Thermal Engineering-I

About the Author

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Dr. Rathore also recently qualified NPTEL Online Certification Examinations in Laws of Thermodynamics and Refrigeration and Air Conditioning with Elite Score. He is a life member of the Indian Society for Technical Education, India, ISHMT (Indian Society for Heat and Mass Transfer), ISHRAE (Indian Society of Heating, Refrigerating and Air Conditioning Engineers) and the Institution of Engineers (India). He is also a faculty of PRINCE (Promoters and Researchers in Non-Conventional Energy) Dhule, Maharashtra.

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Dedicated to my Beloved Parents

Contents

Preface

It gives me immense pleasure to present you this book on 'Thermal Engineering-I'. This text is intended for undergraduate students of Mechanical Engineering of JNTU. It integrates Thermodynamics and Applied Thermodynamics taught in the fifth semester to B. Tech. students of Jawaharlal Nehru Technological University.

Aim

During my teaching span of more than three decades, I felt that the subjects based on thermal engineering are often perceived as difficult by students. I observed that customarily, major problems are faced by students in understanding the text and illustrations. They need a text written in a simple and interesting way which exposes the subject systematically along with a variety of illustrative examples supporting the theoretical concepts.

Through this book, I am making an attempt to overcome the problems of students as well as to impart sound knowledge. The presentation is very simple, lucid and easy to understand. The topics are explained right from the fundamentals with the help of illustrative figures, enabling even a beginner to understand the subject very easily. Solutions for the problems are also explained with the help of illustrative figures so that the logic behind them can easily be understood.

This book discusses the basic concepts first and then supports the theory with applications and solved numerical problems. This approach will help the students in developing an analytical mind. An engineer with an analytical mind and approach would be able to face any problems encountered in the actual engineering field. Moreover, it is my earnest hope that this book will provide a unique combination of features that will make it inviting and effective for both faculty and students.

Salient Features

The salient features of the book are as follows:

- ✓ Complete coverage of courses on Thermal Engineering–I of JNTU
- ✓ Tutorial approach of problem solving
- ✓ Solved examples based on questions from numerous universities all over India as well as competitive examinations like GATE, IES, etc.
- ✓ Diverse and useful pedagogical features like Summary, Glossary, Review Questions, and Problems
- ✓ Well-labelled schematic diagrams supporting theoretical and mathematical explanations.

Organisation

Chapter 1 provides an overview of the basic concepts of internal combustion engines. Concepts are an essential part of any science and my experience has shown that this is an area, where students find difficulty. Several illustrations are put to support the definitions and concepts to help the students.

Chapter 2 gives a detailed information of fuel energy and various fuels used in IC engines. Fuel feeding systems for spark ignition engines and diesel engines are also discussed in this chapter.

Chapter 3 deals with combustion and combustion chambers used for SI and CI engines. Direct and indirect fuel injection combustion chambers with their relative merits and demerits are explained with the support of swirl and squish design.

Chapter 4 deals with testing and performance parameter of internal combustion engines. This chapter covers the variety of numerical problems possible. Notion of heat balance of an internal combustion engine is explained precisely.

Reciprocating air compressor is discussed in *Chapter* 5. This chapter presents importance of compressed air, its volumetric efficiency and minimization of power input.

Chapter 6 deals with thermodynamics of rotary compressors. Isentropic efficiency, Polytropic efficiency, surging, choking in connection with rotary compressors are discussed in this chapter.

In *Chapter* 7 refrigeration cycles and in *Chapter* 8 psychrometry are carefully considered. These two chapters deal with various types of refrigeration and air conditioning processes used for food preservation and human comfort.

Acknowledgements

On completion of this version of the book *Thermal Engineering- I*, I am indebted to all authors who shaped my thoughts on the subject and whose work has been freely consulted in preparation of this text. I also owe an enormous debt of gratitude to my colleagues and students who helped me directly or indirectly in preparation of this treatise.

I take the opportunity to express my sincere thanks to the McGraw Hill Education editorial team: Ms Vaishali Thapliyal whose repeated persuasion made it possible to prepare the text for JNTU under the banner McGraw Hill Education. She has provided required support from time to time. I am pleased to acknowledge the contribution of Mr. P L Pandita and the whole production staff, who put their consistent efforts for making the text as best as it could be within a very short span of time.

I would like to extend my gratitude to administration and Executive Management of SNJB's K B J College of Engineering, Chandwad, (Nashik) who extended all the facilities and full cooperation during the preparation of this manuscript.

I would also like to thank Ch.V.K.N.S.N. Moorthy, Institute of Aeronautical Engineering, Hyderabad for his valuable feedback on draft chapters of this book.

I wish to place on record my earnest gratitude to my parents, my caring wife Meera; sons; Dr Ankit and Prateek, and above all to my sweet daughter-in-law Dr. Pallavi for their wholehearted support and patience, and taking care of my health which helped me indirectly in completing this project.

A human creation can never be perfect. Some mistakes might have crept in the text. My efforts in writing this book will be rewarded, if readers send their constructive suggestions and objective criticism with a view to improve the usefulness of the book. For any suggestion, query or difficulty, you are most welcome to write to me at mmrathore@gmail.com.

Mahesh M. Rathore

Roadmap to the Syllabus

JNTU Hyderabad

Unit-I

I.C. Engines: Classification – Working principles of Four & Two stroke engine, SI & CI engines, Valve and Port Timing Diagrams, Air – Standard, air-fuel and actual cycles – Engine systems – Carburetor and Fuel Injection Systems for SI engines, Fuel injection systems for CI engines, Ignition, Cooling and Lubrication system, Fuel properties and Combustion Stoichiometry.

