6 = \sqrt{[2000 \times 1.005 \times (1214.3 - 637.6)]} = 1076.64$ m/s

Jet speed ration, $\omega = 300/1076.646 = 0.2786$

Propulsive efficiency $= 2\omega/(1 + \omega) = 43.58\%$.

EXAMPLE 2.4

An advanced fighter aircraft engine operating at 250 m/s at an altitude of 10 km (Temperature = 230 K) has the following performance characteristics: Thrust force = 50 kN, mass flow rate of air = 40 kg/s, mass flow rate of fuel = 2.5 kg/s, exit pressure = ambient pressure. If calorific value of the fuel is 40000 kJ/kg, find (i) specific thrust (ii) thrust specific fuel consumption (iii) jet velocity (iv) thrust power (v) propulsive power (vi) heat supplied (vii) thermal efficiency (viii) propulsive efficiency (ix) overall efficiency (x) jet speed ratio.

SOLUTION

Mass flow rate of gas through the nozzle = mass flow rate of

(air + fuel) = 50 + 2.5 kg/s = 52.5 kg/s.

Specific thrust = thrust/mass flow rate of gas = 50000/52.5 = 952.4 N/kg/s

Thrust specific fuel consumption = fuel consumption / thrust

$= 2.5/50000 = 5 \times 10^{-5}$ kg/N/s = 0.18 kg/N/hr

From thrust equation, $F = m.U_j - m_a.u_i$

$= 52.5 \times U_j - 50 \times 250 = 50$ kN

So, Effective jet velocity, $U_j = 1190.5$ m/s.

Thrust power, $P = F \times u_i = 50 \times 250 = 12500$ kW.

Propulsive power, $PP = ½\ [m \times U_j^2 - m_a \times u_1^2] = 35641.4$ kW.

Heat supplied to the system = 2.5 × 40000 kW = 100000 kW

Thermal efficiency = Propulsive power/Heat supplied

= 35641.4/100000 = 35.64%

Propulsive efficiency = Thrust power/Propulsive power

= 12500/35641.4 = 35.07%.

Overall efficiency = Thrust power/Heat supplied = 12.5%

Jet speed ration, $\omega = 250/1190.5 = 0.20999$

Propulsive efficiency = $2\omega/(1 + \omega)$ = 34.71%.

2.4 PROPELLER PROPULSION

Reciprocating engines are developed for automobiles, but as power requirements increase, the reciprocating engine turns out to be more voluminous. Typical power to volume ratio of most of the reciprocating engines lies within 0.3 to 0.5 kW/m^3. For application in aircrafts, this may not be a suitable alternative. However before 1940s, the systems based on piston-engine propeller dominated the propulsion needs for aircrafts. The system is designated as propeller propulsion because main thrust producing element in this case is propeller, which receives power generated by the engine and whose rotation causes increase in the speed of the air stream also. Later on turbojets took over rein. In recent times, propeller propulsion is used for low power low mass high efficiency engines of small single seater aircraft and unmanned aerial vehicles.

Another problem with use of reciprocating engines in aircraft is altitude correction. As altitude rises, density of air reduces and power output also reduces. At high altitudes, the system gives lower energy. Of course, running engine at higher rpm can offset this drawback to some extent.

Expression for power output is given below.

$P = K \cdot N \cdot V_c \cdot \rho_{air} \cdot (1 + f) \cdot Q_m \cdot f \cdot \eta_{ov}$, where K = 1 for 2-stroke engine and 0.5 for 4-stroke engine, N = rpm of engine, V_c = volume of cylinder, ρ_{air} = density of air, f = fuel-air ratio, Q_m = heat released per unit mass of fuel (calorific value), η_{ov} = overall efficiency.

EXAMPLE 2.5

The diameter of the propeller of an aircraft is 2.5 m. if flies at a speed of 540 kmph at an altitude of 8000 m, where density of air is 0.528 kg/m^3. If jet speed ratio is 0.75, determine (i) jet velocity (ii) mass flow rate of air (iii) thrust produced (iv) thrust power (v) specific thrust (vi) specific impulse.

SOLUTION

Air intake area, $A = \pi d^2/4 = 4.91$ m^2

Flight speed, u_i = 540 kmph = 150 m/s

Jet velocity, U_j = 150/0.75 = 200 m/s

Average flow velocity = $(U_j + u_i)/2$ = 175 m/s

Mass flow rate of air = density × area × velocity

= 0.528 × 4.91 × 175 = 453.68 kg/s

Thrust produced, $F = m_a (U_j - u_i)$ = 22.684 kN

Specific thrust = thrust/mass flow rate = 50 N.s/kg

Thrust power, $P = F \times u_i$ = 3402.6 kW

Specific impulse = specific thrust/g = 50/9.81 s = 5.097 s.

2.5 TURBOPROP PROPULSION

Turboprop propulsion is a combination of propeller and turbojet. In this case thrust is partially produced by propeller and partially by jet. First incoming gas expands in turbine and residual expansion takes place in nozzle. Performance of such engines lie between turbojet and propeller propulsion engines. It is fuel efficient as compared to turbojet. This is achieved because large mass of air undergoes small velocity increments. Additionally, propulsive efficiency increase and specific fuel consumption reduce. It offsets limitations of altitude-speed envelope prevalent in propeller engines.

Schematic details of a typical turboprop propulsion engine are depicted in figure 2.4. All components of simple turboprop engines are shown. Turboprop engines find application in large size aircrafts due to their very large power-to engine mass ratio. For operational turboprop engines, this ratio is of the order of 4 to 10 kW/kg. This value is 2-10 times that of a typical reciprocating engine. Similarly deliver power-to-volume ratio of turbo props are 8 times higher than that of reciprocating engines.

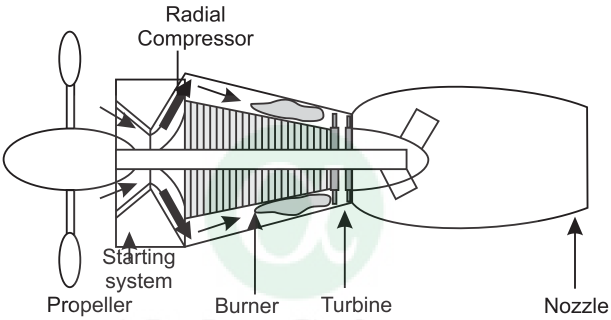


Fig. 2.4: A Simple Turboprop with Radial Compressor

In this case, partitioning of power of turbine is carried out. Some part of output power from the turbine is utilized for running compressor and other auxiliary units. Rest is utilized for generating thrust. The propeller employed develops propulsive thrust. Since turbine operates at high rotational speeds, a gear reduction is needed to power the propellers, which cannot be rotated at such high speeds. The gear box is employed for this operation.

Another variation of turboprop engine is turbo shaft engines, where all the delivered power is shaft power and no thrust is obtained because of expansion of outgoing air-jet. This is other extreme of power extracted from turbine of turbojets. This type of engine is used in helicopters and it does not need any runway for lift-off.

Turboprop engines have high propulsive efficiency. Contrary of turbojet engines, it has high thrust at low speed. As a result thrust specific fuel consumption is also low. One major advantage of this method of propulsion is automatic braking provision by thrust reversal. By changing blade angle, thrust reversal and reduction in speed of the aircraft is possible. It has high thrust per unit frontal area. It has high propulsive efficiency as compared to turbofan engines.

However, this system has several disadvantages also. It works well at low altitude and at low speeds, and operation at high altitude and high speeds are not very efficient. Chances of shock, vibration, and flow separation are observed in turboprop engines. The propeller is attached with a gear box, which adds weight to the unit. It has higher weight per unit thrust produced. This leads to reduction in payload capacity of the engine. One major disadvantage of this engine over turbojet engine is speed of operation. Although turbojets can be operated at both subsonic and supersonic speeds, but turboprop engines can be operated at subsonic speeds only. Turbines in turboprops generates more power, as in addition to turbine (as in turbojets), it has to run a propeller also. Propeller is much bigger in size than main body of the engine. Air flow through main engine is much smaller and ratio of air flow through propeller and main engine is 25 to 50. It acts at very high bypass ratio.

Sonic velocity in air can be given, $C = \sqrt{\gamma . R . T}$, where, γ = ratio of specific heats, R = Gas constant for air, T = temperature at point of interest.
Mach number is ratio of flow velocity to local sonic velocity.
Mach number, $M = U/C$, where U is flow velocity, C = Sonic velocity.

Two specific heats of ideal gases are related to each other as depicted below.

$$C_p - C_v = R$$
$$C_p/C_v = \gamma$$
$$\text{So, } C_p = \gamma \, . \, R/(\gamma - 1)$$
$$C_v = R/(\gamma - 1)$$

For flow of air through diffuser or compressor, the flow is generally assumed to occur in isentropic manner ideally. Incoming jet has certain higher velocity and compared to this outlet velocities are negligible. The incoming jet velocity is converted to pressure and stagnation pressure after passage through the devices is higher than pressure of high velocity incoming air. Inlet and outlet conditions are depicted by subscripts 1 and 2, respectively. Change in enthalpy through a system undergoing isentropic change (compression) from state 1 to state 2, can be equated to change in velocity air through the system.

$$
\begin{aligned}
H_1 - H_2 &= (U_2^2 - U_1^2)/2 \\
C_p\,(T_1 - T_2) &= 0 - U_1^2/2 : \quad \text{Neglect inlet velocity} \\
1 - T_2/T_1 &= -U_1^2/2 \, . C_p \, . \, T_1 \\
&= -[(\gamma - 1)/2]\,[U_1^2/(\gamma \, . \, R \, . \, T_1)] : \text{Put value of } C_p \\
&= -[(\gamma - 1)/2]\,[U_1^2/C_1^2] : \text{Put sonic velocity} \\
&= -[(\gamma - 1)/2]\,M_1^2 : \text{Put Mach number} \\
T_2/T_1 &= 1 + [(\gamma - 1)\,M_1^2/2].
\end{aligned}
$$

Air standard efficiency of a cycle operating between two constant pressures can be calculated from pressure ratio. This resembles air standard efficiency of an Otto cycle, which is calculated from compression ratio. In case of constant pressure processes, air standard efficiency depends on pressure ratio (final pressure to initial pressure), only. Compression ratio (r) is ratio of volumes, whereas pressure ratio (r_p) is ratio of pressures. The process is generally assumed to occur in isentropic manner and both ratios are inter-related.

Compression ratio, $r = (V_1/V_2) = (p_2/p_1)^{1/\gamma} = (r_p)^{1/\gamma}$
Air standard efficiency, $\eta = 1 - 1/r^{\gamma-1} = 1 - 1/r_p^{(\gamma-1)/\gamma}$.

EXAMPLE 2.6

A turboprop engine operates at an altitude of 3000 m where pressure = 70 kPa, temperature = 265 K and air density = 0.909 kg/m^3. Air craft speed is 150 m/s and velocity of air at compressor entry is 90 m/s. Ideal temperature rise in compressor is 230 K. find (i) Sonic velocity and Mach number at inlet (ii) temperature and pressure at compressor inlet (iii) pressure ratio in compressor and diffuser (iv) pressure rise in compressor and diffuser (v) power needed by compressor (vi) air standard efficiency. Assume $\gamma = 1.4$, $C_p = 1.005$ kJ/kg/K.

SOLUTION

Refer figure 2.1

Nomenclature of processes

1-2: Actual diffuser operation

1-2′: Ideal diffuser operation

2-3: Actual compressor operation

2-3′: Ideal compressor operation

Given data

$P_1 = 70$ kPa, $T_1 = 265$ K, $\rho_1 = 0.909$ kg/m^3, $u_1 = 150$ m/s

$u_2 = 90$ m/s, $T_3 - T_2 = 230$ K.

$\gamma = 1.4$, $C_p = 1.005$ kJ/kg/K

Calculation

Gas constant, $R = C_p - C_v = C_p\,(1 - 1/\gamma) = 0.287$ kJ/kg/K

Sonic velocity at inlet = $\sqrt{\gamma.R.T}$

$= \sqrt{1.4 \times 287 \times 265} = 326.31$ m/s

Mach number at inlet = aircraft speed/sonic velocity at inlet

$= 150/326.31 = 0.4597$

Stagnation temperature at diffuser exit = $T_1 \times \{1 + [(\gamma - 1)M_1^2/2]\}$

$= 276.2$ K.

Since there is certain residual velocity at diffuser exit, stagnation condition does not exist and actual temperature at turbine exit is lower than stagnation temperature by the amount calculated from residual velocity

Temperature rise due to residual velocity = $U^2/2.C_p$

$= 90^2/(2 \times 1005) = 4.03$ K

Ideal temperature at diffuser exit, $T_2' = 276.3 - 4.03$ K

$= 272.17$ K.

Temperature at compressor inlet = temperature at diffuser exit

$= 272.17$ K.

Flow through diffuser can be assumed to be isentropic.

Pressure at diffuser exit, $p_2 = p_1 \times (T_2/T_1)^{\gamma/(\gamma-1)} = 76.856$ kPa

Pressure ratio in diffuser, $p_2/p_1 = 1.0979$

Pressure rise in diffuser, $p_2 - p_1 = 6.856$ kPa

Ideal temperature at the exit of compressor, $T_3 = T_2 + 230$

$= 502.17$ K

Pressure at compressor exit, $p_3 = p_2 \times (T_3/T_2)^{\gamma/(\gamma-1)} = 655.714$ kPa

Pressure ratio in compressor, $p_3/p_2 = 8.45$

Pressure rise in compressor, $p_3 - p_2 = 578.858$ kPa

Power needed by compressor = $C_p\,(T_3 - T_2) = 231.15$ kW/kg/s

Air standard efficiency, $\eta = 1 - 1/r_p^{\,(\gamma-1/\gamma)} = 45.65\%$.

2.6 BYPASS JET PROPULSION

Turbofans are further developments over turboprop engines to enhance flight speeds. In this case, front section is placed inside a duct, which acts as diffuser (velocity reduces and pressure rises). This alteration reduce speed in the front section for a given flight speeds. Arrangements are also made to bypass some of the incoming air through an annular duct placed outside the core engine and supply it directly to nozzle for extraction of extra power. These two differences may give higher thrust in same volume. This system may be treated as bypass jet propulsion. The ratio of the air handled by the fan to that going through the core engine is called bypass ratio, denoted by '$\propto$'. Straight turbojet has zero bypass-ratio. Typical value of bypass ratio for civil aircraft may be 6.

Basically, propeller becomes a fan in presence of a duct and it has capacity to compress incoming air. The compressed air on expansion has enhanced velocity and enhanced thrust. The enhancement is a function of amount of air being handled and pressure ratio developed. There are two types of engines on the basis of mixing of bypass part of air. In non-mixing bypass engines, cold air expansion takes place, which does not mix with air passing through core engine. Similarly mixing bypass engines can also be designed.

Bypass ratio is single important factor affecting performance of this class of engines. If bypass ratio is increased, velocity increment is low and higher thrust can be achieved. However, very large increase in bypass ratio results in higher frontal area and enhanced drag. For larger aircrafts, this drawback is relegated to secondary status due to proportionally higher enhancement in thrust. For supersonic aircrafts, even low bypass ratio can deliver sufficient enhancement in thrust. For military aircrafts, the bypass ratio may lie between 0.3 and 1.0. This system has short take off roll and has low weight per unit thrust produced. It has more efficient flow in fan. After passing through fan, air is divided into two streams – first is primary stream which goes through main sub-system of the power generation system and secondary stream of inducted air passes through side channels and expands in annular nozzle as cold air. This makes system viable for use in large jumbo jet airplanes weighing more than 600 tons and carrying more than 250 passengers. Bypass ratio in these units are small (of the order of 5-6). There is no requirement of gear and fan is also placed inside covered duct or flow passage. However, this engine is heavier and drag is also on higher side. It has low propulsive efficiency and low weight per unit thrust. Thrust specific fuel consumption is high and it operates at lower speeds.

2.7 RAMJET PROPULSION

If power is provided at higher flight speeds, compressors can be dispensed with. It is observed that at speeds above Mach 2, the incoming air through air-intake can alone give a 7-12 times rise in pressure by placement of a diffuser. This is comparable to pressure obtained by putting compressors at low flying speeds. So, this system of propulsion derived from turbojet, where compressor is not used is called ramjet propulsion. This system has diffuser, combustor and an exit nozzle.

But such systems work at high flying speeds and cannot be executed in cold-start conditions. These systems are used in missiles, because of their simplicity and high efficiency. They do not produce any thrust at zero speed and an auxiliary system is needed to bring complete system to a certain minimum velocity before ramjet propulsion is operational. This additional propulsion system is called boosters. In missiles, boosters may be based on solid or liquid propellants based rockets. A typical ramjet engine is shown in figure 2.5.

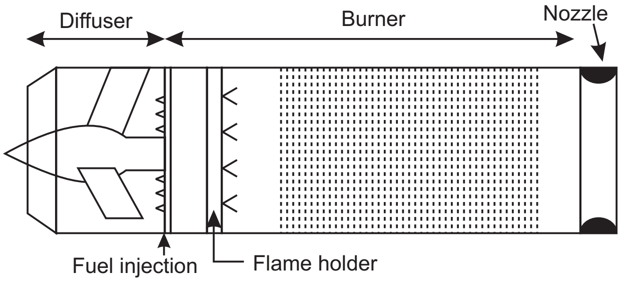


Fig. 2.5 : A Simple Ramjet Propulsion System

The system does not have a moving part and has simple construction. It is economical, has light weight and needs less maintenance. Fuel consumption is lower at high speeds. System can be designed for a variety of fuels. At supersonic speeds, it produces very high thrust and high efficiency. This makes it attractive for high speed aircrafts and missiles. At high speeds and altitude, it is better than turbojets.

However, for this system careful assessment of parameters and design of each sub-component becomes critical. It cannot be started from stationary condition and some auxiliary propulsion unit is needed to propel it to certain speed for successful operation of ramjet. These systems are unstable in subsonic flights. It has low thermal efficiency and high thrust specific fuel consumption. As velocity of incoming air is reduced by ram compression alone, air stream has high velocities. For efficient combustion of fuel, it needs flame holder and flame stabilizers. It has limited altitude of operation, because reduction in density of air affects performance adversely.

SUMMARY

In this Chapter, typical aircraft and missile propulsion engines are discussed. The turbojets are workhorse propulsion concept for aircraft propulsion. Turboprop and bypass propulsion concepts also find application for specific requirements. For missile propulsion, ramjet propulsion is invariably used but it needs additional booster propulsion to fulfill initial high velocity requirements.

QUESTIONS

1. What is jet propulsion? Why they are suitable for aircraft propulsion?
2. What are different generations of aircrafts as per vintage?
3. What are different requirement of materials of selection for turbine blades? What are common materials of construction for the turbine blades?

4. How is propulsive efficiency calculated for turbojets?

5. Explain various processes in operation of turbojet on a temperature entropy plane.

6. What are main advantage and disadvantages of propeller engine?

7. Whether turbojet can be powered without supply of any fuel to the main engine? Explain conditions for the same.

8. What are variants of turboprop engines for aircraft applications?

9. Why helicopters do not need a runway?

10. What are advantages of ramjet propulsion? What are its drawbacks?

11. Differentiate between the following:

(*a*) Rotary piston engine and radial engine

(*b*) Turboprop engine and by-pass jet engine

12. Write short notes on the following:

(*a*) Bypass ratio

(*b*) Propeller

(*c*) Ramjet propulsion

(*d*) Afterburner

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CHAPTER 3

Internal Combustion Engines

STRUCTURE

- Introduction
- Objective
- Type and Process
- Working of Spark Ignition Engines
- Working of Compression Ignition Engines
- Working of 2-stroke Engines
- Working of 4-stroke Engines
- Parts of Engines and their Materials
- Combustion Process
- Abnormal Combustion
- Arrangements for Multi-cylinder aircraft Engines
- Intake and exhaust manifolds
- Aircraft SI Engines
- Ignition Systems
- Effect of Altitude and Speed
- Power required and Power Available
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3.1 INTRODUCTION

Internal combustion engines belongs to heat engines, where chemical energy of fuel is transformed into thermal energy by combustion and this energy is used to generate mechanical work. Obviously, there will be something called external combustion engines also. In an external combustion engine, heat generated by combustion of fuel is used to heat the main working fluid before extracting work. Main advantages of external combustion engines are use of cheaper fuel including solid fuels, high starting torque etc. An internal combustion engine utilizes the product of combustion directly as motive fluid or working fluid. Although internal combustion engines suffers from many deficiencies as compared to external combustion engines, it has several advantages over external combustion engines like simple design, lower cost, less cooling requirement, less bulky, high overall efficiency etc. this makes it suitable for transport vehicles. This Chapter gives a brief description of internal combustion engine and its components. Types, process, material of construction, components, combustion and utilization aspects are discussed.

Objective

After studying this Chapter, you should be able to understand:

- Definition of internal combustion engines.
- Operations of internal engines.
- Spark ignition and compression ignition engines.
- 2-stroke and 4-stroke engines.
- Material of construction of various components of an engine.
- Combustion process inside an internal combustion engine.
- Arrangements of multi-cylinder reciprocating engines.
- Elements of Intake and exhaust manifold.
- Aircraft SI engines.

3.2 TYPES AND PROCESS

Internal combustion engines are used mainly in transport vehicles, but have several other applications also. As the name suggests, in internal combustion engine, combustion of fuel takes place in the engine itself and product of combustion forms the working fluid. If internal combustion engine exists, then there must be another counterpart called an external combustion engine. In case of external combustion engine, fuel is burnt outside the engine and product of combustion heats the working fluid. The combustion gases cannot be expanded to do work. In this case, cheap fuel can be used for combustion and working fluid may be water. This type of system is useful for stationary plants; coal based locomotive railway engines etc. Internal combustion engine is more efficient, but it cannot be made self-starting and initial cranking is needed. Contrary to this external combustion engines are self-starting. These two types of engines can be differentiated on several grounds.

Internal Combustion Engine (ICE)	*External Combustion Engine (ECE)*
Fuel burns in the engine cylinder.	Fuel burns in separate boiler.
Combustion gases act as working fluid.	Separate working fluid is employed. Most of the time, it is water (steam).
Working fluid can be costly.	Cheap fuel can be used.
Maximum temperature and pressure are higher. Material of construction should withstand the same. Generally iron alloy with nickel and molybdenum is used.	Maximum temperature and pressure are limited. Material of construction can be relatively cheaper. Generally cast iron can be used for the making engines.
Size is much smaller, as only fuel tank is needed.	Separate boiler to store water needs more floor space.
Efficiency is higher.	Efficiency is lower.
Engine is cheap.	Engine is costly.
Engine is lighter and can be used in transport vehicles.	Engine is bulky and heavy it is used in stationary plants.
Engine has less starting torque and can be started quickly.	Starting an engine is prolonged activity.
Engine cannot be made self-starting.	Engines are self-starting.
Piston is directly connected to connecting rod.	Stuffing box is generally employed to prevent leakage of steam.
Used in automobiles, aircrafts etc.	Used in power plants, locomotives etc.

Internal combustion engine is reciprocating in nature and piston movement inside a cylinder result in power generation, to impart motion to transport vehicle and do useful work. Fuel is injected along with certain quantity of air in the engine cylinder, where they burn and produce large quantity of hot combustion gases. Because of confinement, high pressure is generated and expansion of high pressure gases to lower pressure gives power output from the internal combustion engine. All these engines are means for propulsion, which means forward pull and are based on chemical propulsion, where combustion of fuel is responsible for creating motion in the body. The word propulsion is derived from two Latin words: ‘pro’ meaning before or forwards and ‘pellere’ meaning to drive. Propulsion means to push forward or drive an object forward. A propulsion system is a machine that produces thrust to push an object forward. On airplanes, thrust is usually generated through some application of Newton’s third law of action and reaction. A gas, or working fluid, is accelerated by the engine, and the reaction to this acceleration produces a force on the engine. A general derivation of the thrust equation shows that the amount of thrust generated depends on the mass flow through the engine and the exit velocity of the gas. Different propulsion systems generate thrust in slightly different ways.

Ground propulsion is a different term than transport, because it refers to solid bodies being propelled. The primitive and most natural type of ground propulsion is the use of muscle power but vehicles drawn by an animal have nearly disappeared now a days. With progress of technology, quest of higher velocity and advent of new systems, steam engines of James Watt ruled the ground propulsion till 1970s. Now the main focus is on (*i*) internal-combustion engines (*ii*) electric motors (which includes linear motors being part of the track) or combinations of those. Turbines are not used because of the small part load efficiency. Also other types with external combustion like the Stirling engine or without combustion like the fuel cell are only used in planes.

The propulsion requirement changes entirely, if medium changes to waterways. In marine propulsion, propulsion systems for ships and boats vary from the simple paddle to the largest diesel engines in the world or even nuclear propulsion. These systems fall into three categories: human propulsion, sailing, and mechanical propulsion. Human propulsion includes the pole, still widely used in marshy areas, rowing which was used even on large galleys, and the pedals, which are used for merry-drives in lakes. In modern times, human propulsion is found mainly on small boats or as auxiliary propulsion on sailboats. Propulsion by sail generally consists of a sail hoisted on an erect mast, supported by stays and spars and controlled by ropes. Sail systems were the dominant form of propulsion until the nineteenth century. They are now generally used for recreation and racing, although experimental sail systems, such as the kites/royals, turbo sails, rotor sails, wing sails, windmills and Skysail's own kite buoy-system have been used on larger modern vessels for fuel savings.

Air propulsion is the act of moving an object through the air. The most common types are propeller, jet engine, turboprop, ramjet, rocket propulsion, and, experimentally, scramjet, pulse jet, and pulse detonation engine. Animals such as birds and insects obtain propulsion by flapping their wings. This is also being imitated in making modern flying equipments like hovercrafts. A variety of propulsion concepts are used in air-propulsion. Basically, an airplane propulsion system must serve two purposes. First, the thrust from the propulsion system must balance the drag of the airplane when the airplane is cruising, and second, the thrust from the propulsion system must exceed the drag of the airplane for it to accelerate. In fact, the greater the difference between the thrust and the drag, called the excess thrust, the faster the airplane will accelerate.

Some aircraft, like airliners and cargo planes, spend most of their life in a cruise condition (constant altitude constant velocity flying). For these airplanes, excess thrust is not as important as high engine efficiency and low fuel usage. Since thrust depends on both the amount of gas moved and the velocity, we can generate high thrust by accelerating a large mass of gas by a small amount, or by accelerating a small mass of gas by a large amount. Because of the aerodynamic efficiency of propellers and fans, it is more fuel efficient to accelerate a large mass by a small amount. That is why we find high bypass fans and turboprops on cargo planes and airliners. Some aircraft, like fighter planes or experimental high speed aircraft require very high excess thrust to accelerate quickly and to overcome the high drag associated with high speeds. For these airplanes, engine efficiency is not as important as very high thrust. Modern military aircraft typically employ afterburners on a low bypass turbofan core. Future hypersonic aircraft will employ some type of ramjet or rocket propulsion. Spacecraft propulsion is any method used to accelerate spacecraft and artificial satellites. There are many different methods. Each method has drawbacks and advantages, and spacecraft propulsion is an active area of research. However, most spacecraft today are propelled by forcing a gas from the back/rear of the vehicle at very high speed through a supersonic de Laval nozzle. This sort of engine is called a rocket engine.

All current spacecraft use chemical rockets (bipropellant or solid-fuel) for launch, though some (such as the Pegasus rocket and Spaceship One) have used air-breathing engines on their first stage. Most satellites have simple reliable chemical thrusters (often monopropellant rockets) or resistojet rockets for orbital station-keeping and some use momentum wheels for attitude control. Soviet bloc satellites have used electric propulsion for decades, and newer Western geo-orbiting spacecraft are starting to use them for north south station keeping. Interplanetary vehicles mostly use chemical rockets as well, although a few have used ion thrusters and Hall Effect thrusters (two different types of electric propulsion) to great success.

Propulsion is an act of changing the motion of a body by either bringing it to motion from rest or changing a velocity or overcoming retarding forces during motion of a body through a media.

Invariably for propulsion, engines, motors, motion producing devices are needed. Depending on external environment, a large variety of engines are devised. Such engines can be classified on different parameters.

Depending on **application in real world**, IC engines can be classified as automobiles, truck, locomotive, light aircraft, and marine engines. Working principles of one or more may be same, but change of utility can give different types of engines. It is basically medium in which these engines are operational, makes them different from each other. Basically, all these overcome resistance and impart acceleration to the vehicle for transportation of a given payload (humans, cargo) from one place to other in economical way.

Depending on **power output**, compactness and mode of operations, these engines are classified in various ways. On the basis of cycle of operations, IC engines are of two classes. First is Otto cycle engines and second is Diesel cycle engines. Otto cycle engines are also called spark-ignition (SI) engines because they operate on petrol, where fuel and air is fed to engine and combustion is initiated by using a spark plug. Diesel cycle engines are called compression ignition (CI) engine. In this case, diesel acts as fuel. Only air is compressed in the engine and at the end of compression, fuel is injected, which ignites due to spontaneous combustion of fuel droplet in aggressive atmosphere (high temperature and pressure) of engine. Ignition in this case is achieved by compression of air alone and thus nomenclature of compression ignition is given to the system.

Depending on **fuel used**, SI engines are classified as gas engine or petrol engine. Out of these two, petrol engine can be classified in several ways. On the basis of **fuel introduction in the engine cylinder**, the engines are of two varieties – (*i*) carbureted type and (*ii*) injection type. Yet another method of classification is on the basis of **ignition process**. The SI engines may have battery ignition or magneto ignition. **Cooling provision** can also be basis for classification of SI engines and water-cooled or air-cooled engines are prevalent. In fact method of **arrangements for execution of various strokes** in an engine may give reciprocating engine or rotary engines. Multi cylinder reciprocating engines are classified on the basis of **cylinder arrangements** and may be opposed cylinder, inline, inclined cylinder, V engine, H engine etc. Rotary engines are classified as single rotor or two rotor systems.

Diesel cycle engines or CI engines are classified on the basis of **fuel system used**. It may be using diesel as working fluid in some cases, while in other cases a combination of diesel (liquid) fuel with gas-fuel is also utilized.

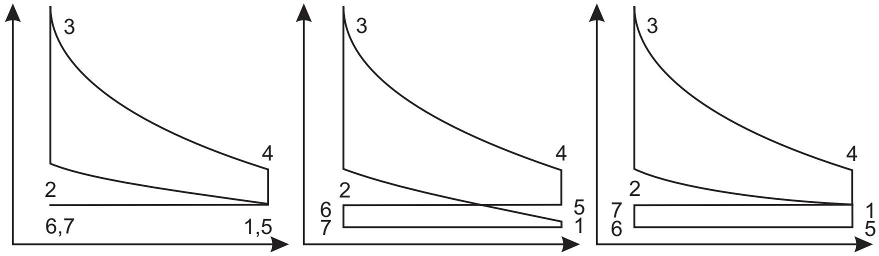
Another method of classification IC engines is based on **application of engines**. Automotive engines are used for land transport like scooter, motorcycles, cars, bus, truck etc. Propulsion of ship is executed by marine engines and that of aircrafts by aircraft engines. Power generation on land uses industrial engines. Based on application, IC engines may be classified as automobile engine, truck engine, Locomotive engine, Light aircraft engines, Portable power station etc.

Valve or port design and location is another basis for classification of IC engines. Overhead valves, under-head valves, rotary valves, cross-scavenged porting, loop-scavenged porting etc may be various types of IC engine based on this approach. Overhead design may have an I-head, while under-head design will be of L-type. In cross-scavenged porting, inlet and exhaust ports are located on opposite side of cylinder at one end, while in loop-scavenged type, they are located on same side of the cylinder.

Combustion chamber design is also a criterion for classification of IC engines. Open chamber engines, divided chamber engines and special chamber engines are operational. In open chamber disc, wedge, hemisphere, bowl-in-piston are popular designs. In divided chamber constructions, swirl chamber and prechamber designs are popular.

Whatever may be the nomenclature and type of IC engine, it invariably uses 4 operations. First stroke is called **suction stroke**, in which a vacuum is created inside engine cylinder by outward movement of the piston and fuel is sucked inside cylinder. Second stroke is **compression stroke**, where fluid inside cylinder is compressed by piston movement. Third stroke is **power stroke**. During this process, fuel is ignited and mechanical output is obtained as crankshafts. Final stroke is **exhaust stroke**, where products of combustion are thrown out of the cylinder and engine becomes ready for subsequent suction of fuel. This process was first developed by **Beau de Rochas in 1862** and it became part of all subsequent engine development as guiding principle. He advocated that maximum pressure should be achieved at the beginning of expansion and maximum expansion ratio should be aimed during construction of any engine. This led foundation for various subsequently developed engines. To minimize heat losses from the engines, largest possible cylinder volume with minimum boundary surface and rapid working is mandatory for such engines.

The complete cycle for a general internal combustion engine is shown in figure 3.1. First figure (*a*) represents a constant volume (Otto) cycle. Intake or Suction (6-7-1) is associated with entry of certain volume of air and/or fuel in the engine cylinder at low pressure. The low pressure is generally atmospheric pressure. This is associated with increase in volume (from V_6 to V_1) inside engine cylinder at constant pressure. Mass of charge also increases inside engine cylinder. Next process occurs in a confined chamber and is associated with reduction of volume (from V_1 to V_2). This is accompanied with realization of high pressure and temperature. At the end of this compression stroke, power stroke (2-3) occurs where combustion of fuel takes place. The heat addition process may occur at constant pressure, constant volume or a combination of both. In the figure 3.1a, it takes place (2 3) at constant volume. Work output is derived in the subsequent expansion stroke (3-4), where expansion of combustion gases takes place and useful work is derived from the cycle. Power generated during this stroke is stored in the flywheels or large rotary masses attached to the crankshaft of the reciprocating engines. Later on this stored power is utilized for execution of other strokes. After expansion, low pressure and temperature and high volume is realized (at condition 4). This is followed by exhaust of combustion gases (process 4-5-6), where combustion gases are thrown out and engine is ready for executing the cycle once again.



a. Constant Volume Cycle b. Throttled Cycle c. Supercharged cycle

Fig. 3.1 : Operation of Various Operational Cycles

In ideal situation, suction or intake and exhaust can occur at same pressure, but this balance is generally not maintained in the engines. In a throttled engine, pressure at the exhaust condition (5) is higher than atmospheric pressure. This condition results in flow of combustion-gases out of the cylinder automatically due to pressure differential. However, this gives a loss in the capability of the combustion gases. Energy that can be extracted from the expanding gases is not fully utilized and there is certain loss of efficiency due to throttled engines. This is also obvious from the negative loop created at the bottom of the cycle in figure 3.1b due to pressure mismatch.

Supercharging is one way to get rid of this efficiency loss and complete utilization of energy from the combustion gases is observed, if engines are supercharged (figure 3.1c). In case of supercharged engines, exhaust pressure is lower than intake pressure. It creates a positive loop of process thus adding to already derived power from normal cycle. However, additional attachments and devices are needed to execute supercharging. Exhaust in this case occurs at point ‘5’, which has pressure lower than intake pressure (point 1). Conversely, suction of air occurs in supercharged engine unaided due to pressure difference favoring suction thus reducing efforts in suction stroke. This reduces extraction of power from flywheel and more efficient utilization of combustion energy is observed in supercharged engines.

A reciprocating IC engine based on above cycle is described in figure 3.2. In reciprocating engines, piston moves back and forth inside a cylinder and transmits power or mechanical work to crank shaft or drive shaft through a connecting rod. Arrangement for sealing between cylinder and piston is ensured through provision of piston rings and cylinder liners. Internal diameter of cylinder is called bore and extent of longitudinal movement of piston in cylinder is called stroke. Volume traversed by the piston in traveling from one end to other end of the cylinder is called swept volume. Cylinder has a suitably shaped cylinder head, which closed end of the cylinder. Cylinder heads have provisions for valves, which includes both inlet and exhaust valves. Extreme position of the piston is described by dead centers. They are named so because at these positions reversal of direction of movement occurs for the piston and momentarily, piston comes to a stand-still position. Dead is indicator of stationary position of the piston at extreme positions. When piston is fully inside the cylinder, it is called top position and in this position, piston speed is zero. This position is also called **top dead center (TDC)** or **inner dead center (IDC)** position of the piston. At TDC, volume in cylinder is called **clearance volume**. Once piston movement starts downward from TDC, vacuum or low pressure is created above piston.

This causes opening of inlet valve and entry of fuel starts. This is called suction stroke and it continues till piston reaches bottom-most position called **bottom dead center (BDC)** or **outer dead center (ODC)**.

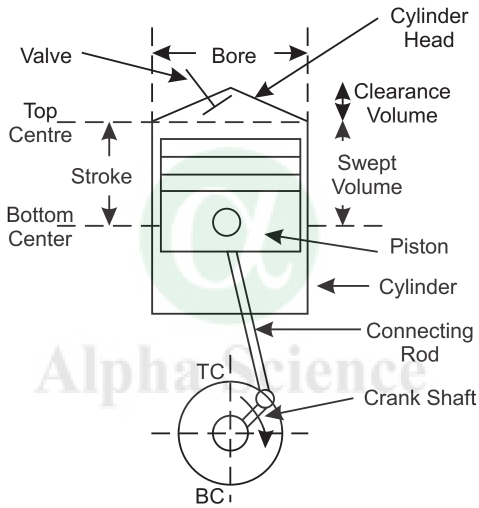


Fig. 3.2 : Operation of a Reciprocating Engine

After that inlet valve closes and piston movement in upward direction starts. This causes compression of accumulated mass above piston in the cylinder. This is compression stroke and near end of compression stroke, combustion process is initiated. This causes rapid rise in pressure and temperature inside cylinder. The rapid generation of high temperature gases, forces piston in downward direction, executing a power stroke. The work done by expanding gases is around 4-5 times work of compression. As piston comes in vicinity of BDC, exhaust valve opens and exhaust stroke starts. Due to inertia of crankshaft and connecting rod, piston pushes combustion gas out of the cylinder through exhaust valves. Again at the end of exhaust stroke, exhaust valve closes. Movement of piston in downward direction starts with opening of inlet valve and initiation of fresh suction stroke.

3.3 WORKING OF SPARK IGNITION ENGINES

Spark ignition engine operates on Otto cycle. In a typical spark ignition (SI) engine, air and fuel are mixed in intake manifold prior to their entry to the engine cylinder using carburetor or fuel injection system. For reliable combustion ratio of mass flow rate of air to that of fuel is maintained at around 15:1. Although cycle is explained earlier, SI engine is being described in terms of crank angle in figure 3.3.

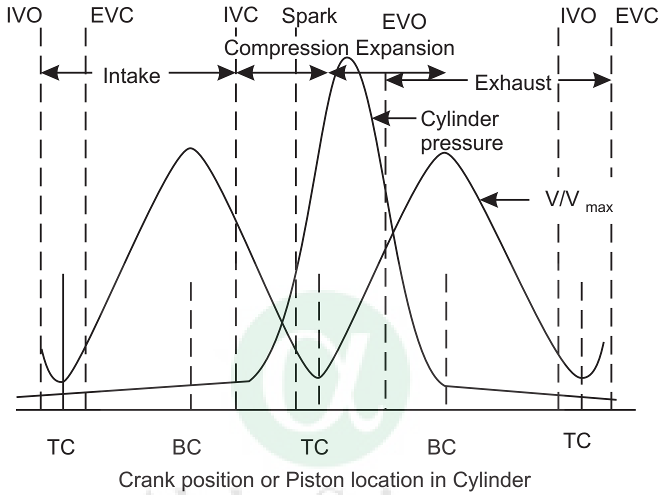


Fig. 3.3 : Working Principles of SI Engine

Since crank of a reciprocating engine has a periodic motion and same process is repeated in each cycle or period, operation of SI engine is illustrated in terms of piston location in cylinder or crank position. If rotation speed of crank is known, *x*-axis can be converted to time also. Variation of volume ratio with crank position is obvious. Maximum volume in the cycle is obtained at the beginning of suction stroke. This is equivalent to piston position at bottom center (BC). Volume ratio is 1 at BC. As piston approaches top center (TC), volume is reduced to clearance volume only and the value of volume ratio is reciprocal of compression ratio at TC. This is depicted by cyclic periodic variation of V/Vmax with crank position in figure 3.3.

To maintain sufficient fuel flow, inlet valve opening (IVO) takes place slightly before top center (TC) position of the piston. This takes care of sluggish opening of intake valve and by the time piston reaches top center position, valve is fully open. Similarly, to take advantage of inertia of flow of fuel, inlet valve closing (IVC) occurs substantially after bottom center (BC). Total duration during which intake valve is open is called intake stroke. During intake stroke, incoming mixture of fuel and air pushes combustion product out of cylinder and also mixes with them. During intake stroke pressure rise is almost negligible.

Once inlet valve closes, compression stroke starts. Between 10° to 40° crank angle, before TC, an electric discharge through spark plug starts the combustion. A turbulent flame develops from the spark-discharge and propagates across the mixture. This augments compression process by increasing volume of mixture. This leads to rapid rise in pressure as well as temperature. Flame starts from spark plug and terminates at the cylinder walls. The combustion process covers around 40° to 60° crank angle. Crank-angle at spark is ahead of TC and advancement of spark is dependent on several engine variables. If spark advanced is high, incomplete compression occurs and if it is less, it causes combustion related problems. However, spark advance is generally planned for maximum torque.

Exhaust valve opening starts at around two-third of the way through expansion. As cylinder pressure is higher than exhaust port and manifold pressure, discharge of combustion gases starts. Energy of expansion of gases is utilized for initial exhaust of gases and subsequently after piston reaches BC, piston motion also forces combustion gases out of the cylinder. Exhaust valve remains open just after TC and inlet opens just before TC.

Cycle efficiency for SI engines is discussed in chapter 1. In ideal air standard cycle, air standard efficiency is dependent on compression ratio alone. As compression ratio increase, air standard efficiency also increase for Otto cycle. For actual fuel-air cycle, the equivalence ratio or amount of available air with respect to actual air requirement also play a major role. Equivalence ratio is amount of fuel-air used to amount of chemical correct fuel-air ratio needed for complete combustion of fuel. As equivalence ratio is reduced below unity (lean mixture), the efficiency for a given compression ratio increases. This increase in efficiency by changed equivalence ratio is achieved due to changed value of ratio of specific heat. As amount of air is less than chemically correct quantity, burnt gas temperature realized after combustion is lower. This reduces specific heat of burnt gas and increases effective ratio of specific heats for the expansion stroke. For a given compression ratio, higher value of ratio of specific heats results in larger temperature difference over the process. For example, if compression ratio of SI engine is 8 and expansion starts from a temperature of 1000 K, two value of ratio of specific heats are considered – 1.35 and 1.4. Temperatures at the end of expansion are calculated to be 482.97 K and 435.27 K respectively. Definitely, temperature difference at the end of expansion is lower, if ratio of specific heats increases thus giving a larger temperature difference during expansion. As equivalent ratio is increased beyond 1, combustion mixture become more and richer and efficiency suffers due to lack of sufficient air for complete combustion. Decreased burned gas temperature and decreased specific heats of combustion gases gives not only reduction in efficiency but also in work output and energy released in combustion. Mean effective pressure for an ideal gas exhibits a maximum value in the range of 1 to 1.1 of equivalence ratio. For less than the given range of equivalent ratio, mass of fuel inducted is less, while at higher equivalence ratio reduced combustion efficiency (conversion efficiency) causes adverse effects on mean effective pressure.

Since combustion in SI engine takes place by spark plug, the energy available with arc created at spark plug point is important. The temperature in spark discharge is of the order of 10000°C. However, this should be viewed in conjunction with spark duration. Such high temperature leaves a thin line of flame as is seen in sky in lightning during heavy rain and thunderstorms. This evaporates fuel and air in the vicinity and subsequently expansion of flame front takes place. This is called preparation phase and is actually the time-delay between injections of fuel to creation of self-propagating flame front. It is desirable to have lower ignition delay so that enough time is available for combustion to complete in a fast moving engine. There are several factors which contribute significantly to the ignition lag. If fuel has a higher ignition temperature, ignition lag will be higher. Ignition delay is drastically reduced if fuel air ratio is tuned to give maximum temperature by working at slightly richer mixture. Higher temperature and pressure at the time of ignition also reduces ignition lag. Reduction in ignition lag is also possible by increasing compression ratio and retarding spark time. Electrode gap is important for the ignition delay. If spark plug gap is less, generated discharge in the gap is too small and despite giving a very high local temperature, the same is not sufficient to propagate through the mixture. There is certain minimum air gap needed between the spark plug points. Richer mixture to the extent up to an equivalent ratio of 1.4 can give a stable range of minimum spark plug gap for better combustion. Lower compression ratio needs higher spark plug gap. Minimum spark plug gap needed for higher compression ratio needs lower equivalente ratio. Conversely higher compression ratio works satisfactorily at relatively lower electrode gap and equivalente ratio.

The expansion of this small nucleus results in so called flame spreads through out the cylinder. The flame propagation is dependent on degree of turbulence present in the engine cylinder. Higher turbulence breaks continuous fire front into a ragged front thus increasing exposed area tremendously and increasing flame velocity. This gives lower combustion period. Ideally pressure in an SI engine should rise steadily during compression and when piston reaches top center, combustion should give a sudden jump in the pressure. This reduces isentropically during expansion stroke.

3.4 WORKING OF COMPRESSION IGNITION ENGINES

Compression ignition (CI) engines generally operate on Diesel Cycle. Contrary to SI engines, where mixture of fuel and air is injected, in CI engines, air alone is inducted into the cylinder. Fuel is injected just before the scheduled time of combustion. Air flow in the engine is virtually invariant and engine power is generally controlled by quantity of fuel supplied. In SI engine, quantity control of mixture is exercised to get different powers while in CI engine, quality of fuel-air mixture gives different loads.

Depending on type of inducted air, CI engines can be

- Naturally aspirated – induction of atmospheric air.
- Turbocharged – induction of compressed air by exhaust-driven turbine-compressor combination.
- Supercharged – induction of compressed air from mechanically driven pump or blower.

Turbo charging and supercharging is advantageous because it allows induction of more mass of air in same cylinder volume. This paves way for induction of more fuel and thereby increasing work output from the engine. However, these methods are suitable for large engines only for giving a compact design.