GO TO

Chapter 1 – **Internal Combustion Engines**

Unit-II

Normal Combustion and Abnormal Combustion in SI Engines – Importance of flame speed and effect of engine variables – Abnormal combustion, pre-ignition and knocking in SI Engines – Fuel requirements and fuel rating, anti knock additives – combustion chamber – requirements, types of SI engines.
Four stages of combustion in CI engines – Delay period and its importance – Effect of engine variables – Diesel Knock – Need for air movement, suction, compression and combustion induced turbulence in Diesel engine – open and divided combustion chambers and fuel injection – Diesel fuel requirements and fuel rating.

GO TO

Chapter 2 – **Fuel and Engine Systems**
Chapter 3 – **Combustion and Combustion Chambers**

Unit-III

Testing and Performance: Parameters of performance – measurement of cylinder pressure, fuel consumption, air intake, exhaust gas composition, Brake power – Determination of frictional losses and indicated power – Performance test – Heat balance sheet and chart Classification of compressors – Fans, blowers and compressors – positive displacement and dynamic types – reciprocating and rotary types.

Reciprocating Compressors: Principle of operation, work required, Isothermal efficiency volumetric efficiency and effect of clearance volume, staged compression, under cooling, saving of work, minimum work condition for staged compression.

GO TO

***Chapter 4* – Testing and Performance of IC Engines**
***Chapter 5* – Reciprocating Air Compressor**

Unit-IV

Rotary Compressor (Positive displacement type): Roots Blower, vane sealed compressor, Lysholm compressor – mechanical details and principle of working – efficiency considerations.

Dynamic Compressors: Centrifugal compressors: Mechanical details and principle of operation – velocity and pressure variation. Energy transfer-impeller blade shape-losses, slip factor, power input factor, pressure coefficient and adiabatic coefficient – velocity diagrams – power.

Axial Flow Compressors: Mechanical details and principle of operation – velocity triangles and energy transfer per stage degree of reaction, work done factor – isentropic efficiency-pressure rise calculations – Polytropic efficiency.

GO TO

***Chapter 6* – Rotary Compressor**

Unit-V

Refrigeration: Mechanical Refrigeration and types – units of refrigeration – Air Refrigeration system, details and principle of operation – applications of air refrigeration, Vapour compression refrigeration systems – calculation of COP – effect of superheating and sub cooling, desired properties of refrigerants and common refrigerants – Vapour absorption system – mechanical details – working principle, Use of p-h charts for calculations.
Air-Conditioning: Concepts of Psychrometry – Properties of moist air – Usage of Psychrometric Chart – Calculation of moist air properties.
Types of air – conditioning systems – Requirements – schematic layout of a typical plant.

GO TO

***Chapter 7* – Refrigeration**
***Chapter 8* – Psychrometry**

1 CHAPTER

Internal Combustion Engines

Introduction

An internal combustion engine is a machine that converts chemical energy in a fuel into mechanical energy. Fuel is burnt in a combustion chamber, releases its chemical energy in the form of heat, which is converted into mechanical energy with the help of a reciprocating piston and crank mechanism. Two principal types of reciprocating internal combustion engines are in general use: the Otto-cycle engine and the Diesel engine. The Otto-cycle engine, named after its inventor, the German technician Nikolaus August Otto, is known as gasoline engine used in automobiles and airplanes. The Diesel engine, named after the French-born German engineer Rudolf Christian Karl Diesel, operates on a different principle and usually uses oil as a fuel. It is employed in electric-generation and marine-power plants, in trucks and buses, and in other automobiles. Both Otto-cycle and Diesel-cycle engines are manufactured in two-stroke and four-stroke cycle models.

1.1 DEFINITIONS

Figures 1.1 and 1.2 show sketches of an internal combustion engine, a piston that reciprocates within a cylinder fitted with two valves and a spark plug or fuel injector. The sketches are lebelled with some special terms, defined below:

1. Bore It is the internal diameter of the cylinder of the reciprocating engine. It is denoted by d and expressed in mm.

2. Stroke It is the linear distance through which the piston travels between the top dead centre (*TDC*) and the bottom dead centre (*BDC*) in the cylinder. It is also called *stroke length*. It is denoted by L and expressed in mm.

3. Swept Volume It is also called *piston displacement volume* or *stroke volume.* It is the volume created or displaced by the piston during its one stroke travel, i.e., travel of the piston from one dead centre to other dead centre. It is denoted by V_s and calculated for one cycle as

$$V_s = (\pi/4)\, d^2 L \qquad ...(1.1)$$

4. Cubic Capacity or Engine Capacity It is the product of swept volume of a cylinder and the number of cylinders in an engine, i.e., total volume displaced in each cycle of an engine. For k number of cylinders,

$$Cubic\ Capacity = k\, V_s$$

5. Clearance Volume It is the volume left in the cylinder when the piston reaches the top dead centre. It is denoted by V_c.