CI engines generally operate at compression ratio of 12 to 24 depending on type of fuel, type of air induction, cylinder head, power requirements etc. The valve timing of CI engine almost similar to that of SI engine as depicted in figure 3.3. Induction or suction stroke involves pure air at normal atmospheric condition. Inducted air is compressed to around 4 MPa pressure resulting in temperature rise to 800 K. Around 15-20° ahead of TC fuel injection starts. Because of high resistance of compressed air in the cylinder, the fuel is atomized.

Atomization is associated with vaporization and mixing with air simultaneously. Fuel is distributed indifferent concentration throughout the cylinder and fuel-air ratio is different locally. As ignition temperatures is achieved at several locations inside cylinder at the available fuel-air ratio, spontaneous ignition or auto-ignition takes place. However, this occurs after some delay. To compensate for this delay, fuel injection starts before TC. The flame spreads to entire mixture. This is followed by expansion process, but combustion continues during some part of expansion stroke also. This is followed by exhaust stroke. At the end of exhaust stroke, fresh cycle starts again.

CI engines are generally recognized by valves, positioned in the engines for inlet and exhaust operations. Although there is no standard criteria for valve timing, valve lift profiles and valve open areas, system design of engines are carried out with practical consideration to carry out intended operation with adequate efficiency. The valves may be a mechanical one or a hydraulic mechanism. Valve timing as shown in figure 3.3 are valid in this case also. Inlet valve opens 10° to 25° before piston reaches top center. Engine performance is not affected by this timing. But this should occur sufficiently before top centre, so that back flow during engine suction stroke is avoided. Inlet valve closes at 40° to 60° after piston reaches bottom center to take advantage of ram effect and provide more time to incoming gases to fill chamber. This affects volumetric efficiency of the engine. As far as exhaust valves are concerned, they are opened 50° to 60° before piston reaches bottom center in the power stroke so that exhaust process of gas can continue with the assistance of expanding gases. This affects expansion ratio in the engine and affects cycle efficiency also. Exhaust valve closes after piston reaches top center and creates larger pressure drop in the engine due to flow inertia of outgoing gases. Comparison of SI and CI engine is tabulated below.

Spark Ignition (SI) Engine	*Compression Ignition (CI) Engine*
It is based on Otto cycle.	It is based on Diesel Cycle.
Fuel used has high self-ignition temperature and it should not auto-ignite on compression.	Fuel used should have low self-ignition temperature, because it has to ignite by compression alone.
It needs a separate ignition system like spark plug.	It does not need separate ignition system.
A mixture of fuel and air is introduced in the cylinder. It needs a carburetor	Fuel is injected directly in the cylinder. It needs a fuel pump and injector.
The engine utilizes quantity governing.	The engine utilizes quality governing.
Control on acceleration through throttle valve is difficult.	Quantity governing gives best acceleration.
To control power, quantity of charge is controlled.	It is a quality control engine and amount of fuel injected is controlled to control power output.
Compression ratio is of the order of 6 to 11. Upper limit is knocking controlled. Limited efficiency.	Compression ratio is of the order of 14 to 22. Upper limit is engine strength controlled. Higher efficiency.
Supercharging is not very effective and is limited by knocking.	Supercharging is employed in CI engines.
It operates at less compression pressure (700 - 1500 kPa).	It operates at high compression pressure (3000 - 5000 kPa).
Higher maximum rotational speed (5000 rpm).	Lower maximum rotation speed (3000 rpm).

Spark Ignition (SI) Engine	*Compression Ignition (CI) Engine*
Light weight engine due to low compression ratios.	Heavy weight engine due to high operating pressures.
High exhaust gas temperature.	Low exhaust gas temperature.
Low cranking efforts in starting.	Difficult in starting.
Low weight per unit power (0.4 to 3.5 kg/hp).	High weight per unit power (2.5 to 10 kg/hp).
Higher power per unit displacement (40 hp / liter)	Lower power per unit displacement (20 hp/liter).
Fuel used is petrol, which is volatile and has high fire hazard.	Fuel used in diesel, which is less volatile and has less fire hazard.
Low initial cost but high running cost.	High initial cost but low running cost.
Less operating life.	Higher operating life.
Less noise and vibration.	More idle noise.
Emitted odour is tolerable.	Objectionable odour and smoke.
Air fuel ratio limited from 10 to 17. It has poor fuel economy.	Air fuel ratio may be 18-100. Higher compression ratio gives lower specific fuel consumption.

SI engines are generally preferred in automobiles and smaller airplanes. They are relatively lighter in weight and have better passenger comfort. Motor cycles, scooters, mopeds, motor boats, lawn movers, mobile generator sets, water pumps, air compressors etc utilizes SI engine.

CI engines have better fuel economy at both part load and full loads and less expensive fuel. It finds uses in heavy duty applications like buses, trucks, locomotives, bulldozers, tractors, earth moving machines etc. reduced fire hazard from fuel makes it ideal for marine use and confined environments.

3.5 WORKING OF 2-STROKE ENGINES

One movement of piston from any center to other center for example from top center to bottom center or bottom center to top center is called a stroke in the operation of a reciprocating engine. In 2-stroke engines, one power stroke is derived in 2 motions of the piston. This is equivalent to 1 revolution of the crank-shaft. Although, 2-stroke engines are available in both CI and SI classes, but SI engines are most preferred. It gives higher power and has simple valve design. Operation of a simple 2-stroke engine is given in figure 3.4.

The explanation of cycle starts when both the ports are closed by piston, required quantity of fuel is injected in the cylinder and piston is moving towards top center (TC). This is equivalent to compression stroke. When piston comes near TC, combustion is initiated depending on type of engine. In SI engines, spark plug is activated and in CI engine, fuel is injected. This process is also associated with suction of mixture (in SI engine) or air (in CI engine) in the crankcase through inlet valve.

On initiation inside cylinder chamber, expansion of exhaust product takes place and piston is forced towards bottom center (BC). This is power cum expansion stroke and mechanical work is done to rotate the crank. As piston moves downwards, first exhaust port opens and exhaust products are discharged. Expansion and exhaust takes place simultaneously. Piston top is suitably configured for proper scavenging of combustion gases. After some movement of piston, transfer port opens and fuel and/or air is injected in cylinder. The entry is deflected towards cylinder head and pushes combustion products towards exhaust port, thus augmenting scavenging. After reaching BC, upward movement of piston starts and compression stroke is executed after both the ports are open.

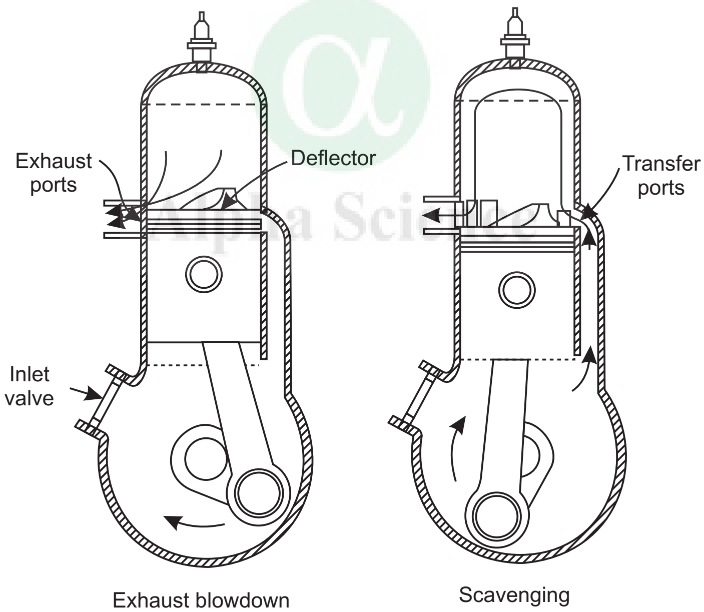


Fig. 3.4 : Working Principles of 2-Stroke Engine

In 2-stroke engines, power is derived in each rotation of crank. Since during operation, both inlet and outlet ports are opened for sometime, together; there are chances of unburned fuel directly moving to exhaust port. Additionally, dilution of fresh charge is always observed in the 2-stroke engines.

3.6 WORKING OF 4-STROKE ENGINES

In a 4-stroke engine, all processes are executed in well defined separate motions. The working principles of a typical 4-stroke engine are depicted in figure 3.5. In this case both intake and outlet ports are located near cylinder head. The description of process in a 4-stroke engine is initiated from intake or suction stroke. During this process, piston moves from TC to BC. When piston moves downwards from TC, suction head is created in cylinder near head. This causes inlet port to open and fresh charge is inducted in cylinder. After reaching BC, piston motion in upward direction starts. This raises pressure above piston resulting in closure of inlet port. With both the ports closed, piston moves upwards executing compression stroke.

At the end of compression stroke, when piston is nearing TC, combustion process is initiated. Combustion of fuel causes generation of large volume of gases at high temperature. This process forces piston towards BC, executing a power stroke. This is also associated with expansion of combustion gases inside cylinder. At the end of expansion, exhaust port opens and upward moving piston forces combustion gases out of the cylinder. To take advantage of inertia of motion, exhaust valve remain open slightly after TC is reached.

After piston reaches TC, it moves towards BC again, resulting in execution of suction stroke again. So in two revolution of crank or in four strokes of the piston one power stroke is derived. This cycle is called 4-stroke cycle. It has lower power per cycle but has high thermal efficiency.

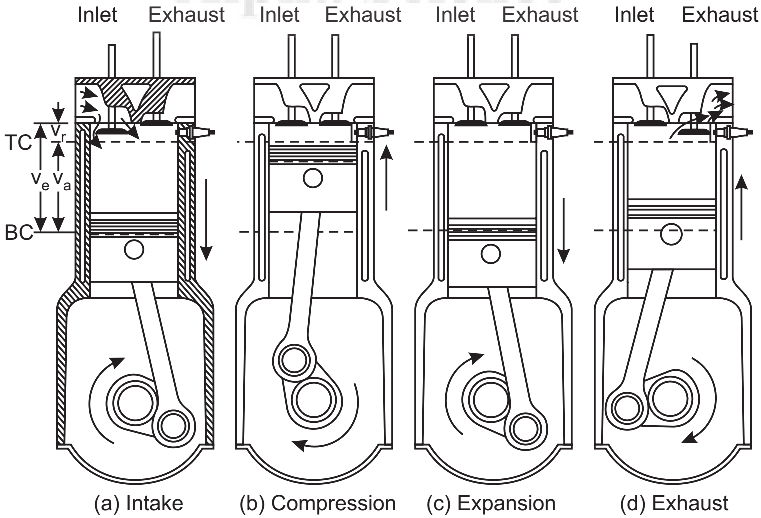


Fig. 3.5 : Working Principles of 4-Stroke Engine

To measure overall effectiveness of a 4-stroke cycle engine, volumetric efficiency (as defined in section 1.10) is used. It is equal to actual to ideal mass flow rate of the air. Ideal mass flow rate of air is obtained as product of density (at atmospheric conditions), swept volume (piston displacement) and effective number of revolution (half of rotation per minute for 4-stroke engines). This term is also used to measure the intake and exhaust systems as a pumping device. In SI engines, less air flow occurs due to restriction imposed by throttle valve. Volumetric efficiency is dependent on several parameters:

- Fuel type (chemical constituents, heat of vaporization, oxygen balance)
- Fuel: air ratio taken inside the cylinder of the engine
- Fraction of fuel vaporized in the intake system.
- Mixture temperature and engine heat transfer
- Ratio of exhaust to inlet manifold pressures
- Compression ratios
- Speed of the engines
- Design of intake and exhaust manifold
- Geometry, size, lift and timings of intake and exhaust valves.

Larger fuel-air ratio, lower exhaust pressure and larger compression ratio give lower volumetric efficiency. If fuel vaporizes in intake manifold, volumetric efficiency improves. During vaporization of fuel, heat is absorbed from the mixture and in absence of heat transfer to the inlet mixture of fuel and air, mixture temperature reduces. As fuel vaporizes partially in the inlet manifold, the deviation in volumetric efficiency is more due to reduction in partial pressure of air rather than due to cooling. As far as temperature is concerned, square root relation generally holds. Compression ratio affects clearance volume and thereby the amount of residual gases left in the cylinder at the end of each exhaust stroke. As exhaust to inlet gas temperature increases, volumetric efficiency reduces. Ram effect is another important concern and due to inertia of air during suction stroke, air continues to flow even after piston starts travel towards top dead center for compression stroke. If inlet valve is open in the initial part of piston movement for compression stroke, air more than cylinder volume can be sucked in the engine. This enhances volumetric efficiency of the engine. At higher engine speeds this effect is more pronounced. Similarly creation of pressure wave in inlet and exhaust manifolds due to valve action also affects actual amount of air taken inside the cylinder. In general volumetric efficiency of SI engine is lower than that for a diesel engine due to flow losses in carburetor and throttle (air flow not directly to engine cylinder), intake manifold heating, presence of fuel vapor (partial pressure of air lowers), and high residual gas fraction (lower compression ratio). With tuning diesel engines can maintain higher volumetric efficiency for a wider range of piston speeds.

For any engine, ideally volumetric efficiency should be 100%. Due to presence of fuel vapour, partial pressure of air reduces. Additionally there are certain other speed independent effects (quasi-static) which reduces volumetric efficiency. However, the reduction is uniform at all engine speeds. Another effect called charge heating in intake manifold affects volumetric efficiency at lower engine speeds and higher speeds it becomes less effective but measurable. At low engine speeds, longer time is available for heating and effect becomes dominant. Further reduction in volumetric efficiency is observed due frictional flow losses, which are less at low speeds but high with almost stable value at higher speeds. Practically frictional losses are proportional to square of the engine speeds. Further to this, at high engine speed, choking occurs and further enhancement in intake volume of air is not possible by higher pressure differential. Volume flow rate cannot be enhanced beyond certain level and volumetric efficiency reduces drastically at higher engine speeds. Ram effect takes advantage of inertia of flow of gases and enhances volumetric efficiency at higher engine speeds. However, at low engine speeds back flow reduces volumetric efficiency. If suitable tuning is applied, volumetric efficiency for a part of engine speeds can be enhanced. Finally, the curve for variation of volumetric efficiency with engine speed has a peak at certain speed-ranges and on both sides, the volumetric efficiency is lower.

2-stroke and 4-stroke engines are compared in the table below.

4-Stroke Engine	*2-Stroke Engine*
One power stroke in every two rotation of crankshaft.	One power stroke per rotation of the crankshaft.
Requirement of heavy flywheel to control non-uniform turning moment.	Requirement of lighter flywheel to control relatively uniform turning moment.
Power for same size of engine is less.	Power from same size of engine is more.
Less cooling and lubrication requirements. Less wear and tear.	Greater cooling and lubrication requirements. More wear and tear.
It has valve and valve mechanisms.	It has ports.
It has high initial cost	It is cheap as valves are absent.
High volumetric efficiency due to greater time of induction.	Low volumetric efficiency due to lower time of induction.
It has higher thermal efficiency.	It has lower thermal efficiency.
It has better part load efficiency.	It has lower part load efficiency.
There is no fuel loss as strokes are well separated.	Some unburned fuel is directly forced out through the exhaust port.

3.7 PARTS OF ENGINES AND THEIR MATERIALS

Piston and cylinder are two important parts of an IC engine. Engine cylinders are generally contained in an engine block, which are made of grey cast iron because of low cost and good wear resistance. Sometimes removable cylinder sleeves are pressed into the block for replacement on wear. Aluminium is being used for smaller SI engines blocks to reduce engine weight. However, for sleeves or cylinder liner, iron is general employed. Cylinder head seals off the cylinder and is made of cast iron or aluminium. The cylinder head contains spark plug, fuel injector, overhead valves and valve mechanisms.

Crankcase is generally an integral part of cylinder block. It houses crank shaft, crank and big and small end bearings. Crankshaft is made of steel forging. Bearings at both ends of crank are made of steel backed precision inserts with bronze, babbit or aluminium.

Piston is generally made of Aluminium in small engines and cast iron in larger engines. Piston seals the cylinder and transmits the pressure to the crank pin via connecting rod. Sealing is attained by piston rings which are made of silicon cast iron. The connecting rod is usually made of steel or alloy forgings. Material of construction and method of manufacturing various engine parts are tabulated below.

Sl. No.	*Name of the parts*	*Material of construction*	*Method of manufacturing*
1.	Cylinder	Cast iron, Alloy steel.	Casting
2.	Cylinder head	Cast iron, Aluminium alloy	Casting, Forming
3.	Piston	Cast iron, Aluminium alloy	Casting, Forging
4.	Piston rings	Silicon cast iron	Casting
5.	Valves	Specialty alloy steel	Forging
6.	Connecting rod	Steel	Forging
7.	Crank shaft	Alloy steel, SG iron	Forging
8.	Crank case	Aluminium alloy, steel, Cast iron	Casting
9.	Cylinder liner	Cast iron, Nickel alloy steel	Casting
10.	Bearing	White metal, Leaded bronze	Casting

It is important to understand and discuss reasons of deviation in performance of actual engines from a fuel-air cycle. As discussed earlier, fuel-air cycle is different from air standard cycle and for an Otto cycle, the efficiency is dependent on equivalence ratio also, in addition to compression ratio. Similarly some practical considerations are necessary to derive power efficiently from actual engines. One such operation is heat transfer through cylinder walls. All valves, and contact surfaces are assumed, in the ideal engines to be leak-proof and they allow mass transfer only when allowed. However, in real system, a leak-proof system is difficult to make and execute. During compression stroke, temperature levels are not very high and heat transfer to engine cylinder are not a major concern, but during expansion stroke after combustion process, temperature of expanding inside engine cylinder is very high. This result in unwanted heat transfer to the walls of the cylinder and walls inadvertently cools the engine content. This results in lower pressure than expected at the end of expansion and pressure always lags actual isentropic pressure at various expansion volumes. This reduces efficiency due to heat loss. Incomplete combustion not only reduces power output but also enhances pollution by introducing carbon monoxide and hydrogen in the combustion products. The combustion process is assumed to occur in infinitesimal-small time in the fuel-air cycle, but in real situation, for optimum efficiency, spark timing is adjusted to before arrival of piston to clearance volume level. Spark timing may be adjusted earlier to piston coming to top dead center, to a little after it proceeds towards expansion stroke. However, giving spark is not sufficient to ensure completion of combustion. Combustion always takes some finite time and burning process, which starts before piston reaches top dead center to slightly after piston retraces away from top dead center. Last part of combustion occurs at lower pressure and is generally sluggish. Peak pressure value in actual cycle is also smaller, but pressure during expansion stroke will be higher than in optimum timing cycle. At the end of expansion stroke, combustion products are sent out of cylinder.

This process also starts a little before piston arrival at bottom dead center to give gases sufficient time to go out. So gas pressure drops below isentropic line in exhaust stroke and work transfer during expansion stroke is further reduced.

For a normal diesel engine, various losses contribute to reduction in actual efficiency to around 30-40%. Combustion losses may be of the order of 20-25%, exhaust losses may be of the order of 12-18%, heat transfer losses to the cylinder walls may vary in almost equal proportion as exhaust losses. Aerodynamic and mechanical friction losses may account for a loss of around 5% each. In general, efficiency higher than this is never achieved from a practical engine.

3.8 COMBUSTION PROCESS

Despite difference in mode of initiation, combustion process remains same for SI and CI engines. Let us have a look at the combustion process in the two types of engines one after the other.

In SI engines, mixture of fuel and air is injected, compressed and then ignited by spark discharge. Under normal operating condition, at the end of compression stroke, flame develops, propagates through the mixture and extinguishes as soon as it reaches wall of the cylinders. The moment spark discharge takes place, for initial duration, marked rise in pressure is not observed because energy release from the flame is very small. As flame continues to grow, progressive heat addition causes deviation from normal isentropic compression curves. The pressure reaches maximum after TC, but before total charge in the cylinder is fully burned.

It is observed that flame development and subsequent propagation do not replicate in each cycle. The variation is observed in pressure, volume fraction, mass fraction of burned charged, volume fraction enflamed etc. Basically flame development depends on local mixture motion and composition, which cannot be made same in each cycle. Total combustion process in an SI engine can be classified under 4 heads–(*i*) Spark ignition (*ii*) Early flame development (*iii*) Flame propagation (*iv*) Flame termination. For maximum power or torque, combustion must be located on TC. Out of the four processes, second and third occurs between 30 to 90° of crank angle.

Combustion generally starts before end of compression stroke and ends after attainment of peak pressure. If start of combustion process is advanced progressively before TC, work transfer in compression stroke increases. Conversely, if end of combustion process is progressively delayed, the peak cylinder pressure occurs at later stage in expansion stroke and has lower magnitude. This may reduce work transfer from cylinder gas to piston. The maximum power situation is arrived when both these effects nullify each other. The optimum spark setting depends on rate of flame development and flame propagation, length of flame travel and flame termination process. It also depends on fuel ignitability.

Combustion process is meant for energy release and these are characterized by different values. One is flame development angle, which can be referred as ignition delay also. Since in SI engine, combustion is initiated by spark, there is virtually no delay. However, flame development angle is interval between spark discharge and the time when chemical energy of 10% fuel (10% mass fraction) has been released. After flame development angle, rapid burning angle is postulated. It is interval when bulk of the charge is consumed. Combination of both the angles is called overall burning angle.

Combustion mechanism in an SI engine takes place in turbulent flow field. The flow field is produced by high shear flows set up during the intake process and modified during compression. Turbulence enhances flame propagation speed and complete combustion process inside SI engine cylinder depends on charge motion, charge composition and chamber geometry. Flame front inside cylinder of SI engine is generally irregular; however it grows in spherical fashion with center at spark plug location. Flame front surface area is governed by geometry of combustion chamber and the spark plug location. Bigger flame front surface area allows entry of large quantity of mixture to flame zone and thereby rapid combustion process is ensured.

Burning rate is affected greatly by the composition, quantity and thermodynamic state of unburned mixture. If inlet pressure is reduced, burning will progress slowly resulting in increase of both flame development and rapid burning angles. Highest burning rate is achieved at slightly richer mixture due to non-homogenous mixing. As mixture becomes leaner, both the specified angle increases and burning rate reduces. If quantity of burned gas remains on higher side in the cylinder, burning rate reduces and both flame development and rapid burning angle increases. Similarly effects of chamber geometry, gas motion, gas composition, inlet conditions, fuel type, air quantity etc affect combustion process significantly in an SI engine.

Ignition is one of the critical requirements for initiation of flame in an SI engine. Spark between the electrodes creates high temperature plasma kernel, which gets transformed into self-sustaining and propagating flame front. Spark is generally created by application of sufficiently high voltage across electrode gap. As voltage is raised across the gap, breakdown of intervening medium occurs and ionizing plasma passes from one electrode to the other. The impedance of gap reduces when the plasma reaches other electrode and current through gap increases rapidly. This stage is called breakdown phase of discharge. This is followed by arc phase, where thin cylindrical plasma expands largely due to heat transfer and diffusion. This leads to exothermic reaction in the fuel-air mixture and visible glow is created. This phase is called glow discharge phase.

Breakdown phase precedes arc and glow phase. It creates electrically conducting path between the electrodes. It is a zero current phase. During this phase gas in the gap is fully dissociated and ionized. The arc phase is associated with low voltage and high current. In this phase dissociation may be on higher side but degree of ionization is much lower. Voltage drop is significant and current reduces gradually during the phase. As arc needs hot cathode spot, it is invariably associated with erosion of cathode material. Gas temperature in arc is limited to 6000 K due to heat conduction and mass diffusion.

Conventional ignition system includes coil ignition systems, which is battery operated and is used in automotive engines. This has reduced maintenance, extended spark plug life, improved ignition of lean and diluted mixture and increased reliability and life. For ignition of SI engine, if homogenous mixture is supplied, spark energy of the order of around 1 mJ only is needed and the spark should last for a few microseconds. Since air-fuel mixture is not homogenous throughout the cylinder volume, and presence of combustion gases may act as inhibitor to flame propagation, the energy requirement becomes higher than ideal energy. Variations in temperature, density, load, and engine speed also affect spark energy requirements and in most of the systems, spark energy of the order of 50 mJ and duration of 0.5 ms are taken as reliable figure of merit. The main requirements of ignition includes–(*i*) high ignition voltage to break down the gap between the plug electrode, (*ii*) low impedance of source and step (not ramp) voltage rise, (*iii*) high energy storage capacitor to ensure ignition, and (*iv*) sufficient duration of voltage pulse to ensure ignition.

Coil ignition system fulfills these requirements adequately. It is a breaker operated inductive ignition system. The system has a battery, switch ignition coil, distributor and spark plug, which are connected in series by suitable wiring. Under normal condition, when switch is on, current flows from battery through resistor (to increase voltage) to the primary winding of ignition coil. This sets up a magnetic field within the iron core of the coil. When ignition is needed, breaker point is opened by action of distributor cam (for providing spark to different cylinders). This interrupts primary current flow and decay in magnetic flux of the coil. This reduces voltage in both primary and secondary windings. The stepped up voltage in secondary winding is fed to spark plugs of different cylinders in sequence as per requirement. As primary current requires time to build up, at low speed time of contact is sufficient to build-up the maximum current. However, at higher speeds, current in primary winding may not reach its maximum. When circuit is open, no current flows through the primary winding and voltage across secondary winding comparable to battery potential is induced. If spark plug is not connected, induced voltage has damped variation of voltage. However, when spark plug is connected, voltage in the secondary coil will rise to breakdown potential of spark plug and discharge between the electrodes of the spark plug occurs. After spark, voltage reduces to lower value. This system has a major limitation. As engine speed increases, available voltage decreases. This is in fact limitation of current switching capability of breaker system. This decreases time available to build up energy storage in the primary coil. If there is some variation in the insulator of the spark plug or gap is not adequately tuned, performance of spark plug deteriorates. Due to high voltage, point of the spark plug is subjected to electrical wear leading to frequent maintenance. Further improvements in the form of transistorized coil ignition (TCI) or capacitive-discharge ignition (CDI) or magneto ignition systems are also observed. There are certain alternative ignition approaches like plasma jet ignition, flame jet ignition etc.

Spark plug is invariably used in SI engine and is in fact identification mark of the engine. Sometimes for better and quicker ignition of fuel inside engine cylinder two spark plugs are used. In old days, two spark plugs were used for redundancy purpose. If there is trouble of fouling of spark plug point, supply can be given to second spark plug and the same becomes active spark plug. However, with advent of digital technology, for larger vehicles, two spark plugs are of no used for efficient combustion. It is observed that because of very less time available for combustion of fuel in SI engine, combustion of fuel cannot be completed with single spark plug. So, two spark plugs at different locations in the engine are placed and ignited simultaneously. This results in two flame front from two different sides and consumes entire mixture in much less time. This improves engine efficiency by complete combustion of fuel in less time. A patent for use of this technology by Bajaj is effective in the name of Digital Twin Spark Ignition (DTSi) technology and is used in Pulsar class of vehicles.

Compression ignition engine has slightly different combustion mechanism. It does not have a spark plug for initiation of combustion. In this case, combustion is initiated by achievement of auto-ignition temperature for fuel air mixture locally, which spreads throughout the cylinder volume. In CI engine, fuel is injected in the compressed air at the end of compression stroke. Fuel is injected at high velocity through a small orifice, where atomization, vapourisation and mixing with compressed high temperature air takes place. As mixture is initiated, temperature rises and further compression of unburned mixture takes place. This reduces auto-ignition temperature of unburned mixture and ultimately, entire cylinder volume is ignited. In a typical CI engine, atomization, vaporization, mixing and combustion occurs simultaneously. Combustion in CI engine is highly complex and is affected by fuel, cylinder design, fuel-injection system and engine operating conditions. The entire process is three-dimensional, heterogeneous, unsteady, and transient and a quantitative analysis of combustion in CI engine is still not possible.

As injection starts before combustion, there is no knock limit in case of CI engine. In fact, combustion in CI engine starts at multiple points and is a knocking action in itself. To improve combustion efficiency, higher compression ratio can be adopted comfortably without any ignition problem. CI engine can be operated in unthrottled condition because airflow is unchanged and only control over fuel injection process is exercised. Part load mechanical efficiency of CI engine is better than SI engine, whose performance deteriorates considerably on part loads. Increase in fuel supply beyond limit for extracting more power invariably may result in formation of soot. So, generally lean mixture is used in CI engines due to improper mixing. Lean mixture results in unutilized air, in substantial quantity, during each cycle.

This results in higher value of specific heat ratio as compared to SI engine. Ultimately, fuel conversion efficiency is higher for the given expansion ratio in CI engine as compared to SI engine.

Commercial diesel engines are made in large ranges of cylinder sizes. Maximum piston speed at maximum rated power is constant over cylinder size. So, maximum rated engine speed is inversely proportional to the stroke. As engine size reduces, more vigorous air motion is needed because fuel penetration length needed in compressed air is on lower side.

For initiation of combustion in CI engine, injection system can be direct injection, where single open combustion chamber into which fuel in injected directly is realized. In larger engines, mixing requirements are not very critical and fuel injection is sufficient to achieve required degree of mixing and distribution of fuel with air. Direct injection is best choice for such situations. Combustion chamber in this case is a shallow bowl and central multi-hole injector is installed. As engine size decreases, air swirl gradually becomes important for better mixing. Air swirl can be achieved by forcing air towards cylinder axis.

Indirect injection engine is another class, where fuel is first injected into the pre-chamber which is connected to the main chamber via a nozzle. Indirect injection engines are used in smallest engines. Here, air swirl by injection is not sufficient to give required degree of mixing. During compression, air is forced from the main chamber into an auxiliary chamber, which has provision for rapid rotation. Fuel is injected into the auxiliary chamber at lower injection system pressure than direct injection systems. Combustion starts in the auxiliary chamber, the pressure rises and forces fluid back to main chamber, where the jet mixes with main chamber air. This is followed by further combustion of remaining mixture.

3.9 ABNORMAL COMBUSTION

Abnormal combustion process in engines results in loss of performance, stress on components, premature failures and abnormal power outputs. Abnormal combustion is generally associated with unexpected release of energy and it invariably result in rapid heat transfer and abnormal sound. In SI engines, two types of abnormal combustion behaviour are observed – Knock and surface ignition.

Knock has got its name from the type of sound created by engines, when under this abnormality. Normal flame propagation should occur from spark plug towards wall of the cylinder. However, as combustion proceeds, volume of gases increases, resulting in compression of unburned mixture inside cylinder. This causes rise in pressure and temperature of the unburned mixture. When auto ignition temperature is locally attained at any point in the unburned mixture, flame erupts at that point, before flame from spark plug reaches there. When this happens, mixture burns very rapidly towards end of the compression stroke resulting in high frequency oscillation inside cylinder, causing sharp metallic noise called knock.

When engine is operating for many numbers of cycles, valve, cylinder head and adjoining area becomes heated. Mixture in the vicinity of such heated components gets ignited without any stimuli from spark plug. Uncontrolled ignition is most evident but in many cases this may result in pre-ignition also. Although spark discharge is effective but it does not have any control over combustion process.

Knock is governed by temperature and pressure history of gas, rate of development of flame etc. If auto-ignition occurs repeatedly, the phenomenon is called spark-knock. It can be controlled by spark-advance. Advancing spark results in increase in severity and intensity of knock and vice-versa. Knock can be initiated by surface ignition and accordingly can be classified as knocking surface ignition and non-knocking surface ignition. Several terminologies are prevalent by combination of various effects of abnormal combustion.

Wild ping causes sharp cracking sound and is supposed to occur due to combustion of loose deposited particles at the inner surface of the cylinder. It disappears when particles are consumed and reappears on further deposition. For wild ping, these particles become local source of additional energy or flame creation and causes disturbances in the normal combustion process. Rumble is another phenomena associated with deposit caused surface ignition It is a relatively low- frequency noise (600 Hz – 1200 Hz) and is associated with surface ignition in high compression ratio engines. Rumble and knock can occur together.

Run-on is another abnormal combustion phenomenon in SI engines. This occurs when mixture inside cylinder continues to burn even after ignition system has been switched off. This occurs due to compression of fuel air mixture and role of surface ignition is negligible in this process. It also emits knock like sound. Heated spark plugs and valves may result in ignition and causes run away surface ignition. It is destructive type of surface ignition and can lead to severe heating and structural damage to the engine.

Amongst surface ignition pre-ignition is potentially the most damaging. If start of combustion is advanced by any process from the conditions of maximum torque delivery, amount of heat rejection increases. Burned gas pressure and temperature increases. Higher heat rejection advances pre ignition point further. Pre-ignition can also be initiated at the point of deposits build up, which causes thermal insulation and a heat reservoir for pre-ignition in absence of spark plug.

In SI engines, knocking is associated with fuel molecules characteristics. Parrafins as fuel show knocking tendency as their chain length increases. Addition of side chains results in decreased knocking tendency. In olefins, if number of double bonds is more, knocking tendency reduces. Exceptions to this rule, for olefins are acetylene, ethylene and propylene, which knock much more readily than the corresponding saturated hydrocarbons. Napthenes have higher knocking tendency than aromatics. Lengthening of side chain in basic ring increases knocking tendency while branching of side chains result in decreased knocking tendency.

In CI engine, major problem of combustion is associated with method to achieve rapid mixing between injected fuel and compressed air. In fact mixing rates control the fuel burning rate.

3.10 ARRANGEMENTS FOR MULTI-CYLINDER AIRCRAFT ENGINES

In the piston engine-propeller combination, production of hot gases by combustion and/or heat transfer takes place in separate unit and thrust generating element is separately placed. In practice, multiple-cylinder configurations are used. These systems utilize reciprocating engines almost similar to engines used in automobiles on the roads. However, flight versions of these engines are much lighter for the same delivered power. In the early part of twentieth century, the water cooled engine developed by the Wright Brothers delivered a power to engine mass ratio of 0.1 kW/kg. Now, most of the operational aircraft engines of piston engine-propeller combination have power to weight ratio in the range of 0.8 kW/kg to 1.2 kW/kg.

Another limitation put forth by such combination in reciprocating engine is requirement of volume. For flight-compliant engines, power to volume ratio should also be considered. Reciprocating engines becomes too long for higher power requirements and their volume rises. Typically power-to-volume ratio of reciprocating engines would be 0.3 to 0.5 kW/m^3.

In such multi-cylinder aircraft engines, several arrangements are envisaged. Most simple system is inline engines. All cylinder banks are arranged linearly and transmit power to a single shaft. It is popular in automobiles where 4 and 6 cylinder inline engines are quite common. Another type of engine has two cylinder banks (two inline engines) inclined at an angle to each other and with one crank shaft. Bigger automobiles use this type of cylinder arrangement.

For the smaller aircrafts, most popular design is opposed cylinder engines. It has two cylinder banks located in same plane on opposite sides of the crankshaft. It is virtually two inline cylinder bank arrangements or ‘V’- engine arrangements separated by 180° angle. It is well balanced and has the advantage of a single crankshaft.

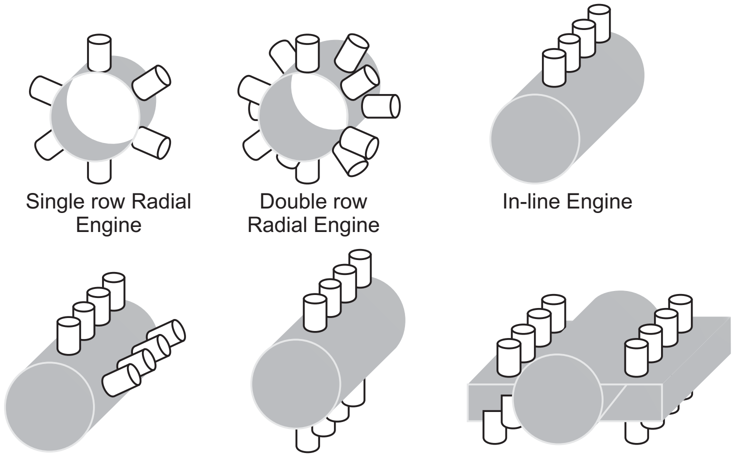


Fig. 3.6 : Arrangements of Cylinders in Multi-cylinder Reciprocating Engines

Radial engines have more than two cylinders in each row equally spaced around the crankshaft. It is most commonly used in air-cooled aircraft engines where three, five, seven or nine cylinders may be used in one bank and two or three banks can be used. An odd number of cylinders per bank is necessary with alternate cylinders firing in successive revolutions for 4-stroke cycle radial engines, but any number of cylinders can be used for 2-stroke engines. The radial engine presents the problem of fastening 3, 5, 7 or 9 connecting rods to a single crank. A master rod is guided by the crank and articulated rods are attached to the master rod. It should be noted that master rod executes the same motion as the connecting rod in other conventional engines, while articulated rod follows a slightly different path since the point of attachment is not at the center of the crankpin.

Similarly horizontal-opposed configuration, H-configuration, X-configuration or Y-configuration can also be adopted. Some of the arrangements are shown in figure 3.6.

3.11 INTAKE AND EXHAUST MANIFOLDS

In a reciprocating engine, flow of mixture or air through the engine is highly pulsating, the pressure variation takes place during movement of piston and both intake and exhaust manifold observes a time-varying fluid flow.

The Inlet Manifold carries the air-fuel mixture from the carburetor/fuel injector to the cylinders. The shape and size of the inlet manifold must be such that it should prohibit the formation of fuel droplets without restricting the flow of the air-fuel mixture. The manifold must be large enough to allow for sufficient flow for maximum power and yet it has to be small enough to maintain adequate flow velocities to keep the fuel droplets suspended within the air that flows through it. The Inlet manifold must not have any sharp bend because they tend to increase fuel separation. Rough interior surfaces within the manifold can increase resistance to airflow and hence must be made smooth.

The intake manifold of an SI engine has an air-filter, a carburetor and throttle in the intake port. During suction or induction phase of operation, mixture passes through each of the components and there is always some pressure loss in the flow through the system. Valves and intake ports are further source of pressure reduction. The pressure drop in intake manifold is dependent on engine speed, the flow resistance offered by walls of the flow passage, the cross-sectional area of the flow and the charge density.

The exhaust manifold is the set of pipes that lead the exhaust gases out of the combustion chamber then the upward traveling piston pushes it all out. The gases are all led out to the exhaust system from the cylinder. The exhaust manifold usually handles flow of gases very hot since they are coming in just after being burnt and hence have to be built to withstand that heat. For this reason, most of the exhaust manifolds are made of cast iron. Just like the inlet manifolds, they have their rib/cage arrangements to give them the much required structural rigidity. Since exhaust manifolds lead directly to the exhaust pipes at the rear of an automobile, the exhaust manifolds have to be designed keeping the space constraints in mind since the underside of the vehicle is precious. The exhaust side of a typical SI engine has exhaust pipe, a catalytic converter for emission control, a muffler or silencer.

During valve timing diagram of engine, it is observed that for some part of the crank angle, both intake and exhaust valves are open. The overlapping period of both the ports leads to flow of exhaust gases to the engine cylinder and that in the engine cylinder to inlet port. However, major advantage of valve overlap is realized in engines with high operational speeds. At high speeds, volumetric efficiency improves and more fresh charge than swept volume is inducted in the cylinder and effective scavenging of exhaust gases also takes place.

3.12 AIRCRAFT SI ENGINES

Aircraft SI engines are similar in construction as compared to engines. Only care is taken to reduce the weight of engine and reduce specific fuel consumption. Aircraft engines are almost always either light weight piston engines or gas turbines.

An aircraft engine must possess certain features over and above automobile internal combustion engines. They should be :

- Light weight, as a heavy engine increases the empty weight of the aircraft and reduces its payload. Light weight makes engine flight worthy.
- Powerful, to overcome the weight and drag of the aircraft. The engine has to develop high acceleration and velocity on ground during take off and maintain drag and lift combination during flight.
- Small and easily streamlined; large engines with substantial surface area, when installed, create too much drag. Aircraft engines must be small and should have preferably an aerodynamic shape for reduced skin friction drag.
- Reliable, as losing power in an airplane is a substantially greater problem than in an automobile. Aircraft engines operate at temperature, pressure, and speed extremes, and therefore need to perform reliably and safely under all reasonable conditions.
- Field repairable, to keep the cost of replacement down. Minor repairs should be relatively inexpensive and possible outside of specialized shops.
- Fuel efficient to give the aircraft the range the design requires.
- Capable of operating at sufficient altitude for the aircraft.

The design of aircraft engines tends to favor reliability over performance. Long engine operation times and high power settings, combined with the requirement for high-reliability mean that engines must be constructed to support this type of operation with ease. Aircraft engines tend to use the simplest parts possible and include two sets of anything needed for reliability. Independence of function lessens the likelihood of a single malfunction causing an entire engine to fail.

Aircraft spend the vast majority of their time traveling at high speed. This allows an aircraft engine to be air-cooled, as opposed to requiring a radiator. With the absence of a radiator, aircraft engines can boast of lower weight and less complexity. The amount of air flow an engine receives is usually carefully designed according to expected speed and altitude of the aircraft in order to maintain the engine at the optimal temperature.

3.12.1 Ignition Systems

Ignition system of automobile engines relies on coil ignition system and power is extracted from battery. Since it is difficult to arrange a battery on aircraft because it adds extra weight on flight worthy configuration, magneto ignition system is utilized in aircrafts. Magneto replaces all components of the coil ignition system except spark plug. It is lighter and more compact than battery ignition system and is preferred choice for aircrafts. A schematic magneto ignition system is shown in figure 3.7. It has a magneto, breaker point's condenser, ignition switch, distributor and spark plug leads. Magneto replaces battery and ignition coil.

A soft iron core is placed through an annular permanent magnet. This magnet is rotated by engine. This causes reversal of direction of magnetic flux through the iron core of coil due to closed breaker point. The magnetic flux alternately builds up and breaksdown resulting in generation of current in primary and secondary coils. Normally the voltage is not sufficient to jump over the gap over spark plug. For generation of high voltage rapid breaker mechanism for magnetic flux is needed. This is achieved by breaker point, which is generally operated by cam. Bringing condenser in line causes increase in voltage. When breaker point is open, condenser is charged. Rapid discharge occurs almost instantaneously to produce almost instantaneous breakdown of magnetic flux. This rapid breakdown causes very high voltage of the order of 10000 to 20000 volts in secondary winding. This high voltage is fed to spark plug through suitable wiring to create spark. In this case, timing of various events like breaking of primary circuit, the motion of the rotor and rotation of distributor cam are controlled for firing of different cylinders appropriately. When ignition switch is closed the primary circuit prevents generation of high voltage. This earths primary circuit and engine stops.

Magneto is rotated by the engine and is capable of very high voltage and it does not need any battery as source of external energy. In the magneto system, spark discharge is independent of battery or generator. They are commonly used in small 4-stroke or 2-stroke engines. In this case, a permanent magnet is rotated, which generates current in the closed primary winding. This generates additional flux, which interacts with the flux of the permanent magnet.

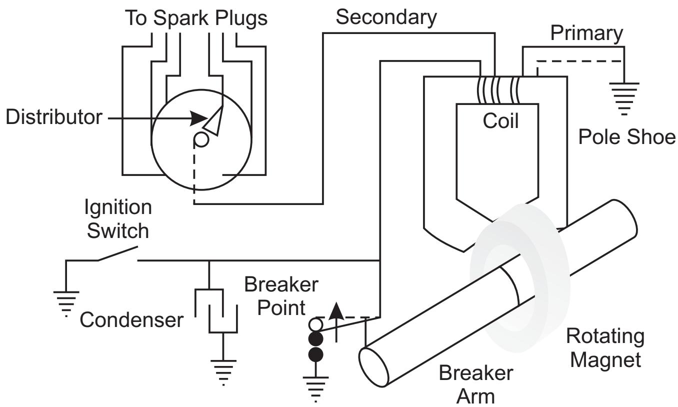


Fig. 3.7 : Magneto Ignition System

Magneto system depends on cranking speed. Magneto system gives a lower current at low engine speeds but at higher speeds, it gives higher current as compared to coil igniters. So, magneto system will always be suffering from starting difficulty and a separate battery is needed for starting. However, at high speeds, it is very reliable. That is why; it is widely used in sports cars, racing cars and aircraft engines. This system is costly but highly reliable. It needs less maintenance and is lighter than coil ignition systems. Aircraft reciprocating engines have two independent magneto ignition systems, and the engine's mechanical engine-driven fuel pump is always backed-up by an electric pump.

3.12.2 Effect of Altitude and Speed

Aircraft operates at higher altitudes where the air is less dense than at ground level. As engines need oxygen to burn fuel, a forced induction system such as turbocharger or supercharger is especially appropriate for aircraft use. This does bring along the usual drawbacks of additional cost, weight and complexity.

As altitude of operation for an aircraft SI engine is higher, intake air density reduces. So, carburetor of an SI engine designed to deliver correct ratio of fuel to air will provide a richer mixture at higher altitudes. Fuel-air ratio (mass based) is proportional to square root of fuel-air density ratios. For example, at 7000 meters of altitude, air density becomes half of that at ground level. So air-fuel ratio becomes around $1/\sqrt{2}$ *i.e.* 0.707 times the value at the ground level and mixture becomes around 40% ($\sqrt{2} - 1$) richer. So it is necessary to place altitude mixture correction device in the aircraft carburetors. This device progressively reduces the amount of fuel with altitude. Carburetor is a device in SI engine, which meters and supplies fuel-air ratio as per instantaneous requirements. However, it is dependent on air volume flow rate, which in turn governs fuel flow. Automatic fuel richness of mixture at high altitude due to reduction in air density is obvious. At 1500 m altitude, air pressure is around 83% of that at mean sea level. Additionally pressure variation as per weather is also significant. Air-fuel ratio varies as square root of density of air. The curve depicted in figure 6.2 can be modified for high altitude and entire curve will shift towards enriched mixture.

A number of methods are prevalent for altitude correction in aircraft SI engine. Sometimes a bypass venturi system is incorporated, which supplies additional volume of air at higher altitudes to offset fuel richness. An auxiliary air pressure feedback system is placed in the fuel line to reduce fuel flow at reduced pressure of air.

As normal aircrafts operation takes place through various maneuvers resulting in much higher tilt in the aircraft engines, simple hydrostatic float system of carburetors is highly inadequate for such operations. Special provisions are made to keep float chamber or equivalent reservoir full of fuel during all such pitch, yaw and roll motions.

At higher altitudes formation of ice in choke tube and on throttle valve is always observed due to low temperature of incoming air. An automatic deicing unit is placed to take care of such eventualities.