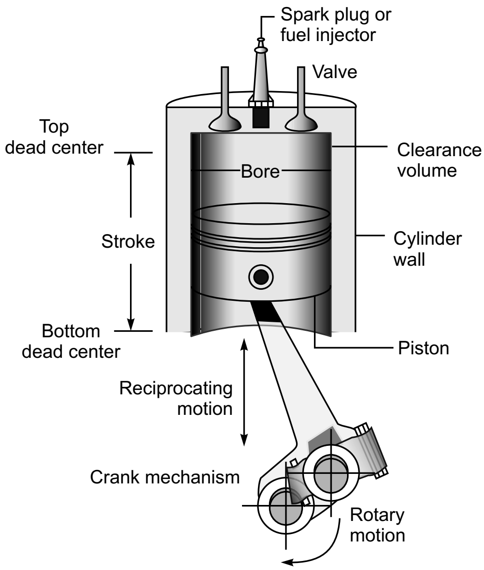
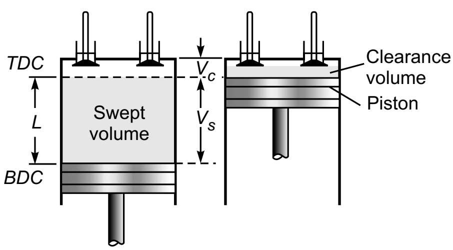


Fig. 1.1 *Nomenclature for reciprocating volume engine*



(a) Swept volume *(b) Clearance volume*

Fig. 1.2 *Swept volume and clearance of a reciprocating engine*

6. Total Volume It is the maximum volume in the cylinder and is the sum of swept volume and clearance volume. It is denoted by V_1.

$$V_1 = V_s + V_c \quad ...(1.2)$$

7. Compression Ratio It is the ratio of maximum possible volume to clearance (minimum) volume in the cylinder. It is denoted by r and expressed as

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{V_s + V_c}{V_c} \quad ...(1.3)$$

8. Clearance Ratio It is the ratio of clearance volume to swept volume. It is usually kept 3–5% of the swept volume. It is denoted by c and expressed as

$$c = \frac{V_c}{V_s} \quad ...(1.4)$$

9. Cut-off Ratio It is the ratio of cut-off volume to swept volume. It is denoted by ρ. This term is used in Diesel-cycle and dual-cycle-engines.

10. Expansion Ratio It is the ratio of maximum volume to minimum volume during expansion in the cylinder. It is denoted by r_e.

11. Dead Centres There are two fixed positions in the cylinder, between which the piston reciprocates. The uppermost position is called *top dead centre* (*TDC*), while the bottom-most position is called the *bottom dead centre* (*BDC*).

12. Charge It is air-petrol mixture for petrol engines and only air for Diesel engines, which is introduced during suction stroke of the engine.

13. Mean Effective Pressure (p_m) It is a *hypothetical average pressure*, which if acted on the piston during the entire power stroke, will produce the same power output as produced during the actual cycle.

Net work done in a cycle,

$$\begin{aligned} W_{net} &= p_m \times (\text{piston area}) \times (\text{stroke}) \\ &= p_m \times (\text{displacement volume}) \\ &= p_m \times V_s \end{aligned}$$

Thus $$p_m = \frac{W_{net}}{V_s} \quad ...(1.5)$$

The actual indicator diagram of an engine and its corresponding mean effective pressure for the same power output is shown in Fig. 1.3. For any particular engine, under specific operating conditions, there will be an indicated mean effective pressure (*imep* or p_{mi}) and a corresponding brake mean effective pressure (*bmep* or p_{mb}). From a given indicator diagram, the indicated mean effective pressure can be obtained as

$$p_{mi} = \frac{\text{Area of indicator diagram (mm}^2)}{\text{Length of the indicator diagram (mm)}} \times \text{Spring constant (kPa/mm)} \quad ...(1.6)$$

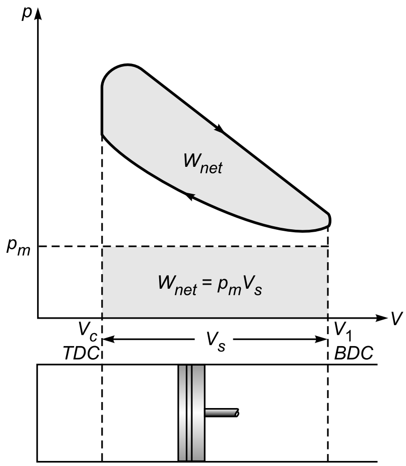


Fig. 1.3 *The network of a cycle is a product of the mean effective pressure and swept volume*

The mean effective pressure is used to compare the output of similar engines of different sizes.

1.2 CLASSIFICATION OF IC ENGINES

The internal combustion engines are usually of reciprocating type. The reciprocating internal combustion engines are classified on the basis of the thermodynamic cycle and mechanical method of operation, type of fuel used, type of ignition, type of cooling system and cylinder arrangement, etc. The detailed classification is given below:

1. *According to piston strokes in the working cycle*
   (a) Four-stroke engine
   (b) Two-stroke engine
2. *According to fuel used in the cycle*
   (a) Petrol engine (b) Diesel engine
   (c) Gas engine (d) Multi–fuel engine
3. *According to method of ignition*
   (a) Spark ignition
   (b) Compression ignition
4. *According to fuel-feeding system*
   (a) Carburetted engine
   (b) Engine with fuel injection
5. *According to charge feeding system*
   (a) Naturally aspirated engine
   (b) Supercharged engine
6. *According to cooling system*
   (a) Air-cooled engine
   (b) Water-cooled engine
7. *According to number of cylinders*
   (a) Single cylinder engine
   (b) Multi-cylinder engine
8. *According to speed of the engine*
   (a) Low-speed engine
   (b) Medium-speed engine
   (c) High-speed engine
9. *According to position of engine*
   (a) Horizontal engine
   (b) Vertical engine
   (c) V-engine