High speed invariably depends on higher fuel requirements. It needs richer mixture to augment power. This is effectively done by special devices, which enriches mixture and they are mandatory part of aircraft SI engines.

EXAMPLE 3.1

Determine the air fuel ratio supplied at 5000 meter altitude by a carburetor, which is adjusted to give an air-fuel ratio of 14:1 at sea level where air temperature is 27°C and pressure is 1.03 kg/cm^2. The temperature of air decreases with altitude as given by expression $T = T_s - 0.0065 \times h$, where, T = temperature at height h (in meters) in °C and T_s = temperature at sea level in °C. The air pressure reduces with altitude as per relation $h = 19200 \times \log_{10}(p_s/p)$, where p_s = pressure at sea level in kg/cm^2, p = pressure at height 'h' (in m) in kg/cm^2.

SOLUTION

Given h = 5000 m, p_s = 1.03 kg/cm^2, T_s = 27°C = 300 K.

$$(A : F)_s = 14.$$

From the temperature formula, T = – 5.5°C = 267.5 K.

From the pressure formula, p = 0.5655 kg/cm^2.

It is known that fuel density does not change with altitude but air density varies and air-fuel ratio is proportional to square root of density of air.

$$(A : F)/(A : F)_s = \sqrt{\rho} / \sqrt{\rho_s}.$$

Assuming behaviour of air as that of an ideal gas $p/\rho T$ = constant.

$$(A : F)/(A : F)_s = \sqrt{p} \times \sqrt{T_s} / \sqrt{p_s} \times \sqrt{T}.$$

So, air-fuel ratio at given altitude, (A:F) = 10.98.

3.12.3 Power Required and Power Available

Unlike automobile engines, aircraft engines are often operated at high power settings for extended periods of time. In general, the engine runs at maximum power for a few minutes during taking off, then power is slightly reduced for climb and then spends the majority of its time at a cruise setting–typically 65% to 75% of full power. In contrast, an automobile engine might spend 20% of its time at 65% power while accelerating, followed by 80% of its time at 20% power while cruising. The power of an internal combustion reciprocating engine is rated in units of power delivered to the propeller (typically horsepower) which is torque multiplied by crankshaft revolutions per minute (RPM). The propeller converts the engine power to thrust horsepower or thp in which the thrust is a function of the blade pitch of the propeller relative to the velocity of the aircraft.

The power output of an engine depends on amount of air intake, the degree of utilization of air and the thermal efficiency of the engine. The air intake can be increased by two methods. First is by increasing engine speed. If speed is increased, inertia load on engine increases and engine becomes robust and rigid. The engine friction and bearing load also increase. Volumetric efficiency is also adversely affected by engine speed. So this method is seldom applied for high power engines. Second way to enhance amount of intake air per unit time is by supercharging. In this case, air is supplied at a higher pressure than pressure of a naturally aspirated engine.

The power output of engine can also be increased by increasing the thermal efficiency of engine through increased compression ratio. This is limited by engine robustness and is invariably associated with maximum cylinder pressure. But supercharging may offset rise of maximum cylinder pressure by increasing rate of brake mean effective pressure. Supercharging leads to lower maximum temperature also. This results in lower structural and thermal loads on supercharged engines.

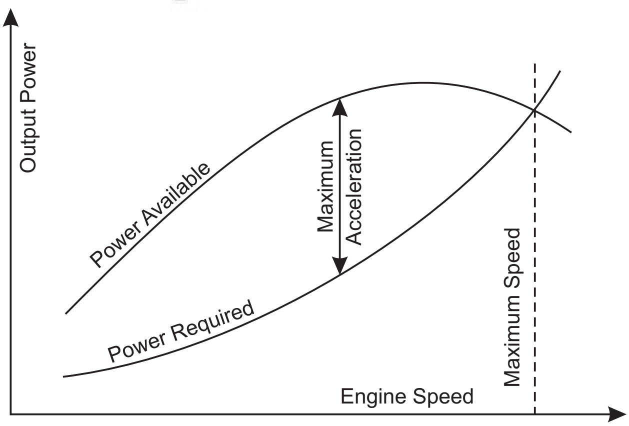


Fig. 3.8 : Power available and Power Required Curves

Power requirement of an engine rises with increase in engine speed. The rate of rise is higher at higher speeds. Contrary to this, power available also increases with speed, but it is limited by energy available from combustion of fuel, conversion efficiency and mechanical efficiency. The variation is shown in figure 3.8. It is clear that maximum speed will be obtained where both the curves intersect. Beyond this point, power available is less than power requirements and further enhancement in speed is not possible. Similarly, maximum acceleration is possible, when difference between both the curves is highest.

3.12.4 Supercharging

Supercharging is method of inducting large quantity of air per unit time, so as to burn more fuel in a given engine dimensions to increase power output. Supercharging action can have three actions. It can be employed to get more power fro a given weight and bulk of the engine. This is utilized for aircraft, marine and automotive engines, if weight and space constraint are predominant. Aircraft engines loose power at the rate of 1% per 100 meters altitude. Supercharging can compensate for the loss of power due to altitude. Lastly supercharging can help in obtaining more power from an existing engine.

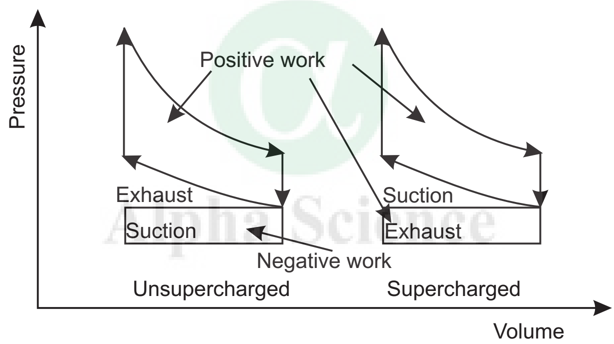


Fig. 3.9 : Thermodynamic Cycle for Supercharged Engines

A typical pressure-volume diagram for supercharged SI engine is shown in figure 3.9. In naturally aspirated unsupercharged engines, exhaust occurs at higher pressure than suction pressure resulting in negative work loop. However, in supercharged engines, suction pressure is on higher side as compared to exhaust pressure. This makes this loop with positive work and extra work output is obtained due to supercharging. However, gain in output energy is mainly due to amount of air induced for the same swept volume. Additional amount of air is also induced due to compression of residual volume to a higher pressure. Supercharging increases mechanical efficiency.

In SI engine, supercharging is employed for aircraft and racing car engines only, because increase in supercharging pressure increases tendency to detonate and pre-ignite. Increase in supercharging pressure increases intake pressure, temperature and flame speed. All these effects result in detonation and limits supercharged SI engine to lower compression ratios, only. Detonation in supercharged SI engine can be controlled by injection of water in the combustion chamber. Inter-cooling of charge is another alternative method to control detonation.

High density means higher value of specific heats and dissociation losses at higher temperatures result in lower thermal efficiency. This leads to higher fuel consumption of supercharged engine as compared to naturally aspirated engines. Increasing flame speed means more sensitive engine to fuel-air ratio. If lean mixture is supplied, engine knocking becomes an essential feature. To offset this effect, rich mixtures are generally supplied. This further enhances fuel consumption of supercharged engines.

In general, supercharging of SI engine is limited to 30% to 50% only. Supercharger pressure of the pressure of 1.3 to 1.5 kg/cm^2 is generally employed.

3.12.5 Types of Superchargers

Various types of superchargers are employed in SI engines. It can be reciprocating compressor, rotary blower, roots type blower, centrifugal compressor, turbochargers. Reciprocating compressors are generally employed for stationary installations of internal combustion engines, because they are heavy and bulky. Because of positive compression, it has higher pressure ratio. Vane type rotary blower is another popular alternative for supercharging. In a circular casing with inlet and outlet ports, an eccentric rotor is placed. The rotor contains multiple radial vanes, which slides in recess

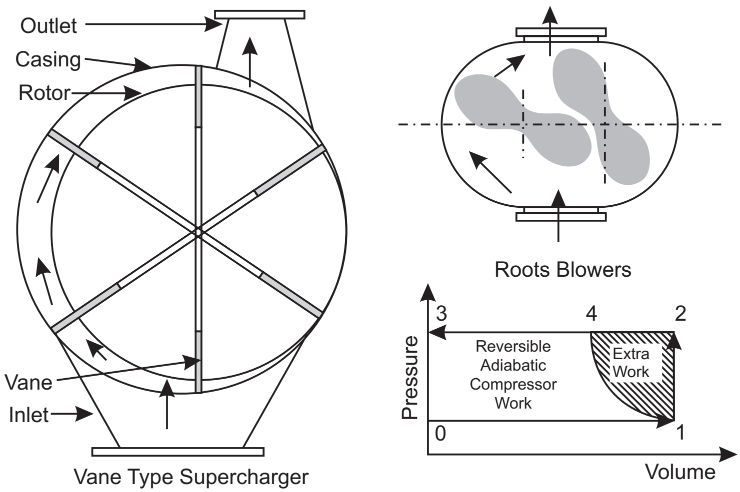


Fig. 3.10 : Vane type Supercharger and Roots Blower

in radial outward and inward direction. Air is inducted in the space between rotor and casing. As the space changes, more air is inducted in the space. Once vanes rotate so as to cut-off supply from inlet port, the space continues to decrease. This causes compression and compressed gas is diverted to outlet port, which is connected to engine cylinder. A schematic is given in figure 3.10. This supercharger gives pulsating and noisy flow and this type of supercharger is also obsolete now.

Another variety of positive displacement rotary supercharger called Lysholm compressor. This structure has helical rotor and the casing. The screw action of rotor displaces the air axially. This compressor produces a constant compression. Although it is mechanical complex but air is compressed throughout the screw.

Roots blower (figure 3.10) is another variety, where two double dumbbells shaped lobes rotate in opposite direction inside a common casing. Air enters through inlet port and is carried around the rotors to the outlet port. There is no pressure rise and the process is shown by 0-1 in figure 3.10. When discharge port is opened, pressure rises almost instantaneously (process 1-4). When outlet port opens, extra work is required to compress air due to back flow. Leakage between lobe and casing creates noise. Number of lobes can be increased to reduce noise. These types of superchargers are suitable for low and medium speed engines for stationary and marine installations. It has low cost, simple design, good mechanical efficiency and lubrication-free operation. These roots blowers are suitable for pressure ratio of 1.1 to 2.0.

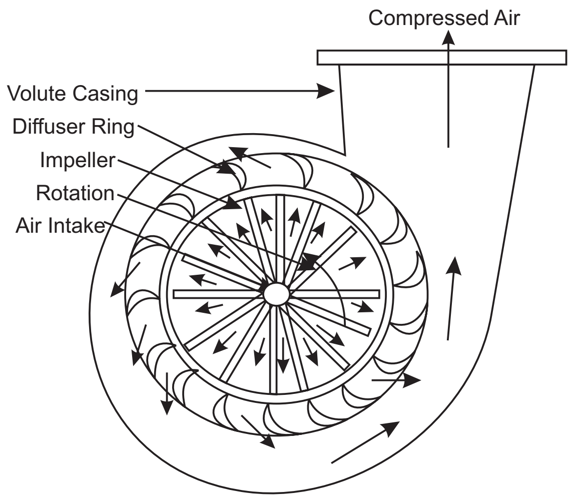


Fig. 3.11 : Centrifugal Compressor for Supercharger

For aircraft operation, centrifugal compressors are mostly used. It has an impeller rotating in a close-fitted casing. The air enters the center of volute casing axially and turns at right angle in radial direction.

Due to high velocity of rotating impeller, pressure rises. The centrifugal compressor runs at very high speed of 10000 to 15000 rpm for low speed engines and 15000 to 30000 rpm at high speed engines. This engine is simple, small, and cheap and has a good efficiency in the pressure range of 1.5 to 3.0. About 70-80% isentropic efficiency is obtained at a pressure ratio of 2:1. Here pressure ratio varies with the square of the speed.

When volume flow changes, pressure ratio remains constant in the high efficient range. Because of constant speed, it is suitable for aircraft applications. The schematic diagram of centrifugal compressor is given in figure 3.11.

The compressor has a volute casing where flow passage is variable. It has a rotating impeller, which resembles vanes of vane type supercharger. Rotation of these vanes creates suction head at the center, from where air is sucked. Sucked air is thrown in radial outward direction through rotating impeller and is directed towards stationary diffuser ring. During this, air is compressed. The compression is augmented by increasing flow passage. Volute casing directs air towards outlet port.

SUMMARY

This chapter gives a brief account of operational internal combustion engines. The chapter gives an idea about spark ignition and compression ignition engines and explains their working principles and salient features. It compares them also. Both 2-stroke and 4-stroke engines are explained in detail and their differences are also highlighted. Combustion mechanism in all types of engines are explained and combustion abnormalities like knock, wild ping, surface ignition, auto-ignition are deliberated and methods to prevent them are also elaborated. The chapter has a brief description of of aircraft SI engines and their critical requirements. Altitude compensation, speed and power requirements are deliberated. The method of feeding extra mass of air in same cylinder volume by supercharging is explained and their utility is established in this Chapter.

QUESTIONS

1. What are general uses of internal combustion engine? Why is it called internal combustion engine?
2. What are various methods of classification of internal combustion engines?
3. Explain operation of a simple reciprocating internal combustion engine?
4. Explain various components of a reciprocating internal combustion engine?

5. Explain operation of a spark ignition engine by valve timing diagram?
6. What is turbocharging? Why is it employed?
7. Explain operation of a 2-stroke engine?
8. What is role of crankcase in 2-stroke engine?
9. What is material of construction of different parts of the reciprocating engine?
10. Explain combustion process in a spark ignition engine?
11. How combustion takes place in compression ignition engine?
12. What are factors affecting ignition and combustion in an IC engine?
13. What is knocking? What is its dimension in SI and CI engine?
14. What are results of abnormal combustion in the internal combustion engine?
15. Why multiple cylinders are installed in cars?
16. What is opposed cylinder engine? How is it different from opposed piston engines?
17. What are critical requirements in an aircraft engine? How are they addressed by internal combustion engine?
18. What are various ignition systems in an internal combustion engine?
19. What are effects of altitude on operation of an internal combustion engine?
20. What are devices installed in internal combustion engines to offsets its limitations while operating at high altitudes?
21. Why simple float based carburetors ineffective in aircrafts? What are arrangements made in carburetor for their effective operation in aircrafts?
22. Why is de-icing unit placed in aircraft fuel intake line?
23. What are effects of speed on operation of an aircraft running on internal combustion engine?
24. How does balance of power required and power available made in internal combustion engine of an aircraft?
25. What is purpose of supercharging?
26. Explain operation of a centrifugal compressor as supercharger of internal combustion engine?
27. Explain operation of a positive displacement rotary supercharger?
28. Differentiate between the following
    (*a*) Internal combustion engine and External combustion engine.
    (*b*) Top dead center and Bottom dead center.
    (*c*) Turbocharged engine and supercharged engine.
    (*d*) 2-stroke engine and 4-stroke engine.
    (*e*) Compression ignition engine and Spark ignition engine.
    (*f*) Coil ignition and Magneto ignition.

29. Write short notes on the following:

(*a*) Rotary internal combustion engine

(*b*) Ignition mechanism in IC engine

(*c*) Cylinder arrangements in reciprocating engines

(*d*) Radial engines

(*e*) Intake and exhaust manifold

(*f*) Supercharging

(*g*) Air SI engine

CHAPTER

4

Engine Performance

STRUCTURE

- Introduction
- Objective
- IHP
- BHP or SHP
- FHP
- Corrections Factors
- Variable Speed and Constant Speed Test
- Summary
- Questions

4.1 INTRODUCTION

Although theories to extract mechanical work from chemical energy of fuel is a well known fact, but making a practical engine, implementing the principles of thermodynamic cycles is a major challenge. Once engines are developed certain performance parameters are fixed, which are used as guiding criteria for their comparison. For conventional two-wheelers, specific fuel consumption or petrol consumption are specified as performance parameter and is advertised also. On similar lines any new development is always tested for performance. This chapter is dedicated to testing and measurement of performance parameters of engines.

Objective:

After studying this Chapter, you should be able to understand

- Engine performance parameters.
- Indicated horse power calculation.
- Measurement methods for BHP.

- Various test method for comparison of engines.
- Performance parameters evaluation for internal combustion engines.

4.2 IHP

Indicated horse power (IHP) is the maximum ideal power available as mechanical work in a cycle. It is measured by two methods. First is by using indicator diagram and second is by measuring BHP and FHP and adding them. Second method is an indirect method for IHP measurement. First method is generally employed for the measurement of IHP. This is an indicator of how effectively heat is converted to mechanical work. In addition, rate of pressure rise, ignition lag etc are also indicated clearly on indicator diagram. Two types of indicator diagram exist. First is pressure-volume diagram and second is pressure-crank angle plot. Both are mutually convertible and can be traced on paper with some scale factor.

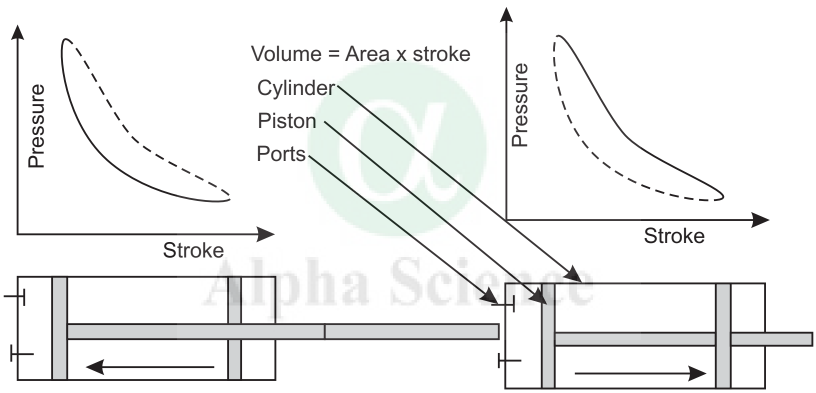


Fig. 4.1 : Indicator Diagram for a Reciprocating Engine

On a tracing paper, pencil is placed and movement of the piston is traced along one rectangular axis of the indicator diagram. An indicator piston cylinder is also attached in such a way that pressure in the cylinder is transferred to indicator cylinder. Arrangements are made to move pencil along another rectangular axis to indicate pressure levels achieved in the cylinder. Before actual tracing, a pressure reference line is drawn on the paper to indicate atmospheric pressure. A sample trace for both inward and outward motion of the piston is depicted in figure 4.1. The area inside the loop created is proportional to work done in one cycle. Mean effective pressure (mep) for the indicated diagram is used to calculate indicated horsepower.

IHP = mep × A × L × N, where A = bore area, L = length or stroke of piston, N = rotational speed of the engine. The units of each of the parameters are to be taken correctly to give output in horsepower.

For generating indicator diagram for measurement of indicated horse-power, several systems are conceived. Piston type of indicators uses a three way cock, which can be connected to either atmosphere or cylinder. When this is connected to cylinder, pressure forces a tension spring and movement of a stylus parallel to drum is proportional to cylinder pressure. The drum is rotated with the help of a card by the reduced link to depict piston displacement motion. This piston type indicator arrangement is simple but has several limitations. The inertia of piston, piston linkage and reducing link are quite high and it gives a lag between cylinder pressure and movement of stylus. This system is not accurate for smaller engines. Severe pressure pulsation at high speed is also observed. Usage of this system for diesel engines is restricted due to presence of carbon particles in passage. However, some of the errors can be nullified by using stiffer springs, by lighter indicator parts and by reduced drum travel.

Balance diaphragm type indicator is attempted to develop a low inertia indicator. In this case a pressure pick-up device, a small disc valve and a diaphragm is attached to the engine cylinder.

The disc valve is free to move between two seats. Diaphragm separates an external pressure and engine pressure. When pressures are unequal, disc sits on any one of the two seats. The movement of disc is connected to an electric circuit, which makes and breaks on each contact and separation of the disc. Pencil point is given motion proportional to the external balancing pressure and the rotating drum is given motion proportional to piston displacement. Sometimes electronic indicators are also employed for such measurements.

A simple mechanical system for drawing indicator diagram of a reciprocating engine is shown in figure 4.2. In this case, two special arrangements are made. Paper is placed in appropriate plane with pencil point lying on it ready to draw lines. The location of pencil point changes with change in the pressure as well as location of the piston. Through a level type arrangement, a rigid rod connects cross-head and paper of the indicator. As piston moves forward or leftwards, due to lever arrangement shown in the figure, paper moves rightwards. Scaling of length is observed and length of the indicator diagram (represented by movement of paper) is not same as stroke of the piston. Length of both the hands connected to piston side and paper side gives proper scaling factor. Arrangements are also made to transmit pressure in front of the piston to an auxiliary cylinder, whose piston is spring loaded to resist any upward movement. First a reference line is drawn with atmospheric pressure. As pressure in front of the piston in main cylinder rises, it pushes piston of the auxiliary cylinder in upward direction. This shifts pencil point upwards as indicated in the figure 4.2. Thus both pressure and piston positions are plotted on the paper with proper scaling. Area of the indicator diagram (shown in figure 4.2 by hatches area) is equivalent to work done per cycle by the engine. If a unit length of indicator diagram represents change in pressure by 'K' units, from the figure 4.2, mean effective pressure is given by the following expression.

Mean effective pressure (mep) = (Ai/Li) × K

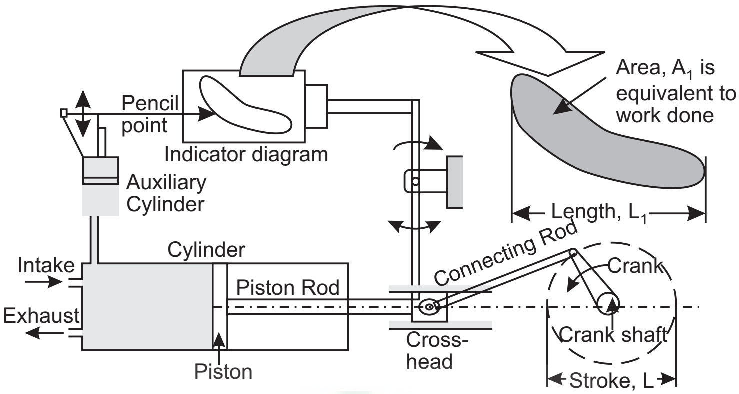


Fig. 4.2 : Mechanism for Generation of Indicator Diagram

The indicator diagram shown in the figure 4.2 is highly idealized. Actually, if indicator diagram is plotted for a two stroke single acting engine, there is a missed cycle after every power stroke, which is also plotted on the indicator plane. During this process no power is developed and the same is represented on the indicator diagram as negative loop. Similarly charging and discharging loops are also represented on the indicator diagram as negative loops.

4.3 BHP OR SHP

Brake horsepower (BHP) or shaft horse power (SHP) are used interchangeably and both are representative of power developed at the output shaft of the engine. Measurement of BHP is one of the most sought after work in the test schedule of an engine. It involves determination of torque and the angular speed of the engine. BHP is given as product of torque and angular speed of the engine. The torque measurement device of the engine is called dynamometer and it can be classified as absorption dynamometer and transmission dynamometer. In absorption dynamometer, power output from the engine is resisted and resistance to this power is measured in prony brake, rope brake or hydraulic dynamometer. In transmission dynamometer, power is delivered after getting transmitted through a device, which indicated the transmitted power on some scale. This type of device is also called torque meter.

Brake power = Torque × Angular Speed = $T \times w$.

In the case of absorption type dynamometer, the rotor or flywheel driven by engine is coupled with a stator or brake. The braking power is improved by increasing load acting on the rotor. The torque is calculated as product of force and distance of lever arm where force acts from the center of flywheel or rotor. The friction between stator and rotor is dissipated as heat. In prony brake dynamometer, attempt is made to stop a flywheel attached to the engine by increasing pressure on flywheel progressively. It is similar to braking the wheel and power is dissipated as heat due to friction. Friction can be increased by spring-loaded bolts. As friction at flywheel is high, this type of system needs adequate cooling to control temperature rise. A simple prony brake dynamometer is shown in figure 4.3. It is clear that torque output is equal to weight multiplied by arm length as shown in the figure 4.3.

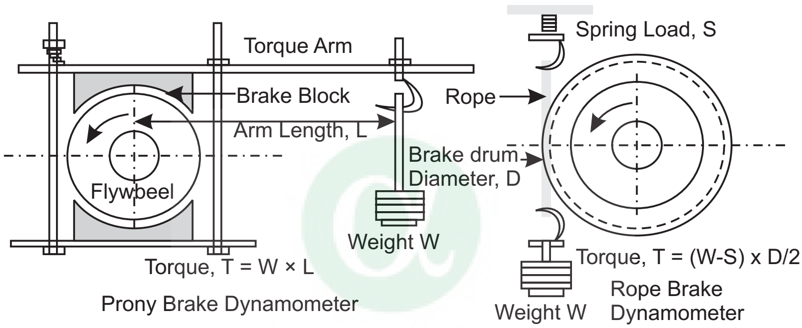


Fig. 4.3 : Mechanism for Generation of Indicator Diagram

Rope brake dynamometer is a simple device to measure BHP. In this case rope is wound around rotating drum of the output shaft. One side of rope has provision for putting loads and the other side of the ropes is connected to stationary support through a spring. The power is absorbed as friction between rope and the drum. Here also drum needs proper cooling. It is a cheap and easy in construction device, but is not very accurate.

A rope drum dynamometer is also shown in the figure 4.3 and expression for the torque is given by product of (W-S) with radius of brake drum.

Hydraulic dynamometer works on the principle of dissipating power in fluid friction and provision of cooling is dispensed with in this case. The principle of operation is similar to fluid flywheel. In this case an impeller connected to the engine is rotated in a casing filled with fluid. The outer casing also tend to rotate with the impeller, but is resisted by a torque arm supported the balance weight. The frictional force between impeller and the fluid is measured by the spring balance fitted on the casing. The heat developed due to dissipation of power is carried away by a coolant supply.

Similarly, electric principles are also utilized in design of dynamometers. Eddy current dynamometers are one such device, where eddy current is generated in stators due to rotation of rotor. The eddy current opposes rotor motion, thus loading the engine. The load is controlled by regulating current in the electromagnets used for creating magnetic flux. This device is easy to handle, control and programme. It can give high hp per unit weight of the system and gives smooth torque profile. However, this also needs cooling devices. Similarly, swinging field dc dynamometer, which is nothing but a dc shunt motor can also be utilized to measure torque of engines practically.

Transmission type dynamometers or torque meters has a set of strain gauges fixed on the rotating shaft. Angular deformation of the shaft is measured and is considered representative of torque. To avoid axial transverse loads on strain gauges, gauges are arranged in pairs. Use of 4-arm bridges negates effect of temperature also. Transmission dynamometers are very accurate and are used when continuous transmission of load is mandatory. They are essential parts of automatic units.

The way, in which indicated power is calculated from mean effective pressure, sometimes brake power is also calculated in similar manner. In this case, a term called brake mean effective pressure (b_{mep}) is defined and brake power is given by an expression similar to indicated power.

$$\textbf{Brake power} = b_{mep} \times L \times A \times N.$$

4.4 FHP

Friction horsepower (FHP) is difference of IHP and BHP. Frictional losses are main governing criterion for selection of an engine and preference of an engine over the other. Frictional losses are generally observed in the form of heat and are dissipated to cooling liquids. High FHP reduces BHP or output power and ultimately for same power fuel consumption rises. FHP is generally assumed to be dependent on engine speed and at constant speed, FHP is found to be constant. Although measurement of FHP is convenient by difference of BHP and IHP, but other methods are also possible for measurement of FHP.

Direct motoring test measures frictional horse power directly without calculation of either brake power or indicated power. In case of direct motoring test, engine is run at desired speed by its own power and allowed to stabilize in terms of water heating, oil consumption and temperature terms. Power under such condition is absorbed by a swinging field type electric dynamometer. The fuel supply is then cut-off and by electric switching device; the dynamometer is converted into a generator to run the engine at same speed at which it was previously running. The power supply to the motor is measured, which is direct measurement of friction power. During motoring test water supply is also cut off to maintain actual operating temperature. During test, first full engine is tested under motoring condition and then slowly components are dismantled one after the other progressively to get the frictional losses in various components.

Although this is a direct measurement of friction power, several conditions lead to measurement errors. The temperature of motored engine is different because in absence of fuel supply there is no power stroke or fuel combustion. In absence of combustion, temperature inside engine cylinder may not rise very high. The incoming air, in fact, cools cylinder during motoring. The temperature of lubricating oil reduces, which increases its viscosity and finally engine friction increases. In addition to this lower temperature of piston and cylinder leads to more clearance and reduced friction. The temperature affects pressure levels also. Expansion line during motoring is lower than compression line on pressure volume plane due to supply of cool air. In addition to this, during exhaust back pressure is more because during exhaust stroke gases do not get sufficient kinetic energy to expel more gases as occurs in the firing engines. Loads on bearings in absence of power stroke are also lower. The special dynamometer requirement is also a major drawback of this method.

For compression ignition (CI) engine Willian's line method is used for determination of FHP. In this case curve is plotted at a constant speed for fuel consumption against BHP. A representative curve is shown in figure 4.4. The curve has an intercept at fuel consumption axis, indicting that even at zero output power, some fuel is consumed. On extrapolating the curve in backward direction, the curve intercepts BHP axis at some negative value. The value of this negative BHP at zero fuel consumption is called FHP. The represents combined effects of mechanical friction, pumping and blowby. This is a simple method but is dependent on correctness of extrapolation.

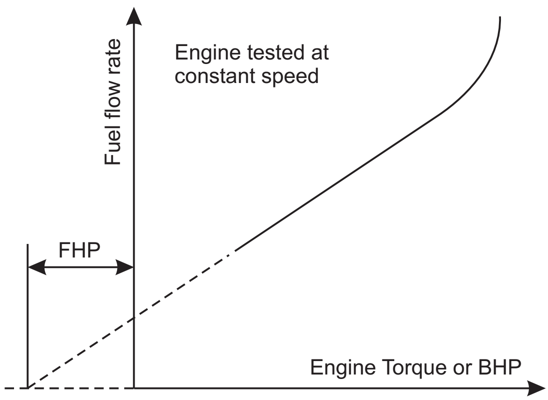


Fig. 4.4 : Willian's Line for an Engine

Willian's line method is limited to compression ignition engine only. Generally reliable results are possible for higher loads and the range of loads for which measurements are reliably done range from 5% to 40%. The extrapolation from such higher loads to zero loads and further to negative side of the BHP needs a linearity assumption, which is not so accurate. Actually changing slope of the curve indicates different efficiencies at different part loads. The pronounced change in the curve at full load (right side of the curve) depicts quality of combustion and negligible role of air-fuel ratio. Slight curvature in the curve exists even at low loads. This represents difficulty in accurate fuel injection consistently over a range of cycles. So nature of curve must be established only after a number of tests are conducted and linearity at low loads is ensured. It is also observed practically that the curve is straight for swirl chamber CI engine and linearity is compromised for direct injection type engines.

Another method for measurement of FHP is Morse test, which is valid for multi-cylinder engines only. The engine is to be tested at single speed. In this case, first output with all engines operational is measured. One engine is cut-off and output is measured. Difference of two outputs (BHPs) gives IHP of the disconnected engine, as FHP in both the cases is same.

Similarly, IHP for all the engines can be obtained by disconnecting them one by one. Finally, total IHP of multi-cylinder engine can be obtained and difference of IHP and BHP gives FHP of the engine. This is an indirect method for friction power determination.

Let us assume a four cylinder engine, with engine nomenclature as 1, 2, 3 and 4. Brake power of the engine is measured. If all four cylinders of the engine are working, then brake power of the engine is given by P1. Indicated power and friction power are inter-related by the relation given below.

P1 = BP1 + BP2 + BP3 + BP4 = IP1 – FP1 + IP2 – FP2 + IP3 – FP3 + IP4 – FP4 = (IP1 + IP2 + IP3 + IP4) – (FP1 + FP2 + FP3 + FP4).

If firing of first cylinder is stopped by shunting or shorting the spark plug of the cylinder, then first cylinder will not generate any power, but movement of piston will cause some frictional loss as earlier situation. In absence of firing, no indicated power is generated by the first cylinder. Under the situation refereed, if brake power developed is given by P2, then expression for P2 is as follows.

P2 = – FP1 + IP2 – FP2 + IP3 – FP3 + IP4 – FP4
= (IP2 + IP3 + IP4) – (FP1 + FP2 + FP3 + FP4)
Now, P1 – P2 = IP1.

Thus indicated power from the first engine is calculated by measurement of two brake powers. Similarly, indicated power from each of the cylinder can be calculated. Summation of all the indicated power from each of the cylinders gives total indicated power. If P1 is subtracted from the total indicated power, total friction power from the multi-cylinder engine can be obtained.

In general four methods evolve for measurement of friction horsepower:

- Measurement of IHP and BHP from indicator diagram
- Direct motoring test
- Willian's line method
- Morse test

However, out of the four, only first method gives direct measurement of friction. Other three methods actually measures power and losses are derived from that. In fact, motoring cannot be treated as actual representative of frictional losses in actual engines. During motoring, firing does not take place inside the engine and only four strokes are executed. This indirectly indicates that there is no combustion stroke and thus peak pressure is not realized in the engine. Lower gas pressure will always lead to lower rubbing friction. Similarly, temperature realized during motoring is also lower than actual engine. Higher temperature realized in actual operation of the engine leads to reduction in lubrication efficiency due to thinning. Due to such lower temperature and pressure, radial expansion of piston is also less and so fiction is less than whatever is observed in actual engine operation. Another variation occurs due to absence of exhaust stroke in the motoring engine. This means gases are discharged at higher density than that in the firing engine. Pumping friction is accordingly altered. During motoring operation, piston displacement occurs during compression and expansion. This work is not a part of frictional losses and should not be deducted from the total indicated horsepower. Although brake mean effective pressure or brake horsepower for motoring and firing engine are practically found to be same, but frictional horsepower changes with brake power. Higher brake power has more difference between total frictional losses of firing and motoring engine. For a firing engine, total frictional horsepower is more or less constant. However, frictional horse power of motoring engine drops as brake power increases. Contrary to this rubbing power for firing engine enhances with increase in brake power but in motoring engine rubbing friction power is more or less constant.

4.5 CORRECTIONS FACTORS

Based on performance curves and actual testing of engines, several general observations and implemented facts emerge. The percentage of heat rejected to coolant is more at lower speed and it reduces at higher speeds. So at high speeds, more heat is carried by the exhaust. Maximum torque position matches well with maximum air charge or maximum volumetric efficiency point. If size of engine is doubled, torque will also doubled, but mep will remain constant. For engines, performance test also include mention of certain efficiency terms.

First type of efficiency is air-standard efficiency, which is defined in first chapter. It is ratio of work done in a cycle to the amount of heat input. This is also called thermodynamics efficiency and is mainly a function of compression ratio and other ratios of thermodynamic parameters. The value of air-standard efficiency for operational engines is within 40-60%. Second type of efficiency is thermal efficiency. This is defined as ratio of power to amount of heat input or thermal energy by combustion of fuel. The thermal energy by combustion of fuel is given by product of mass flow rate and calorific value of the fuel. If indicated power is considered, the ratio is called indicated thermal efficiency and if brake power is taken in calculation, the ratio is called brake thermal efficiency. The value of thermal efficiency is around 30% for most of the operational engines. Third term is called mechanical efficiency, which is ratio of brake power to indicated power and is a representative of friction power. The value of mechanical efficiency is around 75-80%. Sometimes these efficiencies are also compared. A term called relative efficiency is ratio of brake thermal efficiency and air standard efficiency. For most of the engines, it remains between 75% and 90%, if air supply is adequate.

Fourth type of efficiency is volumetric efficiency. It is indicator of breathing capacity of the engine. In third chapter, it is explained that inlet valve remains open for more than 180° crank rotation to induct more air than the swept volume of the engine. It is very important parameter and is defined as actual mass of air drawn inside engine during suction stroke to theoretical mass of air that would be inducted in the same period based upon total piston displacement. Theoretical quantity of air is calculated from speed of the engine, density of incoming air, swept volume and number of cylinders. Actual mass is measured by experiments. For a two stroke engine, each cycle of crank rotation has a power stroke and speed of crank in rpm indicates number of charging per minute. But in case of a four stroke engine, every alternate cycle results in a charging stroke and number of charging stroke is half the rotational speed of the crank shaft.

For 2-stroke engines, scavenging efficiency is defined. It is ratio of amount of air or gas-air mixture, which remains inside cylinder at the beginning of the compression stroke to the amount of air available at the beginning of the compression stroke. Depending on design, scavenging efficiency is of the order of 40% to as high as 95%. Another efficiency term is combustion efficiency, which is ratio of actual heat liberated to theoretical heat output from combustion of the fuel. It ranges from 95-97%. This indicates efficiency of combustion and it indicates poor mixing of air-fuel, dissociation of gases etc.

In addition to these efficiency terms, engine operating parameters also affect engine performance. The spark timing must be adjusted to give maximum pressure at the beginning of the power stroke. As a general rule spark timing is adjusted such that around half of the total pressure rise due to combustion occurs before piston reaches top center. In fact combustion rate plays a major role in adjustment of this parameter and spark timing is related to combustion rate of the fuel. Further combustion rate is a function of temperature, pressure and above all the air-fuel ratio. Generally air-fuel ratio is set for best fuel economy situation. In first chapter, it is elaborated that higher compression ratio always give higher air standard efficiency. Using anti knocking agent in the fuel may enhance performance of SI engine, whose performance is restricted by knocking. Higher compression ratio results in increased friction in the cylinder components and it is generally not feasible to increase the compression ratio beyond certain limit. At low speed heat transfer from engine wall to the surrounding is more because of higher time available for heat dissipation. At relatively higher speeds, the heat loss is drastically reduced. Inducting more charge in a given cylinder volume results in more power from the engine. Improvement in engine performance by control of heat transfer is a suitable method employed in operational engines. This is accompanied with induction of charge at higher pressures. Charge must be inducted at high densities.

4.6 VARIABLE SPEED AND CONSTANT SPEED TEST

Torque and mean effective pressure do not strongly depend on the speed of the engine. High horsepower is derived at high speed and doubling the speed, doubles the power. At low speed, FHP is relatively low and BHP matches well with IHP. As engine speed increases, IHP increases continuously and FHP increases continuously at greater rate such that BHP reaches a peak. After this, BHP reduces despite IHP is increasing. As engine speed increases beyond normal operating speed, FHP increases very rapidly. Parallel to this IHP also reduces. At some point IHP equals FHP and BHP becomes zero.

Variable speed and full throttle test of petrol engine show that mechanical efficiency reduces with rise in speed. Specific fuel consumption reduces with increase in speed but later it increases due to higher power requirements. For every engine, there is a normal speed range in which indicated and brake thermal efficiency is the highest and specific fuel consumption is the lowest. Brake torque, bmep and bsfc shows optimum value at certain speed ranges. However, peak BHP is realized at higher speeds and it matches well with peak IHP. FHP continuously increases with speed. The variation of various parameters with speed of the engine is shown in figure 4.5.

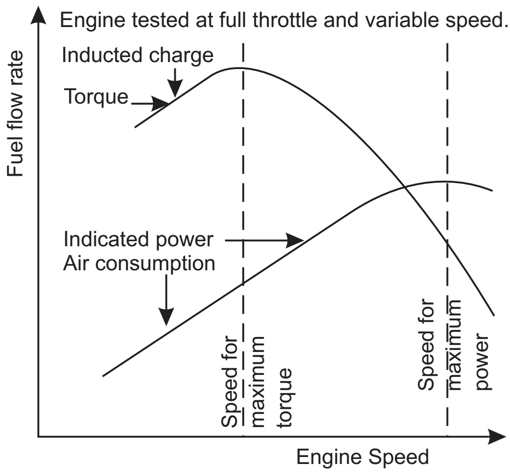


Fig. 4.5 : Performance Plot of a Typical Engine

As speed of engine increases, inducted charge per cylinder per cycle increases. It reaches a maximum after which due to small induction time at higher speed, the inducted charge per cycle per cylinder reduces. At this speed where inducted charge (air-fuel mixture) is maximum, torque is also maximum for the given engine. This is represented schematically in figure 4.5. Please note that both inducted charge and torque is shown by same curve. These are representative curves and both the parameters are plotted at different secondary axes. Only trend is same, which is depicted in the figure. Although after this maximum, charge inducted in the engine reduces with increases in speed but power output increases because of more number of cycles per unit time. However, the rise in indicated power ceases once reduction in inducted charge surpasses rise in number of cycle per unit time. So, maximum power is observed at a little higher speed after which indicated power reduces. The maximum indicated power also corresponds to maximum air consumption. In fact higher air induction gives spare capacity to the engine to induct more fuel and increase power output. As engine speed increases, quantity of fuel consumed increases and brake specific fuel consumption (bsfc) drops at lower speeds. However, as speed increases, the variation of bsfc becomes neutral and at further higher speeds, bsfc increases with engine speed.

At constant speed and variable load tests, performance parameters are expressed with respect to percentage load. As load is raised, torque, exhaust temperature, BHP and b_{mep} all rises monotonically. Except exhaust temperature, all three increases linearly with percentage load. Exhaust temperature initially increases slowly, and then increases rapidly. Compared to this, bsfc reduces continuously with load.

EXAMPLE 4.1

A 4-cylinder 2-stroke gasoline engine has bore and stroke as 10 cm and 15 cm respectively. It runs at 1500 rpm. Areas of positive and negative loops of the indicator diagram are 6 sq cm and 0.5 sq cm respectively. Length of indicator diagram is 110 mm and spring constant is 0.6 MPa/cm. Find indicated power of the engine.

SOLUTION

Given that 55 mm on indicator diagram is equivalent to 15 cm stroke length and 1 cm on vertical side is equivalent to 3.5 bar. Net area of the indicator diagram = 6 – 0.5 sq cm = 5.5 sq cm. As length of indicator diagram is 110 mm, mean height of the indicator diagram is = 5.5/11 = 0.5 cm. Mean effective pressure

$$= 0.5 \times 0.6 \text{ MPa} = 0.3 \text{ MPa}.$$

$$\text{Indicated power} = 0.3 \times 0.15 \times (\pi/4) \times 0.12 \times 1500 \times 4/60 \text{ MW}$$

$$= 35.343 \text{ kW}.$$

EXAMPLE 4.2

A single cylinder engine runs at 1600 rpm and develops a torque of 10 Nm. If indicated power is 2 kW, find friction horsepower and mechanical efficiency.

SOLUTION

$$\text{Brake power of the engine} = 2\pi \times 1600 \times 10/60 \text{ W}$$

$$= 1.6755 \text{ kW}.$$

$$\text{Friction power} = \text{indicated power} - \text{brake power}$$

$$= 2 - 1.6755 \text{ kW} = 0.3245 \text{ kW}$$

$$\text{Mechanical efficiency} = \text{brake power/indicated power}$$

$$= 1.6755/2 = 83.77\%.$$

EXAMPLE 4.3

A single cylinder 4-stroke engine has bore and stroke as 75 mm and 100 mm respectively. If rating of the engine is 4 kW at 1500 rpm, find mean effective pressure and torque.

SOLUTION

Since it is 4-stroke engine, effective rpm is half of engine rpm from power output point of view because after two revolutions only one power stroke is executed. Additionally indicated and brake power of the engine is taken same for further calculation.

Power from the engine = mean effective pressure × stroke × bore area × effective rps

$4 = P_m$ (in MPa) × 1000 × 0.1 × (π/4) × 0.0752 × 1500/(2 × 60)

Mean effective pressure, Pm = 0.7243 MPa

4000 = 2π × 1500 × T (in Nm)/60, So, Torque = 25.46 Nm.

EXAMPLE 4.4

An indicator diagram taken from a single cylinder 4-stroke CI engine has a length of 100 mm and an area of 2045 mm^2. The indicator pointer deflects a distance of 10 mm for a pressure increment of 2 kgf/cm^2 in the cylinder. If the bore and stroke of the engine cylinder are both 100 mm and the engine speed is 900 rev per minute, calculate the mean effective pressure and the indicated power. If the mechanical efficiency is 75%, what is brake power developed?

SOLUTION

Mean height of the indicator diagram = area/length

= 2045/100 = 20.45 mm.

Mean effective pressure = height × scale factor

= 20.45 × 2/10 kgf/cm^2 = 4.09 kgf/cm^2.

Since it is a 4-stroke engine, one power stroke is delivered at every 2 revolution of crankshaft. This indicates that per minute, out of 900 revolutions, only 450 revolutions will result in a power stroke. Operation is assumed single acting and per second 450/60 = 7.5 power strokes are possible.

Indicated horsepower,

ihp = 4.09 × 0.10 × (π/4) × 102 × 7.5/75 = 3.21 hp.

Brake horsepower = ihp × mechanical efficiency = 2.40 hp.

EXAMPLE 4.5

A four cylinder petrol engine has an output of 70 bhp at 2000 rpm. A Morse test was carried out and the brake torque were 17.97, 17.28, 17.00 and 17.70 kgf-m respectively, when each of the engines were made inoperative one by one. For normal running at this speed, specific fuel consumption is 0.272 kg/bhp/hr. the calorific value of the fuel is 10500 kcal/kg, calculate the mechanical efficiency and the brake thermal efficiency of the engine.

SOLUTION

Conversion relation used,

1 hp = 75 kgf-m/s, 1 kcal = 427 kgf-m

Rotational speed = 2 × π × 2000/60 rad/seconds

= 209.44 rad/s

Brake horsepower, when cylinder 1 is not firing = torque × rotational speed = 17.97 × 209.44/75 hp = 50.18 hp

So, ihp of cylinder 1 = 70 – 50.18 = 19.82 hp

Brake horsepower, when cylinder 2 is not firing = torque × rotational speed = 17.28 × 209.44/75 hp = 48.25 hp

So, ihp of cylinder 2 = 70 – 48.25 = 21.75 hp

Brake horsepower, when cylinder 3 is not firing = torque × rotational speed = 17.00 × 209.44/75 hp = 47.47 hp

So, ihp of cylinder 3 = 70 – 47.47 = 22.52 hp

Brake horsepower, when cylinder 4 is not firing = torque × rotational speed = 17.70 × 209.44/75 hp = 49.43 hp

So, ihp of cylinder 4 = 70 – 49.43 = 20.57 hp

Total ihp of all the engines = 19.82 hp + 21.75 hp + 22.52 hp + 20.57 hp = 84.66 hp

So mechanical efficiency = bhp/ihp = 70/84.66 = 82.68% Specific fuel consumption = 0.272 kg/bhp-hr

Energy ratio of heat input and bhp = mass of fuel/bhp-hr × calorific value = 0.272 × 10500 = 2856 kcal /bhp-hr

= 2856 × 427/(3600 × 75) = 4.52 (made dimensionless)

Brake thermal efficiency = 1/(energy ratio of heat input and bhp) = 22.14 %.