The petrol engines use low compression ratio. The fuel and air mixture as a charge is ignited by a high intensity spark. Therefore, they are also called *spark ignition (SI) engines.* The Diesel engines use high compression ratio, and the compressed charge is autoignited. Therefore, they are also called *compression ignition (CI) engines.*

1.3 COMPONENTS OF ENGINES

The essential parts of Otto-cycle and Diesel-cycle engines are the same. Actually an internal combustion engine consists of a large number of parts and each part has its own function. A few of them are shown in Fig. 1.4 and listed below:

1. Cylinder It is the heart of the engine. The piston reciprocates in the cylinder. It has to withstand high pressure and temperature, and thus it is made strong. Generally, it is made from cast iron. It is provided with a cylinder liner on the inner side and a cooling arrangement on its outer side. For two-stroke engines, it houses exhaust and transfer port.

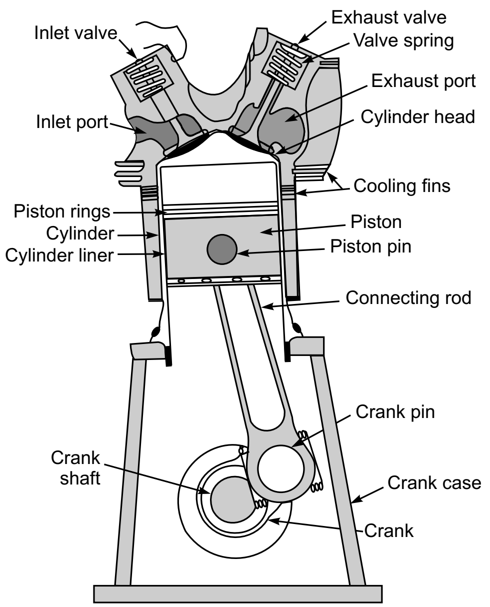


Fig. 1.4 *Details of an internal combustion engine*

2. Cylinder Head The top cover of the cylinder, towards *TDC*, is called *cylinder head*. It houses the spark plug in petrol engines and fuel injector in Diesel engines. For four stroke cycle engines, the cylinder head has the housing of inlet and exhaust valves.

3. Piston It is the reciprocating member of the engine. It reciprocates in the cylinder. It is usually made of cast iron or aluminium alloys. Its top surface is called *piston crown* and bottom surface is called *piston skirt*. Its top surface is made flat for four-stroke engines and deflected for two-stroke engines.

4. Piston Rings Two or three piston rings are provided on the piston. The piston rings seal the space between the cylinder liner and piston in order to prevent leakage (blow by losses) of high-pressure gases, from cylinder to crank case.

5. Crank It is a rotating member. It converts reciprocating motion of connecting rod to rotary motion of crankshaft. Its one end is connected with a shaft called *crank-shaft* and the other end is connected with a connecting rod.

6. Crank Case It is the housing of the crank and body of the engine to which cylinder and other engine parts are fastened. It also acts as a ground for lubricating oil.

7. Connecting Rod It is a link between the piston and crank. It connects its one end with a crank and on the other end with a piston. It transmits power developed on the piston to a crank shaft through crank. It is usually made of medium carbon steel.

8. Crank Shaft It is the shaft, a rotating member, which connects the crank. The power developed by the engine is transmitted outside through this shaft. It is made of medium carbon or alloy steels.

9. Cooling Fins or Cooling Water Jackets During combustion, the engine releases a large amount of heat. Thus the engine parts may be subjected to a temperature at which engine parts may not sustain their properties such as hardness, etc. In order to keep the engine parts within safe temperature limits, the cylinder and the cylinder head are provided with a cooling arrangement. The cooling fins are provided on light duty engines, while a cooling water jacket is provided on medium and heavy duty engines.

10. Cam Shaft It is provided on four-stroke engines. It carries two cams, for controlling the opening and closing of inlet and exhaust valves.

11. Inlet Valve This valve controls the admission of charge into the engine during a suction stroke.

12. Exhaust Valve The removal of exhausted gases after doing work on the piston is controlled by the valve.

13. Inlet Manifold It is the passage which carries the charge from carburettor to the engine.

14. Exhaust mainfold It is the passage which carries the exhaust gases from the exhaust valve to the atmosphere.

15.(*a*) *Spark Plug* It is provided on petrol engines. It produces a high-intensity spark which initiates the combustion process of the charge.

(*b*) *Fuel Injector* It is provided on Diesel engines. The Diesel fuel is injected in the cylinder at the end of the compression through a fuel injector under very high pressure.

16.(*a*) *Carburetor* It is provided with a petrol engine for preparation of a homogeneous mixture of air and fuel (petrol). This mixture, as a charge, is supplied to engine cylinder through suction valve or port.

(*b*) *Fuel Pump* It is provided with a diesel engine. The diesel is taken from the fuel tank, its pressure is raised in the fuel pump and then delivered to fuel injector.

17. Fly wheel It is mounted on the crank shaft and is made of cast iron. It stores energy in the form of inertia, when energy is in excess and it gives back energy when it is in deficit. In other words, it minimizes the speed fluctuations on the engine.