EXAMPLE 4.6

An engine consumes 5 grams of fuel (calorific value = 45 MJ/kg) per second and delivers 80 kW with mechanical efficiency of 80%. Find (i) brake specific fuel consumption, (ii) indicated specific fuel consumption (iii) brake specific energy consumption (iv) indicated specific energy consumption.

SOLUTION

Brake specific fuel consumption = fuel consumption/brake power = $5 \times 10^{-3} \times 3600/80$ kg/kW-hr = 0.225 kg/kW-hr.

Brake specific energy consumption = 0.225 × 45000/3600 = 2.8125

Indicated fuel specific consumption = Brake specific fuel consumption × mechanical efficiency = 0.225 × 0.80 kg/kW-hr

= 0.18 kg/kW-hr.

Indicated specific energy consumption = 0.18 × 45000/3600 = 2.25.

EXAMPLE 4.7

A 4-stroke gas engine running at 500 rpm has a bore and stroke of 20 cm and 30 cm respectively. If air-fuel ratio is 5:1 by volume and volumetric efficiency is 80%, determine the volume of gas taken in per minute. If calorific value of fuel is 6 MJ/m^3 at NTP and the brake thermal efficiency is 25%, determine the brake power of the engine.

SOLUTION

Swept volume per stroke = $(\pi/4) \times 0.22 \times 0.3$ m^3

$= 0.009425$ m^3.

Since volumetric efficiency is 80%, actual gas volume taken in is less than swept volume. So, volume of charge taken in per cycle = 0.8×0.009425 m^3 = 0.00754 m^3.

Since it is a four stroke engine, charge is taken in every alternate revolution. For a speed of 500 rpm, per minute only 250 times charge will be taken in.

So, volume of charge taken in per minute = 0.00754 m^3 $\times$ 500/2

$= 1.885$ m^3/min.

This volume is a mixture of air and fuel both. Amount of fuel taken in is one-sixth of the total volume of gas taken in.

So, volume of fuel taken in = 1.885/6 m^3/min.

$= 0.31416$ m^3/min.

Heat input of the system = volume of fuel $\times$ calorific value of the fuel = 0.31416×6 MJ/min = 1.855 MJ/min

Brake power = heat input $\times$ brake thermal efficiency = 0.47124

MJ/min = 7.854 kW.

EXAMPLE 4.8

A gas engine working on the constant volume cycle gave the following results during a one-hour test run - Cylinder diameter 24 cm; stroke 48 cm; effective diameter of brake wheel 1.25 cm; net load on brake 126 kgf; average speed 226.7 rpm; average explosions per minute 77; mep of indicator card 7.5 kg/cm2; gas used 13 m^3 at 15°C and 771 mm of Hg pressure; lower calorific value of gas 5250 kcal/m^3 at NTP; Cooling water used 625 kg; inlet temperature 25°C; outlet temperature 60°C.

Determine (i) The mechanical efficiency, (ii) The gas consumption in m^3 at NTP per ihp hour, (iii) The indicated thermal efficiency. Draw heat balance sheet.

SOLUTION

Given

D = 24 cm, L = 48 cm, D_b = 1.25 cm, W = 126 kgf, N = 226.7 rpm, n = 77, mep = 7.5 kg/cm^2, V = 13 m^3, T = 15°C = 288 K, p = 771 mm Hg, CV = 5250 kcal/m^3, W_c = 625 kg, t = 1 hour, T_{win} = 25°C = 298 K, T_{wout} = 60°C = 333 K.

Conversion relations used

1 hp = 75 kgf-m/s; 1 kcal = 427 kgf-m.

Torque on the brake = $W \times D_b/2$ = 0.78125 kgf-m. Angular speed = 2 × π × N/60 = 23.74 /s.

Brake horsepower. bhp = torque × angular speed = 1854.685 kgf-m/s = 24.73 hp.

Swept volume = (π/4) × $D^2 \times L$ = 0.0271 m^3.

Indicated horsepower, ihp = mep × swept volume × power stroke per second = 2090.038 kgf-m/s = 27.86 hp.

Mechanical Efficiency = bhp/ihp = 88.74% **Ans.** **(*i*)**

Gas consumption at 15°C and 771 mm of Hg pressure is 15 m^3. This is to be converted to gas consumption at NTP using simple ideal gas equation. NTP is given by 27°C (300 K) and 760 mm Hg.

Gas consumption at NTP, = 15 × (771/760) × (300/288)

= 15.85 m^3. **Ans.** **(*ii*)**

Thermal energy of fuel at NTP = gas consumption × CV = 83218.5 kcal

Since test is carried out for 1 hour, the heat output is also for 1 hour. Converting it to horsepower gives value of equivalent heat as (83218.5 × 427)/(75 × 3600) 131.6 hp.

Indicated thermal efficiency = ihp/heat energy of fuel

= 21.17% **Ans.** **(*iii*)**

Since total testing is carried out for 1 hour, all heat content is converted to per second basis. All calculations are based on NTP.

Heat supplied per second = 15.85 × 5250/3600

= 23.11 kcal/second

Heat equivalent to bhp = 24.73 × 75/427

= 4.34 kcal/second

Heat to cooling water = 625 × (60-25)/3600

= 6.07 kcal/second

Heat going to exhaust = calculated by difference

= 15.7 kcal/second

EXAMPLE 4.9

A 4-stroke petrol engine has 6 single acting cylinders of 7.5 cm bore and 9 cm stroke. The engine is coupled to a brake having a torque arm radius of 38 cm, at 3300 rpm, with all cylinders operating the net brake load is 33 kgf. When each cylinder in turn is rendered inoperative, the average net brake load produced at the same speed by the remaining 5 cylinders is 25 kgf. Estimate the mean effective pressure of the engine.

With all cylinders operating, the fuel consumption is 0.3 kgf/min (calorific value of the fuel = 10000 kcal/kg); the jacket water flow rate and temperature rise are 65 kg/min and 12°C; On test, the engine s enclosed in a thermally and acoustically insulated box, through which the output drive, water, fuel, air and exhaust connections pass. Ventilation air blown up through the box at the rate of 14 kg/min enters at 10°C and leaves at 55°C. Draw up a heat account of the engine indicating percentage of each term.

SOLUTION

Conversion relations used

1 hp = 75 kgf-m/s; 1 kcal = 427 kgf-m.

Data used, Specific heat of water = 1 kcal/kg K

Specific heat of air = 0.24 kcal/kg K

When all the cylinders are working, bhp = $2 \times \pi \times N \times T/4500$ hp = $2 \times \pi \times 3300 \times 33 \times 0.38/4500 = 57.78$ hp

When one cylinder is cut-off, bhp = $2 \times \pi \times N \times T/4500$ hp = $2 \times \pi \times 3300 \times 25 \times 0.38 / 4500 = 43.77$ hp

As it is 6-cylinder engine and Morse-test is being conducted,

Total ihp of engine = $6 \times (57.78 - 43.77) = 84.06$ hp.

Piston displacement or swept volume = $(\pi/4) \times D_2 \times L = 0.0003976\ \text{m}^3$.

Revolution per second, = 3300/60 = 55.

As it is single acting 4-stroke engine, power stroke is executed every two revolutions of crank. As it is a 6-cylinder engine and each cylinder delivers power in 2 rotation, (6/2 =) 3 times power delivered in one crack case revolution is sufficient to account for total power delivered.

Indicated mean effective pressure, (imep) = ihp/(piston displacement × rps)

$$\text{So, imep (kg/cm2)} = 84.06 \times 75/(3.976 \times 55 \times 3)$$
$$= 9.6\ \text{kgf/cm}^2.$$

Heat input to the engine = 0.3×10000

$$= 3000 \text{ kcal/min } (100\%).$$

$$\text{Heat equivalent to bhp} = 43.77 \times 75 \times 60/427$$

$$= 461.6 \text{ kcal/min } (15.38\%).$$

$$\text{Heat cooling water} = 65 \times 12 = 780 \text{ kcal/min } (26.00\%).$$

$$\text{Heat to ventilating air} = 14 \times 0.24 \times (55 - 10)$$

$$= 151.2 \text{ kcal/min } (5.04\%).$$

Heat to exhaust gases and other losses is calculated by difference = 1607.2 kcal/min, which is 53.58%.

EXAMPLE 4.10

A Morse-test on twelve cylinder 2-stroke CI engine of bore 38 cm and stroke 50 cm gave following readings. Speed = 200 rpm. Brake load is measured in kgf.

Condition	*Brakeload*	*Condition*	*Brakeload*	*Condition*	*Brakeload*
All firing	204	No 1 out	183	No 2 out	185
No 3 out	185	No 4 out	183	No 5 out	184
No 6 out	185.5	No 7 out	183.5	No 8 out	186
No 9 out	182	No 10 out	184	No 11 out	185
No 12 out	183	All firing	206		

The law of brake is hp = WN/13.5, where W = load in kgf, N = rpm. Calculate the brake mean effective pressure in kgf/cm^2 and mechanical efficiency with all cylinders firing.

SOLUTION

We know that $\quad bhp_{total} = ihp_{total} - fhp$ and

$$bhp_{except\ 1} = ihp_{except\ 1} - fhp.$$

So, $ihp_{left\ cylinder} = bhp_{total} - bhp_{except\ 1}$.

Since there are two values of all firing condition, average value is taken for analysis. Calculation for each engine is tabulated below.

Engine cylinder	*Brakeload (kgf)*	*bhp = WN/13.5*	*Cylinder ihp*
All firing	205	3037.04	—
No 1 Out	183	2711.11	325.93
No 2 Out	185	2740.74	296.30
No 3 Out	185	2740.74	296.30
No 4 Out	183	2711.11	325.93
No 5 Out	184	2725.92	311.12

Engine cylinder	*Brakeload (kgf)*	*bhp = WN/13.5*	*Cylinder ihp*
No 6 Out	185.5	2748.15	288.89
No 7 Out	183.5	2718.52	318.52
No 8 Out	186	2755.55	281.49
No 9 Out	182	2696.29	340.75
No 10 Out	184	2725.92	311.12
No 11Out	185	2740.74	296.30
No 12 Out	183	2711.11	325.93
Total ihp from all the engines			3718.58

Mechanical efficiency = bhp/ihp = 81.67%.

Swept volume = $(\pi/4) \times D^2 \times L = 0.0567\ \text{m}^3$.

Brake mean effective pressure = bhp/(no of cylinder × swept volume × rpm) = 3037.04 × 75 × 60/(12 × 567 × 200) kg/cm^2

$$= 10.04\ \text{kg/cm}^2.$$

EXAMPLE 4.11

In a trial on four-cylinder 4-stroke petrol engine of 10.16 cm bore and 12.7 cm stroke, the net dynamometer load was 18.67 kgf at a radius of 50.8 cm when the speed was 2500 rpm. At this speed and throttle opening the engine required 6.285 hp to motor it with ignition off. (i) Calculate the mechanical efficiency and the indicated mean effective pressure, (ii) During a 3 minute run at this speed and power, the engine used 0.589 kg of petrol of calorific value 10840 kcal/kg and 22.68 kg of cooling water with a temperature rise of 55.5°C. Draw a heat balance chart for the test in kcal/min.

SOLUTION

Conversion relations used

1 hp = 75 kgf-m/s; 1 kcal = 427 kgf-m.

Data used Specific heat of water = 1 kcal/kg K

In 4-stroke engine, power stroke is delivered at each 2 revolutions and in 2500 rpm, only 1250 revolutions per engine per minute will be giving power. Additionally there are four cylinders, so overall 5000 power strokes per min are possible from the given engine. This is utilized for calculation of ihp. Friction horsepower is given as 6.285 hp.

Torque from the engine = 18.67 × 0.508 = 9.48436 kgf-m.

Rotational speed of the engine = 2 × π × 2500/60 rad/second

$$= 261.8\ \text{rad/s}.$$

So, bhp of the engine = torque × rotational speed

$= 9.48436 \times 261.8/75$ hp $= 33.107$ hp.

Now ihp = bhp + fhp = 33.107 = 6.285 = 39.392 hp

Mechanical efficiency = bhp/ihp = 33.107/39.392 = 84.04%.

Swept volume of the given engine = $(\pi/4) \times D_2 \times L$

$= (\pi/4) \times 10.162 \times 12.7 = 0.00102963\ m^3$.

Indicated horsepower can be given by the expression imep × 2 × revolutions × swept volume. From this indicated mean effective pressure can be calculated.

So, imep = ihp/(swept volume × revolutions per seconds × 2)

$= 39.392 \times 75 \times 60/(10.2963 \times 2500 \times 2) = 3.443\ kgf/cm^2$.

Heat balance chart for the engine

Heat supplied by combustion of fuel = 0.598 × 10840/3

= 2160.77 kcal/min (100%).

Heat carried away by cooling water = 22.68 × 55.5/3

= 419.58 kcal/min (19.42%).

Heat converted to work (bhp) = 33.107 × 60 × 75/427

= 348.90 kcal/min (16.15%).

Heat going to exhaust and other losses (by difference)

= 1392.29 kcal/min (64.43%).

SUMMARY

Test of internal combustion engines is necessary to compare performance of engines and assess relative worth of various engines. Practical measurement of performance parameters needs special set-up and measurement principles. Using these set-ups, IHP, BHP and FHP measurement for various types of engines can be executed. Performance parameter at constant speed and variable speed are conducted to assess effect of speed on various engine parameters. Numerical included at the end of the chapter establishes importance of various test methods.

QUESTIONS

1. What is indicator diagram? How is it obtained?
2. What is Morse-test for engine performance?
3. What is William's line? What is its utility?
4. What are various methods to measure friction horsepower?
5. What are various methods for measurement of BHP?

6. What is brake mean effective pressure?
7. What is meaning of constant speed and variable speed test?
8. What is normal calorific value of fuel used in IC engine?

   Which engine parameters are dependent on calorific value of fuel?
9. Whether air standard efficiency is same as thermal efficiency in an engine? Give reasons in support of your answer.
10. How is friction horsepower measured?

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CHAPTER

5

Elements of Heat Transfer

STRUCTURE

- Introduction
- Objective
- Heat Transfer Process
- Heat Conduction
- Thermal Conductivity
- General Equation for Heat Conduction
- 1-D heat Conduction
- 2-D heat Conduction
- Convective Heat transfer Process
- Free Convection Heat Transfer
- Convection on Flat plate
- Convection on Planes
- Convection over Cylinders
- Convection on Spheres
- Thermal Radiation
- Emissive Power
- Plank's Distributive Law
- Radiation Properties
- Summary
- Questions

5.1 INTRODUCTION

Heat transfer is one of the important means to understand energy transfer in chemical propulsion. Although energy is generated in chemical propulsion by combustion, this is excluded from the scope. However, efficient mode of heat transfer needs thorough understanding of governing equation, heat transfer mechanism and salient parameters. All three modes of heat transfer namely conduction, convection and radiation are discussed in this Chapter with intention to analyze and calculate heat losses. Heat transfer to cooling water, exhaust, finned surfaces and heat sinks/sources need a deliberation on heat transfer. This Chapter fulfills this requirement of propulsion module.

Objective

After studying this Chapter, you should be able to understand

- Various modes of heat transfer,
- Conductive heat transfer in 1-D and 2-D,
- Thermal conductivity of materials,
- Convective heat transfer at different surfaces,
- Thermal radiation heat transfer,
- Radiation properties of materials.

5.2 HEAT TRANSFER PROCESS

Heat transfer is a mode of energy transfer, which takes place due to temperature difference. This is thermodynamic definition of heat transfer. Heat generally flows from higher temperature to lower temperature. It is associated with prediction of rate of heat transfer, where as thermodynamics is different and it deals with systems in equilibrium. Thermodynamics cannot predict the quickness with which heat is transferred but it concentrates only on amount of energy needed to go from one energy state to the other. Thermodynamics deals with amount of heat transfer, while heat transfer estimates rate of heat transfer. In heat transfer transient and non-equilibrium states are discussed and static equilibrium is discussed in classical thermodynamics. If a hot iron ball is placed in cold water, thermodynamics gives us detail about final equilibrium temperature attained by both water and the hot ball. Obvious at equilibrium, both the temperatures are same. However, thermodynamics cannot give variation of temperature with time. Heat transfer does so and is a rate process. Time varying temperature and exchange of energy in the form of heat is mainly discussed in heat transfer. Although, heat transfer takes placed as per both first and second law of thermodynamics, second law of thermodynamics is more relevant. As per second law, heat flows from high temperature to low temperature. First law of thermodynamics is only building block for the heat transfer and it simply gives equivalence of heat and work transfer. Increase in internal energy of the system is equal to sum of heat received and work done by the system.

There are several requirements of daily life, where heat transfer calculations are done and then systems are devised for meeting human needs. There are many varieties of heat transfer equipments used in industrial and domestic arena like boiler, condenser, solar collector, radiator, heat exchangers, air conditioners, refrigerators, insulators, stoves etc. All these equipments are different in size, shape, need and heat transfer modes. Their analysis is also different from each other, which is mainly dependent on mode of heat transfer. The heat transfer takes place in three modes – conduction, convection and radiation. All three modes of heat transfer occur as per different mechanisms and heat transfer devices may utilize one or more modes of heat transfer in conjunction.

Conduction is that mode of heat transfer, which occurs through a medium from high temperature to low temperature. It needs a medium and is faster in metallic solids. Temperature is associated with molecular motions and high temperature means more vibration/motion of molecules about mean free position. Each molecule transfers energy to its adjacent molecule by molecular motion and retains their mean position. As per steady flow energy equation of first law of thermodynamics, rate of work transfer during conduction is generally negligible. The mechanism of conduction through solids is by molecular interaction, while in fluids, it is due to direct impact. In solids, main mechanism is lattice vibration, which is augmented further by drift of free electrons in metals, resulting in good conductivity of metals. This is the reason why good electric conductors (free electron transfer) are also good heat conductors. This free electron interactions and molecular or lattice vibration are well established in physics.

This is governed by Fourier's law, who was a French mathematical physicist. Joseph Fourier published his pioneering work in 1822 in the form of a book, where he explained his simple law of heat conduction. As per Fourier's law of heat conduction, heat transfer rate per unit area is proportional to the normal temperature gradient. Mathematically, this is given as $(q/A) \propto (\partial T/\partial x)$, or $q = -k.A.(\partial T/\partial x)$, where q is heat transfer rate, A is heat flow area perpendicular to the direction of heat transfer, $(\partial T/\partial x)$ is thermal gradient in the direction of heat flow and k is constant of proportionality called thermal conductivity of the material. This law is valid for solids, liquids and gases. Negative sign is introduced to indicate that heat flow takes place in the direction of negative thermal gradient.

Convection is another mode of heat transfer, where material diffusion or flow plays a major role. The name convection is derived from the fact that in this case heat is conveyed through a fluid by motion of its particles. In this case, two separate media is needed, one solid and another fluid. Heat transfer from or to solid occurs from fluid media due to movement of fluid over solid. So convective heat transfer is very complex and role of fluid flow velocity cannot be neglected in this case. If steady flow energy equation of first law of thermodynamics is considered, work done by fluid pressure dominates. This is different from mechanism explained in conduction. In convection, change in kinetic energy and potential energy of the fluid is generally negligible as compared to rate of heat transfer. If fluid flows over a solid, due to viscous action, velocity of flow appears to be zero at contact. Since velocity of flow at contact is zero, main mode of heat transfer is by conduction. With this explanation, the heat transfer is simulated using heat conduction equation, but heat transfer is governed by temperature gradient and temperature gradient is dependent on velocity of fluid over the solid. Physical mechanism of heat transfer at the wall is conduction but beyond that it is fluid motion and so heat convection. This coefficient is also called film conductance and is determined experimentally. In addition to the earlier specified parameters, like conductivity, density and specific heat, heat convection is strongly dependent on viscosity of fluid.

The heat convection process is governed by Newton's law of cooling. It states that heat transfer rate is proportional to overall temperature difference between wall and fluid and the surface area of exposure.

In this case, $q \propto A \times (T_s - T_f)$, or $q = h \times \text{A} \times (T_s - T_f)$, where T_s is temperature of the surface, Tf is temperature of the fluid and h is convective heat transfer coefficient. The unit of convective heat transfer coefficient is $\text{W/m}^2\text{.K}$.

Heat convection may be classified as natural or free, if movement of fluid is energized by density gradient alone. Contrary to this, if some external means are employed to move or blow fluid over solid, the mode of heat transfer is called forced-convection. For a temperature difference of 30°C between solid and fluid convective heat transfer by free convection on a cylinder for air and water are 6.5 $\text{W/m}^2\text{.K}$ and 900 $\text{W/m}^2\text{.K}$ respectively. If forced convection with air velocity of 50 m/s is employed, the convective heat transfer coefficient may attain a value as high as 180 $\text{W/m}^2\text{.K}$ from 6.5 $\text{W/m}^2\text{.K}$ for free-convection. Boiling and condensation are grouped under this mode of heat transfer. In fact heat transfer in radiators of automobiles takes place by convection.

The relations predicted for convection mode of heat transfer cannot give rate of heat transfer explicitly. It require certain other parameters of the system like geometry, material properties, fluid flow rate, surrounding temperature fluid temperature etc. The overall heat transfer coefficient is difficult to express universally and generally empirical relations are proposed for various known situations. These requirements are in addition to inputs to first law of thermodynamics. The materials properties may be density, viscosity, surface tension, coefficient of thermal expansion, specific heat etc. the cases where phase change are involved, latent heat of melting, vaporization an sublimation are also required for complete analysis.

In both the above mentioned mode of heat transfer, presence of medium is mandatory for heat transfer. Radiation is a mode of heat transfer where intervening medium is not a facilitator for heat transfer, but it acts as an obstruction. In fact, heat transfer by radiation occurs best in vacuum. Heat transfer through glass or other participating gases is restricted. All the solids are opaque to radiative mode of heat transfer. All opaque bodies above absolute zero temperature radiate energy. A perfect or blackbody surface emits radiation at a maximum rate and additionally, it absorbs all the incident radiation. The nomenclature of blackbody is derived from the absorption of incident radiation by the perfect bodies. As they do not reflect any light of visible region, it appears black for the naked eye and nomenclature is derived from this fact. Actually, total rate of heat flux exchange is given by difference of total emitted and reflected energy and total absorbed energy from the surrounding. This mode of heat transfer occurs in the form of electromagnetic waves emitted by atomic and subatomic agitation at the surface of the body. It is surface phenomena and not a bulk phenomenon. Since, this mode of heat transfer occurs as electromagnetic radiation, the thermal radiation also travels at the speed of light.

This form of heat transfer includes electromagnetic energy transfer and is strongly dependent on temperature. Pioneering work has been carried out in the area of radiation mode of heat transfer and the governing equation is formulated using experiments by J. Stefan (1879) and theoretical treatment by L. Boltzmann (1884). This mode of heat transfer is governed by Stefan-Boltzmann Law, which states that an ideal thermal radiator or blackbody will emit energy at a rate proportional to fourth power of the absolute temperature of the body and is directly proportional to its surface area. In this case constant of proportionality is called Stefan-Boltzmann Constant (σ) and it has a value of 5.669×10^{-8} W/m^2 K^4. The law is given as $q = \sigma \,.\, A \,.\, T^4$. This law is valid for thermal radiation only and it cannot be used for all types of electromagnetic radiations.

Any non-black body emits radiation less than blackbody maintained at same temperature. The ratio of emission by a non-black body and a blackbody is called emissivity. This is a dimensionless parameter and has a value between zero and unity. Emissivity is a function of temperature and surface features like roughness, texture, colour, coating, curvature etc. the variation of emissivity with temperature is very rapid for some of the materials and this becomes one of the governing parameters for heat transfer by radiation for non-black bodies. Since emissivity of a material can vary with as high as six power of temperature and as low as temperature raised to power 2, the heat transfer by radiation may take place at higher than fourth power of temperature. If emissivity effects are considered, heat transfer by radiation in selected range for selected materials may be as high as temperature raised to power 10 to as low as temperature raised to power 2. If emissivity of a material is constant, it is called a gray body.

Since radiation heat transfer varies with fourth power of temperature, this becomes dominating heat transfer mode at high temperatures. Radiation is dominant in combustion chambers, furnaces, incandescent filaments and other high temperature zones. The temperature of sun is also calculated using radiation heat transfer equation. At room temperature, radiation can be dominant only if conduction and convection are negligible. This type of situation may occur if heat transfer through a medium less volume is calculated like in vacuum bottle insulator. Although air temperature is higher than freezing point of water, but due to radiation losses from ground, ground may reach a temperature, which is lower than freezing point of water. This is the reason for formation of ice on ground at night despite higher atmospheric temperature.

Although governing equations are proposed by various researchers for all modes of heat transfer, any real life problem of heat transfer is generally solved using empirical relations because of various reasons. The equations rely on certain properties of matter and situation like conductivity, convective heat transfer coefficient and so on. Exact determination of these parameters reliably for the operating condition of the problem may not be possible. In addition to this the heat transfer may not exactly follow the governing equations discussed above. The effect of emissivity is illustrated above in this connection. The heat transfer experiments are also difficult and uncertain due to measurement inaccuracies, semi-empirical theories, boundary conditions, finite volume effects etc. at high heat flux, all bodies' deviates from Fourier's law of heat conduction. The convective heat transfer coefficient may have non-linear variation with temperature difference and temperature levels. Some of the materials also behave in an anomalous way in certain range of operating conditions.

▮▮ EXAMPLE 5.1

One face of an aluminium plate of 5 mm thick is maintained at 400°C and the other face is maintained at 50°C. How much heat is transferred through the plate? Take thermal conductivity of aluminium as 202 W/m.K.

SOLUTION

Heat transfer by conduction, $q/A = -k \cdot (\partial T/\partial x)$.

Given, k = 202 W/m.K, Temperature difference = 400°C – 50°C

= 350°C = 350 K, Thickness = 5 mm.

So, $q/A = 202 \times 350/5 \times 10^{-3} = 1.414 \times 10^7$ W/m^2 = 14.14 MW/m^2.

Heat transfer through the plate is 14.14 MW/m^2.

EXAMPLE 5.2

A 150 mm brick wall has an area of 23 m^2 and is made of fire brick (thermal conductivity = 1.04 W/m.K). Inside and outside temperatrues are 525°C and 20°C. How much heat is lost through the wall? If heating value of fuel oil is 40 MJ/kg, how much fuel oil is needed to balance this loss?

SOLUTION

Heat transfer by conduction, $q/A = -k\ .\ (\partial T/\partial x)$.

Given, k = 1.04 W/m.K, A = 23 m^2, Temperature difference = 525°C – 20°C = 505°C = 505 K, Thickness = 150 mm.

So, $q = 23 \times 1.04 \times 505/150 \times 10^{-3} = 8.053 \times 104$ W

$= 80.53$ kW.

Heat loss through the wall is 80.53 kW.

Quantity of fuel required to compensate the heat loss = 80.53/40000 kg/s = 0.002013 kg/s = 0.002013 × 3600 × 24 kg/day

= 173.94 kg/day

Amount of fuel oil needed per day is 173.94 kg.

EXAMPLE 5.3

A foam plastic box is 460 mm × 380 mm × 310 mm with a thickness of 15 mm. heat loss through the box is 70 W, when inner and outside temperatures are respectively 32°C and –6°C. Estimate thermal conductivity of the foam.

SOLUTION

Heat transfer by conduction, $q/A = -k\ .\ (\partial T/\partial x)$.

Given, Temperature difference = 32°C + 6°C = 38°C = 38 K,

Thickness = 15 mm, heat transfer = 70 W.

Heat conduction area = 2 × (460 × 380 + 380 × 310 + 310 × 460) mm^2 = 0.8704 m^2

So, $k = q \times \partial x/(A \times \partial T) = 70 \times 0.015/(0.8704 \times 38)$ W/m.K

= 0.03175 W/m.K.

Thermal conductivity of the foam is 0.03175 W/m.K.

EXAMPLE 5.4

A sphere of diameter 3 m has 50 mm thick wall of super insulation ($k = 1 \times 10^{-4}$ W/m.K). It contains liquid oxygen at 90 K. if the outside temperature is 20°C, how much liquid oxygen evaporates in g/day if heat of vaporization is 240 J/g?

SOLUTION

Heat transfer by conduction, $q/A = -k \cdot (\partial T/\partial x)$.

Given, $k = 1 \times 10^{-4}$ W/m.K, Temperature difference = 273 + 20 – 90 K = 203 K, Thickness = 50 mm.

Surface area of the sphere, $A = \pi D^2 = \pi \times 32\ m^2 = 28.27\ m^2$.

So, $q = 28.27 \times 1 \times 10^{-4} \times 203/0.050 = 11.479$ W.

Quantity of liquid oxygen evaporated = 11.479/240 g/s = 0.04783 g/s = 0.04783 × 3600 × 24 g/day = 4132.58 g/day.

EXAMPLE 5.5

Estimate internal temperature for a stainless steel sphere (thermal conductivity = 45 W/m.K) of 15 mm thick and 1 m outer diameter, which looses heat at the rate of 100 kW, when outside surface is maintained at 30°C.

SOLUTION

Heat transfer by conduction, $q/A = -k \cdot (\partial T/\partial x)$.

Given, $k = 45$ W/m.K, Thickness = 15 mm, Area for heat transfer = $\pi D^2 = \pi \times 12\ m^2 = 3.14\ m^2$, Heat loss = 100 kW.

So, $\partial T = q \times \partial x/(A \times k) = 100000 \times 0.015/(3.14 \times 45)\ K = 10.62$ K

Since there is a heat loss, outside temperature must be lower than inside temperature. Internal surface temperature of the sphere = 273 + 30 + 10.62 K = 313.62 K.

Internal surface temperature of the sphere is 313.62 K.

EXAMPLE 5.6

A 50 mm diameter 500 mm long cylinder originally at 60°C is to be cooled by immersing in a cross flow stream at 10°C. If rate of cooling is 300 W, find convective heat transfer coefficient.

SOLUTION

Heat transfer by convection, $q = h \times A \times (T_s - T_f)$.

Given, $q = 300$ W, $A = \pi \times 0.05 \times 5 = 0.7854\ m^2$, $T_s = 60°C$, $T_f = 10°C$.

So, $h = 300/[0.07854 \times (60 - 10)]\ W/m^2.K = 7.64\ W/m^2.K$

Convective heat transfer coefficient is 7.64 $W/m^2.K$.

EXAMPLE 5.7

Air at 27°C blows over a copper flat plate of dimension 20 cm × 30 cm × 6 cm is maintained at 250°C. The convection heat transfer coefficient is 50 W/m².K. Calculate the heat transfer. If 100 W of heat is lost by radiation, calculate *temperature difference across thickness of the plate. Take thermal conductivity of copper as 385 W/m.K.*

SOLUTION

Heat transfer by convection, $q = h \times A \times (T_s - T_f)$.

Given, h = 50 W/m^2.K, A = 0.2 × 0.3 = 0.06 m^2, T_s = 250°C, T_f = 27°C.

So, q = 50 × 0.06 × (250 – 27) = 669 W.

Total heat lost by convection and radiation = 669 W + 100 W
= 769 W.

This heat loss must be compensated by conductive heat transfer through the thickness.

Heat transfer by conduction, $q/A = -k \cdot (\partial T/\partial x)$.

So, Temperature difference = q × thickness/(A × k)
= 769 × 0.06/(0.06 × 385)
= 1.99°C.

EXAMPLE 5.8

Electric current is passed through a wire 1 mm in diameter and 10 cm long, submerged in water at atmospheric pressure and maintained at 100°C. For this situation h = 5000 W/m^2.K. How much electric power must be supplied to the wire to maintain the wire surface at 114°C?

SOLUTION

It is a problem of heat convection and power is supplied to compensate the heat loss taking place in wire, when submerged in relatively cooler water.

Heat transfer by convection, $q = h \times A \times (T_s - T_f)$.

Given, h = 5000 W/m^2.K, $A = \pi \times 0.001 \times 0.1$ = 0.000341 m^2, T_s = 114°C, T_f = 100°C.

Heat transfer, q = 21.99 W. This is amount of electric power required to accomplish the given task of maintaining surface temperature of wire as 114°C in wire submerged in water maintained at temperature of 100°C.

EXAMPLE 5.9

An immersion heater coil is 4 mm in diameter and 200 mm in length. It delivers 25 W. Find surface temperature, if it is immersed in surrounding temperature of 20°C in (a) water (convective heat transfer coefficient = 80 W/m^2.K) (b) air (convective heat transfer coefficient = 10 W/m^2.K).

SOLUTION

Heat transfer by convection, $q = h \times A \times (T_s - T_f)$.

Given, q = 25W, h = 80 W/m^2.K (a) and 10 W/m^2.K (b), $A = \pi \times 0.004 \times 0.2 = 0.002513$ m^2, T_f = 20°C.

From the given equation $T_s = T_f + q/Ah$

For water, surface temperature = 20 + 25/(0.002513 × 80)

= 144.34°C

For air, surface temperature = 20 + 25/ (0.002513 × 10)

= 1014.72°C.

EXAMPLE 5.10

A thin plate has one side insulated and other side is exposed to air flow at 20°C (convective heat transfer coefficient = 40 W/m^2.K). The plate is electrically heated at a rate of 12000 W/m^2. If radiation is neglected, find the equilibrium temperature of the plate.

SOLUTION

Heat transfer by convection, $q = h \times A \times (T_s - T_f)$.

Given, q/A = 12000 W/m^2, h = 40 W/m^2.K, T_f = 20°C.

From the given equation $T_s = T_f + q/Ah$

Equilibrium surface temperature of the plate = 20 + 12000/40

= 320°C.

EXAMPLE 5.11

If the radiant flux from a surface is 1350 W/m^2, what would be its equivalent blackbody temperature?

SOLUTION

Heat transfer by radiation, $q = \sigma \cdot A \cdot T^4$.

Given, q/A = 1350 W/m^2, $\sigma = 5.669 \times 10^{-8}$ W/m^2 K^4.

So, $T^4 = 1350/5.669 \times 10^{-8} = 2.38 \times 10^{10}$ and T = 392.83 K.

EXAMPLE 5.12

Electric current is passed through a wire 1 mm in diameter and 500 mm long. The electric power delivered is 20 W. If the wire is a blackbody radiating in a cold environment, estimate temperature of the wire, neglecting conduction and convection.

SOLUTION

Heat transfer by radiation, $q = \sigma \cdot A \cdot T^4$.

Given, q = 20 W, σ = 5.669 × 10^{-8} W/m^2 K^4. A = π × 0.001 × 0.5 m^2 = 0.00157 m^2

So, T^4 = 20/5.669 × 10^{-8} × 0.00157 = 2.246 × 1011, and T = 688.4 K.

Temperature of the wire is 677.4 K.

EXAMPLE 5.13

Differentiate between heat transfer by conduction, convection and radiation.

SOLUTION

Parameter	*Conduction*	*Convection*	*Radiation*
Requirement of media	Yes	Yes	No
Movement of media	No	Yes	NA
Governing law by	Fourier	Newton	Stefan Boltzman
Physical contact	Yes	Yes	No
Heat transfer proportional to	Temperature difference	Temperature difference	Fourth power of Temperature
Nature of heat transfer	Electronmotion and lattice vibration	Flow of fluids over a solid surface.	Electromagnetic radiation due to temperature of body.
Dependence on thermal gradient	Yes	Yes	No

In an internal combustion engine, maximum temperature of the combustion gases may be of the order of 2000 K to 2500 K. However, engine wall temperature is limited by several parameters – intermittent alternate high and low temperature generation, heat transfer coefficient between combustion gases and wall of the engine, specific heat to thermal conductivity of the engine material, heat loss to surrounding or taken away by cooling water. Study of heat transfer for the internal combustion engine becomes important from the severity of temperature to which cylinder material is subjected to while engine is operating. The temperature at the gas-side of the cylinder surface is restricted to less than 200°C to prevent deterioration or thinning of lubrication films. In fact high temperature may result in higher thermal stresses in the material also. Spark plug or valves must be cool enough to avoid any abnormal combustion phenomena. Heat transfer affects engine performance, efficiency and emission. During engine operation, if more heat is transferred to wall of the cylinder, average temperature of combustion gases is reduced and ultimately works done per cycle is reduced. Friction is another factor, which is omnipresent and is always associated with increase in temperature.

In the intake manifold, intake mixture is generally at lower temperature than wall of the engine cylinder and flow velocity are also higher. During this period, heat flows from wall to the incoming mixture or air. Sometimes provisions are also made to heat the incoming mixture by the high temperature exhaust for easy evaporation. During next stroke, mixture is compressed and both temperature and pressure rises. Rise in temperature results in reversal of heat transfer direction and now mixture or air becomes at higher temperature then engine wall. Heat transfer occurs from gas to cylinder wall. During combustion further heat addition occurs and the gas temperature rises further. This results in further rise in heat transfer (loss) from gas to cylinder walls. Next step is expansion stroke, where gases in the cylinder gain velocity and heat transfer is enhanced. Heat transfer to the wall is highest during this period due to higher temperature difference and higher overall heat transfer coefficient due to higher velocity. As expansion proceeds, gas temperature reduces and heat transfer is also restricted accordingly. During exhaust or blow-down process, again gases gain velocity but temperatures are relatively lower. However, exhaust gases will definitely transfer heat to all components of exhaust manifold.

As far as mode of heat transfer is concerned, all three modes of heat transfer are generally present in engine cylinder. Conduction is present in all the solid components of the engine namely cylinder head, cylinder walls and pistons etc. Convection is major mode of heat transfer inside cylinder from combustion gases to cylinder walls. Forced convection is observed between combustion gases and cylinder solid components coming in contact with the hot gases. The same mode of heat transfer is present from cylinder walls to coolant, which is generally in liquid or gaseous state. Lubrication, heat transfers in inlet and exhaust manifolds, heat loss to environment etc are all examples of forced convection heat transfer. Radiative heat transfer needs presence of high temperature, which is observed during combustion stroke (after compression stroke). Heat loss to environment through radiation is observed in engines and heat transfer from flame or hot gases to other exposed components always take place through radiation.

Various factors affect heat transfer process in an internal combustion engine. Heat loss during engine operation to environment reduces work output. It is observed that with increase in speed and load, importance of heat transfer is reduced. This is due to the fact that at higher speeds, time available for heat transfer reduces and actual heat transfer per cycle may be negligible during operation. However, if heat transfer per unit time is calculated, number of cycle increases per unit time at higher speeds and accordingly heat transfer rises as speed increases. Similar effects are observed for increment in load also. The effect of varying air: fuel ratio is directly taken as variation of load and heat transfer rates can be ascertained. Similarly if compression ratio is increased in an SI engine, heat transfer rate decreases. However the percentage change is not very significant. As compression ratio is associated with several other changes, other factors affect heat transfer rate from engine rather than direct effect of compression ratio on heat transfer rates. Increasing compression ratio increases (*i*) gas pressure (*ii*) gas temperature (*iii*) gas velocity (*iv*) combustion rate (*v*) surface to volume ratio and reduces gas temperature during exhaust stroke. These factors in combination give final estimate of heat transfer rates.

If spark timing in an IC engine is retarded, heat transfer rate reduces. Retarding spark timing makes fuel/air mixture burn at later instance in the cycle (larger chamber volume). This results in lower temperature during and at the end of combustion. Accordingly heat transfer rate is affected and mostly temperature change in piston and spark plug are affected by such changes. Due to turbulence inside the engine cylinder, gas velocity rise locally and heat transfer rate increase. Contrary to this, if temperature of circulating coolant is increased, the temperature at various components of the engine increases and accordingly heat transfer rates are affected. Heat transfer rate rise with increase in inlet temperature of incoming gases and knock.

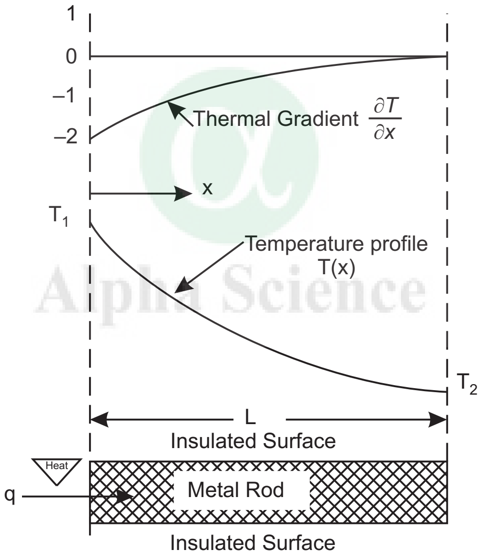
One of the important factors affecting heat transfer rate is material of construction of cylinder wall. Although by various means wall temperature is restricted to within 200°C, but it is observed that from heat transfer point of view and for fatigue cracking cast iron (conductivity = 54 W/m.K) can withstand temperature up to 400°C and aluminium alloys (conductivity = 155 W/m.K) can withstand up to 300°C. To restrict heat transfer, thermal insulation materials are applied at the inner surface of the cylinder walls to reduce temperature rise and heat transfer. Use of silicon nitride (conductivity = 5-10 W/m.K) and zirconia spray (conductivity = 1.2 W/m.K), is reported.

Although heat transfer is an auxiliary requirement for internal combustion engines, the design of cooling system needs thorough understanding of heat transfer. The efficient heat removal from the engine and complications due to elaborate cooling arrangement governs the final engine design. Other auxiliary systems like lubrication system also affect the same. Subsequent sections deals with these heat transfer processes in detail for understanding the actual heat transfer process in the engine cylinder.

5.3 HEAT CONDUCTION

The early development of heat conducting is largely due to the efforts of the French mathematician Fourier (1822). Although process of heat flow from a material at high temperature to that at low temperature was known, but relevant parameters and their dependence was established as Fourier's law.

As per Fourier's law, heat transfer by conduction is proportional to surface area and thermal gradient. A constant of proportionality 'k' is introduced, which is referred as thermal conductivity. This is explained in next section. A schematic of heat flow by conduction in a metallic rod is shown in figure 5.1. The rod is insulated at its lateral surface and heat conduction is taking place in its longitudinal direction depicted as 'x'. As heat flow takes place from higher temperature to lower temperature, direction of heat flow (q) in the given figure (5.1) is from left to right. In a given length 'L' of the rod, temperature variation from T_1 to T_2 is envisaged and temperature is dependent on distance from left end of the rod. As temperature reduces as 'x' increases, thermal gradient ($\partial T/\partial x$) is negative as shown in the figure. To make heat transfer value positive, a negative sign is introduced in heat transfer expression. The overall thermal gradient in a length 'L' of the rod can be given by ratio of gross difference of temperature and length ($T_1 - T_2/L$).



Heat transfer by conduction,

$$q = -k \cdot A \cdot \frac{\partial T}{\partial x} = k \cdot A \cdot \frac{T_1 - T_2}{L}$$

Fig. 5.1 : Heat Conduction through an Insulated Metal Rod

In fact the temperature profile and temperature gradient profiles are depicting a transient situation. When heat transfer process starts by increasing temperature of right end of the laterally insulated rod by brining it to certain high temperature, entire rod except a very small length will experience high temperature. Affected thickness will be small and for same temperature difference, heat transfer rate will be high. Temperature rise percolates to more part of the rod as time progresses. As affected thickness rises, heat transfer rate reduces. As time progresses a constant temperature gradient sets in the rod making heat transfer rate constant. In heat conduction work effects are negligible.

Heat conduction is mainly governed by thermal conductivity of material and it is explained in next section.

▮▮ EXAMPLE 5.14

A thermally insulated glass window 60 cm by 30 cm is made of two 8 mm thick pieces of glass sandwiching an 8 mm thick air space. Determine the conduction heat loss through the window, if it's inside and outside surface temperature are 20°C and –20°C, respectively. Determine the temperature at both internal glass-air interfaces. Neglect convective heat transfer. Thermal conductivity of air = 0.018 W/m.K and thermal conductivity of glass = 0.78 W/m.K.

SOLUTION

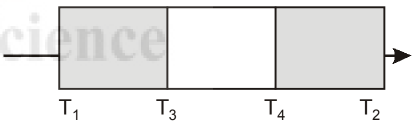
Heat transfer by conduction, $q = k \,.\, A \,.\, (T_1 - T_2)/L$

Given area of cross-section = 0.6 m × 0.3 m = 018 m^2.

Here same heat transfer takes place through glass and air. Here there are two interfaces of air and glass.

Temperature at outer surface of glass, T_1 = 20°C = 293 K.

Temperature at outer air-glass interface = T_3



Temperature at inner air-glass interface = T_4

Temperature at inner surface of glass, T_2 = – 20°C = 253 K.

For outer glass, $T_1 - T_3 = q \times 0.008/(0.18 \times 0.78) = 0.05698\, q$

For air gap, $T_3 - T_4 = q \times 0.008/(0.18 \times 0.018) = 2.4691\, q$

For inner glass, $T_4 - T_2 = q \times 0.008/(0.18 \times 0.78) = 0.05698\, q$

Adding three equations, $T_1 - T_2 = q \times 2.5831$.

So, heat transfer, q = (293 – 253)/2.5831 = 15.4853 W

Using equation for outer glass, T_3 = 293 – 0.88235 = 292.1176 K = 19.11°C.

Using equation for inner glass, T_4 = 253 + 0.88235 = 253.88235 K = – 19.11°C.

For cross-checking, $T_3 - T_4$ = 38.22°C and 2.4691q = 38.22°C.

5.4 THERMAL CONDUCTIVITY

Thermal conductivity is a parameter to indicate the quickness of heat transfer. A larger thermal conductivity means faster rate of heat transfer by conduction. The unit of thermal conductivity in SI unit is watts per meter per degree Kelvin (W/m.K). Mechanism of heat conduction is different in different phases.

In gases, thermal conduction is governed by kinetic theory of gases. As temperature of gas rises, molecules move with higher velocity. As molecules of gas collide randomly with each other and with wall of the container depending of their molecular motion, higher temperature means more number of collisions resulting in higher heat transfer. In fact molecular collisions are elastic and are associated with energy and momentum transfer. The molecular collision remains even in absence of temperature gradient, but heat transfer occurs only when high velocity molecules are colliding with low velocity molecules. Thermal conductivity of gases is dependent on temperature and for all practical purposes, it is independent of pressure. As per kinetic theory of gases, thermal conductivity of gas varies with square root of the absolute temperature. Gases show higher conductivity at higher temperatures. If density of gases is low, number of collisions of molecules will be less and thermal conductivity will be low. Kinetic theory of gases correlates thermal conductivity with molecular weight of gases. Thermal conductivity is proportional to square root of the molecular weight of gases. This is reason why helium (molecular weight = 4) has higher thermal conductivity than Argon (molecular weight = 40). For air thermal conductivity is around 0.024 W/m.K.

Heat conduction through liquid has same physical mechanism as gases, but molecules are not free to move that freely as gases. In liquids, molecules are more closely packed and are not free to collide and exchange energy. Molecular forces are predominant in these cases. The phenomena of heat conduction in liquids are very complex. Dependence on temperature is also very difficult to predict. Water has low thermal conductivity at room temperature, which rises with rise in temperature till around 125°C. Water has thermal conductivity of 0.556 W/m.K at 0°C. Raising temperature beyond this, results in reduction in thermal conductivity of water. Glycerin and benzene show slight reduction in thermal conductivity with certain rise in temperature till around 50°C. Since mercury is liquid metal, it has very high thermal conductivity (8.21 W/m.K) as compared to liquids.

In solids, heat transfer by conduction is mainly by lattice vibration and free electrons. In good conductors, main mechanism of heat conduction is transportation of heat by free electrons. The electron gases in conductors are responsible for transmission of electricity also. So, thermal conductors are invariably good electric conductors. This is the reason that copper (385 W/m.K), aluminium (202 W/m.K) and silver (410 W/m.K) are good electric as well as heat conductors. Iron and steel has varying thermal conductivity depending on alloying. Pure iron has a thermal conductivity of 73 W/m.K, but 1% C Carbon steel has a value of only 43 W/m.K. Alloying generally reduces thermal conductivity. Materials showing this mode of heat conduction have high value of thermal conductivity. If energy is transported through solid by lattice vibration, heat flow rate is low and these materials although solid but are not showing high heat conduction. If heat conduction through a material is not effective, they are called thermal insulators. Glass wool (0.038 W/m.K) and window glass (0.78 W/m.K) are examples of such materials.

Thermal conductivity is generally a material property and it depends on temperature. As temperature rises, thermal conductivity of gases increases. For liquid the values are relatively more or less stable. However for benzene it reduces and for water it attains a peak value at around 100 to 150°C and then reduces on both sides of the temperature. Amongst solids, variation of thermal conductivity with temperature is more or less not seen for iron and its alloys like stainless steel and carbon steel. However, for aluminium, it increases with temperature and for copper, it reduces with temperature. These results have pure practical consequences and are controlling parameters for selection of materials for various heat transfer applications.

As far as practical measurement of thermal conductivities is considered, it has several limitations. If range of thermal conductivity for solids is considered, copper has thermal conductivity of 385 W/m.K, while Rock wool has thermal conductivity of 0.04 W/m.K. no doubt copper is homogenous and value of its conductivity is reproducible. Contrary to this rock wool is a heterogeneous mass made of fiber and air voids. In this case apparent conductivity can only be measured, which depends on temperature, density or compaction and partial pressure of air space. Diamond has highest known conductivity. This makes diamond covering an ideal choice as heat sink on sensitive electronic semiconductors.

There is another thermal property, which depicts heat storage capacity of the material. Thermal conductivity is a measure of quickness with which heat is transmitted through material, while a complementary property called heat capacity depicts the amount of energy absorbed per unit volume of the material per unit time in temperature of the material. Heat transfer is generally associated with free expansion of material; the property is generally depicted at constant pressure. The base property is called specific heat and when multiplied with density of the material, it is called heat capacity. A material with higher density exhibits low specific heat. This is the reason why heat capacities of solids and liquids are almost in same range. Nickel and iron show high heat capacity while water also shows heat capacity in almost same range. The unit of heat capacity is $J/m^3.K$.

EXAMPLE 5.15

A plane slab is 120 mm thick and has its outside temperature as 0°C. With the inner temperature at 100°C, 200°C and 300°C, the measured heat fluxes are 15, 38 and 72 kW/m² respectively. Comment on variation of thermal conductivity of slab with temperature.

SOLUTION

Heat transfer by conduction, $q/A = k\ .\ (T_1 - T_2)/L$

So, $k = q \times L/[A \times (T_1 - T_2)]$

For the first situation, $k = 15 \times 0.12/(100 - 0)$ W/m.k

$= 0.018$ W/m.K

For the second situation, $k = 38 \times 0.12/(200 - 0)$ W/m.k

$= 0.0228$ W/m.K

For the third situation, $k = 72 \times 0.12/(300 - 0)$ W/m.k

$= 0.0288$ W/m.K

The conductivity of slab rises with rise in temperature difference.

5.5 GENERAL EQUATION FOR HEAT CONDUCTION

If this heat conductivity has directional property, then heat conduction equation for three dimensional Cartesian coordinate system is given below.

$$\frac{\partial}{\partial x}\left(k_x \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_y \frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k_z \frac{\partial T}{\partial z}\right) + \dot{q} = \rho_c \frac{\partial T}{\partial t}.$$

Here $\dot{q}$ is heat generated per unit volume, ρ is density of material and c is specific heat of the materials. If thermal conductivity of the material is isotropic, then all three values of thermal conductivity are same (say given by k), then above equation changes.

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} = \frac{1}{\alpha}\frac{\partial T}{\partial t}.$$

Here α is called thermal diffusivity of the material. This is a measure of rate of change of temperature of the body for a given heat flux. Larger the value of α, the faster heat will diffuse through the materials. High value of α means either rapid energy transfer rate (high k) or low value of heat capacity ($\rho.c$). A low value of heat capacity means less part of energy in the form of transmitted heat is absorbed by the material and used to raise the temperature of material. Metals and gases show high value of thermal diffusivity. Liquids (liquid metals), refractories (aluminium oxides) and insulating materials have low diffusivity. Diffusivity is one of the chief governing parameter in unsteady heat conduction, as obvious from the above mentioned equation.

If steady state heat conduction is considered, no variation of temperature with time is effective. Left hand side of the equation is not present and the resulting equation is called Poisson equation.

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}+\frac{\dot{q}}{k}=0.$$

For steady state conduction without heat generation, right most term of left hand side is also not present. This gives Laplace form of equation.

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=0$$

Similarly, heat conduction equation in cylindrical and spherical coordinate systems can also be written. For the cylindrical coordinates system, variables are r, θ and z. These variables are related to Cartesian coordinate system variables by the following relations.

$x = r \text{ Cos } \theta$, $y = r \text{ Sin } \theta$ and $z = z$.

Using this relation, the equation in Cartesian coordinates can be converted to cylindrical coordinates. The same is given below.

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \phi^2}+\frac{\partial^2 T}{\partial z^2}+\frac{\dot{q}}{k}=\frac{1}{\alpha}\frac{\partial T}{\partial t}.$$

The spherical coordinate system has a single radius R and two angles θ and ϕ. They are also related to Cartesian coordinate system and the relations are given below.

$x = R \text{ Sin } \theta \text{ Cos } \phi$, $y = R \text{ Sin } \theta \text{ Sin } \phi$ and $z = R \text{ Cos } \theta$.

Using above relations, the equation of heat conduction in Cartesian coordinates can be converted into spherical coordinates. The same is given below.

$$\left[\frac{1}{r}\frac{\partial^2 (rT)}{\partial r^2}+\frac{1}{r^2 \sin\theta}\frac{\partial}{\partial \theta}\left(\sin\theta\frac{\partial T}{\partial \theta}\right)+\frac{1}{r^2\sin^2\theta}\frac{\partial^2 T}{\partial \phi^2}+\frac{\dot{q}}{K}=\frac{1}{\alpha}\frac{\partial T}{\partial t}\right]$$

The equation of heat conduction in three dimensions is of second order in space and first order in time. For solution of this equation boundary conditions are to be specified carefully. Temperature is represented as $T(x, y, z, t)$, which shows dependence of temperature (T) on spatial (x, y, z) and temporal (t) variables. There are many forms of boundary conditions possible for such situations. To tackle temporal variation of temperature, initial boundary conditions are specified, which are nothing but specification of temperature in the beginning. This is given as $T(x, y, z, 0)$. On similar lines temperature can be specified at any other time-step also like $T(x, y, z, t_1)$. First form of boundary condition is known temperature at certain spatial coordinates. Another form of boundary condition is expressed as exposure to fluid at certain temperature. In this form convective heat transfer coefficient at the interface surface must be known. The heat conduction in the body is made equal to heat convection at the surface. Yet another type of boundary condition is present, when insulated walls are encountered. Definitely there is no heat conduction through the wall and first derivative of temperature with respect to spatial coordinate perpendicular to the wall is zero. Sometimes heat flux at the boundary is also prescribed.

EXAMPLE 5.16

A slab of thickness 100 mm has conductivity of 25 W/m.K, density 2700 kg/m^3 *and specific heat of 510 J/kg.K. It has an instantaneous temperature distribution,* T *(K) = 500 – 2500x + 6000x*^{2}*, x in m and x = 0 denotes left hand side surface. How much energy is instantaneously received by the slab in W/m*^{2}*? At what rate is the temperature at the center of the slab changing in K/s?*

SOLUTION

Temperature on left side of the slab, $T(0) = 500$ K

Temperature on the right side of the slab, $T(0.1) = 310$ K

Energy received by the slab, $q/A = -k\,(T_1 - T_2)/L$

$$= -25 \times (310 - 500)/0.1 \text{ W/m}^2 = 47500 \text{ W/m}^2$$

Thermal diffusivity, $\alpha = k/\rho.c = 25/(2700 \times 510)$ m^2/s

$$= 1.815 \times 10^{-5} \text{ m}^2\text{/s}$$

$$(\partial^2 T/\partial x^2) = 1/\alpha\ (\partial T/\partial t)$$

$$(\partial T/\partial t) = \alpha\ (\partial^2 T/\partial x^2)$$

$$= 1.815 \times 10^{-5} \times 12000 \text{ K/s} = 0.21786 \text{ K/s.}$$

5.6 1-D HEAT CONDUCTION

If heat conduction is taking place predominantly in one direction, it is called 1-D heat conduction. Alternatively, if one dimension is much bigger than other two dimensions of the body, heat transfer may be assumed to be taking place in one direction only. The situation may be achieved by properly insulating the material surface to restrict heat conduction. For 1-D heat conduction, concept of thermal resistance is very important. Analogy can be derived from electrical current flow, which needs a potential difference. It is clear that heat flow takes place due to temperature difference, which may be considered as thermal potential difference. For 1-D heat conduction in Cartesian coordinate, Fourier law is depicted below.

Heat transfer, q = Thermal potential difference/ Thermal resistance = $[T_1 - T_2]/(L/k\ .\ A)$
Thermal resistance, $R = L/k.A$.

Heat conduction through the thickness of a wall is governed by 1-D heat conduction equation. Heat conducted is proportional to average conductivity, wall area and the temperature difference and it is inversely proportional to the wall thickness.

The temperature distribution across a 1-D thick wall depends on thermal conductivity variation with temperature. If thermal conductivity of the wall is independent of temperature, the temperature distribution across wall is linear. Temperature becomes proportional to distance from any side of the wall. The temperature variation across walls of such 1-D heat conduction is as follows.

$$T - T_1/T_2 - T_1 = x - x_1/x_2 - x_1.$$

Non-linearity in variation of temperature with distance is introduced by variation of thermal conductivity of material of the wall with temperature. If conductivity increases with temperature, more part of material thickness is maintained at higher temperature and the variation of temperature with distance from one edge is concave downward. Reverse is situation, if thermal conductivity reduces with temperature.

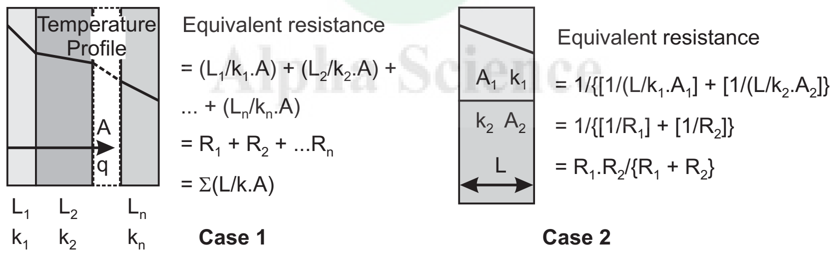


Fig. 5.2 : Thermal Resistances for Heat Conduction in Cartesian Coordinate

If multiple materials are employed for heat conduction, figure 5.2 can be referred for calculation of equivalent resistance. If materials of same cross-section area are stacked one beside the other and heat flows by conduction through them, they are found to act as thermal resistance in series (case 1 of figure 5.2). In this case equivalent resistance is given by sum of individual thermal resistances. Contrary to this if same thickness of two materials is stacked in such a way that heat conduction is split between the given materials (depicted in case 2 of figure 5.2), the reciprocal of equivalent thermal resistance is given by sum of reciprocals of individual resistances.

Heat conduction in one-dimension can take place in a cylinder in radial direction. If cylinder has a length of 'L', then area of cross-section, $A_r = 2\pi r.L$. If inner surface at radius of r_1 is maintained at temperature of T_1 and outer surface at temperature of r_2 is maintained at T_2, heat flows in radial direction. In this case of heat transfer, area of cross-section, for heat conduction, increases with increase in radius. This results in lower temperature variation at higher radii for same radial thickness.

The Fourier equation for cylindrical coordinate in one dimension is as follows.

Heat conduction, $q = -k \,.\, A_r \,.\, dT/dr = -2\pi r.L.k.dT/dr$. on integration, this results in equation of heat conduction in cylindrical coordinate as below.

Heat conduction, $q = 2\pi.L.k.(T_1 - T_2)/\ln(r_2/r_1)$. This clearly indicates that

Thermal resistance = $\ln(r_2/r_1)/2\pi.L.k$.

$T - T_1/T_2 - T_1 = \ln(r/r_1)/\ln(r_2/r_1)$.

On similar lines, for spherical coordinates,

$$q = -k \,.\, A_r \,.\, dT/dr = -4\pi R^2.k.dT/dr.$$

$$\text{Heat conduction, } q = 4\pi.k\,(T_1 - T_2)/(1/R_1 - 1/R_2)$$

$$\text{Thermal resistance} = 1/R_1 - 1/R_2/(4\pi.k).$$

$$T - T_1/T_2 - T_1 = (1/R_1 - 1/R)/(1/R_1 - 1/R_2).$$

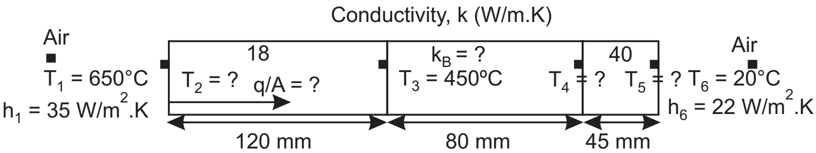
The concept of thermal resistance can be used in case of convective heat transfer also. As per Newton's law of cooling,

$$q = h \times A \times (T_s - T_f) = (T_s - T_f)/(1/h.A).$$

$$\text{So, thermal resistance} = 1/(h.A).$$

▮▮ EXAMPLE 5.17

Structure of a composite wall is given in the figure. If T_3 measures 450°C, find (i) heat transfer, (ii) conductivity of central material and (iii) T_2, T_4, T_5.



SOLUTION

Between station 1 and 3, thermal resistance, $R_{1\text{-}3}$ = 1/35 + 0.12/18 m^2.K/W = 0.0352 m^2.K/W.

Heat transfer, q/A = [650 – 450]/0.0352 W/m^2

= 5675.67 W/m^2.

Between station 3 and 6, thermal resistance, $R_{3\text{-}6} = 0.08/k_B$ + 0.045/40 + 1/22 m^2.K/W

Heat transfer, $q/A = 5675.67 = [450 - 20]/ R_{3\text{-}6}$. Solving for unknown conductivity, $k_B = 2.74$ W/m.k between station 1 and 2, q/A = 5675.67 = 35 × [650 – T_2]. So, $T_2 = 487.84$°C.

Between station 5 and 6, q/A = 5675.67 = 22 × [T_5 – 20]. So, T_5 = 277.98°C.

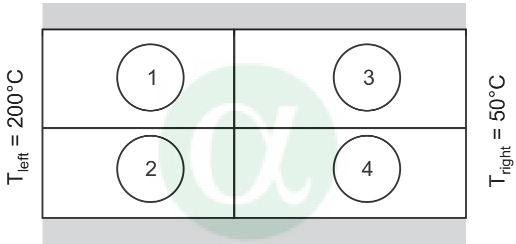
Between station 4 and 5,

q/A = 5675.67 = 40 × [T_4 – 277.98]/0.045. So, T_4 = 284.37°C.

The calculations can be checked for accuracy for T_4 from left hand side also.

Between station 3 and 4, q/A = 5675.67 = 2.74 × [450 – T_4]/0.08. So, T_4 = 284.37°C.

▮▮ EXAMPLE 5.18



Parameters	1	2	3	4
K, W/m.K	5	8	12	3
A, m^2	0.4	0.6	0.4	0.6
L, mm	50	50	80	80

A grid of four materials is shown in figure heat transfer takes place in one dimension in horizontal direction. Top and bottom surfaces are insulated. Calculate heat transfer assuming (i) two series thermal resistances in parallel, (ii) 2 parallel thermal resistances in series and (iii) explain reason in variation and which is correct?

SOLUTION

Thermal resistances in series gets added up to get equivalent resistance, while inverse of equivalent thermal resistance in parallel are obtained by sum of reciprocals of individual thermal resistances.

Thermal resistance of 1, R_1 = L/A.K = 0.0250 K/W

Thermal resistance of 2, R_2 = L/A.K = 0.0104 K/W

Thermal resistance of 3, R_3 = L/A.K = 0.0166 K/W

Thermal resistance of 4, R_4 = L/A.K = 0.0444 K/W (*i*) In first case thermal resistances of 1 and 3 are in series and so is case with 2 and 4. Both these resistances are in parallel.

$R_{1\text{-}3} = R_1 + R_3 = 0.0416$ K/W and $R_{2\text{-}4} = R_2 + R_4 = 0.0548$ K/W

$R_{eq} = R_{1\text{-}3} \times R_{2\text{-}4}/[R_{1\text{-}3} + R_{2\text{-}4}] = 0.02367$.K/W

Heat transfer, q/A = [200 – 50]/0.02367 W = 6335.016 W.

(*ii*) In second case thermal resistances of 1 and 2 are parallel and so is case with 3 and 4. Both these resistances are in series.

$R_{1\text{-}2} = R_1 \times R_2/[R_1 + R_2] = 0.0073446$ K/W and

$R_{3\text{-}4} = R_3 \times R_4/[R_3 + R_4] = 0.0121$ K/W

$R_{eq} = [R_{1\text{-}2} + R_{3\text{-}4}] = 0.019427$.K/W

Heat transfer, q/A = [200 – 50]/0.019427 W = 7721.11 W

(*iii*) Both schemes are solution of the same problem, but equivalent thermal resistance and heat transfer are different. In my opinion, first case depicts the situation depicted in the figure more correctly. In second case, the interface of 1-3 and 2-4 are assumed to be at the same temperature, which is not true. This violates 1-D heat conduction and the expressions used are erroneous.

EXAMPLE 5.19

A house wall consists of 20 mm plaster (conductivity = 0.5 W/m.K), 120 mm glass fiber (conductivity = 0.035 W/m.K) and 40 mm of oak wood (conductivity = 0.17 W/m.K). Inner and outer heat transfer coefficients are 10 and 25 W/m^2.K and temperatures are 30°C and – 15°C. Find heat loss.

SOLUTION

The configuration has five thermal resistances – convection at inner surface + conduction by plaster, glass fiber and oak wood in sequence + convection at outer surface. The convective resistance is 1/h, while conductive resistance is thickness/k, where h = convective heat transfer coefficient and k = thermal conductivity.

Thermal resistance, R = 1/10 + 0.02/0.5 + 0.12/0.035 + 0.04/0.17 + 1/25 m2.K/W = 3.84386 m^2.K/W.

Heat loss per unit area = temperature difference/thermal resistance = [30 – (– 15)]/3.84386 W/m^2 = 11.71 W/m^2.

EXAMPLE 5.20

A furnace is 3 m by 4m by 5 m with a composite wall of 150 mm silica brick (conductivity = 1.1 W/m.K), 50 mm glass fiber (conductivity = 0.035 W/m.K) and 10 mm carbon steel (conductivity = 45 W/m.K). Inner conditions are T_1 =

450°C and h_1 = 15 W/m^2.K; Outer conditions are T_2 = 20°C and h_1 = 20 W/m^2.K; Find heat loss per unit area and temperature at brick glass fiber interface.

SOLUTION

The configuration has five thermal resistances – convection at inner surface + conduction by silica brick, glass fiber and carbon steel in sequence + convection at outer surface. The convective resistance is 1/h, while conductive resistance is thickness/k, where h = convective heat transfer coefficient and k = thermal conductivity.

Thermal resistance, R = 1/15 + 0.15/1.1 + 0.05/0.035 + 0.01/45 + 1/20 m^2.K/W = 1.6818 m^2.K/W.

Heat loss per unit area = temperature difference / thermal resistance = [450 – 20] / 1.6818 W/m^2 = 255.675 W/m^2.

Resistance from inner side till silica brick glass fiber interface

= 1/15 + 0.15/1.1 m^2.K/W = 0.203 m^2.K/W.

Heat transfer as calculated earlier is valid here

255.675 = [450 – T]/0.203, or T = 398°C.

▮▮ EXAMPLE 5.21

For the problem above, maximum temperature in fiberglass is limited to 320°C. How much extra silica brick should be added?

SOLUTION

The data is taken from previous problem. But it must be considered that, calculated heat transfer of previous problem cannot be used here. If thickness of silica brick is increased to restrict temperature in glass fiber, the heat transfer calculated in previous problem will not be valid. Increase in thermal insulation leads to reduction in heat transfer. For calculation of heat transfer, the temperature at the inner surface of fiber glass is given and thermal resistance external to this plane is considered.

Thermal resistance, R_{outer} = 0.05/0.035 + 0.01/45 + 1/20 m^2.K/W = 1.47879 m^2.K/W.

Heat transfer, q/A = [320 – 20]/Router = 202.868 W/m^2.

(This heat transfer is less than value of 255. 675 W/m^2 of the previous problem)

This is heat transfer for the internal surface also. Let 'L' be thickness of additional silica brick added, then thermal resistance R_{inner} = 1/15 + (0.15 + L) /1.1 m^2.K/W.

$202.868 = [450 - 320]/R_{inner}$. Or, $L = 0.4815\ m = 481.5$ mm. Thickness of additional silica brick is 481.5 mm.

EXAMPLE 5.22

A water-heater tank is 1000 mm in diameter and 2000 mm high. It is covered with 50 mm thick rock wool insulation (thermal conductivity = 0.04 W/m.K). Metal wall resistance is negligible. If temperature of water is 60°C and surrounding is at 15°C, find heat loss.

SOLUTION

Heat transfer by conduction, $q = 2\pi.\text{L.k}.(T_2 - T_1)/\ln(r_2/r_1)$

Given $k = 0.04$ W/m.K, $D = 1000$ mm, $L = 2000$ mm, $t = 50$ mm, $T_1 =$ 60°C, $T_2 =$ 15°C.

Here, internal radius, $r_1 = 1000/2$ mm = 500 mm, $r_2 = (1000+50)/2$ mm = 525 mm

So,
$$q = 2\pi.\text{L.k}.(T_1 - T_2)/\ln(r_2/r_1)$$
$$= -2\pi \times 2000 \times 0.04 \times (60 - 15)/\ln(525/500)\ \text{W}$$
$$= -463607.1\ \text{W} = -0.4636\ \text{MW}$$

Heat loss is 0.4636 MW.

EXAMPLE 5.23

A thick walled tube of stainless steel (k = 19 W/m.K) with 2 cm inner diameter and 4 cm outer diameter is covered with a 3 cm layer of asbestos insulation (k = 0.2 W/m.K). If the inside temperature of the pipe is maintained at 600°C and outside temperature is restricted to 100°C, calculate the heat loss per meter of length.

SOLUTION

Thermal resistance of steel tube = $\ln(r_2/r_1)/2\pi.\text{L.k} = \ln(2/1)/2\pi \times 19 = 5.806 \times 10^{-3}$ W/K.

Thermal resistance of insulation = $\ln(r_2/r_1)/2\pi.\text{L.k} = \ln(5/2)/2\pi \times 0.2 =$ 0.72916 W/K.

Since these two materials are in series, equivalent resistance is sum of individual resistances = 0.73496 W/K.

Heat loss per meter length = temperature difference/thermal resistance = (600 – 100)/0.73496 W = 680.3 W.

EXAMPLE 5.24

A hollow cylinder with inner and outer radius as 40 mm and 60 mm respectively is maintained at 30°C and 100°C at inner and outer surface respectively. If is surrounded with fluid at 250°C. Find average thermal *conductivity of the cylinder of length 2000 mm, if convective heat transfer coefficient is 50 W/m^2.K.*

SOLUTION

Given r_1 = 40 mm, r_2 = 60 mm, L = 2000 mm, T_1 = 30°C, T_2 = 100°C, T_∞ = 250°C, h = 50 W/m^2.K.

Heat transfer at the surface, $q = h \times \text{A} \times (T_\infty - T_2) = h \times 2\pi r_2 L \times (T_\infty - T_2)$ $= 50 \times 2\pi \times 0.06 \times 2 \times (250 - 100)$ W = 5654.86 W.

Heat conducted through the tube = heat convected at the surface 5654.86 $W = 2\pi.\text{L.k.}(T_2 - T_1)/\ln(r_2/r_1) = 2\pi \times 2x\ .k.\ (100 - 30)/\ln(0.06/0.04)$.

So, k = 8.1887 W/m.K.

▮▮ EXAMPLE 5.25

A hollow stainless steel sphere has D_1 = 400 mm, D_2 = 600 mm, T_2 = 50°C. It is cooled by a fluid with T_∞ = 15°C and h = 145 W/m^2.K. What is inner temperature?

SOLUTION

For stainless steel, thermal conductivity is taken as 45 W/m.K.

Heat convection at the outer surface, $q = h \times A \times (T_2 - T_\infty) = 145 \times 4\pi \times 0.3^2 \times (50\text{-}15)$ W = 5739.69 W.

Heat conducted through the sphere = heat convected at the surface

$4\pi.k(T_1 - T_2)/(1/R_1 - 1/R_2)$ = 5739.69 W

Or $4\pi \times 45 \times (T_1 - 50)/(1/0.2 - 1/0.3) = 5739.69$

Or, T_1 = 66.9°C

Inner temperature is 66.9°C.

▮▮ EXAMPLE 5.26

A hollow sphere of uniform conductivity has inner, mid-point and outer temperature of 5°C, 25°C and 40°C. Outer diameter is 150 mm, find inner diameter.

SOLUTION

Let inner, midpoint and outer conditions are depicted by subscripts 1, 2, 3 respectively.

Given, that conductivity of material is uniform and T_1 = 5°C, T_2 = 25°C, T_3 = 40°C, D_3 = 150 mm, find D_1.

Since conductivity of the material is uniform, the heat transfer between inner and midpoint is same as that between midpoint and outer surface.

$T_2 - T_1/(1/R_1 - 1/R_2) = T_3 - T_2/(1/R_2 - 1/R_3)$

As terms are uniform, radius can be replaced with diameters on both sides. It is given that $D_2 = (D_1 + D_3)/2$, because of 2 is subscript for the mid-point.

$$(25 - 5)/(1/D_1 - 1/D_2) = (40 - 25)/(1/D_2 - 1/D_3)$$
$$20/(1/D_1 - 1/D_2) = 15/(1/D_2 - 1/D_3)$$
$$3/D_1 - 3/D_2 = 4/D_2 - 4/D_3$$
$$3/D_1 + 4/D_3 = 7/D_2$$
$$3/D_1 + 4/D_3 = 14/(D_1 + D_3)$$
$$(3D_3 + 4D_1) \times (D_1 + D_3) = 14\ D_1D_3$$
$$3D_3D_1 + 3D_3^2 + 4D_1^2 + 4D_1D_3 = 14\ D_1D_3$$
$$4D_1^2 - 7D_1D_3 + 3D_3^2 = 0$$
$$(4D_1 - 3D_3) \times (D_1 - D_3) = 0$$
$$D_1 = 3D_3/4 \text{ or } D_3$$

Second value is inconsistent. So, inner diameter, $D_1 = 3 \times 150/4 = 112.5$ mm.

■ ■ EXAMPLE 5.27

Frustum of a cone has $T_1 = 150°C$ at the base with diameter 100 mm. The top surface at a distance of 150 mm, has diameter of 50 mm and temperature $T_2 = 30°C$. Assume one dimensional heat flow in axial direction with lateral surface fully insulated. If thermal conductivity of the material is 45 W/m.K, find (a) heat transfer rate and (b) temperature at a distance of 75 mm from the bottom.

SOLUTION

This is a problem of 1-D steady state heat conduction in axial direction through variable area of cross-section. Let lateral direction is denoted by 'x' and bottom is maintained at $x_1 = 0$. Top surface is at $x_2 = 150$ mm. $D_1 = 100$ mm and $D_2 = 50$ mm. General equation of diameter, $D = 100 - x/3$.

Heat conduction through the cone frustum, $q = A.k.\ dT/dx = k.\ dT/[dx/A] = k.(T_1 - T_2)/[4.dx/\pi.D^2]$

(integrating along length)

$$= 45\pi \times (150 - 30)/[4 \times 3 \times (1/0.05 - 1/0.1)]$$
$$= 141.37 \text{ W.}$$

If insulation is added on a wall, thermal resistance generally increases. This rise in resistance results in less heat transfer and prevents reduction in inside temperature by heat loss. Such situations are valid for Cartesian coordinate system where surface area through which heat loss is taking place is constant. The rise on thermal resistance is due to increase in thickness encountered by heat flow. However, if surface area for heat transfer changes as in the case of cylinders in radial outward direction, same situation may not hold good. Adding insulation increases thickness encountered by heat flow, but simultaneous increase in area may result in a lump-sum effect of reduced thermal resistance. Such situations are seen in cylindrical and spherical coordinate systems.

Let us consider a tubular pipe of outer diameter r_1 placed in a fluid stream at temperature of T_f. If insulation layer is placed so as to make outer diameter r_2 such that interface temperature of tube and insulation is maintained at T_i, heat will flow from inside to outside through the thermal insulation. Heat transfer through insulation takes place through conduction while at the outer interface, convection is effective. Heat transfer through insulation can be expressed by concept of thermal resistance.

Heat transfer, $q = T_i - T_f/[\{\ln (r_2/r_1)/2\pi.L.k\} + \{1/(2\pi.r_2.L.h)\}]$.

For maximization of heat transfer through the insulation, above expression is differentiated with respect to r_2 and equated to zero. This gives an expression as r_2 = k/h. This indicates that, if outer radius of tube after application of insulation is less than the value given by the expression, heat transfer will increase after addition of insulation-layer. However, if outer layer is more than critical thickness of insulation, addition of insulation layer results in decrease of heat transfer and insulation will, in true sense will give the insulation effect. This is called critical radius of insulation and for lower values of r_2, convective heat transfer coefficient should be high. At this critical radius of insulation, heat loss is maximum and addition of insulation till this thickness decreases thermal resistance and increases heat loss. However, for radii higher than this radius, insulation layers behave in same way as they behave for Cartesian coordinates.

From practical point of view, insulation has thermal conductivity of around 0.04 W/m.K and lowest value of convective heat transfer coefficient is 5 W/m^2.K. For such values, critical thickness of insulation is around 8 mm obviously; industrial pipes and other fluid flow channels in heat exchangers, condensers or other devices are larger than this thickness. So, chances of increased heat transfer by adding insulation is not seen for such situations. But electric wires are some times smaller than this critical radius and adding insulation may further enhance the heat loss. So, such wires are left uninsulated (bare) for reduced heat loss. Of course adding insulation may change the convective heat transfer coefficient and make the system more complex to analyze by such simple assumptions.

Critical thickness of insulation
= k/h for cylindrical coordinate system
= 2.k/h for spherical coordinate system.

EXAMPLE 5.28

Calculate the critical radius of insulation (k = 0.1 W/m.K) surrounding a pipe and exposed to room air at 20°C with h = 2 W/m^2.K. Calculate heat loss at 200°C, 5 cm diameter pipe, when covered with the critical radius of insulation and without insulation.

SOLUTION

Given, k = 0.1 W/m.K, h = 2 W/m^2.K, T_f = 20°C, T_i = 200°C, r_1 = 2.5 cm = 0.025 m.

Critical radius of insulation, r_2 = k/h = 0.1/2 m

= 0.05 m = 5 cm.

Heat transfer per unit length of pipe without thermal insulation, assuming same convective heat transfer coefficient is valid for pipe and surrounding fluid is

Given by, $q = 2\pi.r_1.h.\ (T_i - T_f)$

$= 2\pi \times 0.025 \times 2 \times (200 - 20) = 56.548$ W/m.

Thermal resistance per unit length of pipe with insulation layer, placed on the outer surface of the pipe is given by [{ln (r_2/r_1)/2π.L.k} + {1/(2π.r_2.L.h)}] = {ln(5/2.5)/(2π × 0.1)} + {1/(2π × 0.05 × 2)} = 2.6947 K.m/W.

Heat transfer, q = (200 – 20)/2.6947 W/m = 66.797 W/m.

In this case placing insulation increases heat transfer through the pipe and increase is of the order of 18% more in effective heat transfer by applying insulation.

EXAMPLE 5.29

An aluminium wire (conductivity = 238 W/m.K) has a diameter of 3 mm and a surface temperature of 150°C with ambient condition T_∞ = 20°C and h_0 = 8 W/m^2.K. What is maximum allowable conductivity of the insulation that can be added on this wire that will cause a reduction in heat flux? If conductivity of the insulation is 0.035W/m.K, what are critical radius and the maximum heat flux for that condition?

SOLUTION

The maximum allowable conductivity of the insulation that can be added to the wire without adversely affecting the heat transfer is basically critical thickness of insulation. The problem demands calculation of thermal conductivity for which critical thickness of insulation is equal to wire radius. It is assumed that convective heat transfer coefficient does not change by addition of thermal insulation.

Required conductivity, $k = r \times h_o = 0.0015 \times 8$ w/m.K

$= 0.012$ W/m.K.

Any thermal conductivity of insulation material higher than this will reduce critical thickness of insulation to less than diameter of the wire and insulation addition will reduce heat flux.

Critical radius for the given conductivity, $r_c = k/h_o = 0.035/8$ m = 0.004375 mm.

Maximum heat flux occurs, when critical thickness of insulation is applied on the surface.

Thermal resistance, $R = \ln (r_c/r_o)/[2\pi.L.k] + 1/[2\pi.r_c.h_o] = \ln (0.004375/0.0015)/[2\pi \times 0.035] + 1/[2\pi \times 0.004375 \times .8]$ m.K/W = 9.41488 m.K/W

Heat flux, $q = [150 - 20]/9.41488$ W/m = 13.81 W/m.

In automobiles, application of fins for rapid cooling is regularly employed. This also acts on combined principle of conduction and convection. Fins are designed to increase heat transfer, so that engines can be cooled faster. This is opposite of thermal insulation. Fins increase heat transfer by providing additional surface area. Electronic equipments are finned for cooling. Gas side of the heat exchangers is also finned. They are extended surfaces from the base surface and are also called heat transfer augmenters.

For fins, effective control is a combination of heat conduction area, heat convection perimeter, thermal conductivity and convective heat transfer coefficient. A combined parameter, 'm' is defined. Parameter, **$m = \sqrt{(h.P/A.k)}$, where h = convective heat transfer coefficient, P = perimeter area or lateral area for heat convection, A = end area for heat conduction, k = thermal conductivity of fin.** This parameter is not dimensionless. It has a unit of per unit radius and is generally written as per unit length. End surface of the fin protruding in fluid is considered insulated. This also means that either surface area reduces to zero or it is physically insulated. This is to ensure simplified calculation. Additionally, it also signifies that compared to convective heat transfer through lateral surface is much higher than that at the end faces. Some of the fin configurations are shown in figure 5.3.

To analyze heat transfer through fins, a rectangular projecting surface from a plane is considered as shown in figure 5.4. Base of the projection is maintained at certain wall temperature (T_w) and outside atmosphere is maintained at other temperature (T_∞). The area of protruding rectangular cross-section is A (= $Z.t$) and its thickness and width are 't' and 'Z' respectively. Length of protruding surface is 'L'. For the figure shown, there are two modes of heat transfer – one is conduction through the protruded volume and other is heat convection through lateral and end surfaces. An infinitesimal length (dx) of the protrusion is taken for analysis. Net heat transfer in 'x'-direction is by conduction alone and is governed by Fourier's equation. Net heat transfer by conduction through the protrusion is equal to $-AK.dx.d^2T/dx^2$.

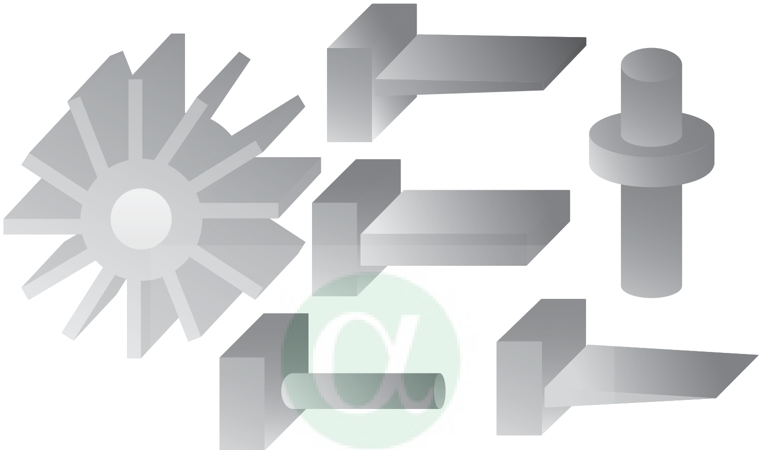


Fig. 5.3 : Different Fin Configurations

Heat convection is also depicted in the figure 5.4. Heat balance equation gives the following equation.

$$d^2T/dx^2 - (hP/AK)\,(T - T_\infty) = 0$$
$$d^2\theta/dx^2 - m.\,\theta = 0, \text{ where, } \theta = T - T_\infty \text{ and } d\theta = dT.$$

This is a simple second order differential equation and solution of the same is given by

$$\theta = C_1.e^{-mx} + C_2.e^{mx}.$$

It is given that

When $x = 0$ (at the base of the protrusion),
$T = T_w$ and $\theta_0 = T_w - T_\infty$.

Several cases of the protrusion environment can be considered from the solution of above mentioned equation with suitable boundary conditions.

First case is considering a very long protrusion. This invariably indicates that open end of the protrusion attains atmospheric temperature. The boundary conditions are (*i*) at $x = 0$, $\theta = \theta_0$ and (*ii*) at $x = \infty$, $\theta = 0$. Second boundary condition gives $C_2 = 0$ and equation changes to $\theta = C_1.e^{-mx}$. First boundary condition gives $C_1 = \theta_0$. This gives solution as $\theta = \theta_0\ e^{-mx}$. Total heat transfer through the protrusion is given by $\boldsymbol{\theta_0\ x\ \sqrt{(hPkA)}}$.

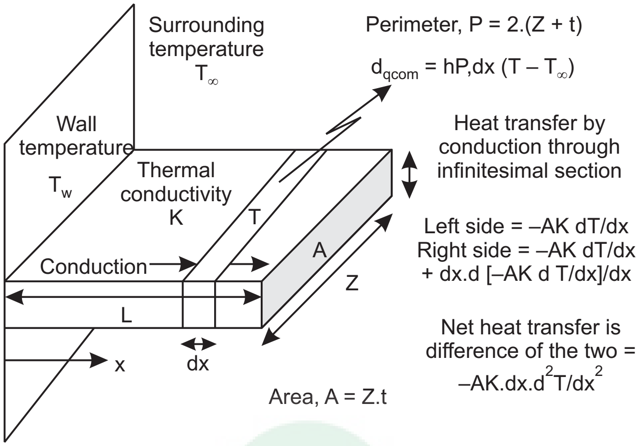


Fig. 5.4 : Heat transfer through protrusions Depicting Fins

Second situation is when protrusion length is finite, but ends of the protrusion are insulated and there is no heat conduction through the outer surface of the protrusion. First boundary condition remains same (at $x = 0$, $\theta = \theta_0$) as it was earlier, and the boundary condition gives $C_1 + C_2 = 0$. However second boundary condition changes. At finite length '*L*', thermal gradient ($d\theta/dx$) has to be zero to make zero heat conduction ($-\text{AK}d\theta/dx$) at outer wall. This gives boundary condition differently. $\boldsymbol{d\theta/dx = m.[-C_1.e^{-mx} + C_2.e^{mx})}$ is mathematical representation of this boundary condition. The final solution is $\theta = \boldsymbol{\theta_0 \cosh [m(L - x)]/\cosh mL.}$ Total heat transfer is given by $\boldsymbol{\theta_0 \tan h\ (mL) \times \sqrt{(hPkA)}}$. This is most promising case observed in practical situations.

However, a more general case is observed, which needs complicated mathematical operations. If protrusion of finite length is considered and there are finite heat losses from the outer ends of the protrusion is also considered, then second boundary condition changes and solution needs more calculations. The final result is given as

$\theta = \theta_0$ {cos *h* [m(*L* – *x*)] + (h/mK).sinh [*m*(*L* – *x*)]}/{cos *h* mL + (h/mK).sin *h* mL}.

Total heat transfer is given by θ_0 [(h/mK).cos *h* mL + sin *h* mL]/[cos *h* mL + (h/mK).sin *h* mL} × √(*h* PkA).

Numerical evidence for importance of fins is expressed by certain ratios. Usefulness of heat transfer through such protrusions can be expressed by ratio of actual heat transfer with fins and heat transfer through fin when entire protrusion area is maintained at the base temperature (T_b). This term is called fin efficiency and is a number between zero and one. It is given by following expression for second situation (infinite fin length) depicted above.

Fin efficiency = tan *h* (mL)/mL.

It is observed that fin has maximum efficiency, when fin length ($L = 0$) is zero (no fin situation). This is a trivial case but is true. It indicates that in absence of fins maximum fin performance is possible. So, fin performance cannot be improved using fin of higher length. To maximize efficiency fin material (mass, volume, conductivity, cost, surface-roughness) is selected judiciously. In fact sometimes heat transfer with fin and without fin is also considered as a performance parameter of the fin. This is called fin effectiveness.

Fin effectiveness = tan *h* (mL)/√(hA/kP).

Since hyperbolic functions are used in expressions, their expansions are given below for ready reference.

Sin *h* x = $(e^x - e^{-x})/2$;
Cos *h* x = $(e^x + e^{-x})/2$;
Tan *h* x = $(e^x - e^{-x})/(e^x + e^{-x}) = [2/(1+ e^{-2x})] - 1$.

Addition of fin will increase heat transfer is the general perception but there are several situations in which fins are not very helpful. Heat transfer rate may not increase by addition of the fins. If convective heat transfer coefficient is very lager as depicted in case of high fluid flow velocity or boiling liquids, reverse situation may occur by addition of fins. In this case conduction becomes faster and convective heat transfer is not that effective. Let us consider a circular fin of stainless steel (conductivity = 43 W/m.K) with diameter 10 mm and length 100 mm protruding out from a plane surface in the atmosphere of boiling water (convective heat transfer coefficient = 5000 W/m^2.K).

Ratio of area to perimeter = A/P = Diameter/4

$= 0.01/4 = 0.0025$ m

Fin parameter, $m = \sqrt{(hP/AK)}$

$= \sqrt{(5000 /0.0025 \times 43)}/m = 215.665$/m.

Denominator of fin effectiveness = $\sqrt{(hA/KP)}$

$= \sqrt{5000 \times 0.0025/43)} = 0.53916$

Fin effectiveness = tan h (mL)/$\sqrt{(hA/KP)}$

$=$ tan h $(215.665 \times 0.1)/0.53916$

$=$ tan h $(21.5665)/0.53916 = 1.8$

So, heat transfer with fin is 1.8 times higher than heat transfer without fin. However, if conductivity of the material is reduced to 10 W/m.K, fin effectiveness can be calculated on similar lines as depicted above. The value of '*m*' is 447.213 /m for the given case and fin effectiveness is 0.894. This indicates that transmission of heat without fin is higher than transmission of heat with fins and addition of fin does not enhances the heat transfer rate. Hence under such situations, fins should not be added to the system.

EXAMPLE 5.30

An aluminium (k = 200 W/m.K) rod 2.5 cm in diameter and 15 cm long protrudes from a wall which is maintained at 260°C.

The rod is exposed to an environment at 16°C. The convection heat transfer coefficient is 15 W/m^2.K. Calculate heat loss by the rod.

SOLUTION

Given, d = 2.5 cm = 0.025 m, L = 15 cm = 0.15 m, T_b = 260°C, T_f = 16°C, h = 15 W/m^2.K, k = 200 W/m.K.

Assumption: Fin end is insulated in view of $d << L$.

Perimeter area of fin = $\pi.d$ = 0.07854 m.

End-face area of the fin = $(\pi/4).d^2$ = 0.000491 m^2.

Parameter $m = \sqrt{(h.P/A.k)}$

$= \sqrt{(15 \times 0.07854/0.000491 \times 200)} = 3.46365$/m

Heat transfer, $q = (T_b - T_f) \times \tan h\,(mL) \times \sqrt{(h.P.A.k)} = (260 - 16) \times \tan h\,(3.46365 \times 0.15) \times \sqrt{(15 \times 0.07854 \times 0.000491 \times 200)} = 4.311$ W/m.