1.4 OTTO CYCLE ENGINES: PETROL ENGINES

The ordinary Otto-cycle engine is a four-stroke engine; that is, its piston makes four strokes, two toward the cylinder head (TDC) and two away from the head (BDC). By suitable design, it is possible to operate an Otto-cycle as a two-stroke cycle engine with one power stroke in every revolution of the engine. Thus, the power of a two-stroke cycle engine is theoretically double that of a four-stroke cycle engine of comparable size. These engines are also called *spark ignition engines.*

1.4.1 Two-stroke Petrol Engine

All essential operations are carried out in one revolution of the crank shaft or two strokes of the piston. Therefore, the engine is called a two-stroke or two-stroke cycle engine.

(*a*) *Construction Details*

A two-stroke petrol engine is shown in Fig. 1.5. It consists of a cylinder, cylinder head, piston, piston rings, connecting rod, crank, crank case, crank shaft, etc. The charge (air–fuel mixture) is prepared outside the cylinder in the carburetor. In the simplest type of two-stroke engine, the ports are provided for charge inlet and exhaust outlet, which are covered and uncovered by the moving piston. The suction port *S* with a reed-type valve is used for induction of charge into the crank case, the transfer port *T* is used for transfer of charge from the crank case to the cylinder and the exhaust port *E* serves the purpose of discharging the burnt gases from the cylinder. The spark plug is located in the cylinder head.

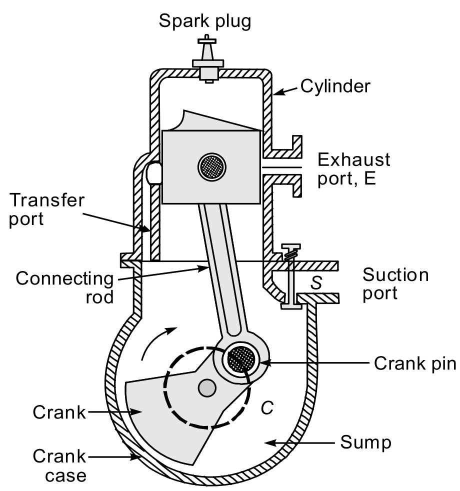


Fig. 1.5 *Two stroke petrol engine*

(*b*) *Operation*

In a two-stroke engine, the inlet, transfer and exhaust ports are covered and uncovered by a moving piston. The following operations take place in one cycle of a two-stroke engine.

(*i*) *Charge Transfer and Scavenging* When the piston is nearer to the crank case (bottom dead centre), the transfer port and exhaust port are uncovered by the piston as shown in Fig. 1.6(a). A mixture of air and fuel as a charge, slightly compressed in the crank case, enters through the transfer port *T* and drives out the burnt gases of the previous cycle through the exhaust port *E*.

In a two-stroke engine, the piston top is made deflected. Therefore, the incoming charge is directed upward, and aids in sweeping of the burnt gases out of the cylinder. This operation is known as *scavenging* (a gas-exchange process).

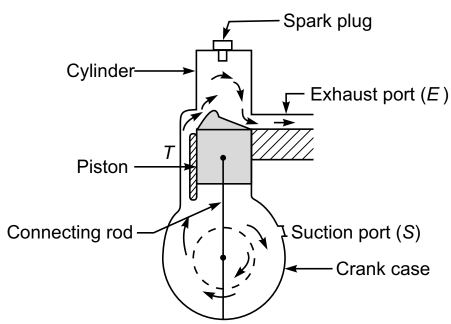
As the piston moves upward, the fresh charge passes into the cylinder for $1/6^{th}$ of the revolution and the exhaust port remains open a little longer than the transfer port.

(*ii*) *Compression and Suction* As the piston moves upward, both the transfer port and exhaust port are covered by the piston and the charge trapped in the cylinder is compressed by the piston's upward movement as shown in Fig. 1.6(b). At the same time, a partial vacuum is created into the crank case, the suction port *S* opens and the fresh charge enters the crank case [Fig. 1.6 (c)].

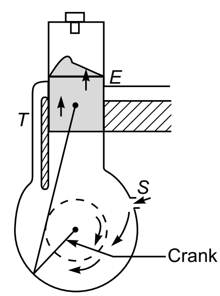
(*iii*) *Combustion* When the piston reaches at its end of stroke nearer to the cylinder head or at the top dead centre, a high-intensity spark from the spark plug ignites the charge and initiates the combustion in the cylinder. The burning of the charge generates the pressure in the cylinder.

(*iv*) *Power and Exhaust* The burning gases apply pressure on the top of the piston, and the piston is forced downward as a result of pressure generated.

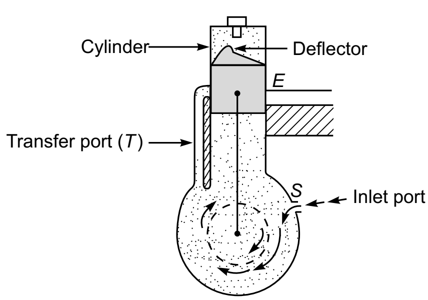
As the piston descends through about 80% of the expansion stroke, the exhaust port *E* is uncovered by the piston, and the combustion gases leave the cylinder by pressure difference and at the same time, the underside of the piston causes compression of charge taken into crank case as shown in Fig. 1.6(d).



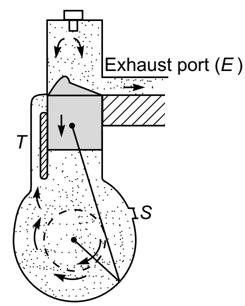
(a) Charge transfer and scavenging



(b) Start of compression



(c) Compression and Suction



(d) Power and exhaust

Fig. 1.6 *Operations of a two-stroke petrol engine* (*v*) *Charging* The slightly compressed charge in the crank case passes through the transfer port and enters the cylinder as soon as it is uncovered by the descending piston and when it approaches the bottom dead centre, the cycle is completed.

The *p-V* diagram and port-timing diagram for a two-stroke petrol engine are shown in Figs. 1.7 and 1.8.

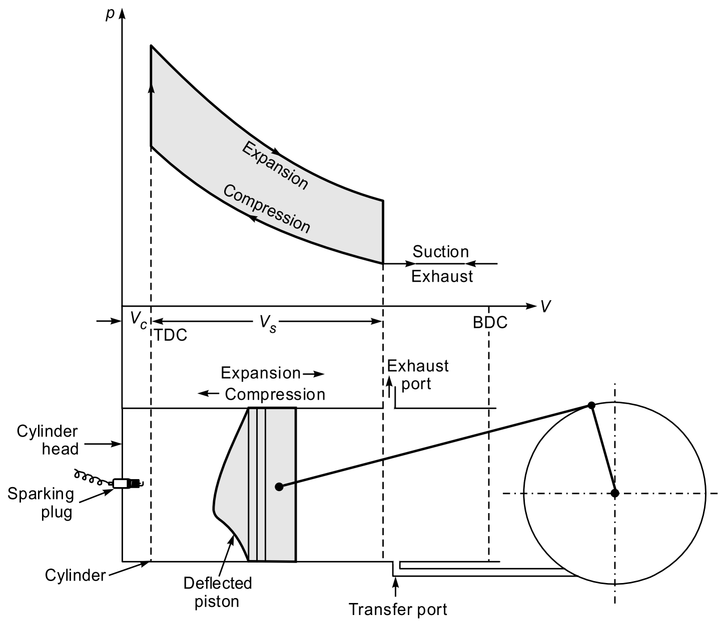
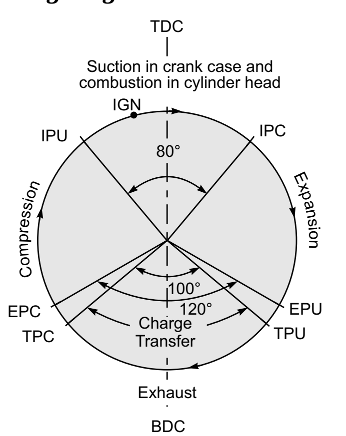


Fig. 1.7 *p–V diagram and schematic of a two-stroke petrol engine*

(*c*) *Port-Timing Diagram*



IPU = Inlet port uncovered 40° before TDC
IPC = Inlet port closed 40° after TDC
EPU = Exhaust port uncovered 60° before BDC
EPC = Exhaust port closed 60° after BDC
TPU = Transfer port uncovered 50° before BDC
TPC = Transfer port closed 50° after BDC
IGN = Spark ignition 15-20° before TDC

Fig. 1.8 *Port-timing diagram for a two-stroke petrol engine*

(*d*) *Applications*

Two-stroke gasoline engines are used where simplicity and low cost are main considerations. These engines have a little higher specific fuel consumption.

1. 50 cc–70 cc engines are used in mopeds, lawn moovers and non-gear vehicles.
2. 100–150 cc engines are commonly used in scooters and motor cycles.
3. 250 cc two-stroke engines are used in high-powered (racing) motor cycles.
4. These engines can also be used in small electric generator sets, pumping sets and motor boats.

1.4.2 Four-Stroke Cycle Petrol Engine

All operations are carried out in four strokes of the piston, i.e., two revolutions of the crank shaft.