5.7 2-D HEAT CONDUCTION

In previous section, temperature gradient and are is expressed in term of one space coordinate only. This may be actual situation or dimensions are such that effective mode of heat transfer can be represented well by considering variation of temperature in one direction only. However, in real life and for exact calculation, multi-dimensional approach is generally adopted. Even for the simplest case of steady state heat transfer by conduction in multiple dimension, close form solutions are difficult to derive. Most of the time two dimensional heat transfer is expressed and calculated analytically, numerically or graphically.

Although analytical solution is most favoured, most of the time, analytical solution is not possible for the given problems. With suitable boundary conditions, differential equation in two dimensions is considered. For the simplest case of steady state conduction in two dimensions without any heat generation, Laplace equation $\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0$ will be result. This equation can be solved by variable separable method and it needs knowledge of orthogonal functions, Fourier series, Bessel functions and series solution. Depending on geometry, boundary conditions and thermal gradients, the solution procedure varies.

Another way of calculation is resorting to conduction shape factor. In two dimensional heat conduction, heat transfer can be defined as q = k.S.ΔT, where k = thermal conductivity, S = conduction shape factor depends on geometry and is equal to reciprocal of product of thermal resistance and thermal conductivity, ΔT = temperature difference. Shape factor for 1-D heat conduction in Cartesian coordinate is A/L. However, if a thin plate of width W and Length L ($W > L$) is placed with one surface exposed to isothermal medium, the conduction shape factor if given by πW/ln (4W/L). If plate is buried inside the medium then conduction shape factor becomes double.

In the graphical approach for prediction, estimation and calculation of temperature distribution, isothermal and heat flow lines are sketched over the geometry after drawing several small elements on the geometry. Aggregation of heat flow across each heat flow line is accumulated to arrive at the total heat transfer. Most of the time heat flow is assumed to take place as per Fourier's law. However, this depends on skill of the individual in drawing various isothermal and heat flow lines. This method has only historical significance and is now replaced with advanced numerical methods. Currently finite difference method is mostly employed for solving heat flow and temperature distribution in the given irregular geometries.

In finite difference method, whole domain is divided into equal increments in both the planar directions. The entire domain is mapped by array of dots or nodes arranged in rectangular fashion. For any dot (m, n), in horizontal direction, two adjacent nodes at locations $m - 1$ and $m + 1$ exists, similarly in vertical direction, the node lies between nodes at $n - 1$ and $n + 1$. Temperature at each node can be given by two subscript formulations with first subscript showing horizontal location and second showing vertical location. The domain mapping is shown in figure 5.5. Although nodes here are shown by single subscript notation but temperatures can be depicted by two subscript notation also. The finite difference formulation for each of the element represented by a square is invoked. The grid sizes in both the directions are same. As far as boundary conditions are concerned two significant cases are prevalent– (*i*) temperature at external boundary of the structure is known and (*ii*) external boundary is exposed to certain temperature. For first boundary condition, temperatures at nodes are directly available while in second case convective heat transfer formulation is used to ascertain temperature at external nodes. Temperatures at internal nodes are calculated from external boundary conditions. There are basically three types of nodes (refer figure 5.5) – (*i*) internal nodes which are surrounded by four adjacent nodes (*e.g.* 6, 7, 10, 11), (*ii*) line nodes lying on any outer side and have three internal nodes in the vicinity (*e.g.* 2, 3, 5, 8, 9, 12, 14, 15) and (*iii*) corner nodes which are connected to only two nodes lying on sides of the structure (*e.g.* 1, 4, 13,16).

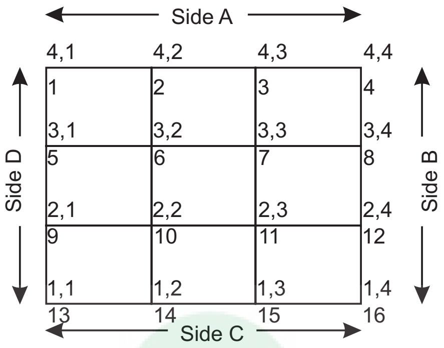


Fig. 5.5 : Numerical Methods for Temperature Calculation

Heat generation can be considered as 'q'. For equidistant distribution of nodes in both the directions, the governing equation of node of first type can be given as follows.

$$T_{m+1,n} + T_{m-1,n} + T_{m,n+1} + T_{m,n-1} + q\,(\Delta x)2/k - 4\,T_{m,n} = 0.$$

This formulation basically indicates that in absence of heat generation, temperature at any internal node is average of adjacent 4 nodes. For other two types of node, the coefficient of temperature in the equation is modified by a term $h\ \Delta x/k$, which is important for second type of boundary condition. For the nodes at the boundary (type 2), if convective heat transfer coefficient is given as 'h', then for those nodes, mathematical formulation varies and is given below.

$$2T_{m,n}\,[2 + (h\,\Delta x/k)] - 2(h\,\Delta x/k)T_{\infty} - 2T_{m-1,n} - T_{m,n-1} - T_{m,n+1} = 0.$$

The above-mentioned equation indicates that only three nodes participate in the mathematical equation of temperature. The internal adjacent node has coefficient 2 (*e.g.* $T_{m-1,n}$), which can be on vertical or horizontal grid distance from the node under consideration. For nodes on vertical side, horizontally spaced node has coefficient 2, while for nodes on horizontal boundary; nodes at grid distance in vertical direction are considered with multiplier 2. Other two adjacent nodes, which share same boundary with the node under consideration, have multiplier 1. Role of atmospheric temperature is also significant in this case.

For the corner nodes, only two adjacent nodes are available and mathematical formulation is depicted below.

$$2\,T_{m,n}\,[1+(h\,\Delta x/k)] - 2\,(h\,\Delta x/k)\,T_{\infty} - T_{m-1,n} - T_{m,n-1} = 0.$$

Once temperatures at various nodes are available, the heat transfer can be obtained using Fourier equation of heat transfer for each of the boundaries.

EXAMPLE 5.31

Calculate the temperature at various nodes of the symmetrical square array shown in figure 5.5 at internal four nodes, where side A is maintained at 400°C and other three sides are maintained at 100°C. Neglect heat generation terms and take grid size as 1 m. If thermal conductivity is 20 W/m.K, find heat transfer per unit thickness also.

SOLUTION

Since boundary condition of first type is given, temperatures at all the boundary nodes are available and need not be calculated. For the internal nodes 6, 7, 10, 11, equations are to be written and solved simultaneously. As heat generation is not present, temperature at any internal node is average of temperature of the four adjacent nodes. In addition to this symmetry about a fictitious vertical line passing through centre of the geometry is also present. Symmetry for geometry, temperature and boundary condition exists and thus $T_6 = T_7$ and $T_{10} = T_{11}$. So, ultimate aim is to get only two temperatures. Temperature equations are written for node 6.

$$4T_6 - T_2 - T_5 - T_{10} - T_7 = 0;$$

$$\text{Or } 4T_6 - 400 - 100 - T_{10} - T_6 = 0;$$

$$\text{Or } 3T_6 - T_{10} = 500.$$

Similar equation for node 10 can also be derived and the equation becomes $3T_{10} - T_6 = 200$. Solving both the equations simultaneously, $T_6 = 212.5$°C and $T_{10} = 137.5$°C. It is clear that temperature variation is not uniform in vertical direction. Starting from side *D*, it is clear that value of in first grid distance temperature rise is only 37.5°C, while for second grid distance it is 75, which is double of earlier temperature rise. For the next grid in vertical direction temperature rises from 212.5°C to 400°C, resulting in a rise in temperature of 187.5°C, which is 2.5 times rise in temperature of previous grid distance.

For calculation of heat transfer, it is clear that heat flows from side 'A' towards other sides in the structure. Since grid is symmetrical and grid size in both the directions are same, heat transfer per unit thickness can be calculated directly using Fourier's equation.

Heat transfer from side 'A' $= -k\,\Sigma\,\Delta T = -k\,[\Delta T_{6A} + \Delta T_{7A}]$

$$= -20\,[(212.5 - 400) + (212.5 - 400)] = 7500 \text{ W/m.}$$

Heat transfer from other sides $= -k\,\Sigma\,\Delta T$

$$= -k\,[\Delta T_{6D} + \Delta T_{10D} + \Delta T_{10C} + \Delta T_{11C} + \Delta T_{11B} + \Delta T_{7B}]$$

$$= -20\,[(212.5 - 100) + (137.5 - 100) + (137.5 - 100) + (137.5 - 100) + (137.5 - 100) + (212.5 - 100)] = -7500 \text{ W/m.}$$

Since heat transfers are same, the calculation is also validated indirectly.

For the calculation depicted above, it is clear that if one side is maintained at certain high temperature (T_h) and other three sides are at certain other temperatures (T_c) and grids are spaced as depicted in figure 5.5, it is clear that there is symmetry about vertical central line. The temperature rise in vertical direction follows definite pattern. If temperature rise in first grid spacing is T_r, it will be 2 times T_r at next step and 5 times T_r for the next step. With these types of results, it is clear that the temperature profile can be calculated directly by estimation of T_r. For the given conditions, $T_r = (T_h - T_c)/8$. If temperature at side 'A' is changed in the example 5.31 to 500°C, T_r is calculated as 50°C. This gives $T_{10} = 100 + 50 = 150$°C and $T_6 = 100 + 50 + 100 = 250$°C. The same result is obtained by mathematical formulation depicted in example 5.31.

EXAMPLE 5.32

For a simple square grid depicted in figure 5.5, side 'A' and side 'B' are maintained at 500°C, while other two sides are maintained at 100°C. Find temperatures at internal nodes. Neglect heat generation inside the geometry and take equi-distance grid points in both the directions.

SOLUTION

As depicted earlier, there is symmetry of geometry and boundary conditions about the leading diagonal joining point 4 with point 13. So temperature at node 6 is same as temperature at node 11. Now, we have only three temperature variables (T_6, T_7, T_{10}) to be solved simultaneously. Equations of the internal nodes of the grid are written below.

$4T_6 - T_{10} - T_7 = 600$; $4T_{10} - 2T_6 = 200$; $4T_7 - 2T_6 = 1000$. Solving them simultaneously, $T_6 = T_{11} = 300°C$, $T_{10} = 200°C$ and $T_7 = 400°C$.

EXAMPLE 5.33

For the two dimensional heat transfer depicted in figure 5.5, if side 'A' is maintained at 500°C, side 'B' and side 'C' are exposed to fluid at 100°C and side 'D' is maintained at 100°C, find temperature at all the except those lying on sides 'A' and 'D'. Take symmetrical grids in both the directions and neglect any heat generation. Take h = 10 W/m^2.K, k = 10 W/m.k and grid separation = 1 m.

SOLUTION

Since second boundary condition is also invoked, all the three types of equations depicted above are important. Now, calculation is to be made for nine nodes located at 6, 7, 8, 10, 11, 12, 14, 15, and 16. Now, there is no symmetry and nine equations are to be written and solved simultaneously by matrix inversion.

For the internal nodes 6, 7, 10, 11 equations are written below.

$$4\ T_6 - T_7 - T_{10} = 600$$
$$4\ T_7 - T_6 - T_{11} - T_8 = 500$$
$$4\ T_{10} - T_6 - T_{11} - T_{14} = 100$$
$$4\ T_{11} - T_7 - T_{10} - T_{12} - T_{15} = 0.$$

For the nodes lying on the outer boundaries where convective heat transfer is taking place, separate equations are needed. These are nodes 8, 12, 14 and 15. For connective terms $h\ \Delta x/k$ is to be calculated. For the present problem, it is 1.

$$6T_8 - 2T_7 - T_{12} = 700$$
$$6T_{12} - 2T_{11} - T_8 - T_{16} = 200$$
$$6T_{14} - 2T_{10} - T_{15} = 300$$
$$6T_{15} - 2T_{11} - T_{14} - T_{16} = 200.$$

For the corner node at 16, another formulation is valid and equation for the given nodes is given below.

$$4\ T_{16} - T_{12} - T_{15} = 200.$$

These nine equations are to be solved simultaneously by matrix inversion. The equations are written in matrix form as shown in next page.

Nodes Equation	*6*	*7*	*8*	*10*	*11*	*12*	*14*	*15*	*16*	*Constants*
1	4	–1	0	–1	0	0	0	0	0	600
2	–1	4	–1	0	–1	0	0	0	0	500
3	–1	0	0	4	–1	0	–1	0	0	100
4	0	–1	0	–1	4	–1	0	–1	0	0
5	0	–2	6	0	0	–1	0	0	0	700
6	0	0	–1	0	–2	6	0	0	–1	200
7	0	0	0	–2	0	0	6	–1	0	300
8	0	0	0	0	–2	0	–1	6	–1	200
9	0	0	0	0	0	–1	0	–1	4	200

After this, the matrix is inverted and temperatures at various nodes are obtained.

Nodes	*6*	*7*	*8*	*10*	*11*	*12*	*14*	*15*	*16*
T (°C)	268.75	301.5625	243.75	173.4375	193.75	159.375	131.25	140.625	125

EXAMPLE 5.34

For the two-dimensional heat transfer depicted in figure 5.5, if side 'A' is maintained at 500°C, Other three sides are exposed to atmosphere maintained at 100°C, find temperature at all the except those lying on sides 'A'. Take symmetrical grids in both the directions and neglect any heat generation. Take h = 10 W/m^2.K, k = 10 W/m.k and grid separation = 1 m.

SOLUTION

Since three surfaces are under convective heat transfer, temperatures lying at nodes on these three sides are not known.

In general temperatures are not known for 12 nodes on the grid (from 5 to 16). However, the geometry and boundary conditions are symmetrical about vertical central line and temperatures at some of the nodes are identical.

Here $T_5 = T_8$; $T_9 = T_{12}$; $T_{13} = T_{16}$; $T_6 = T_7$; $T_{10} = T_{11}$; $T_{14} = T_{15}$. So, temperature at nodes on one side of line of symmetry is to be evaluated. This makes temperature at six nodes unknown. The six governing equations are written below.

$$3\ T_6 - T_5 - T_{10} = 500$$

$$3\ T_{10} - T_6 - T_9 - T_{14} = 0$$

$$6T_5 - 2T_6 - T_9 = 700$$

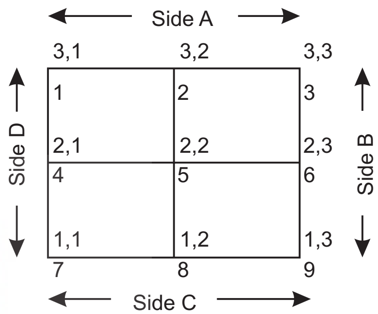
$$6T_9 - 2T_{10} - T_5 - T_{13} = 200$$
$$5T_{14} - 2T_{10} - T_{13} = 200$$
$$4T_{13} - T_{14} - T_9 = 200.$$

Solving them simultaneously by matrix inversion, the temperature at six nodes is given below.

Nodes	*5*	*6*	*9*	*10*	*13*	*14*
T (°C)	248.814	315.8864	161.1111	198.8452	120.1623	119.5381

EXAMPLE 5.35

Shown temperature at all the nodes for boundary conditions. Assume h Δx/k = 1. Boundary conditions are given below. The known temperature at nodes on a particular side of the grid is directly depicted in (°C), while sides exposed to convective heat transfer from atmosphere, maintained at certain temperature (°C), is expressed with 'Exposed to'.



Side→ Case number	*A*	*B*	*C*	*D*
1	500	100	100	100
2	500	Exposed to 100	100	100
3	500	Exposed to 100	100	Exposed to 100
4	500	Exposed to 100	Exposed to 100	100
5	500	Exposed to 100	100	Exposed to 100
6	500	Exposed to 100	Exposed to 100	Exposed to 100

SOLUTION

For all the nodes lying on side 'A', temperatures are known for all the situations. However, for nodes on other sides, boundary conditions govern the temperature equations. Assuming convective heat transfer at all other sides, general equations are written for all the other 6 nodes from number 4 to number 9.

$$6T_4 - 2T_\infty - 2T_5 - T_1 - T_7 = 0$$
$$4T_5 - T_2 - T_4 - T_8 - T_6 = 0$$
$$6T_6 - 2T_\infty - 2T_5 - T_3 - T_9 = 0$$

$$4T_7 - 2T_\infty - T_4 - T_8 = 0$$
$$6T_8 - 2T_\infty - 2T_5 - T_7 - T_9 = 0$$
$$4T_9 - 2T_\infty - T_6 - T_8 = 0$$

Depending on case, equations are solutions will change.

Case 1

Temperature is known at all the points except that at node 5. Using second equation from the set of 6 equations depicted above, T_5 = 200°C. Please note that although geometrically node 5 is at the center of the geometry, but temperature is not average of the two given temperatures.

Case 2

In this case, temperatures are not known at only two nodes namely 5 and 6. The extracted equations from the set of 6 equations depicted above, following two relations are valid.

$$4T_5 - T_6 = 700$$
$$6T_6 - 2T_5 = 800$$

Solving them simultaneously, T_6 = 2300/11 ~ 209°C, T_5 = 2500/11 ~ 227.3°C.

Case 3

In this case, temperature is not known at 3 nodes – 4, 5 and 6. However, since both geometry and boundary conditions are symmetrical about the central vertical line, temperature at node 4 and that at node 6 are same. So temperature is to be calculated at only two nodes. Equations are depicted below for these two nodes.

$$6T_4 - 2T_5 = 800$$
$$4T_5 - 2T_4 = 600$$

Solving them simultaneously, $T_4 = T_6$ = 220°C and T_5 = 260°C.

Case 4

In this case temperatures are not known at 4 nodes namely 5, 6, 8, and 9. The relevant equations are extracted from the above-mentioned set of equations.

$$4T_5 - T_8 - T_6 = 600$$
$$6T_6 - 2T_5 - T_9 = 700$$
$$6T_8 - 2T_5 - T_9 = 300$$
$$4T_9 - T_6 - T_8 = 200$$

Solving them simultaneously, T_5 = 305.556°C, T_6 = 244.44°C and T_8 = 177.778°C and T_9 = 155.556°C.

Case 5

In this case temperature is not known at all the 6 nodes for which set of equations are written and all the equations are to be used for calculation. However, again there is symmetry of boundary condition and geometry both about the vertical central line.

So, $T_4 = T_6$ and

The equations for the given boundary conditions are given below.

$$6T_4 - 2T_5 - T_7 = 700$$
$$4T_5 - T_4 - T_8 - T_6 = 500$$
$$6T_6 - 2T_5 - T_9 = 700$$
$$4T_7 - T_4 - T_8 = 200$$
$$6T_8 - 2T_5 - T_7 - T_9 = 200$$
$$4T_9 - T_6 - T_8 = 200$$

Solving them simultaneously, the unknown temperatures at 6 nodes are given below.

Node	*4*	*5*	*6*	*7*	*8*	*9*
T (°C)	239.2265	290.0552	239.2265	155.2486	181.768	155.2486

5.8 CONVECTIVE HEAT TRANSFER PROCESS

Convective heat transfer is another mode of heat transfer where fluid motion over a solid surface is involved. It involves study on energy balance as well as fluid dynamics simultaneously. During study of conduction, convection is analyzed as a surface phenomenon for fins and for wall heating and cooling. The main criterion is flow of fluid over a solid surface. For analysis of such convection, energy equation (first law of thermodynamics), continuity equation and momentum equations are applied simultaneously. Temperature distribution is correlated with fluid motion and wall heat flux and heat transfer coefficients are predicted or ascertained. Due to complexities, theory is not found adequate for analysis of most of the convective heat transfer problems. Then dimensional analysis, experimental data and empirical relations are used for analysis. To understand convection, it is better to first understand boundary layer and fluid motion and then apply energy balance to find out temperature distribution in fluids. From this heat transfer rate from solid to fluid or vice-versa can be ascertained. There are several ways for classification of convective heat transfer process.

Depending on origin of fluid motion, convection is classified as forced convection and free convection. If the fluid has a nonzero velocity at some distance away from the solid surface, which is created by a pump or fan or other driving forces, situation conform to forced convection. Flow through a duct or bodies immersed in uniform stream of fluid are examples of forced convection. An alternate way is free or natural convection, where fluid motion is created entirely due to local buoyancy difference caused by presence of hot or cold body. Due to heated surface, fluid near the solid surface gets heated and their density reduces. The density difference is responsible for fluid motion in a vertical plane. Although fluid velocities are generally low and resultant heat flux at the wall is also less as compared to that in forced convection. However, free convection heat transfer is enhanced by phase change, if present. Boiling agitates fluid in the form of rising bubbles and condensation disturbs fluid through falling dense liquid droplets. Of course, there may be a situation of mixed convection, where both the forms of heat convection are present.

Another way to classify convection is on the basis of geometry and classification includes internal and external flow. An internal flow is bounded by solid surface on all sides except for an inlet and an outlet. The flow may be forced and free. Flows through nozzle, ducts, diffusers, heat exchangers fall under this category. They are easier to analyze for convective heat transfer. However, temperature and velocity profile has no free stream known conditions and they are computed as average integrated values. Average velocity is criteria for calculation of heat flux in internal flow convection. External convection is present when flow is not bound. A free stream velocity and temperature is known. The free stream velocity is governing parameter for calculation of convective heat transfer coefficient in external flow heat convection.

Fluid flow can be classified as laminar or turbulent based on non-dimensional number called Reynolds Number (Re). This is a number which is dependent on several parameters – geometry, surface roughness, wall temperature, fluctuations in free stream, pressure gradients etc. The Reynolds Number is defined as ratio of inertia force to viscous force and is mathematically expressed as follows.

Re = $u_\infty x/v$, where, u_∞ = free stream velocity, x = distance from leading edge in flow over flat plate, and v = kinematics viscosity = μ/ρ, μ = dynamic viscosity = shear stress in fluid/velocity gradient, ρ = density of the fluid.

Flow at low Reynolds number is laminar. Under such low velocities, smooth streamlines exists in the flow. Hear and conduction effects are due to molecular viscosity and conductivity. Any dye left in the flow stream at the inlet follows a defined path, preferably in a straight line along streamlines. However, as Reynolds number increase, transition creeps in and streamlines gets disturbed. Random fluctuations in flow and flow reversals are also observed. The dye follows and erratic pattern with large lateral fluctuations. Fluctuations rise further to completely turbulent flow on further rise in Reynolds number. Velocity, pressure and temperature show fluctuations. Flow undergoes intense mixing. This leads to high friction and heat transfer.

For internal flow heat transfer, Reynolds number for laminar flow is less than 2300, while for turbulent flow it is above 4000. The regime of Reynolds number in the range of 2300 to 4000 is called transition region. For flow over flat plate, if Reynolds Number is less than 5×10^5, flow is called laminar. As Reynolds Number increases, turbulence increases and at high Reynolds Number, flow is very vigorous. High Reynolds Number means high value of inertia force compared to viscous force and any flow or disturbances created will not subside that fast in the fluid. As fluid flows over flat plat, flow development takes place from the leading edge (first edge of the solid in contact with the fluid) of the plate. The effect of presence of flat plate is felt by fluid due to viscosity of the fluid and flow pattern is affected. Initial length of flow over flat plate is laminar and region of boundary layer will gradually increase. If perpendicular to flat plate is traversed, velocity profile will acquire a parabolic shape in laminar region. As flow progresses over flat plate, thickness of boundary layer rises. After traversing certain distance, flow becomes turbulent. In turbulent region also a thin boundary layer near flat plate exists, where velocity profile is linear. However, further away from flat plate, velocity profile is relatively flatter in comparison to laminar fluid flow regions.

For derivation of boundary layer thickness for steady flow of incompressible fluid over flat plate, fluid viscosity is assumed independent of temperature and pressure. Viscous forces and pressure variation in the direction perpendicular to flat plate is neglected. Equation of motion for the boundary layer of fluid can be obtained by making a force-and-momentum balance. If δ = hydrodynamic boundary layer thickness, x = distance from the leading edge, Re_x = Reynolds Number at distance x from the leading edge,

Then $\delta/x = 5/\sqrt{Re_x}$.
And $u/u_\infty = 1.5\,(y/\delta) - 0.5\,(y/\delta)^3$, where u = velocity at × distance from the leading edge and y distance from flat plate.

▮▮ EXAMPLE 5.36

Air at 27°C and 1 atmospheric pressure flows over a flat plate at a speed of 2 m/s. calculate the boundary layer thickness at distance of 20 and 40 cm from the leading edge of the plate. Calculate the boundary layer thicknesses at these locations. The viscosity of air at 27°C is 1.85×10^{-5} kg/m.s. Assume unit depth of the plate.

SOLUTION

Given, T = 27°C = 300 K, p = 1 atm = 1.01325 × 105 N/m^2, u_∞ = 2 m/s, x_1 = 20 cm, x_2 = 40 cm, μ = 1.85×10^{-5} kg/m.s, Rair = 287 J/kg.K.

Density of air, $\rho = p/R.T = 1.01325 \times 10^5/(287 \times 300)$

$= 1.177$ kg/m^3.

Reynolds Number at '1', $Re_1 = 1.177 \times 2 \times 0.2/1.85 \times 10^{-5}$

$= 27580.$

Boundary layer thickness at '1', $\delta_1 = 5 \times 0.20/\sqrt{27580}$ m

$= 6.02147$ mm.

Reynolds Number at '2', $Re_2 = 1.177 \times 2 \times 0.4/1.85 \times 10^{-5}$

$= 55160$

Boundary layer thickness at '2', $\delta_2 = 5 \times 0.40/\sqrt{55160}$ m

$= 8.51565$ mm.

Similar to fluid flow boundary layer, an analogy can be drawn for the thermal boundary layer also. Instead of fluid velocity, heat transfer can be taken as relevant parameter. Thermal boundary layer is defined as that region in which temperature gradient is present in the flow. These temperature gradients result from a heat exchange process between the fluid and the wall. For defining thermal boundary layer, another non-dimensional number called Prandtl number (Pr) is used. Prandtl number is defined as ratio of kinematics viscosity (ν) and thermal diffusivity (d). It may be interpreted as ratio of viscous effects to conduction effects. It is a correlation between kinematics to thermal flow properties and it indicates relative worth of fluid flow over the other parameter. For Prandtl number greater than 0.7, ratio of hydrodynamic to thermal boundary layer thicknesses is given by $1/(1.026 \times Pr^{1/3})$. Most gases and liquids behave in this fashion. However, liquid metals have Prandtl number of the order of 0.01.

Kinematics viscosity is indicator of rate at which momentum may diffuse through the fluid because of molecular motion. Thermal diffusivity is related to diffusion of heat in the fluid. Larger diffusivity indicates that effect of flat plate temperature is felt farther out in the flow field. Expression for Prandtl number is as follows.

$$Pr = \nu/\alpha = (\mu/\rho)/(k/\rho \,.\, c_p) = c_p \,.\, \mu/k.$$

Another non-dimensional number called Nusselt number (Nu) also gains importance in case of convective heat transfer. It is defined as ratio of convection heat transfer to fluid conduction heat transfer under same condition.

$$Nu = h \times L/K.$$

If Nusselt number is near unity, it indicates sluggish motion (low Reynolds number). The heat transfer will be little more effective than simple conduction. Such situation may arise in laminar flow through a long pipe (internal flow). A large Nusselt number indicates efficient convection. In turbulent flow through pipe, the value of Nusselt number may be of the order of 100 to 1000. This is similar to Biot number used in conduction. In biot number, 'k' indicates conductivity of the solid while in Nusselt number 'k' indicates fluid conductivity.

If flat plate is heated over the entire length,

$Nu = 0.332 \times Pr^{1/3} \times Re^{1/2}$, where Nusselt number, $Nu = h \times l / k$.

The above relation is valid for laminar heat transfer from an isothermal surface. Instead, if heat flux is made constant, another expression for Nusselt number is valid.

$Nu = 0.453 \times Pr^{1/3} \times Re^{1/2}$, where Nusselt number, $Nu = h \times l / k$.

Contrary to this, if flow of fluid inside a tube is considered, laminar fluid flow is realized till Reynolds Number reaches 2300. A transition zone exists up to $Re = 4000$ and beyond this value of the Reynolds Number fluid flow is turbulent. The definition of Reynolds Number in this case changes slightly. Instead of using distance from the leading edge, tube-diameter (d) is taken and $\boldsymbol{Re = u_\infty d/v}$. Under such situation velocity distribution inside the tube has expression $u/u_o = 1 - (r/r_o)^2$, where u_o = maximum velocity at the center of the tube and u is velocity at a radius of r from center of the tube. Similarly, Nusselt number for this case is 4.364 from calculation. This expression is helpful in calculation of convective heat transfer coefficient.

5.9 FREE CONVECTION HEAT TRANSFER

Contrary to forced convection, where fluid is forced over the heat transfer surface, free convection does not need any addition means to force the fluid over solid surfaces. In this case, due to heating process, density of fluid changes and fluid motion sets in. A hot radiator is a good example of free convection. In fact, it is gravitational force, which is exploited to make flow in free convection. In additional to this, centrifugal force can also generate buoyancy force to make free convection possible.

In case of free-convection, velocities are rarely more than 1 m/s. so, heat flux is also very small. However, it gives a very good estimate for lower limit of heat transfer for forced convection for a given geometry. All natural phenomena like atmospheric or oceanic motions are in fact natural for free convection processes. In free convection, there is no free stream velocity (as there is no external source to set fluid in motion) and governing velocity parameter is to be calculated by some other means.

For free convection, non-dimensional number Grashof number (Gr) is of great importance. It is defined as $Gr = g.\beta.\rho^2.\ Lc^3.\Delta T/\mu^2$. Physically, it is ratio of buoyancy force and viscous force. This plays same role in free convection as Reynolds number plays in forced convection. Grashof number is used to finalize flow characteristic as laminar, turbulent or transition. Grash of number higher than 109 indicate flow is turbulentes and less than 108 indicates flow is laminar.

5.9.1 Convection on Flat Plate

When a vertical plate is heated, free convection boundary layer is formed. Velocity of fluid at the wall is zero because of no-slip condition. Fluid close to wall moves up due to density reduction and horizontal component of velocities are negligible. Now, streamwise direction is taken as 'x' and transverse direction is taken as 'y' as shown in figure 5.6. A thin layer exists adjacent to the hot surface of the vertical plate. Velocity and temperature is confined in this boundary layer. Laminar layer exist up to certain distance up from the leading edge. Velocity is zero at plate as well as in free stream and it reaches a maximum in boundary layer. As distance from leading edge increases, turbulent eddies are formed and transition to a turbulent boundary layer begins. Farther up boundary layer may be fully turbulent. Temperature changes monotonically within the boundary layer.

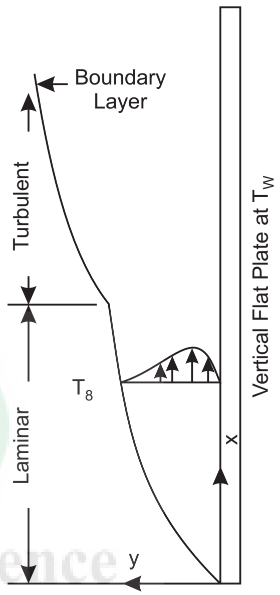


Fig. 5.6 : Free Convection on Vertical Wall

If constant density lines in the fluid flow field are drawn, they are equivalent to constant temperature lines. So, once temperature field is obtained, heat transfer from the surface can be calculated using thermal conductivity of gases and temperature gradient at the surface.

5.9.2 Planes

Since natural convection is mainly due to density difference arising out of temperature variation in the fluid, gravitational forces play a major role. On a horizontal plane, the governing factor is whether direction of heat flow is same as direction of buoyancy force. It also depends on whether surface is hot or cold with respect to ambient fluid. If a plate is inclined from vertical, only gravitational acceleration terms is changed by cosine of inclination angle and vertical plane relations are used. If planes are inclined, transition to turbulent region begins much earlier. Higher inclination leads to turbulence at much lower Grashof numbers. For horizontal planes, the induced convective currents are normal to the plane and are very effective.

5.9.3 Cylinders

Thermal gradient across a cylindrical surface is also examined by several researchers and several empirical relations between various non-dimensional numbers are arrived at. If a vertical cylinder of diameter D and length L is considered, large D resembles vertical flat plate. If boundary layer thickness is not large, compared to diameter of the cylinder, the heat transfer may be calculated with the same relation as used for vertical plates. The general criteria, when vertical cylinder becomes similar to vertical flat plate is when $D/L > 35/Gr^{1/4}$. Here, D = diameter of the cylinder. Small D heat transfer is enhanced due to rise in curvature. For horizontal cylinder, heat convection depends on location on the cylinder. Bottom sides have different correlations and as circumference of the cylinder is traversed, gravitational acceleration changes with inclination angle again. In this case, there is no boundary layer separation, because of negligible pressure gradient. This indicates conformity to boundary layer theory to this present situation. Heat flux data on a horizontal cylinder is approximated by vertical plane correlation, if plane length is replaced by cylinder diameter.

5.9.4 Sphere

Empirical relations are developed for free-convection heat transfer from sphere to air. It is observed that Nusselt number is a function of Grashof number and for laminar flow $Nu = 2 + 0.392\ Gr^{1/4}$. Sometimes Prandtl number is also introduced in the correlation and equation comes out to be $Nu = 2 + 0.43.(Pr.Gr)^{1/4}$. If product of Prandtl number and Grashof number is low, the value of Nusselt number approaches 2. This is value obtained for pure conduction through an infinite stagnant fluid surrounding the sphere. For higher Grashof numbers, $Nu = 2 + 0.50\ .\ (Pr\ .\ Gr)^{1/4}$.

5.10 THERMAL RADIATION

As explained earlier, thermal radiation is electromagnetic radiation emitted by a body due to its temperature. Thermal radiation is one type of electromagnetic radiation, which propagates at a speed of 3×10^8 m/s. The speed is equal to product of wavelength and frequency of the radiation ($c = \lambda.v$, where λ = wavelength, v = frequency of radiation). Thermal radiation lies in the frequency range of 0.1 μm to 100 μm. Within this wavelength range, visible radiation lies within 0.4 μm (Violet) to 0.7 (Red) μm. Human eyes can detect radiation as colours of light within this wavelength band. This means eyes can pick up radiations, form hot sources, which are at temperature of more than 800 K like sun, lamp, etc. Colours seen by naked eyes are reflected radiation from these hot sources. Bodies, cooler than 800 K, emit radiation in infrared region (wavelength between 1 μm and 1000 μm), which can be detected by special optical means only. The visibility should not be confused with black or white surfaces used in thermal radiation. Both black and white surfaces are good emitters and absorbers in infrared regions and they should not be separated on the basis of their performance in visible region. The propagation of thermal radiation takes place in the form of discrete quanta and each quantum is integral multiple of hv, where $h = 6.625 \times 10^{-35}$ J.s and v = frequency of radiation. The quanta can be treated as particles having energy, mass and momentum. So radiation can be thought to be as photon gas, which may place from one place to another. The energy of particles can be equated to $m.c^2$, where m = mass of particle.

The thermal radiation is emitted by atomic excitation of any substance. It propagates more easily through vacuum and does not need any media for transmission. Any intervening media obstructs flow of thermal radiation. It travels at speed of light till it strikes another body. On striking, it gets absorbed, reflected, transmitted or scattered. Thermal radiations are different from other forms of electromagnetic radiations on the basis of there mechanism of creation.

- Thermal radiations are caused by temperature excitations.
- X-rays are created by electron bombardment on metals.
- γ-rays are formed by nuclear reactions.
- Radio-waves are created by excitations of crystals.

As stated in the form of Stefan-Boltzmann equation, total energy emitted is proportional to absolute temperature to the fourth power of temperature. Absolute temperature is used in all the calculations. Law is mathematically expressed as $E_b = \sigma \,.\, T^4$, where E_b is the energy radiated per unit time per unit area by the ideal radiator and to be more specific blackbody. Here, $\sigma = 5.669 \times 10^{-8}$ W/m^2.K^4. Materials, which obey the above-mentioned equation looks black to naked eyes. This happens because these materials do not reflect any radiation. Perception of blackbody is deceptive. A surface coated with lampblack appears black to eyes and turns black for thermal-radiation spectrum. But contrary to this snow or ice appears quite bright and white fro the eyes, but it is black for long-wavelength thermal radiation.

Sometimes a terms emissivity (ε) is added in the Stefan-Boltzmann Law and the relation is written as $E_b = \varepsilon \,.\, \sigma \,.\, T^4$. Emissivity is a dimensionless quantity. For blackbody, its value is unity and for all other surfaces, it lies between 0 and 1. It depends on temperature, roughness, texture, colour, material, degree of oxidation, coating etc. The definition of emissivity is related to that of blackbody, which is considered a reference for all radiation calculations. It is a perfect emitter and perfect absorber. Its emissivity is independent of wavelength and temperature. It emits maximum possible energy at a given temperature.

5.11 EMISSIVE POWER

Energy radiated by blackbody per unit time per unit area is also called emissive power of the blackbody. Blackbody absorbs all radiation incidents upon it, but nothing is reflected. In fact, when a radiation strikes a surface, three things can happen – radiation gets reflected, transmitted or absorbed. If

Reflected radiant incident energy = Reflectivity = ρ
Absorbed radiant incident energy = Absorptivity = α
Transmitted radiant incident energy = Transmissivity = τ
Energy balance gives $\rho + \alpha + \tau = 1$.

Most of the solids do not transmit thermal radiation. So, transmissivity, $\tau = 0$.

Reflection can be classified under two heads. If angle of incidence is equal to angle of reflection, the reflection is specular. Specular reflection may cause mirror image of source. Another type of reflection is diffuse reflection, when incident radiation reflects uniformly in all directions. No real solid surface is either specular or diffuse. However, a rough surface is more diffuse than a highly polished surface. Highly polished surface is more specular. Another aspect is wavelength of radiation. Any surface may be specular for one type of wavelength and may be diffuse for another wavelength of radiation.

As far as emissive power is concerned, if a body is enclosed in a black enclosure, it comes in thermal equilibrium with the enclosure. At equilibrium, energy absorbed by the body must be equal to the energy emitted. Energy emitted by the body = $E \times A$. energy released by black enclosure is absorbed by the body = $q \times A \times \alpha$, where q = energy from black enclosure incident on per unit area of the body. If temperature of the body is not changing, then $E = q \times \alpha$.

If the body is replaced by a blackbody and brought into same equilibrium position with the black enclosure, energy emitted = $E_b \times A$. If blackbody is present, absorptivity is 1. Incident energy on the blackbody = $q \times A$. Combining both situations, $E/E_b = \alpha$. So, ratio of emissive power of a blackbody at the same temperature is equal to absorptivity of the body. Ratio of energy emitted by a body to that from a blackbody is known as emissivity (ε).

So $\alpha = \varepsilon$. This is called Kirchoff's identity. This is derived by Gustav Kirchoff in 1860 using thermodynamic principals. This equation is valid only when emitted and received radiations are at different temperatures. Both the properties namely absorptivity and emissivity are valid for all the wavelengths. Real substances emit less radiation than an ideal black surface. However, in reality emissivity of a material is dependent on wavelength and temperature of the radiation.

5.12 PLANK'S DISTRIBUTIVE LAW

The dependence of emissivity over various parameters defines a new type of body called gray body. For this definition of monochromatic emissivity is essential. It is defined as ratio of monochromatic power of the body to monochromatic emissive power of the blackbody at the same temperature and wavelength. If monochromatic emissivity of the body is independent of wavelength, the body is called Gray Body. For real bodies, emissivity varies widely with wavelength, temperature and surface conditions.

Plank's introduced the quantum concept to electromagnetic radiation and used statistical thermodynamics to derive, what is called Plank's distribution law for monochromatic emissive power of blackbody. Plank presented his paper on blackbody radiation in 1901. He modeled blackbody as a set of dipole oscillators whose energy at a given frequency could be a multiple of discrete or quantum energy level hí. After statistical averaging, Plank arrived at his final equation in the form of Planck's blackbody spectral energy distribution.

As per Plank's law,

$E_{b\lambda} = dE_b/d_\lambda = (C_1 \times \lambda^{-5})/(\exp[C_2/\lambda T] - 1)$, where λ = Wavelength in μ*m*, T = Temperature in *K*, $C_1 = 3.743 \times 10^8$ W μm^4/m^2, $C_2 = 1.4387 \times 10^4$ μm.K.

If monochromatic emissive power of blackbody is plotted against wavelength of radiation at constant temperature, it rises, reaches a maximum and then reduces (figure 5.7). Many characteristics of the curves can be pointed out.

The blackbody spectrum is continuous and is not symmetrically distributed. It is skewed towards right and majority of the energy is emitter at lower wavelength.

At all the wavelengths, the emissive power increases with increasing temperature. A high temperature always gives higher emission, which conform to the definition of radiation, which is electromagnetic radiation in given wavelength due to temperature of the body.

As temperature increases, the emissive power becomes concentrated at shorter and shorter wavelengths.

The maximum emissive power increases as T_5. However the total emitted radiation (area under the curves) increases only as T_4. This is not clear from the curves, but Planck's law is in line with Stefan-Boltzmann law. It is observed that solar radiation (T ~ 5800 K) has its peak in the visible region of wavelength, while for temperature lower than 700 K, the emissive power is infrared and is totally outside the visible range. The maximum occurs at a given wavelength for a given temperature. It is found that when temperature increases, maximum of emissive power shifts to shorter wavelengths. Maximum points in the radiation curves are related by Wein's displacement law.

For a given temperature, maximum of the plotted curve can be obtained by differentiated Planck's equation with respect to wavelength λ and then setting it to zero. The locus of such peaks is governs by Wein's displacement law, whose expression is given below.

$$\lambda_{max} \,.\, T = 2897.6 \text{ μm.K.}$$

W. Wein derived these results using classical thermodynamics in 1894, seven years before Planck's laws were published. If expression of Wein's displacement law is substituted in Planck's distribution law, maximum blackbody emissive power can be calculated as below.

$$E_{b\lambda,max} = 1.2865 \times 10^{-5}\, T^5 \text{ W/m}^3.$$

The naked eyes can detect colour from radiating bodies at an emissive power of about 106 W/m^3. For red wavelengths, λ = 0.7 μm, this occurs at temperature of 950 K for which a blackbody will glow as dull red. For violet wavelengths, λ = 0.4 μm, this occurs at temperature of 1500 K, which is enough for eye to see as white. The emissive power peak of such white hot objects will be in far infrared region.

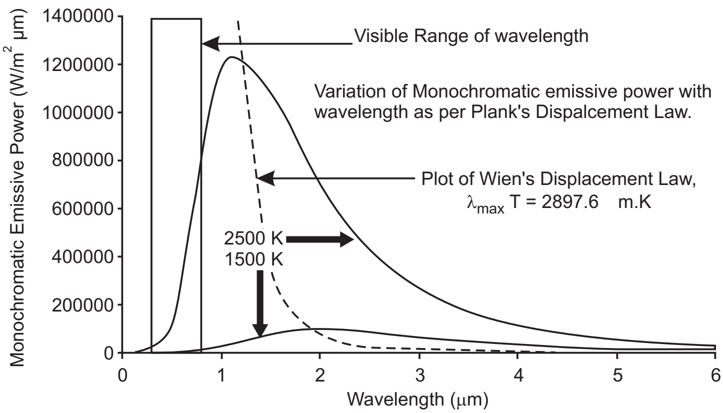


Fig. 5.7 : Variation of Monochromatic Emissive Power with Wavelength

This shift in the maximum point of the radiation curve explains the change in the colour of a body as it is heated. A very small portion of the radiant energy spectrum at low temperature is detected by eyes due to small band of visible wavelength. As body is heated, maximum intensity is shifted to shorter wavelengths and the first visible sign of increase in temperature of the body is dark red colour. With further increase in temperature, the colour appears bright red, then bright yellow and finally to white. The material appears much brighter at higher temperatures because a larger portion of the total radiation falls within the visible range. The variations are shown in figure 5.7.

A real body behaves differently than a blackbody, for which some of the governing rules or equations are available. For a real surface, it is important to know blackbody energy contained within a given wavelength band. Each wavelength band is characterized by two wavelengths and blackbody radiations are given by Planck's law. The components of blackbody radiation (given by Stefan-Boltzmann law) are obtained by integrating this blackbody radiation obtained from Planck's law over the two wavelengths. This integral is cannot be solved in closed form for all wavelength band. Fraction form of this quantity is obtained by dividing the blackbody radiation from the body in the given wavelength band by energy calculated by Stefan-Boltzman law at the body temperature. For this tables are developed and sometimes a series solution with adequate number of terms can be of some use. Lower limit is generally taken as wavelength of zero and since this property is additive, for two different wavelength bands, the values are obtained by simple subtraction of fractional energy for two wavelengths. A term $w = C_2/\lambda T$ is defined and series solution has the form given below.

$$F(\lambda T) = F(C_2/w) = {}_0\int^{\lambda T} E_{b\lambda}(\lambda, T)\, d\lambda/\sigma T^4 = (15/\pi^4) \times {}_{\lambda T}\int^{\infty} w^3\, dw/(e^w - 1) = (15/\pi^4)\, x\, {}_{i=1}\Sigma^{\infty}\, i^{-4} e^{-iw} [(iw)^3 + 3\,(iw)^2 + 6\,(iw) + 6].$$

5.13 RADIATION PROPERTIES

Any opaque surface can be classified on the basis of dependence of absorptivity on wavelength. For the first type of surface called flat absorbers, absorptivity is maintained at high level in the entire wavelength band. It captures or absorbs all the incoming radiation. Surfaces coated with black paints fall in this category. Second variety of surface has low absorptivity over the entire spectrum. They are called flat reflector, as they reflects all the incoming radiation. Polished surfaces, aluminium foils etc fall under this category. Solar absorber is another class of surface. It absorbs lower wavelength spectrum, which is in low wavelength band of solar radiation. Contrary to this, for higher wavelength band in the infrared region, this surface has low absorptivity and it reflects all the incoming radiation in those wavelength levels. Contrary to this reverse is true for surface called solar reflectors. They reflect in low wavelength region and absorbs at high wavelength radiations. Sometimes transparent or semitransparent materials are also encountered. For example glass windows behave as solar reflectors and are a solar transmitter also. These glasses have low absorptivity in solar region (low wavelength). Reflectivity of the surface is also low. Thus transmissibility of glass is high in solar region. However, in infrared region (high wavelength and low temperature), it has high absorptivity. It absorbs most of the infrared radiation and does not allow it to transmit. As a result, solar radiation can enter through glass unhindered, but heat from inside cannot escape window glass. This results in heating. This is the region why car with closed window kept in sun becomes warm from inside. This effect is also called greenhouse effect and is used for farming and keeping rooms warm during winter.