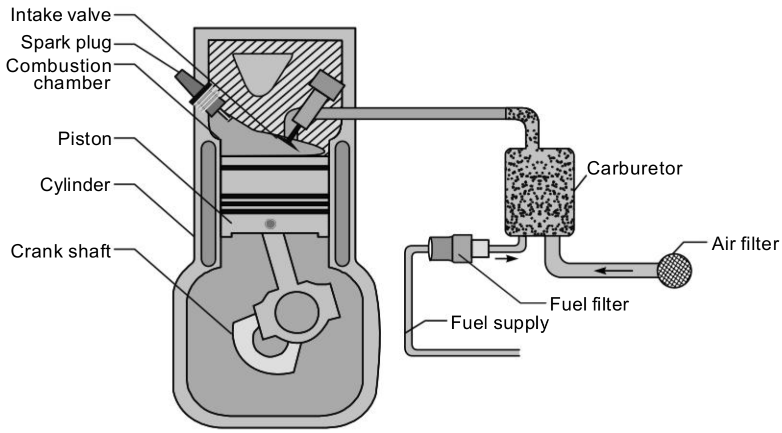


Fig. 1.9 *Four-stroke petrol engine*

(*a*) *Constructional Details*

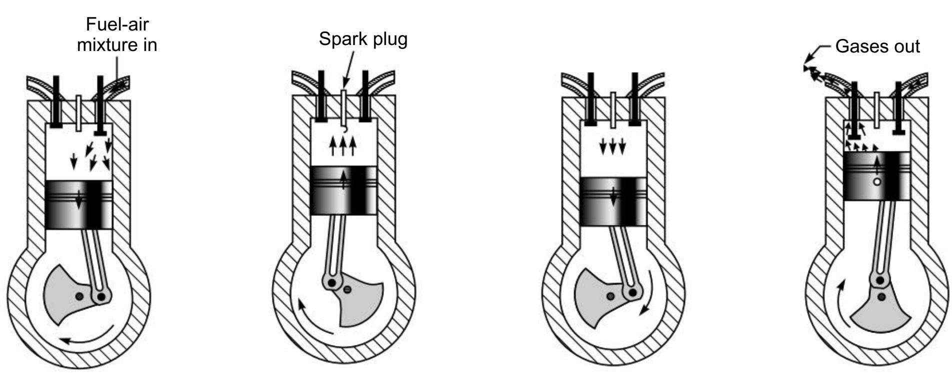
Similar to a two-stroke engine, it also consists of a cylinder, cylinder head attached with spark plug, piston attached with piston ring, connecting rod, crank, crank shaft, etc., as shown in Fig. 1.9. In a four-stroke engine, valves are used instead of ports. There are suction and exhaust valves. These valves are operated by cams attached on a separate shaft, called a *cam shaft*. It is rotated at half the speed of a crank shaft.

(*b*) *Operation*

The travel of the piston from one dead centre to another is called *piston stroke* and a four-stroke cycle consists of four strokes as suction, compression, expansion and exhaust strokes.

1. Suction Stroke The suction valve opens, exhaust valve remains closed as shown in Fig. 1.10(a). The piston moves from the top dead centre to the bottom dead centre, the *charge* (mixture of fuel and air prepared in the carburetor) is drawn into the cylinder.

2. Compression Stroke When the piston moves from the bottom dead centre to top dead centre, and the suction valve is closed, exhaust valve remains closed as shown in Fig. 1.10(b). The trapped charge in the cylinder is compressed by the upward moving piston. As the piston approaches the top dead centre, the compression stroke completes.



(a) Suction *(b) Compression* *(c) Power* *(d) Exhaust*

Fig. 1.10 *Operations of a four-stroke petrol engine*

3. Expansion Stroke At the end of the compression stroke, the compressed charge is ignited by a high-intensity spark created by a spark plug, combustion starts and the high-pressure burning gases force the piston downward as shown in Fig. 1.10(c). The gas pressure performs work, therefore, it is also called *working stroke* or *power stroke*. When the piston approaches the bottom dead centre in its downward stroke then this stroke is completed. In this stroke, both valves remain closed.

4. Exhaust Stroke When the piston moves from the bottom dead centre to the top dead centre, only the exhaust valve opens and burnt gases are expelled to surroundings by upward movement of the piston as shown in Fig. 1.10(d). This stroke is completed when the piston approaches the top dead centre. Thus, one cycle of a four stroke petrol engine is completed. The next cycle begins with piston movement from the top dead centre to the bottom dead centre.

Figure 1.11 shows the *p–V* diagram with a schematic of a four-stroke petrol engine.

(*c*) *Valve Timing*

Theoretically, in a four-stroke cycle engine, the inlet and exhaust valves open and close at dead centres as shown in Fig. 1.12(a).

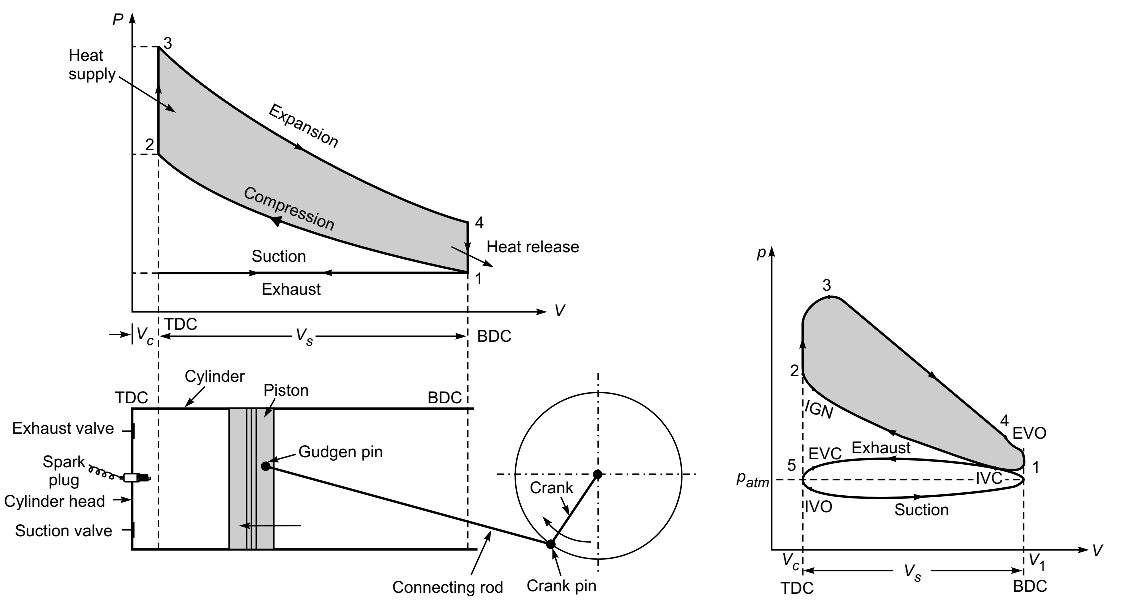
A typical valve-timing diagram for a four-stroke petrol engine is shown in Fig. 1.12(b). The angular positions in terms of crank angle with respect to TDC and BDC position of piston are quoted on the diagram.

When the inlet valve and exhaust valve remain open simultaneously, it is called a *valve operlap*.