Net radiation heat exchange between two surfaces is computed using radiation shape factor. The most important term used to understand radiation heat transfer is called radiation shape factor. This is purely a geometrical quantity. It is also known as view factor, angle factor and configuration factor. To understand this term, a term emissive power is already defined earlier. E_b is defined as energy emitted per unit area from a blackbody. If two blackbodies are in vicinity, then energy emitted from first blackbody is $E_{b1} \times A_1$. The radiation shape factor $F_{1\text{-}2}$ is defined as energy emitted from body '1' and intercepted by body '2'. So, energy released from body '1' and intercepted or received by body '2' is $E_{b1} \times A_1 \times F_{1\text{-}2}$. Similarly, energy released from body '2' and received by body '1' is given by $E_{b2} \times A_2 \times F_{2\text{-}1}$. If both the bodies '1' and '2' are at different temperatures, some energy exchange may take place. If both the bodies are at same temperature, then

$$E_{b1} = E_{b2} \text{ and}$$
$$A_1 \times F_{1\text{-}2} = A_2 \times F_{2\text{-}1}.$$

Radiation shape factor has additive properties and a relation can be derived.

$$F_{1\text{-}2,3} = F_{1\text{-}2} + F_{1\text{-}3}.$$

If two bodies do not see each other, shape factor is zero. For a surface, if all the energy released or emitted by a surface is accounted for, sum of fraction of total energy leaving a surface, which arrives at other surfaces, will be 1.

$$\Sigma_{i=1,n} F_{1i} = 1.0.$$

This is also calculated from geometry of the surfaces. For simple two-dimensional geometry, some of the relations for shape factor are derived. The structure may be seen as having infinite length and cross-sectional area is depicted below.

For cross-section in the form of two equal lines of length 'L' each, forming an angle A with each other – $F_{1\text{-}2} = F_{2\text{-}1} = 1 - \text{Sin}(A/2)$. Shape factor is independent of length. The open are is depicted by 'e', then $F_{1\text{-}e} = F_{2\text{-}e} = \text{Sin}(A/2)$.

For a cross-sectional area in the form of two perpendicular lines, one of length 'L' (surface 1) and other of length 'w' (surface 2) – $F_{1\text{-}2} = 0.5 \times [1 + w/L - \sqrt{(1 + w^2/L^2)}]$.

For cross-section in the form of a triangle with lengths 'L' (surface 1), 'w' (surface 2) and 'h' (surface 3) – $F_{1\text{-}2} = (L + w - h)/2L$.

For cross-section in the form of two parallel lines of equal length 'L' and separated by distance 'w' – $F_{1\text{-}2} = F_{2\text{-}1} = \sqrt{(1 + w^2/L^2)} - \text{w/L}$.

For cross-section in the form of two circle of diameter 'D' each, separated by a center distance of '$L + D$' – $F_{1\text{-}2} = F_{2\text{-}1} = 1/\pi \times [\sqrt{(Z^2 - 1)} + \text{Sin}^{-1}(1/Z) - Z]$, where $Z = 1 + L/D$.

For three dimensional configurations, the value of shape factor is very complex. For example, if center of two discs of radii 'R_1' and 'R_2' are separated by a distance L, The shaped factor $F_{1\text{-}2} = 0.5 \times [A - \sqrt{(A^2 - 4C^2/B^2)}]$, where $A = 1 + (1 + C^2)/B^2$, $B = R^{1/L}$, $C = R^{2/L}$. Other shapes are having more complex formulations, which are not perused in this book.

For blackbodies, such calculations are easy, but real and non-black bodies, all the energy striking a surface is to be absorbed, but some are reflected also. Reflection may be to the original surface or to other surrounding surfaces. To understand such heat transfers by radiations, two more terms are defined. First is irradiation, G, defined as total radiation incident upon a surface per unit time and per unit area. Second is radiosity, J, defined as total radiation which leaves a surface per unit time and per unit area. If bodies are under thermal equilibrium, no radiative heat exchange takes place. Energy leaving a surface may be divided into two parts. One is emitted energy and other is reflected part of the incident energy. For a real body total energy emitted is product of emissivity and blackbody emission.

Radiosity, $J = \varepsilon \times E_b + \rho \times G$.

For solids, transmissivity is zero. So, $\rho + \alpha = 1$. As Kirchoff's law states that $\alpha = \varepsilon$, $\rho = \mathbf{1} - \varepsilon$.

So, $J = \varepsilon \times E_b (1 - \varepsilon) \times G$.

Net energy leaving a surface is the difference between the radiosity and irradiation. $\boldsymbol{q/A = J - G = J - [J - x\,E_b]/(1 - \varepsilon)}$,

So, $q = (E_b - J)/[(1 - \varepsilon)/\varepsilon.A]$.

If two surfaces '1' and '2' are exchanging heat, then total radiation leaving surface 1 and reaching surface 2 is $J_1 \cdot A_1 \cdot F_1 - 2$. Similarly total radiation leaving surface 2 and reaching surface 1 is given by $J_2 \cdot A_2 \cdot F_{2\text{-}1}$. Net interchange between the two surfaces is $\boldsymbol{q_{1\text{-}2} = J_1 \cdot A_1 \cdot F_{1\text{-}2} - J_2 \cdot A_2 \cdot F_{2\text{-}1}}$. As per definition of radiation shape factor $\boldsymbol{A_1 \cdot F_{1\text{-}2} = A_2 \cdot F_{2\text{-}1}}$.

So, heat exchange, $q_{1\text{-}2} = (J_1 - J_2)/(1/A_1 \cdot F_{1\text{-}2})$.

For calculation of net heat exchange between two bodies can be simulated using above mentioned concept. It is observed that the bodies can be treated black bodies, whose net radiation heat exchange is given by Stefan Boltzmann law. However, the radiant thermal resistance will have two terms containing radiocity on both side and a shaped factor term in between. For radiation heat exchange between two bodies 1 and 2, three heat exchanges takes place.

First between E_{b1} to J_1, where thermal resistance is $[(1 - \varepsilon_1)/\varepsilon_1 \cdot A_1)]$.

Second between J_1 to J_2, where thermal resistance is $1/A_1 \cdot F_{1\text{-}2}$ or $1/A_2 \cdot F_{2\text{-}1}$.

Third between J_2 to E_{b2}, where thermal resistance is $[(1 - \varepsilon_2)/\varepsilon_2 \cdot A_2)]$.

Sun is a source of thermal radiation and its radiation is dependent on atmospheric conditions, time of year and angles of incidence fro the sun rays. The total solar irradiation at the outer limit of atmosphere at mean distance of the Earth is 1395 W/m^2. This number is called solar constant. Some part of this energy is absorbed by carbon dioxide and water vapour present in the Earth's atmosphere. More movement through atmosphere is observed, if sun-rays are inclined. During this inclined incidence, less part of solar radiation reaches the earth. Outside atmosphere, solar radiation follows gray body behaviour. Maximum is observed at a wavelength range of 0.5 μm and using Wien's displacement law, equivalent solar temperature fro thermal radiation is obtained as around 5800 K.

EXAMPLE 5.37

Calculate maximum emissive power in W/m^3 and the wavelength at which it occurs in microns by a blackbody surface at (i) 300 K and (ii) 3000 K.

SOLUTION

Maximum emissive power, $E_{b\lambda,\max} = 1.2865 \times 10^{-5}\ T^5$ W/m^3.

Wein's displacement law, $\lambda_{\max} \cdot T = 2897.6$ μm.K.

(*i*) At 300 K, $E_{b\lambda,\max} = 1.2865 \times 10^{-5}\ T^5$ W/m^3

$= 1.2865 \times 10^{-5} \times 3005$ W/m^3 $= 31.2619 \times 106$ W/m^3

The wavelength is given by Wein's displacement law.

$\lambda_{\max} = 2897.6/T$ μm $= 9.6586$ μm.

(*ii*) At 3000 K, $E_{b\lambda,max}$ = 1.2865 × 10-5 T5 W/m3

= $1.2865 \times 10^{-5} \times 30005$ W/m^3 = 3.12619 × 1012 W/m^3

The wavelength is given by Wein's displacement law.

λ_{max} = 2897.6/T μm = 0.96586 μm.

EXAMPLE 5.38

The yellow lines in the visible spectrum have an approximate wavelength of 0.58 μm. estimate (i) frequency of yellow waves (ii) the temperature at which it radiates maximum blackbody emissive power.

SOLUTION

All electromagnetic radiations travel at a speed of light, which are 3×10^8 m/s.

As, Velocity = frequency × wavelength,

So, frequency = velocity/wavelength = $3 \times 10^8/0.58 \times 10^{-6}$ Hz

= 517.241 × 1012 Hz.

Temperature at which maximum blackbody emissive power occurs is given by Wein's displacement law, $\lambda_{max} \cdot T$ = 2897.6 μm.K.

$T = 2897.6/\lambda_{max}\ K = 4995.86\ K.$

EXAMPLE 5.39

At wavelength of 0.7 μm, for what temperature will a blackbody have spectral emissive power of 106 W/m^3.

SOLUTION

The problem is based on Planck's distribution law. $E_{b\lambda} = dE_b/d\lambda = (C_1 \times \lambda^{-5})/(\exp[C_2/\lambda T] - 1)$, where λ = wavelength in μm, T = temperature in K, C_1 = 3.743×10^8 W μm^4/m^2, $C_2 = 1.4387 \times 10^4$ μm.K.

$$e^{C2/\lambda T} = 1 + (C_1/E_{b\lambda} \times \lambda^5) = 1 + [3.743 \times 10^8 \times 10^6/(10^6 \times 0.75)]$$
$$= 2.22704 \times 10^9$$

$$C_2/\lambda T = 21.5239$$

$$T = C_2/(21.5230 \times \lambda) = 1.4387 \times 10^4/(21.5230 \times 0.7) \text{ K}$$
$$= 954.88 \text{ K}.$$

EXAMPLE 5.40

At wavelength of 0.4 μm, for what temperature will a blackbody have spectral emissive power of 109 W/m^3.

SOLUTION

The problem is based on Planck's distribution law. $E_{b\lambda} = dE_b/d\lambda = (C_1 \times \lambda^{-5})/(\exp[C_2/\lambda T] - 1)$, where λ = wavelength in μm, T = temperature in K, $C_1 = 3.743 \times 10^8$ W μm^4/m^2, $C_2 = 1.4387 \times 10^4$ μm.K.

$$e^{C2/\lambda T} = 1 + (C_1/E_{b\lambda} \times \lambda^5) = 36538087.0$$

$$C_2/\lambda T = 17.4138$$

$$T = C_2/(17.4138 \times \lambda) = 2065.6 \text{ K.}$$

EXAMPLE 5.41

What percentage of emissive power is in the wavelength range 0 to 1.2 μm for a blackbody at temperature 5000 K?

SOLUTION

$F(\lambda T) = F(C_2/w) = {}_0\int^{\lambda T} E_{b\lambda}(\lambda, T)\, d\lambda/\sigma T^4 = (15/\pi^4) \times {}^{\lambda T}\int^{\infty} w^3\, dw/(e^w - 1)$

$= (15/\pi^4) \times {}_{i=1}\Sigma^{\infty}\, i^{-4}\, e^{-iw}\, [(iw)^3 + 3\,(iw)^2 + 6\,(iw) + 6]$

for the problem, $\quad w = C_2/\lambda T = 1.4387 \times 10^4/(1.2 \times 5000)$

$= 2.3978333$

First term of expansion $= e^{-w}\,[(w)^3 + 3\,(w)^2 + 6\,(w) + 6] = e^{-w} \times [6 + w\,(6 + w\,(3 + w))] = 4.675$.

Second term of expansion $= (1/16) \times e^{-2w}\,[(2w)^3 + 3\,(2w)^2 + 6\,(2w) + 6] = e^{-2w} \times [6 + w\,(12 + w\,(12 + 8w))]/16 = 0.11058$.

Third term of expansion $= (1/81)\, e^{-3w}\,[(3w)^3 + 3\,(3w)^2 + 6\,(3w) + 6] = e^{-3w} \times [6 + w\,(18 + w\,(27 + 27w))]/81 = 0.00535$.

Further terms are not considered, as terms are becoming smaller as compared to first term.

$$F(\lambda T) = (15/\pi 4) \times (4.675 + 0.11058 + 0.00535\ldots) = 0.7377$$

$$= 73.77\%.$$

EXAMPLE 5.42

Consider a 1 cm OD sphere located in a 2 cm ID sphere. Find all shape factors.

SOLUTION

Let smaller body is designated as body '1' and bigger sphere is body '2'. There are 4 radiation shape factors possible namely $F_{1\text{-}1}$, $F_{1\text{-}2}$, $F_{2\text{-}1}$ and $F_{2\text{-}2}$.

Sphere is a concave surface and no part of smaller sphere can see any other part of the smaller sphere. So, $F_{1\text{-}1} = 0$.

As $\quad F_{1\text{-}1} + F_{1\text{-}2} = 1.0,$

$$F_{1\text{-}2} = 1.$$

This indicates that all energy emitted by body '1' (smaller sphere) is intercepted by body '2' (bigger sphere inner surface).

It is derived that $A_1 \,.\, F_{1\text{-}2} = A_2 \,.\, F_{2\text{-}1}$,

So $F_{2\text{-}1} = (A_1/A_2) \times F_{1\text{-}2} = \frac{1}{4} = 0.25.$

As $F_{2\text{-}1} + F_{2\text{-}2} = 1.0$,

$F_{2\text{-}2} = 1 - 0.25 = 0.75.$

▮▮ EXAMPLE 5.43

For a long semicircular duct (⌓) of radius 'R', diameter is represented as surface 1 and curved surface as surface 2. Compute $F_{1\text{-}2}$, $F_{2\text{-}1}$, and $F_{2\text{-}2}$.

SOLUTION

In this case, surface 1 is convex, so $F_{1\text{-}1} = 0$.

From properties of shape factor,

$$F_{1\text{-}2} = 1 - F_{1\text{-}1} = 1.$$

$$F_{2\text{-}1} = A_1\, F_{1\text{-}2}/A_2 = 2R \times 1/\pi R = 2/\pi.$$

$$F_{2\text{-}2} = 1 - F_{2\text{-}1} = 1 - 2/\pi.$$

▮▮ EXAMPLE 5.44

A long circular duct is in the form of a triangle, geometry is specified. Surface 1 – 30 cm, surface 2 – 20 cm. What should be angle between both these surfaces so that only 40% of the radiant energy from surface 1 is captured by surface 3?

SOLUTION

Given, $L_1 = 30$ cm, $L_2 = 20$ cm, $F_{1\text{-}3} = 0.4$

For the geometry, $F_{1\text{-}3} = (L_1 + L_3 - L_2)/2L_1$

$$= (30 + L_3 - 20)/60.$$

$$L_3 = 14 \text{ cm}.$$

There is an alternate way to find L_3. Since radiation emitted by surface 1 is captured either by surface 2 or surface 3, $F_{1\text{-}2} + F_{1\text{-}3} = 1$. So $F_{1\text{-}2} = 0.6$.

$$F_{1\text{-}2} = (L_1 + L_2 - L_3)/2L_1 = (30 + 20 - L_3)/60$$

$$L_3 = 14 \text{ cm}.$$

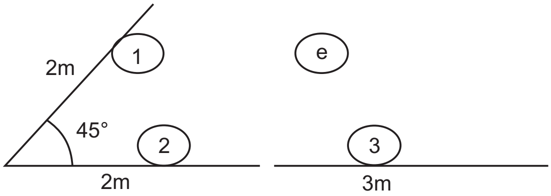
Angle between surface 1 and 2 can be calculated using cosine law, as all three sides are known.

$$\text{Cos } (1 - 2) = (L_1^2 + L_2^2 - L_3^2)/2L_1L_2 = 0.92.$$

$$\text{Required angle} = \text{Cos}^{-1}\ (0.92) = 23.074°$$

EXAMPLE 5.45

For the figure shown find all shape factors, where e is environment and 1,2,3 are plane surfaces.



SOLUTION

This problem also resembles previous problem of triangle.

Some of the identities of the problems are – $F_{1\text{-}1} = 0 = F_{2\text{-}2} = F_{3\text{-}3} = F_{2\text{-}3} =$ $F_{3\text{-}2}$

$$F_{1\text{-}2} = F_{2\text{-}1} = 1 - \text{Sin } (45/2) = 0.6173.$$

$$F_{1\text{-}3,e} = 1 - F_{1\text{-}2} = 0.3827$$

$$F_{2\text{-}e} = 1 - F_{2\text{-}1} - F_{2\text{-}2} - F_{2\text{-}3} = 1 - 0.6173 - 0 - 0 = 0.3827.$$

Let us join end of surface 1 and 3, so as to make a triangle. The

unknown side of the triangle may represent environment. Length of this third side may be calculated from trigonometry.

$$L_e = \sqrt{[2^2 + 5^2 - 2 \times 2 \times 5 \times \text{Cos } 45]} = 3.8546 \text{ m}$$

$$F_{1\text{-}2,3} = (L_1 + L_{2,3} - L_e)/2L_1 = 0.7863$$

$$F_{1\text{-}3} = F_{1\text{-}2,3} - F_{1\text{-}2} = 0.169$$

$$F_{1\text{-e}} = F_{1\text{-}3,\text{e}} - F_{1\text{-}3} = 0.2137$$

$$F_{3\text{-}1} = L_1 F_{1\text{-}3}/L_3 = 0.1127$$

$$F_{3\text{-e}} = 1 - F_{3\text{-}1} - F_{3\text{-}2} - F_{3\text{-}3} = 1 - 0.1127 - 0 - 0$$

$$= 0.8873.$$

Surfaces	*1*	*2*	*3*	*e*
1	0.0000	0.6173	0.1690	0.2137
2	0.6173	0.0000	0.0000	0.3827
3	0.1127	0.0000	0.0000	0.8873

EXAMPLE 5.46

Find all shape factors for a rectangular long duct with sides 'a' and 'b'.

SOLUTION

Let surface represented by dimension 'a' is 1 and opposite to it lays surface 3. Surface with dimension 'b' are called 2 and 4.

Shape factor for surfaces 3 and 4 behave in exactly same manner as that for surfaces 1 and 2 respectively. So calculation for surface 3 and 4 are not needed.

Let us merge surface 3 and 4 into a single surface 'h' and constitute a triangle with sides 'a', 'b' and $\sqrt{(a^2 + b^2)}$. The third side is calculated using Pythagorus theorem and may be designated as 'c' [$= \sqrt{(a^2 + b^2)}$] for this triangle 1-2-h.

$$F_{1\text{-}2} = (a + b - c)/2a = [a + b - \sqrt{(a^2 + b^2)}]/2a.$$

This shape factor is valid for the rectangular duct also for the adjacent surface.

For surface 1,

$$F_{1\text{-}1} = 0.$$

$$F_{1\text{-}2} = (a + b - c)/2a = [a + b - \sqrt{(a^2 + b^2)}]/2\text{a}.$$

$$F_{1\text{-}3} = 1 - F_{1\text{-}1} - F_{1\text{-}2} - F_{1\text{-}4} = 1 - 2F_{1\text{-}2} = (c - b)/a = [\sqrt{(a^2 + b^2)} - b]/a.$$

$$F_{1\text{-}4} = F_{1\text{-}2} = [a + b - \sqrt{(a^2 + b^2)}]/2a.$$

For surface 2,

$$F_{2\text{-}1} = A_1F_{1\text{-}2}/A_2 = (a/b) \times F_{1\text{-}2} = [a + b - \sqrt{(a^2 + b^2)}]/2b.$$

$$F_{2\text{-}2} = 0.$$

$$F_{2\text{-}3} = F_{2\text{-}1} = [a + b - \sqrt{(a^2 + b^2)}]/2b.$$

$$F_{2\text{-}4} = 1 - F_{2\text{-}1} - F_{2\text{-}2} - F_{2\text{-}3} = 1 - 2F_{2\text{-}1} = (c - a)/b = [\sqrt{(a^2 + b^2)} - a]/b.$$

▮▮ EXAMPLE 5.47

A 50 mm diameter sphere is maintained at 600°C and is near an infinite wall maintained at 100°C. Both the surfaces are black. Find net radiant heat transfer between the two.

SOLUTION

Assume sphere and wall as surfaces 1 and 2 respectively.

Net radiant heat transfer $= A_1\, F_{1\text{-}2} \cdot (E_{b1} - E_{b2}) = A_1\, F_{1\text{-}2} \cdot \sigma \cdot (T_1^4 - T_2^4)$

If infinite walls are placed on either side of the sphere, all the emitted radiation from sphere will be irradiating the infinite wall. Since only one infinite wall is present, only half the radiation is seen by wall and $F_{1\text{-}2} = 0.5$. $T_1 = 500 + 273$ K = 773 K, $T_2 = 100 + 273 = 373$ K.

Net heat transfer = $(\pi \times 0.05^2) \times 0.5 \times 5.669 \times 10^{-8} \times (773^4 - 373^4)$ W = 75.17 W.

EXAMPLE 5.48

Calculate the emissive power of a blackbody at (a) 0°C (b) 70°C (c) 200°C (d) 6000°C.

SOLUTION

Use Stefan-Boltzmann's Law, $E_b = 5.669 \times 10^{-8} \times T^4$ W/m^2

(*a*) 314.8 W/m^2 (*b*) 784.66 W/m^2 (*c*) 2837.6 W/m^2 (*d*) 8.778×107 W/m^2.

EXAMPLE 5.49

Two very large plates are maintained at uniform temperatures T_1 = 800 K and T_2 = 500 K and have emissivity ε_1 = 0.2 and ε_2 = 0.7, respectively. Determine net radiation heat transfer between the two surfaces per unit surface area of the plate.

SOLUTION

Neat radiant heat transfer

$$= \sigma \times (T_1^4 - T_2^4)/[(1 - \varepsilon_1)/\varepsilon_1 + 1/F_{1\text{-}2} + (1 - \varepsilon_2)/\varepsilon_2]$$

$$= 5.669 \times 10^{-8} \times (800^4 - 500^4)/[(1 - 0.2)/0.2 + 1/1 + (1 - 0.7)/0.7\ ] \text{ W/m}^2$$

$$= 3624.73 \text{ W/m}^2$$

SUMMARY

This unit gives a brief outline of heat transfer and starts with difference of heat transfer and thermodynamic treatment of energy exchange. First, three modes of heat transfer namely conduction; convection and radiation are introduced along with their governing equations. Conduction for 1-D and 2-D are explained. Heat transfer in fins used for cooling internal combustion engine is also explained. Convection mode of heat transfer and various non-dimensional numbers to explain convection is introduced. Natural/Free convection on various surfaces are explained and empirical relations are produced. Radiation is explained as extension of electromagnetic radiations. Defining terms and radiation heat transfer mechanism are also explained. Numerical examples are given to understand the mode of heat transfers.

QUESTIONS

1. What are various modes of heat transfer? How each mode is different from one another?
2. Why heat transfer is important for engine performance?

3. What are factors affecting heat transfer in the engine?
4. What is thermal conductivity? What is its effect on heat transfer in the operation of an engine?
5. What is mechanism of heat conduction in solids, liquids and gases?
6. What is mechanism of heat transfer through fins? What is fin effectiveness?
7. Derive expression for heat transfer through an infinite fin.
8. What is Reynold's number? What is its significance in convective heat transfer process?
9. How does thermal conductivity of different materials change?
10. What are various methods for calculation of heat transfer in two dimensions?
11. Explain numerical method of heat transfer in two dimensions?
12. Why thermal radiation does not need any media for heat transfer?
13. What is Stefan-Boltzman law? How does it help in deriving heat transfer through radiation?
14. What is Planck's law? Derive Wein's displacement law from Plank's law.
15. What is shape factor for radiation? Which parameters affect it?
16. Give reasons for the following
    (*a*) Conductivity of metals is high.
    (*b*) Good electric conductors are good heat conductors also.
    (*c*) Gases at high temperature and pressure have high thermal conductivity.
    (*d*) Thermal diffusivity of metals and gases are high but they are low for liquids.
    (*e*) Ice forms on ground at night even if air temperature is higher than freezing point of water.
    (*f*) Blackbodies are called so.
    (*g*) Radiation is dominant at high temperature.
    (*h*) Transmission electrical wires are uninsulated (bare).
17. Differentiate between the following :
    (*a*) Free convention and Forced convection
    (b) Hydrodynamic boundary layer and Thermal boundary layer.
    (*c*) Emissivity and Absorptivity.
    (*d*) Blackbody and Gray body.
    (*e*) Radiocity and Irradiation.
    (*f*) Fin efficiency and Fin effectiveness.
    (*g*) Biot number and Nusselt number.
18. Write short notes on the following :
    (*a*) Thermal resistance.
    (*b*) Thermal diffusivity.
    (*c*) Critical thickness of insulation.
    (*d*) Cooling by fins.
    (*e*) Prandtl number.
    (*f*) Nusselt number.
    (*g*) Greenhouse effect.

❖ ❖ ❖

CHAPTER

6

Carburetion

STRUCTURE

- Introduction
- Objective
- Fuel-air Ratio Requirements
- Typical Carburetor Elements
- Calculation of Venturi and Fuel Orifice Size
- Summary
- Questions

6.1 INTRODUCTION

Carburetion is process of preparation of fuel-air mixture in a typical SI engine. In such engines, the mixture is prepared outside the engine cylinder. This chapter gives a brief idea about requirements of fuel-air mixture and it's criticality for power generation by SI engines. This also explores design calculation for sizing vent holes for flow of fuel through air streams. Various elements of carburetor are also discussed in this Chapter.

Carburetor is a device which atomizes the fuel and mixes it with air outside engine cylinder. It is a part of intake manifold of the engines. Some of the atomized fuel vapourises due to suction created in intake manifold during suction stroke and remaining droplets of bigger sizes do the same in engine cylinder before ignition by electric spark. Carburetion is basically a process, which is governed by (*i*) the time for preparation of mixture (*ii*) the temperature of incoming air (*ii*) the quality of fuel supplied (*iv*) design of combustion chamber and induction system.

Objective

After studying this Chapter, you should be able to understand:

Requirements of carburetion in SI engines,

- Critical requirements of carburetion,
- Properties of fuel-air mixtures,
- Governing principles for calculation of venturi and fuel orifice,
- Design of various types of carburetors.

6.2 FUEL-AIR RATIO REQUIREMENTS

SI engine works on petrol and invariably, carburetion is concerned with petrol-air mixture requirements. Carburetion is combination of atomization, mixing and vaporization processes. In high speed engines, time available for mixture formation is very small (of the order of 10-50 ms). This requirement of efficient, complete and distributed mixing makes design of carburetors very critical. Temperature of incoming air dictates rate of vaporization of fuel and high temperature means higher rate of vaporization. It is not temperature alone, but volatility of fuel also decides the rate of vaporization. Finally, design of carburetor, intake manifold becomes a complicated task.

The most important parameter in carburetor design is requirement of optimum fuel-air ratio at all places in the engine cylinders. In SI engines, there is a limited range of fuel-air ratio for which mixture can be ignited. Stoichiometric fuel-air ratio is 1:15. On richer side, the ratio is limited to 1:7, while on lean side; fuel-air ratio must be limited to 1:22. The mixture cannot be ignited, if fuel-air ratio lies beyond this range.

For a very rich mixture ratio of the order of 1:9 to 1:11, low power accompanied with high fuel consumption, incomplete combustion and carbon deposits on exhaust line is observed. For a rich mixture of the range 1:12 to 1:14, good condition for maximum power exists. Exhaust is free from oxygen and maximum temperature is realized giving a high flame propagation speed. On the other side, for a very lean mixture of fuel-air ratio 1:18 to 1:22, minimum power is realized, burning process slow down and erratic running with overheating is observed. Back-fire and popping in carburetor is observed for this condition of fuel-air ratio. For a slightly lean mixture in the range of 1:15 to 1:18, condition for most economical mixture is realized.

Depending on operating conditions of throttle, speed, power and fuel consumption, different fuel-air ratios are preferred. Maximum energy from fuel is released, if slightly excess fuel is introduced in the cylinder, so as to utilize all the oxygen present in the cylinder. Maximum power is realized, if the fuel-air ratio is 1:12.5. At conditions of maximum power, mechanical efficiency is optimum. If more fuel than this condition is supplied, smaller energy per unit mass of fuel is released due to incomplete combustion and formation of carbon monoxide. Minimum specific fuel consumption is realized, if all the supplied fuel undergoes complete combustion. Although, Stoichiometric fuel-air ratio in ideal condition is considered synonym of complete combustion, but under such conditions, some of fuel portion remain without air and at other locations, excess oxygen is observed. Some extra air is supplied to ensure that fuel available at each location inside the engine cylinder gets air. At full throttle, minimum specific fuel consumption occurs at fuel-air ratio of around 1:17. Making fuel-air mixture leaner, results in reduction of flame speed, increase in time. The conditions for a typical engine are shown in figure 6.1.

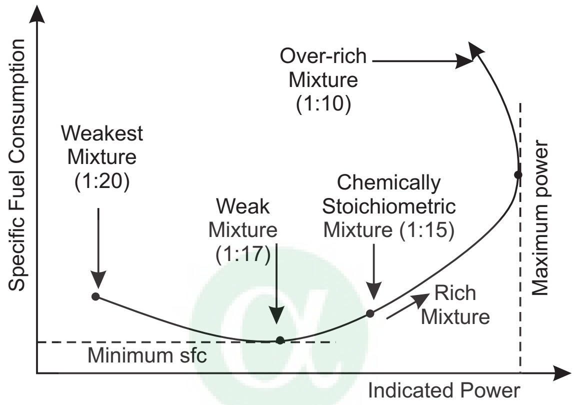


Fig. 6.1 : Fuel Consumption and Indicated Power Curves for SI Engine

Practical conditions are different from ideal working conditions. Depending on load requirements, fuel-air ratio varies. From no loads to around 20% of rated power, the engine is said to be under idling condition and needs rich mixture. At idling condition, air supply is restricted by nearly closed throttle and the suction pressure is very low. The condition of low pressure gives backflow of exhaust gases as well as leakage of air from other intake systems. Under such conditions, dilution of incoming fuel occurs. This leads to erratic combustion with very poor combustion efficiency.

Another factor, which is predominant for idling condition is reduction in exhaust gas temperature at low loads. This increase density and mass of residual gases in engine cylinder. So, during idling condition, fuel-air mixture should be slight rich.

Fuel economy is not important during this period, as amount of fuel burned is very small. Supply of fuel-rich mixture increases probability of contact between fuel and air particles and thus improves combustion.

During normal operating conditions depicted by 20% to around 80% of full load, dilution by exhaust gases and intake manifold leakage, both reduces. The attention is also diverted to fuel economy during this power ranges and fuel lean mixture with fuel-air ratio of around 1:17 is sufficient during normal operation of the engine.

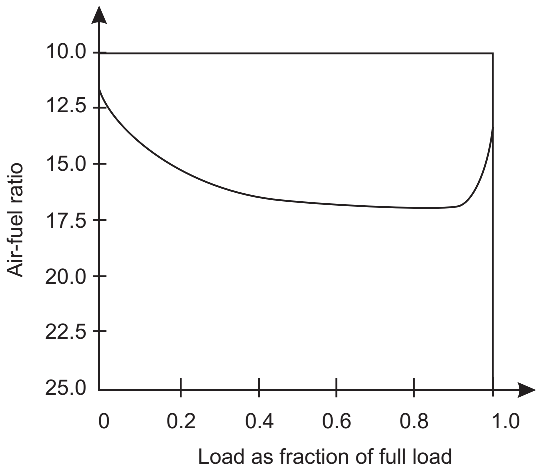


Fig. 6.2 : Operational Ranges of SI Engines

During maximum power range defined by 80% to full load, again rich mixture is needed. In addition to providing maximum power, this condition also prevents exhaust valves from overheating and prevents detonation. In aircraft engines, special provisions are made for enrichment of mixture. All three operational ranges of loads for a typical SI engine are depicted in figure 6.2.

6.3 TYPICAL CARBURETOR ELEMENTS

From construction point of view, carburetors have to facilitate supply of air-fuel mixture to the engine cylinder for cruising speeds or normal loads (20% to 80% of rated power). Further mechanisms to perform other duties like starting, idling, variable loads, variable speeds and acceleration are added to the simple carburetor. Figure 6.3 shows sketch of a simple carburetor.

The most important system in carburetor is float chamber, which is mainly responsible for governing quantity of fuel, supplied to the engine-cylinder. From the inlet fuel flow line, fuel is received from fuel-tanks. It passes through a strainer, which filters incoming fuel. A needle valve attached to a float maintains level of fuel in float chamber. If more fuel is available in the float chamber, needle valve lifts up and stops fuel supply. Level of fuel in float chamber is maintained at same level as the nozzle for discharging jet of fuel is placed in main air-flow line. Float chamber is vented to atmosphere. During suction stroke, air is drawn through the venturi, which resembles a nozzle in construction. It has a variable cross-sectional area, which decreases to minimum at throat. This venturi tube is also known as choke tube. The curved surface of nozzle is maintained for minimum resistance to air flow. As velocity of air moving in venturi increases, pressure at the venturi reduces. The pressure at the fully open throat condition is around 40-50 mm Hg. To avoid wastage of fuel, the level of liquid in the jet is adjusted by the float chamber needle valve to maintain the level a short distance below the tip of the discharge jet. If this condition is not maintained, fuel flow will continue at no-throttle also. As pressure in the float chamber is atmospheric due to venting, a pressure difference called carburetor depression is created and fuel is discharged into air stream. Size of smallest section in the fuel passage controls or meters the rate of flow of fuel. Size of discharge jet is empirically fixed for a definite engine performance.

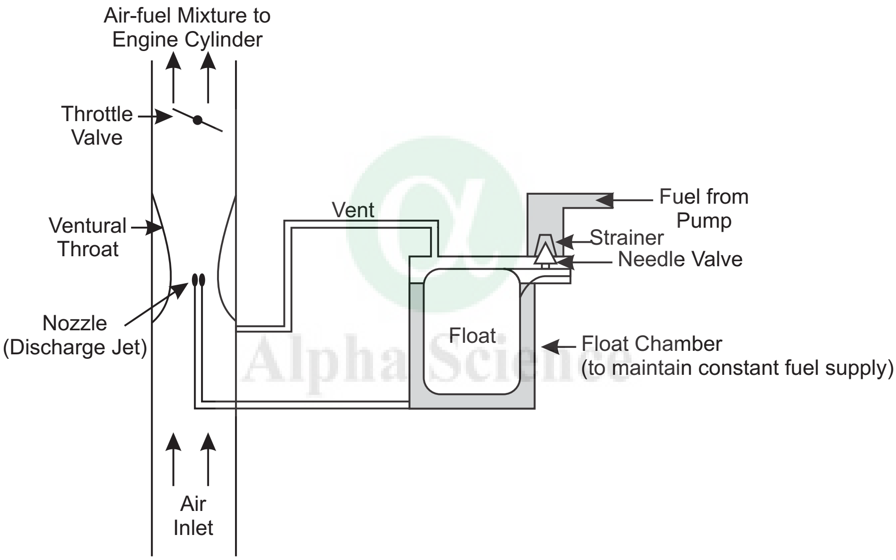


Fig. 6.3 : Sketch of a Simple or Elementary Carburetor

Control on carburetor depression is key parameter to fuel metering. This is controlled by a butterfly valve called throttle valve. As throttle is closed, less quantity of air flows through the venturi and flow velocity of air-stream is also smaller. This gives lower carburetor depression and less quantity of fuel-supply to maintain air-fuel ratio properly. So, carburetor is quantity-governed device. When more power is needed, throttle is opened and more quantity of both air and fuel is sucked-in. However, as throttle is opened more and more, carburetor depression increases and more fuel are inducted in venturi tube. At higher air flow velocity and lower pressure, density of air reduces, resulting in reduction in mass of air for same volume flow rate. This leads to increase in fuel-richness of air-fuel mixture at higher throttle openings. Similarly, for closed throttle or low load condition, lean mixture is supplied to engine, which is contrary to the requirements of fuel-air ratio, depicted in figure 6.2. To overcome these problems, certain other devices are added to simple carburetors. To compensate for the variable fuel-air ratio and supply nearly constant fuel-air ratio to engine at a wide range of speeds and loads, compensating devices are incorporated in the main metering system to correct the anomalies in fuel supply. To control fuel richness at full/higher throttle, devices to supply additional air to venturi tube is attached. In nutshell, a simple carburetor has following deficiencies.

- Engine requires rich mixtures at low loads, but simple carburetor gives leaner mixture. As introduction of fuel depends on venturi depression, which is a function of engines speed, at low loads, low venture depression gives lower venturi depression and less suction of fuel.
- At intermediate loads, equivalence ratio of mixture should be more or less constant. Actually a simple carburetor gives richer mixture at high loads.
- After throttle of the simple carburetor is fully opened, upper limit of equivalence ratio is achieved and it cannot be increased further. However, for maximum power equivalence ratio of 1.1 or above should be achieved.
- During engine start and warm up, rich mixture is needed. A simple carburetor cannot give this.
- An elementary carburetor cannot compensate disturbances, due to Transient flow of fuel-air mixture in the intake manifold of the engine.
- Provision of altitude correction is not present in simple carburetor, which considers reduction in density of air at high altitude.

In an elementary carburetor, as airflow increase, larger vacuum is created at the throat. Actual purpose of venturi is two fold – first is enhancement in flow velocity and second is venturi depression. Higher vacuum or depression is always accompanied with higher airflow. Increasing air velocity without increasing pressure loss significantly multiple nozzles is used as depicted in figure 6.4 (top right corner). An auxiliary boost venturi is placed upstream in the choke tube, such that discharge from additional venturi takes place at the main venturi. Only a fraction of total air flows through annular space available with main venturi. No doubt main venturi pressure depression augments the boost venturi vacuum and more pressure drop is observed at the venturi of the auxiliary venturi. Use of multiple venturi increases airflow velocity, but fuel is inserted at the throat of main venturi only, venturi depression is controlled. This results in supply of more homogenous mixture through the carburetor. Triple venturi systems can be implemented but diameter of main venturi should increase to accommodate discharge from many auxiliary boost venturies. Additionally overall discharge coefficient of multiple venturi system is also lower. Another way to get similar results is by splitting a single carburetor into several choke tubes of lower diameters operating in parallel. This gives better part-load metering, high volumetric efficiency and maximum air flow velocity.

Provision for compensating jet is one such device (depicted in figure 6.4). A compensating well is attached to float chamber through an orifice for supply of fuel and is connected to a compensating jet, opening in the venturi-tube. Compensating jet is open to atmosphere. As carburetor depression increases due to higher reduction in venturi pressure, level of fuel in the compensating well reduces, resulting in reduced fuel supply from the compensating jet. This leads to control of fuel-richness at higher throttle opening. Additionally, it supplies rich mixture in initial conditions of low throttle.

In modern carburetors, the mixture is controlled by emulsion tube or air-bleeding. In this case, main metering jet is fitted around 25 mm below the petrol level and above this a well with provision of direct air-entry is attached. At normal operating conditions, discharge of fuel is normal. However, at higher throttle with more pressure reduction, air is sucked in the well through holes open to atmosphere. The supply becomes a mixture of air and fuel rather than pure fuel. This emulsification of fuel causes leanness to mixture at higher throttle. Similarly, provision of back-suction control, auxiliary valves and auxiliary ports are also helpful in reducing fuel-richness of supplied mixture at fully open throttle.

Another concern is requirement of fuel-rich mixture for idling conditions, which is otherwise not possible with simple carburetors. In fact, at closed throttle, no suction is created in the venturi and no supply of fuel is possible by main metering device. So, idling condition needs a separate idling jet in the main metering system. Additionally, this separate supply line of fuel must become gradually ineffective once throttling starts. Idling jet consist of a direct connection pipe from float chamber to a little downstream in the venturi tube towards engine side (figure 6.4). The opening is generally placed immediately after the throttle valve and the piping is open to atmosphere for supply of air. When throttle is close, there is no flow of air through main pipe and no suction exists for supply of any fuel to the engine. However, due to suction stroke of engine, lower pressure acts directly on the idling jet and lifts the fuel. As throttle is opened gradually, suction created at idling jet inlet reduces and main jet gradually takes over and slowly idling jet becomes ineffective. In automobiles, needle valve with air bleed called idle adjust is provided to regulate air-fuel ratio for idling jet.

Another attachment for all the carburetor devices is called choke, which supports cold start of the engine. At low cranking speed before engine warm up, fuel vaporization is poor and so a mixture 5-10 time more rich than normal is desired. To get such high richness of mixture, choke is placed upstream in venturi tube before the throat section (figure 6.4). Choke is a butterfly valve, which closes air supply completely through the main supply line. Once choke is operated, entire suction of suction stroke acts near throat resulting in higher carburetor depression. This results in drawing higher quantity of fuel and restricting quantity of air. Sufficiently rich mixture is supplied to engine for cold start. However for normal operation, choke has to be closed. Sometimes spring loaded choke is provided in SI engines. Using thermostat or sensor for monitoring condition of engine, operation of choke can be automated also.

In automobiles, carburetors are provided with atmospheric pressure conditions in mind. However, for aircraft operations, atmospheric pressure becomes another variable. As altitude of aircraft rises, density of air reduces and supply of mixture by carburetor at higher altitude for same throttle opening becomes fuel rich. To compensate this fuel richness, earlier explained devices like air bleeding, back-suction control or metering pin are provided. The float system also becomes ineffective in aircrafts because of tilt-conditions of the engine during banking. Special types of devices are mandatory to ensure supply of fuel at all points of aerobatic or banking operations. As temperature reduces with altitude, formation of ice on choke tube is another burning problem due to supply of cold air in suction of an aircraft-carburetor. To overcome this obstacle, de-icing units or hot engine oil supply is resorted to.

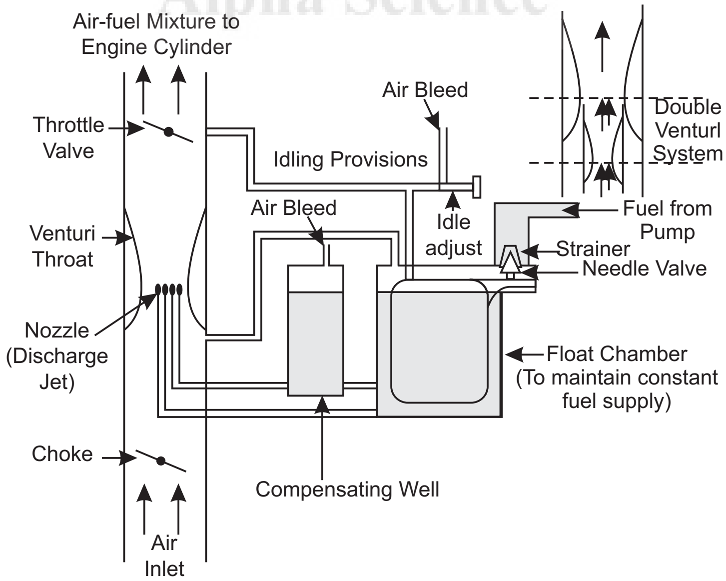


Fig. 6.4 : Modifications to an Elementary Carburetor

Throttle valve can be made hollow and hot engine oil can be re-circulated around to maintain temperature at higher levels.

6.4 CALCULATION OF VENTURI AND FUEL ORIFICE SIZE

For calculation of venturi and orifice sizes in a carburetor, energy equation is used. Since there is no work or heat transfer and flow through choke tube considered isentropic. Inlet condition is considered to have zero velocity of flow for idealized condition. Suction head is '*h*', which is distance to which fuel can be lifted by suction head to get fuel from float chamber.

Comparison of two conditions as depicted by figure 6.5 can give calculation strategy for venturi as well as orifice size. If pressure is denoted by '*p*', volume by '*V*', velocity of flow by '*C*', temperature by '*T*', area of cross-section by '*A*', then for the air supply calculation, following approach is adopted.

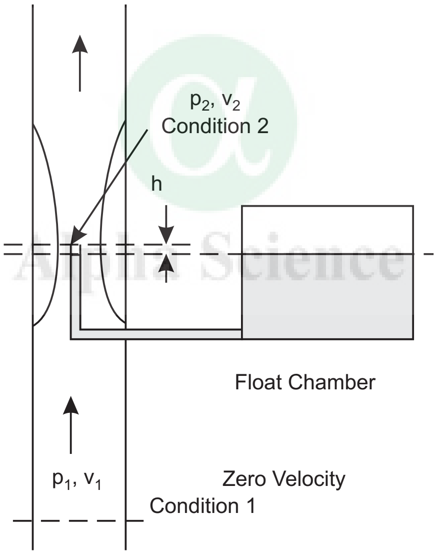


Fig. 6.5 : Principles of a Simple Carburetor

Since air is assumed to be an ideal gas with ratio of specific heats as γ and difference of specific heats as R, a relation exists between these conditions (refer chapter 2).

$$C_p = \gamma \,.\, R/(\gamma - 1)$$
$$C_v = R/(\gamma - 1).$$

Considering isentropic flow between conditions '1' and '2', following relations are valid.

$$p_1/p_2 = (V_2/V_1)^{\gamma} = (T_1/T_2)^{\gamma/(\gamma-1)}.$$

The ideal gas equations given below are always applicable to the air flow.

$$p_1 \,.\, V_1/T_1 = p_2 \,.\, V_2/T_2.$$

Using energy equation between '1' and '2' following equation can be written.

$$C_p\, T_1 = C_p\, T_2 + C_2^2/2g.$$

Mass is conserved between conditions '1' and '2' and mass flow rate of air (ma) is given by equation given below.

$$m_a = A_1 \,.\, C_1/V_1 = A_2 \,.\, C_2/V_2.$$

Fuel is incompressible and there is no change in its density. Applying energy equation again, following equation is valid for fuel.

$$p_1/\rho_f = p_2/\rho_f + C_f^2/2g$$

$$\text{Mass of fuel, } m_f = A_f\, C_f\, \rho_f.$$

Using above relations, design of orifice and venturi can be executed.

EXAMPLE 6.1

A simple carburetor supplies mixture at Stoichiometric fuel-air ratio of 1:12.5. Fuel density is 740 k/m^3 and inlet conditions are depicted by pressure and temperature values of 1.03 kg/cm^2 and 27°C respectively. Calculate throat diameter of the choke for a flow velocity of 100 m/s and air flow rate of 6 kg/min. Velocity coefficient (C_d) is 0.8. If pressure drop across the fuel metering orifice is 0.75 of that at the choke, calculate orifice diameter assuming discharge coefficient as 0.6. Take γ = 1.4, R = 29.27 kgf-m/kg K.

SOLUTION

Refer figure 6.5.

Given data, F:A = 1:12.5, ρ_f = 740 kg/m^3, p_1 = 1.03 kg/cm^2, T_1 = 27°C = 300 K, C_2 = 100 m/s, ma = 6 kg/min, C_d = 0.8, C_{df} = 0.6, γ = 1.4, R = 29.27 kgf-m/kg K.

$$C_p = \gamma \,.\, R/(\gamma - 1) = 102.445 \text{ kgf-m/kg K.}$$

$$C_v = R/(\gamma - 1) = 73.175 \text{ kgf-m/kg K.}$$

For air flow through choke tube, $C_2^2 = 2 \,.\, R \,.\, g. \,(T_1 - T_2) \,.\, C_d$

So, $\quad T_2 = T_1 - C_2^2/(2 \,.\, R \,.\, g \,.\, C_d) = 278.23$ K.

Pressure at venturi, $p_2 = p_1 \times (T_2/T_1)^{\gamma/(\gamma-1)} = 0.79127$ kg/cm^2.

Volume at inlet condition, $V_1 = R \,.\, T_1/p_1 = 0.8525$ m^3/kg.

Volume at venturi, $V_2 = V_1 \times (p_1/p_2)^{1/\gamma} = 0.9758$ m^3/kg.

From mass balance equation, $m_a = A_2 \cdot C_2/V_2$

So, area of cross-section at venturi, $A_2 = m_a \times V_2/C_2$

$= 7.58$ cm^2.

Diameter at venturi $= \sqrt{4 \times A_2/\pi} = 3.1$ cm.

Pressure drop across venturi $= p_1 - p_2 = 0.2387$ kg/cm^2.

Pressure drop across fuel metering orifice $= 0.75 \times 0.2387$ kg/cm^2 $=$ 0.179 kg/cm^2.

Mass flow rate of fuel, $m_f = m_a$/F : A $= 0.48$ kg/min $= 0.008$ kg/s

Flow velocity of fuel, $C_f = \sqrt{2.g.(p_1 - p_2)/\rho_f} = 7.956$ m/s

From the equation, $m_f = A_f \times C_f \times C_{df} \times \rho_f$

So, Area of orifice, $A_f = m_f/(C_f \times C_{df} \times \rho_f) = 0.0226$ cm^2

Diameter of orifice, $d_f = \sqrt{4 \times A_f/\pi} = 0.1698$ cm $= 1.698$ mm.

EXAMPLE 6.2

Determine the sizes of fuel orifice to give a 13.5 : 1 air-fuel ratio, if venturi throat is 3 cm diameter and the vacuum in the venturi is 6.5 cm Hg. The air temperature and pressure at carburetor entrance are 1.03 kgf/cm^2 and 27°C. The fuel orifice is at the same level as that of the float chamber. Take specific gravity of petrol as 0.74 and discharge coefficient as unity.

SOLUTION

Refer figure 6.5.

Given data, F:A = 1: 13.5, $d_2 = 3$ cm, pressure drop in venturi = 6.5 cm Hg, $p_2 = 0.9448$ kg/cm^2, $p_1 = 1.03$ kg/cm^2, $T_1 = 27°C = 300$ K, $\rho_f = 740$ kg/m^3, $C_d = C_{df} = 1$, $\gamma = 1.4$, $R = 29.27$ kgf-m/kg K.

Volume of air, $V_1 = R \cdot T_1/p_1 = 0.8525$ m^3/kg.

Volume at venturi, $V_2 = V_1 \times (p_1/p_2)^{1/\gamma} = 0.9067$ m^3/kg.

Temperature at venturi, $T_2 = T_1 \times (p_2/p_1)^{(\gamma-1)/\gamma} = 292.69$ K.

For air flow through choke tube, $C_2^2 = 2 \cdot R \cdot g \cdot (T_1 - T_2)$

$= 4197.97$ m^2/s^2.

So, flow velocity of air, $C_2 = 64.79$ m/s.

Mass flow rate of air, $m_a = A_2 \times C_2/V_2 = 0.05051$ kg/s.

Mass flow rate of fuel, $m_f = 0.00374$ kg/s.

Fuel flow velocity, $C_f = \sqrt{2.h.(p_1 - p_2)\rho_f}$ = 4.753 m/s.

So, Area of orifice, $A_f = m_f/(C_f \times C_{df} \times \rho_f)$ = 0.01064 cm^2

Diameter of orifice, $d_f = \sqrt{4} \times A_f / \pi$ = 1.16 mm.

EXAMPLE 6.3

A 100 mm × 120 mm four cylinder 4-stroke engine has carburetor venturi throat of 30 mm. if engine has a volumetric efficiency of 70% and is running at 2500 rpm, find venture depression. Assume density of air as 1.2 kg/m^3 and coefficient of air flow as 0.8.

SOLUTION

Refer figure 6.5.

Given data, bore, d = 100 mm, Stroke, l = 120 mm, number of cylinder, n = 4, venture throat d_2 = 30 mm, engine rpm, N = 2500, density of air, ρ_a = 1.2 kg/m^3, flow coefficient, C_d = 0.8.

Area of bore = (π/4) × 0.12 = 0.007854 m^2

Swept volume per cylinder = 0.007854 × 0.12 m^3

= 0.00094277 m^3.

Total stroke volume = 4 × 0.00094277 m^3 = 0.00377 m^3

Actual volume sucked by engine per stroke = 0.7 × 0.00377 m^3

= 0.02639 m^3.

Actual volume sucked per second = 0.002639 × 2500/(60 × 2)

= 0.05497 m^3/s.

Calculating mass flow rate of air = $m_a\ \rho_a = C_d \times A_2 \times \sqrt{(2g\rho_a\ \Delta p)}$ 0.05497 × 1.2 = 0.8 × (π/4) × 0.032 × 100 × $\sqrt{(2 \times 9.81 \times 1.2 \times \Delta p)}$.

Venturi depression, Δp = 0.05779 kg/cm^2.

EXAMPLE 6.4

The diameter of jet tip of a simple carburetor is 1 mm. The venture depression is 0.01 MPa and density of fuel is 770 kg/m^3. If mass discharge coefficient is 0.7, calculate weight of fuel discharged per second.

SOLUTION

Mass discharge of fuel = 0.7 × (π/4) × 0.0012 × $\sqrt{(2 \times 9.81 \times 770 \times 0.01 \times 106)}$ kg/s = 6.757 × 10^{-3} kg/s = 6.7574 kg/s.

EXAMPLE 6.5

A petrol engine consumes 7 kg of petrol (density 770 kg/m^3) per hour through a carburetor having venturi tube diameter at throat equal to 20 mm. Level of petrol in float chamber is maintained 5 mm below the jet tip. If air supply is maintained at 27°C and fuel air ratio is maintained as 1:15, find diameter of the single jet of the carburetor.

SOLUTION

Given air temperature = 27°C = 300 K.

Assume atmospheric pressure = 101 kPa

Gas constant for air = 29.27 kgf-m/kg.K

Density of air = p/RT = 101/(29.27 × 300) = 1.1725 kg/m^3

Air flow rate = petrol flow rate × air-fuel ratio = 7 × 15 /3600 = 0.029167 kg/s.

Assuming discharge coefficient as 1, flow of air through choke tube can be simulated.

Air flow rate = 1 × (π/4) × 0.022 × √(2 × 9.81 × 1.1725 × Δ*p* (in MPa) × 106) kg/s.

So, Venturi depression, Δ*p* (in MPa) = 3.7468 × 10^{-4} MPa.

Petrol is to be lifted up by 5 mm and venture depression at throat is reduced by this amount. 5 mm.

5 mm depression = 0.005 × 770 kg/m^2 = 3.85 kg/m^2 = 3.7756 × 10^{-5} MPa

Flow rate of petrol = 7/3600 = 0.0019444 kg/s

Assuming discharge coefficient as 1, flow of fuel through jet tip can be simulated.

Petrol flow rate = 1 × $A_{jet\ tip}$ × √(2 × 9.81 × 770 × (0.00037468 – 0.000037756) × 106) kg/s.

So, area of jet tip, $A_{jet\ tip}$ = 8.6185 × 10^{-7} m^2

Diameter of jet = √(4 × 8.6185 × 10^{-7}/π) m = 1.0475 × 10^{-3} m

= 1.047 mm.

EXAMPLE 6.6

A simple jet carburetor has petrol (density = 770 kg/m^3) consumption of 0.408 kg/min with fuel-air ratio by weight of 1:15. Air pressure and temperature are 101 kPa and 27°C. If velocity coefficient (C_v) for air flow is 0.85 and speed of air is 100 m/s, find throat diameter of choke tube. If pressure drop across fuel metering orifice is 0.8 times pressure drop in choke tube and coefficient of discharge is 0.7, find jet tip diameter. Take γ = 1.4, R = 29.27 kgf-m/kg K.

SOLUTION

When air is sucked in, velocity rises at the cost of pressure drop. From steady flow energy equation, gain in kinetic energy is equal to loss of enthalpy.

Initial temperature, $T_1 = 300$ K.

$$(V/C_v)^2/2g = C_p\, \Delta T = [\gamma R/ (\gamma - 1)] \times (T_1 - T_2).$$

Note that here C_v is velocity coefficient and not specific heat at constant volume.

So, $T_2 = 293.11$ K

Pressure at the choke = $101 \times (T_2/T_1)^{\gamma/(\gamma-1)} = 93.11$ kPa.

Density of air at the throat, $\rho_a = 93.11/(29.27 \times 293.11)$

$= 1.106$ kg/m^3.

Mass flow rate of air = 0.408×15 kg/min = 0.102 kg/s.

Area of choke tube at throat,

A choke tube at throat = $0.102/(1.106 \times 100) = 9.222 \times 10^{-4}$ m^2.

Diameter of the choke tube at throat = 3.426 cm.

Fuel flow rate,

$0.408/60 = 0.7 \times A_f \times \sqrt{(2 \times 770 \times (101 - 93.11) \times 103)}$

Diameter of jet tip = 1.88 mm.

SUMMARY

In this chapter, a brief idea about carburetion is explained. This chapter starts with assessment of fuel-air ratio for various operational as well as ideal requirements. Essentially, the concept of maintaining desired fuel-air ratio in a typical SI engine is elaborated in this unit. Design of a simple carburetor is illustrated and approach to overcome various specific deficiencies in simple system is discussed. At the end, method of calculation of venturi and orifice size is elaborated with a numerical example.

QUESTIONS

1. What is a carburetor?
2. Why is carburetor needed in SI engine?
3. Explain fuel-air ratio requirements in SI engine?
4. Explain principles of a simple carburetor?
5. What are essential components of a carburetor?
6. What arrangements are made in the carburetor to supply rich mixture during idling?
7. How carburetor accomplishes atomization, mixing and vaporization?

8. Write short notes on the following:

(*a*) Carburetor depression.

(*b*) Fuel metering.

(*c*) Choke tube.

(*d*) Float chamber.

(*e*) Throttle valve.

(*f*) Carburetor setting.

9. Give reasons for the following:

(*a*) Idling needs rich mixture.

(*b*) Choke helps in cold start of engines.

(*c*) Mixture becomes rich at high loads in normal carburetor.

(*d*) Fuel-air requirements for maximum power condition and maximum fuel efficiency conditions in simple SI engine do not coincide.

(*e*) Fuel level in float chamber is lower than discharge nozzle tip in venturi.

(*f*) Normal carburetor cannot be employed in aircrafts.

(*g*) Orifice size is made smaller.

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CHAPTER

7

Lubrication and Cooling Systems

STRUCTURE

7.1 INTRODUCTION

Lubrication and cooling systems are auxiliary but essential systems for engine operations and are normally called part of the basic engine. These systems also consume some power and so power delivered by a fully equipped engine is lower than power of a basic engine. This chapter gives idea about systems necessary for control of temperature generated during action of an engine. Lubrication systems are needed to reduce friction and thereby control temperature rise. Additionally, cooling system is required to take away heat from the engines to reduce chances of overheating different components. This chapter gives complete details about lubrication and cooling systems used in various internal combustion engines.

Objective

After studying this Chapter, you should be able to understand :

- Requirements of lubrication in engines.
- Different types of lubrication systems.
- Salient properties of lubricants and its additives.
- Engine cooling requirements.
- Method to achieve engine cooling.

7.2 TYPE OF LUBRICATION SYSTEMS

Engine friction is one of the prime concerns for performance of engines. It has dissipative effect and is directly associated with heat loss from the engine. It is represented by friction horse power, which is generally 10% of indicated horse power at full load and is 100% at ideal or no-load. This single parameter reduces maximum brake torque and minimum brake fuel consumption. Frictional losses are used as indicator of good and average engine design and are the critical parameter for an engine design and engine performance. It is not performance alone, but size of cooling system is also affected by consideration and effectiveness of friction management in the engine.

In an engine, pumping is an important exercise. Both intake and expulsion of working fluid needs pumping activity. This is essentially associated with pumping losses. This forms a single important factor for both SI and CI engine. In SI engines, it is significant and with rise in load, this reduces. However increase in speed has relatively less effect on frictional losses due to pumping. For CI engine, pumping account for less part in total frictional losses and the contribution enhances with rise in speed of the engine. Friction occurs at the interface of all moving parts including piston, cylinder, bearings, valves, gears etc. They form major part of total friction loss in both types of engines. Load does not affect it in case of SI engine, but in CI engines, it reduces with load. However speed of engine enhanced these interface frictional losses in both the types of engine. Frictional losses to drive auxiliary systems, like secondary air pump, power steering, air conditioning etc. They also needs power and are invariably associated with transmission losses. In fact overall frictional losses reduced if load on the engine is increased and speed enhances total frictional loss significantly. So frictional losses are present in the engine and lubrication system is needed for all types of engines for following operations.

- Reduce frictional resistance and maximize mechanical efficiency.
- Protect against engine wear.
- Contribute for engine cooling.

- Remove impurities in lubricated regions.
- Hold gas and liquid leakage to minimum levels.

Type of lubrication system depends on choice of lubricants and also on functional requirements. Lubricants should have low viscosity for good pumping and flow within rubbing parts. However, viscosity should be high at high temperature to get good lubrication and it must have lower volatility. Protection against wear needs anti-corrosive action, degradation-resistance, decomposition-control, dissolution of deposits and detergent action. Good thermal stability and oxidation resistance are other major properties of good lubricants.

7.2.1 Petrol Lubrication System

In 2-stroke SI engines, which are normally crank charged, lubricant is added in fuel oil itself. The quantity of lubricating oil is 2-3%. This type of lubrication is called mist lubrication system. Oil and fuel mixture is inducted in carburetor. Fuel vaporizes and oil in the form of mist (air entrapped) goes via crankcase into the cylinder. The oil lubricates main and connecting rod bearing when in crankcase. When oil reaches cylinder, it lubricates piston, piston ring and cylinder. Since lubricating oil is directly added to fuel oil, performance of engine is also affected by lubricating oil. Selection of lubricating oil becomes an important requirement and selection criteria must include exhaust smoke, internal corrosion, bearing life, ring and cylinder bore wear, ring sticking, deposits in exhaust and chamber, spark plug fouling. So, specially prepared ash less oil must be used as lubricant, if addition directly to petrol is applied. Fuel to lubricating oil ratio should be within 40:1 to 50:1 for good performance. Higher ratio increases rate of wear and lower results in spark plug fouling.

This system is very simple and it does not need any separate oil-pump. It has low cost. However, several problems may be encountered, which restrict usage of this system to smaller engines only. Some lubricating oil burns in cylinder resulting in heavy exhaust emission, formation of deposits and interference with normal and smooth engine operation. The anti-corrosion properties of lubricating oil are lost the moment it comes in contact with exhaust products which are acidic in nature.

Another problem with this system is lubrication is effective only when fuel supply is continuing. If throttle is closed (downward movement on a hill) lubrication stops. This may cause overheating and piston seizure due to oil starvation. As there is no need based control after addition of oil to fuel, most of the petrol engines are over-oiled.

7.2.2 Splash Lubrication System

This system is also useful for small engines. This is a wet sump lubrication system, where bottom part of the crank case, called sump is filled with lubricating oil. The level of oil is maintained in such a way that while on BC, the dripper of connecting rod strikes the oil. This splashes oil over engine parts like crank bearing, piston skirts and rings, piston pins etc. excess oil drips back to sump. This system also does not use any separate pump for lubricating oil. This system is not suitable at high bearing loads.

7.2.3 Semi-Pressure Lubrication System

For high bearing loads, modified splash wet sump lubricating system is used. In this case a separate oil pump is used for supply of lubricants. The main and camshaft bearings are lubricated by oil from the pump. However other engine parts are lubricated by splash system only.

7.2.4 Pressure Lubrication System

In pressure lubrication system, all engine parts are lubricated by lubricating oil supplied from an oil pump. Drilled passages are used to lubricate connecting rod bearings. The cylinder walls and piston rings are lubricated by the spray thrown from crankshaft and connecting rod. This is most practical and operational system used in automobiles, including Ambassador, Fiat, Jeep and Ashok Leyland vehicles.

7.2.5 Dry-sump Lubrication System

All the lubricating system described above belong to wet sump lubrication system, where crank case itself is used as lubricating oil holder. Contrary to this, in a dry sump uses a secondary external reservoir for oil. It is a lubricating motor oil management method for 4-stroke and large 2-stroke piston internal combustion engines, mainly. In a dry sump, the oil falls to the base of the engine after oiling relevant parts, but rather than being collected into an oil sump, it is pumped into another external reservoir by one or more scavenge pumps, run by belts from the front or back of the crankshaft. Oil is then pumped from this reservoir to the bearings of the engine by the pressure pump. Typical dry sump systems have the pressure pump and scavenger pumps "stacked up", so that one pulley at the front of the system can run as many pumps as desired, just by adding another to the back of the stack.

A dry sump offers many advantages, namely increased oil capacity and a lower center of gravity for the engine. Because the reservoir is external, the oil pan can be much smaller in a dry sump system, allowing the engine to be placed lower in the vehicle; in addition, the external reservoir can be as large as desired, whereas a larger oil pan raises the engine even further. Increased oil capacity by using a larger external reservoir leads to cooler oil. Furthermore, dry sump designs are not susceptible to the oil starvation problems wet sump systems suffer from if the oil sloshes in the oil pan, temporarily uncovering the oil pump pickup tube. Having the pumps external to the engine allows them to be maintained or replaced more easily, as well. Dry sumps are common on larger diesel engines such as those used for ship propulsion. Many racing cars, high performance sports cars, and aerobatic aircraft also utilize dry-sump equipped engines because they prevent oil-starvation at high 'g' loads, and because their lower center of gravity positively affects performance. Dry sump systems add cost and complexity, and the extra pumps and lines require more oil, so maintenance costs may rise accordingly.

7.3 PROPERTIES OF ENGINE LUBRICANTS

Engines use lubricating oil for lubrication, noise control, cleaning, cooling and sealing purposes. To work effectively, these fluid must possess certain salient property. These property requirements of lubricants are explained in this section. Viscosity is one of the important properties of lubricants. It is resistance to flow of a fluid. However, there are some contradictions in the properties requirements. Thin oils are good for cooling but bad for sealing, while reverse is true for thick oils. Oils must possess scavenging and cleaning capacity, but at the same time it should be cleaning to impart good lubrication. For lubricants viscosity index is assigned, which is an arbitrary figure compared at different temperatures. Viscosity index indicates change in viscosity under influence of temperature. If viscosity change with temperature is small, oil has high viscosity index. Lubricating oil should maintain sufficient viscosity at high temperatures and should not be too viscous at cold temperatures. So, lubricating oil should have high viscosity index. Parrafin base oil is assigned a value of zero, while naphthenic base oil is assigned viscosity index of 100. With this nomenclature, high viscosity index lubricating oil should have the value lying above 90. Sometimes, long chain paraffins are added to improve viscosity index of the oil. Oiliness is another property, which is similar in nature to viscosity, but has entirely different meaning. Viscosity may be treated as stickiness, but oiliness indicates properties of oil to cling to the metal surface by molecular action. This creates a thin oil film on the metal surfaces under boundary lubrication conditions. This is measured by coefficient of friction under extreme conditions of operations. This is important at high pressures and small clearances because it governs the squeezing out of oil.

Amongst physical parameters, specific gravity of lubricating oils is monitored only to control weight and volume requirement of lubricating systems. This property has no bearing on performance of lubricants. This generally lies between 0.85 and 0.96. Naphthenic base oils have higher specific gravity than paraffinic base oils.

Thermal stability of oil is another criterion for lubricating oil. Since oil experiences a very high range of temperatures, temperature based data of oils are also importants. Cloud point of oil is that temperature at which it starts solidifying. But more than cloud point, another temperature called pour point is more important for lubricants. Pour point is that temperature below which lubricants will not flow under given condition. Pour point is generally governed by wax content of the oil and wax at reduced temperatures result in honeycomb type of structure due to crystallization. Generally oils derived from paraffinic crude have higher pour point than those derived from naphthenic crude. Pout point can be lowered by adding pour point depressant. Popular pour point depressants are polymerized phenol, esters etc. In general pour point should be at least 8-10°C lower than operating temperature to ensure maximum circulation.

Another thermal property of lubricating oil is flash point. This is temperature at which oil vapour flash when exposed to naked flame. Flash point can be determined in a closed (sealed) or open container and accordingly prefixed in attached in the nomenclature. Contrary to this, fire point is that temperature at which oil, lit by a flame burns for at least 5 seconds steadily.

This varies from 190 to 290°C for lubricants. Fire and flash points are indicators of flammability of oil and are monitored for safety against fire hazards. They are very god indicators of crank case dilution also.

Under chemical properties of lubricating oils, carbon residue and oxidation stability are two important criteria. Carbon reside is quantity of the known mass sample of oil, which on evaporation under certain condition remains as carbonaceous residue. This is although a rough indicator of expected deposits from lubricants, but gives a fair idea about lubricants suitability for a purpose. Paraffin base oils have high carbon residue than asphaltic base oils. Oxidation stability is resistance of oil to oxidation at operating temperature and pressure conditions prevailing in the engine. On oxidation, oils may form deposits and loose lubrication capacity. To enhance this complex compounds of sulphur, phosphorus or amine and phenol derivatives are generally added to lubricants.

Low acid content is also an important requirement for lubricating oils. The neutralization number is a major of acidity or alkalinity of lubricating oils. New oil has low neutralization number. Neutralization number increases as oil is used in engine. Neutralization number is quantity of acid or alkaline solution needed to make the oil neutral.

In addition to this, absence of water and sediments is ensured in lubricating oil. Water, if present may promote corrosion and reduce capacity to lubricate. Transparent lubricating oil indicates good pure and fresh oil, while used oil should be thicker and grayish in colour. Colour has no role except as indicator for degree of use of oil.

7.4 PROPERTIES OF LUBRICANT ADDITIVES

Although simple mineral oil contains most of the essential characteristics needed from lubricating oil. However, to meet al the requirements, certain additives are always added to lubricants. These additives may give or affect one or more of the above-mentioned properties and sometimes add certain specific features to lubricating oils.

Viscosity index improvers reduce sensitivity of oil-viscosity to temperature and thereby make oil useful for a very wide of operating temperatures. Anti-wear additives reduce wear and prevent scoring galling and seizure and marks of other tribological interactions. Anti-rust additives prevent rusting by formation of thin film over exposed heated metallic surfaces. Pour point depressants interfere with crystallization of wax in lubricating oil and prevent both growth and agglomeration of wax crystals. This maintains flow properties of oil even at low temperatures.

To improve chemical properties, oxidation inhibitors are critical. Although they prevent or resist oxidation of lubricating oils but they are ineffective against control of carbonaceous residue and deposits. They cannot prevent sludge formation in engines. To take care of deposits and sludge, detergents and dispersants are used as additives in lubricating oils. Detergents are added to control high temperature deposits by keeping them in suspension and preventing their agglomeration. It can also act as effective acid neutralizer. Dispersant does same as detergent but they act on low temperature sludge and varnish deposits. In addition to these, anti-foam additives are used to reduce oil foaming. This is achieved by collapsing of bubbles formed due to air entrapment.

In addition to above-mentioned chemicals, corrosion preventives, rust preventives, metal deactivators, water repellants, colour, stabilizer, foam inhibitors, emulsifiers, dyes and odour control agents are also added on requirement basis to lubricating oils.

EXAMPLE 7.1

A diesel engine is used in a truck requiring 120 bhp. The mechanical efficiency of the engine is 80%. The brake specific fuel consumption of the engine is 200 gm per bhp-hr. A design improvement is made, which reduced engine friction by 5 hp. assuming the indicated thermal efficiency remains same, calculate (a) the new mechanical efficiency (b) the new bsfc and (c) the saving in fuel per hour.

SOLUTION

Indicated hp = bhp/mechanical efficiency = 150 hp.

Friction hp = ihp – bhp = 30 hp.

After improvement, new fhp = 30 hp – 5 hp = 25 hp

Since bhp is invariant, new ihp = bhp + new fhp = 145 hp. So, new mechanical efficiency = new bhp/new ihp

= 82.75 %. **Ans. (*a*)**

As indicated thermal efficiency remains constant, so isfc is same. This indicates that bsfc is inversely proportional to mechanical efficiency.

So, new bsfc = old bsfc × (ratio of mechanical efficiencies)

= 193.35 gm per bhp-hr. **Ans. (*b*)**

Saving in fuel = bhp × (difference of bsfc)

= 1330 gm/hr. **Ans. (*c*)**

7.5 TYPES OF COOLING SYSTEMS

In the internal combustion engines, very small amount of heat is only utilized or derived as useful power. A large part of energy is either taken away by exhaust gases or is lost to coolants. Figure 7.1 gives a brief balance-sheet of heat extracted from fuel combustion for various types of engines. For completeness, gas turbine engines are also included where there is no heat lost to coolants. However they have lower work output. Invariably, whether 2-stroke or 4-stroke, SI engine or CI engine, around one-third energy is taken away by coolants. The requirement of coolant for an internal combustion engine can be enumerated.

In previous section about lubricant, it is stated that lubricant should have high viscosity index to offset effects of temperature rise. Another method to reduce viscosity variation of lubricant is to keep the operating temperature on lower side by providing proper cooling arrangement so as to restrict temperature rise to maximum 150 to 200°C. From structural point of view, material of construction of engines looses their strength at high temperatures and most of the time temperature rise is governing criteria for assessing structural margin of safety in the engine. Lowering temperature in the cycle results in lower thermal stresses due to uneven expansion of various engine parts. Operational point of view also needs control over temperature. High temperatures result in very hot exhaust valves. This may act as local source of ignition and give rise to pre-ignition, surface ignition and sometimes even detonation. Maintaining cylinder head at higher temperature reduces volumetric efficiency by reduction of air density with rise in temperature. This affects power output from the engine adversely.

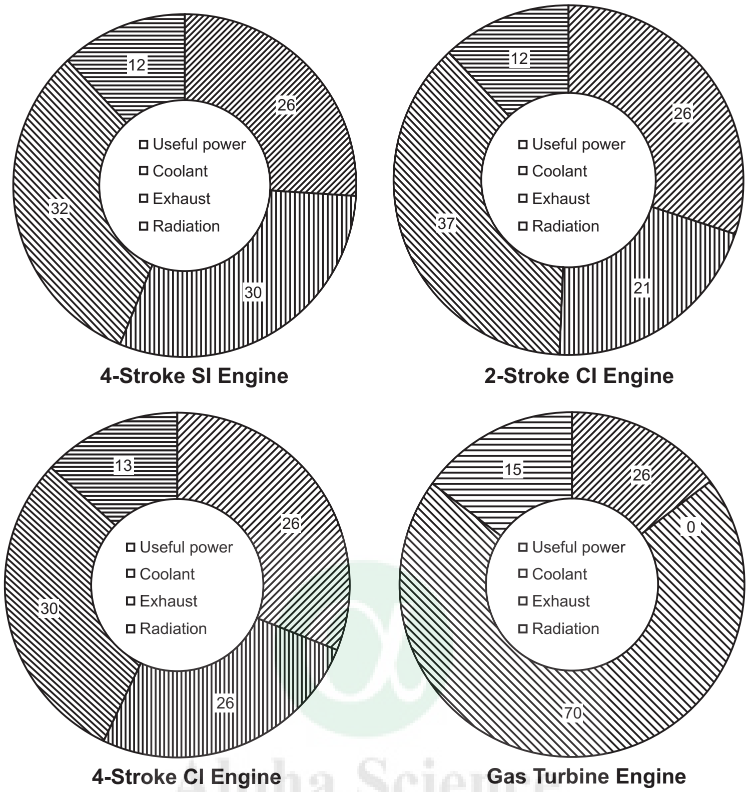

Fig. 7.1 : Heat Balance of Various Internal Combustion Engine

The cooling system is designed based on engine type, area of heat flow available, predominant mode of heat transfer, peak temperature and pressures in the cycle, heat transfer coefficients and combustion system, engine loads, operating conditions, material of construction of heat transfer surfaces, etc. Based on engine cooling provisions, internal combustion engines are classified under two heads namely air cooled engines and water cooled engines. Air is omnipresent and ultimate heat sink. However it has lower thermal conductivity. Water cooling system needs separate water storage, pumping, circulation and refurbishment mechanism and are generally bulky in nature.

7.5.1 Air Cooling

Air cooling is good for small engines. In additional to this for aircraft engines, where weight reduction of engines is major requirements, air cooling is employed. In some industrial or agricultural engines, uses of water are objectionable due to corrosion or solvent related problems. In those cases also air cooling is preferred. For air cooling heat transfer area is increased by arrangement of fins and air is passed over these fins for cooling. Since heat is directly transferred to air, very large temperature gradient exists and surface requirement can be reduced.

Stationary air has lower overall heat transfer coefficient. To effectively execute heat transfer, air is given velocity for distribution around the cylinder and coverage of entire finned area. As far as fins are concerned, their design is important for effective heat transfer and cooling. Fin shape, spacing, projection, orientation and width are decided as a compromise between several competing factors.

When inter-fin spacing is increased, fins offer larger volume for cooling air flow with reduced surface area available for heat transfer. This leads to higher requirements of cooling air. If interspacing is reduced, reverse action takes places, air between fins gets heated up more. This needs higher flow velocity of air through fins and leads to higher pressure drop for flow. If distance between two fins is kept very low, boundary layers of adjoining fins overlap and efficiency of fins reduces. So, fin separation is never kept below 2.5 mm. height of fins is governed by space available between two engines. Usually fin height is kept 15 to 25 mm.

Parts which are more stressed thermally, like exhaust valve, exhaust manifold must be adequately finned. Air cooled results in higher engine temperature and engine parts with air cooling must have higher clearance. Since temperature of engine is higher, strength of parts reduces and clamping must be placed to avoid distortion. Higher specific outputs are possible if material of construction is light alloy.

7.5.2 Pressure Cooling

Engine cylinder and cylinder head is submerged as enclosed by water jacket in water cooled engines. The water jacket is connected to radiator, where major mode of heat transfer is convection. Water flows through the jacket and cools the engine. Water gets heated up in this process and it releases heat to air in the radiator and gets re-circulated. The most critical control parameter is local velocity of water and bulk rate of water passed through the jacket. Generally 3 to 4 m/s local velocity is used for reasonable heat transfer. For better heat transfer, surface in contact with water is properly machined. Drilled coolant passages are also incorporated to ensure cooling of highly stressed parts.

For facilitating in cold starting, anti-freeze solutions are added to water. Anti-freeze agents are kerosene, wood alcohol, denatured alcohol, glycerin, sugar solution, calcium or magnesium chloride, ethylene glycol and propylene glycol. Many methods of water cooling arrangements are possible. Thermo-siphon cooling is one such water cooling system. Water becomes lighter on heating is the basic principle used for thermo-siphon cooling arrangement. The top of the radiator is connected to top of the water jacket and bottom to the bottom of the radiator. Air, while passing through radiator cools the water and sets water in motion in downward direction in the radiator. The flow of air is due to vehicle motion or a fan can be used for this purpose. However this system depends on temperature rise and is independent of the engine speed. The rate of circulation is slow and insufficient. Circulation of water is effective only when engine is hot.

Forced or pump circulation is a modification to thermo-siphon cooling, where water is circulated by a pump. This ensures positive circulation and power is derived from engine for running the pump. It works on all conditions and is independent of temperature. However, this system may result in overcooling. This system has one major limitation. While moving uphill, more fuel is burned, and cooling requirement increases, but pump circulation leads to reduced coolant circulation. This leads to engine overheating. Since power is derived from engine, as soon as engine stops, cooling also stops. This is also undesirable because temperature of engine parts has to be brought to normal values.

If cylinder barrel temperature is low, sever corrosion damage due to condensation of acids on barrel wall may result. To stop coolant flow, after achievement of pre-set temperature, thermostat cooling may be used. This is just a control device and can be applied in both the above class of cooling arrangements. However, there is no method to control overheated engine, as extra cooling cannot be arranged on will. Another method of cooling arrangement uses pressurized water cooling. At normal atmospheric pressure, boiling point is 100°C, but as pressure rises, boiling point of water also raises. At pressure levels of 2 kg/cm^2, 5 kg/cm^2, 10 kg/cm^2, boiling point of water are 121°C, 153°C and 180°C respectively. Use of pressurized water gives larger heat transfer coefficient and it can have a better cooling. Pressurized water needs an additional valve called vacuum valve to avoid formation of vacuum, when water is cooled after stoppage of engine. A safety relieve valve is also placed to release pressure. Water pressurized to around 2 kg/cm^2 so that water remains liquid even above 100°C to absorb heat from engine cylinder.

7.5.3 Steam Cooling

Steam cooling is one of the latest cooling arrangements for engine cylinder. This employs steam as cooling agent and is also called vapour cooling or evaporative cooling. It is well known fact that latent heat of vaporization is higher than sensible heat. This indirectly conveys that with less quantity of water, latent heat can give same heat transfer effectiveness. Water is stored in radiator and heated to 100°C to form steam. In fact in radiator, both water and steam remains in equilibrium in the radiator. Coolant always remains in liquid phase and is pumped to water jacket around the cylinder block. From heat of engine cylinder, water flashes to steam absorbing large quantity of heat from engine cylinder. Steam is returned back to radiator, where it condenses and becomes make-up water for cooling.

This system of cooling is used in many industrial engines. Schematic of steam cooling is shown in figure 7.2.

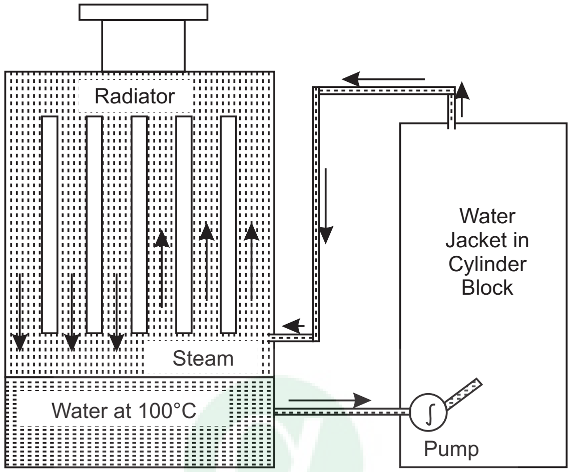

Fig. 7.2 : Steam Cooling System Schematic

SUMMARY

This chapter is meant to give idea about auxiliary but important system of international combustion engines. Both lubrication and cooling systems take care of engine temperature and reduces thermal stresses in engine parts. Lubricants have several additives for specific function and it must be effective to realize cleaning, noise reduction, corrosion prevention also. Many systems are prevalent for effectively carrying out these operations. Cooling of engine components is another prime requirement for trouble-free operation of an engine. Although only two media air and water are used for cooling, but several systems are designed for cooling of engine cylinder heads and heated parts.

QUESTIONS

1. What is main purpose of using lubricants in internal combustion engines?
2. What is lubrication methods followed in petrol engines?

   What are advantages and limitations of the petrol lubrication system?
3. What is splash lubrication system? Whether it needs separate pump for lubrication?
4. What is difference between pressure and semi-pressure lubrication system?

5. What is dry-sump lubrication? What are its advantages?
6. What are salient properties needed by engine lubricants?
7. What are additives in lubricant? What are purposes of using additives?
8. Why are cooling systems needed for engines?
9. How does air cooling helps in engines?
10. What is pressure cooling? Can pressurization helps in increasing heat transfer?
11. What is steam cooling? How it operates?
12. How does lubrication and cooling systems help in improvement of performance of an engine?

Further Reading

- Internal Combustion engine Fundamentals by John B. Heywood, McGraw Hill Book Company.
- Internal Combustion Engine by H.B. Keshwani, Standard Book House.
- Internal Combustion Engine by V.L. Maleev, International Students Edition.
- A Course in Internal Combustion Engines by M.L. Mathur and R.P. Sharma, Dhanpat Rai and Sons.
- Understanding Aerospace Propulsion by H.S. Mukunda, Interline Publishing.
- Heat Transfer by J.P. Holman and P.R.S. White, McGraw Hill Book Company.

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Appendix

SAMPLE QUESTIONS—FIRST-MID SEMESTER

Duration : 90 minutes ***Full marks : 35***

Answer any 5 questions

All questions carry equal marks.

Use of calculator and steam table is permitted.

Assume data suitably, if data given in problem is insufficient.

1. Explain diesel cycle and derive expression for its thermal efficiency.

2. What is mean effective pressure for a cycle? Derive expression for mean effective pressure of an Otto cycle.

3. A dual combustion cycle has an adiabatic compression volume ratio of 15:1. The conditions at the commencement of compression are 1 kg/cm^2, 25°C and 0.15 m^3. The maximum pressure of the cycle is 60 kg/cm^2 and the maximum temperature of the cycle is 1500°C. If $\gamma = 1.4$, calculate the pressure, volume and temperature at the five corners of the cycle.

4. Write short notes on any 3 topics.

(*a*) Specific fuel consumption

(*b*) Supercharging

(*c*) Detonation in Petrol engines

(*d*) Two stroke compression ignition engine

5. What are different parts of an internal combustion engine?

What are their use and common material of construction?

6. Give reasons for any 2 of the following:

(*a*) For a given compression ratio and heat rejection, Otto cycle is more efficient than diesel cycle.

(*b*) Brake thermal efficiency is lower than indicated thermal efficiency of an engine.

(*c*) Detonation is undesirable in a petrol engine but desirable in diesel engine.

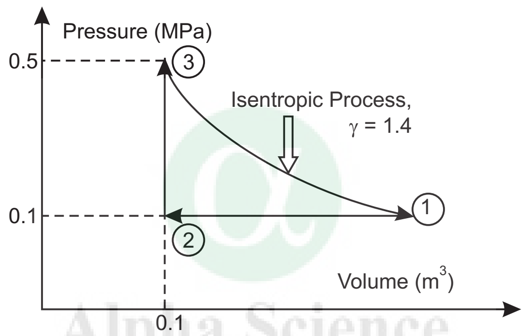
7. Explain different components of a turbojet engine. Demonstrate the processes involved in turbojet propulsion on a thermodynamic plane?

SAMPLE QUESTIONS—FIRST-MID SEMESTER

Duration : 90 minutes ***Full marks : 35***

Answer any 5 questions
All questions carry equal marks.
Use of calculator and steam table is permitted.
Assume data suitably, if data given in problem is insufficient.

1. A cycle consisting of three processes namely isobaric compression (1-2), isochoric heat addition (2-3) and isentropic expansion (3-1) is depicted in the figure beside on a pressure-volume thermodynamic plane. Find work done (in J or kJ) for the cycle 1-2-3-1 depicted in the figure.



2. Bore and stroke of a diesel engine is 10 cm and 15 cm respectively. If clearance volume and cut-off ratio are 0.0001 m^3 and 1.3 respectively, find air standard efficiency of the cycle. Take ratio of specific heats as 1.4.

3. Air flow through a turbojet engine is 40 kg/s and it propels aircraft at a velocity of 250 m/s. If isentropic enthalpy drop in the nozzle is 250 kJ/kg, find (*i*) thrust produced, (*ii*) thrust power, (*iii*) Propulsive power and (*iv*) Propulsive efficiency.

4. Explain working principles of an SI engine using valve timing diagram.

5. Write short notes on any three of the following:
(*a*) Ramjet engine
(*b*) Cut-off ratio
(*c*) Friction power
(*d*) Knocking

6. Give reasons for any three of the following:
(*a*) Pressure and volume are called conjugate properties.
(*b*) Slope of isobaric line on temperature-entropy plane is lower than that for isochoric line.

(*c*) Reciprocating engines are unsuitable for aircraft propulsion.

(*d*) Spark plug fires before piston reaches TDC in SI engine.

7. An engine consumes 5 grams of fuel (calorific value = 45 MJ/kg) per second and delivers 80kW power with a mechanical efficiency of 80%. Find (*i*) Brake specific fuel consumption, (*ii*) Indicated specific fuel consumption, (*iii*) Brake thermal efficiency and (*iv*) Indicated thermal efficiency.

SAMPLE QUESTIONS—SECOND-MID SEMESTER

Duration : 90 minutes ***Full marks : 35***

All questions carry equal marks.
Use of calculator is permitted.
Attempt maximum five questions.
Assume data suitably.

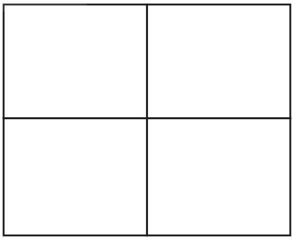
1. Give reasons for any 2 of the following:

(*a*) Thermal conductivity of gases increase with increase in temperature.

(*b*) Choke helps in cold start of engines.

(*c*) Water cooling is not employed in two-wheelers.

2. For 2-D heat transfer through the grid shown beside, nodes on upper boundary are maintained at 100°C, while nodes on left and right boundaries are maintained at a temperature of 200°C. If fluid at temperature 500°C is flowing through bottom boundary and ($h.\Delta x/k$) for the grid is 2, find temperature at all the nodes of the grid.



3. What is Fourier law of heat conduction? Write down 3-D equation of heat conduction in Cartesian coordinates and enlist condition for the derivation of Laplace equation from this. Differentiate between conduction and radiation.

4. The dimension of jet tip of a simple carburetor is 1 mm. The venturi depression is 0.01 MPa and density of fuel is 770 kg/m^3. If mass discharge coefficient is 0.7, calculate weight of fuel discharge per second.

5. What is purpose of fins on engine cylinder? How it improves cooling of the engine?

6. What are requirements of engine lubricants? How they are met by additives?
7. What is pressure cooling? How it helps in cooling of engine?

SAMPLE QUESTIONS – ASSIGNMENT

1. Compare Otto Cycle and Diesel cycle. Comment on their air standard efficiency for identical (*i*) compression ratio and heat input (*ii*) heat input and maximum pressure (*iii*) maximum pressure and work output
2. For a dual cycle, air is taken in at 101 kPa and 25°C. Compression ratio is 16 and maximum pressure in the cycle is 6000 kPa. The heat transfer to air at constant pressure and constant volume are the same. If C_p = 1.008 kJ/ Kg-K, γ = 1.4. Find (*i*) pressure and temperature at all the corners of the cycle (*ii*) cut-off ratio and pressure ratio (*iii*) air standard efficiency (*iv*) work output (*v*) swept volume (*vi*) mean effective pressure.
3. Why is jet propulsion employed as aircraft power plant? Explain working of a turbojet engine.
4. Differentiate between 4-stroke and 2-stroke engines used in automobiles.
5. What are effects of altitude and speed on the performance of aircraft SI engine? How adverse effects are neutralized in aircraft SI engine?
6. What is purpose of fins in engine heads? Derive expression for fin efficiency.
7. Explain Plank's law of radiation. Derive Wien's displacement law from Plank's law of radiation
8. Explain purpose of lubrication in engines. What are various types of engine lubrication methods?
9. Explain a simple carburetor. Derive expression for orifice and venture sizes for the same.
10. What are various cooling systems employed in SI engines?

SAMPLE QUESTIONS – ASSIGNMENT

1. Derive expression for air standard efficiency of standard Diesel cycle. Define compression ratio and cut-off ratio. Take ratio of specific heat as 1.3 to numerically show the effects of compression ratio and cut-off ratio on air standard efficiency of Diesel cycle.
2. A turbojet engine has a compressor, combustor, turbine and nozzle. Compressor, turbine and nozzle have an efficiency of 95%. If overall

pressure ratio is 4, mechanical efficiency is 98%, find (*i*) Effective jet velocity (*ii*) Thrust and specific thrust. Mass flow rate of air = 20 kg/s, Maximum temperature = 1000 K, Inlet temperature = 300 K, Inlet pressure = 101 kPa, $\gamma = 1.4$, $C_p = 1.005$ kJ/kg/K.

3. Explain working of a 2-stroke engine with sketches. Compare it with 4-stroke engine.

4. Explain abnormal combustion processes in an internal combustion engine.

5. What are uses of Superchargers? Why they are essential in Aircrafts? What are various means to get supercharging?

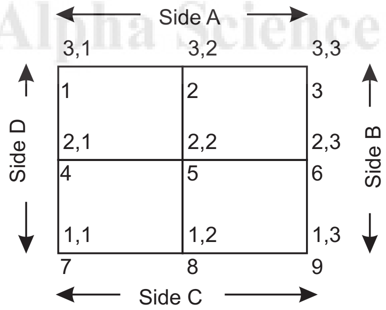
6. Explain various methods for measuring friction horsepower of an engine. Enlist merits and demerits of methods. Give suitable numerical examples.

7. What is fuel-air ratio requirements in an SI engine at different loads and speeds? How these needs are accomplished in modern carburetors?

8. What is purpose of lubricants in IC engine? What are expectations from lubricants and how they are accomplished?

9. Derive expressions for fin efficiency for various boundary conditions?

10. For the grid shown beside, side A and side B are maintained at 100°C and other sides are exposed to temperature of 500°C. If $(h\Delta x/k)$ for the grid is 1, find temperature at all the nodes of the shown grid.



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