(*d*) *Applications*

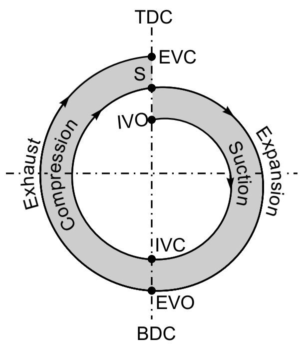
These engines are mostly used on automobiles, motor cycles, cars, buses, trucks, aeroplanes, small pumping sets, mobile electric generators, etc.

Nowadays, the four-stroke petrol engines have been replaced by four-stroke Diesel engines for most applications.



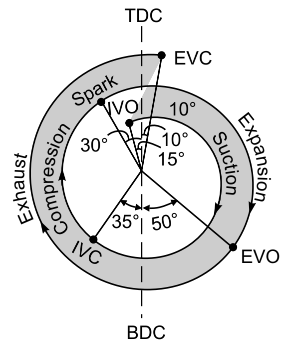
(a) *Theoretical p–V diagram for a four-stroke petrol engine* (b) *Actual p–V diagram for four stroke petrol engine*

Fig. 1.11



(a) Theoretical valve timing diagram of a four stroke cycle

IVO = Inlet valve opens when piston at TDC
IVC = Inlet valve closes, when piston reaches BDC
S = Spark produces, when piston reaches TDC
EVO = Exhaust valve opens when piston at BDC
EVC = Exhaust valve closes, when piston at TDC



(b) The typical valve timing diagram for a four stroke petrol engine

IVO : Inlet valve opens about 15° before TDC
IVC : Inlet valve closes 20° – 40° after BDC to take advantage of rapidly moving gas
S : Spark occurs 20°–40° before TDC
EVO : Exhaust valve opens about 50° before BDC
EVC : Exhaust valve close about 0° to 10° after TDC

Fig. 1.12

1.5 DIESEL ENGINES

All engines using diesel as a fuel operate on the Diesel cycle. They work similar to a petrol engine except they take in only air as charge during suction, and fuel is injected at the end of the compression stroke. The Diesel engines have a fuel injector instead of a spark plug in the cylinder head as shown in Fig. 1.13. The diesel engines use a high compression ratio in the range of 14 to 21. The temperature of intake air reaches quite a high value at the end of compression. Therefore, the injected fuel is self ignited. The Diesel engines use a hetrogeneous air–fuel mixture, ratio ranging from 20 to 60.

1.5.1 Two-stroke Diesel Engine

The operation of a two-stroke diesel engine is similar to a petrol engine, except it takes air as charge and fuel is injected at the end of the compression stroke. It uses a high compression ratio. Therefore, the injected fuel is self-ignited.

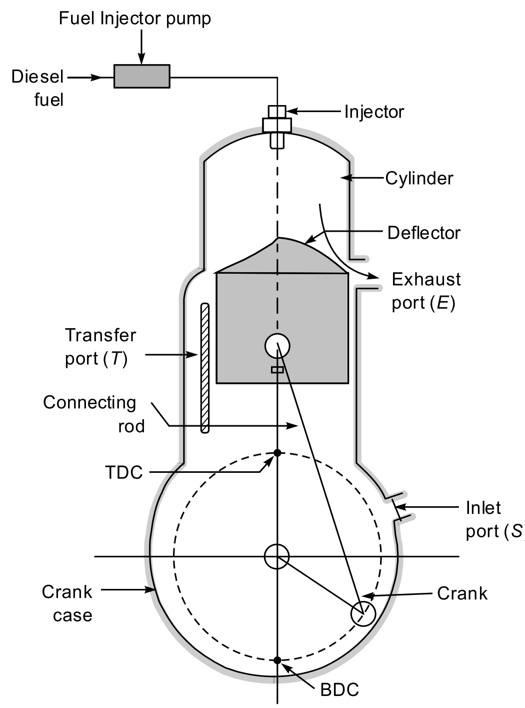


Fig. 1.13 *Schematic of two-stroke Diesel engine*

Operation

Both inlet and exhaust take place through the cylinder ports which are covered and uncovered by the piston.

(*i*) *Charge Transfer and Scavenging* When the piston is nearer to the crank case (bottom dead centre), the transfer port and exhaust port are uncovered by the piston and the slightly compressed air enters into the cylinder through the transfer port and helps to scavenge the remaining burnt gases from the cylinder as shown in Fig. 1.14(a). The charge transfer and scavenging continue till the piston completes its downward stroke and further, it moves upward and covers the transfer port.

(*ii*) *Compression and Suction* After covering the transfer port, the exhaust port is also covered by the upward moving piston. As both ports are covered by the piston in Fig. 1.14(b), the air trapped in the cylinder is compressed during the forward stroke of the piston. As the piston moves towards the cylinder head, a partial vacuum is created in the crank case, the inlet port opens and fresh air enters the crank case, Fig. 1.14(c).

(*iii*) *Combustion and Power* Near the end of the compression stroke, the fuel is injected at a very high pressure with the help of the fuel pump and injector. The injected fuel is self ignited in the presence of hot air and combustion starts. The piston is forced downward by very high pressure of burnt gases and power is transmitted to the crank shaft.

(*iv*) *Exhaust* Near the end of the power stroke, the exhaust port is uncovered first by the piston and the products of combustion start leaving the cylinder as a result of pressure difference as shown in Fig. 1.14(d).

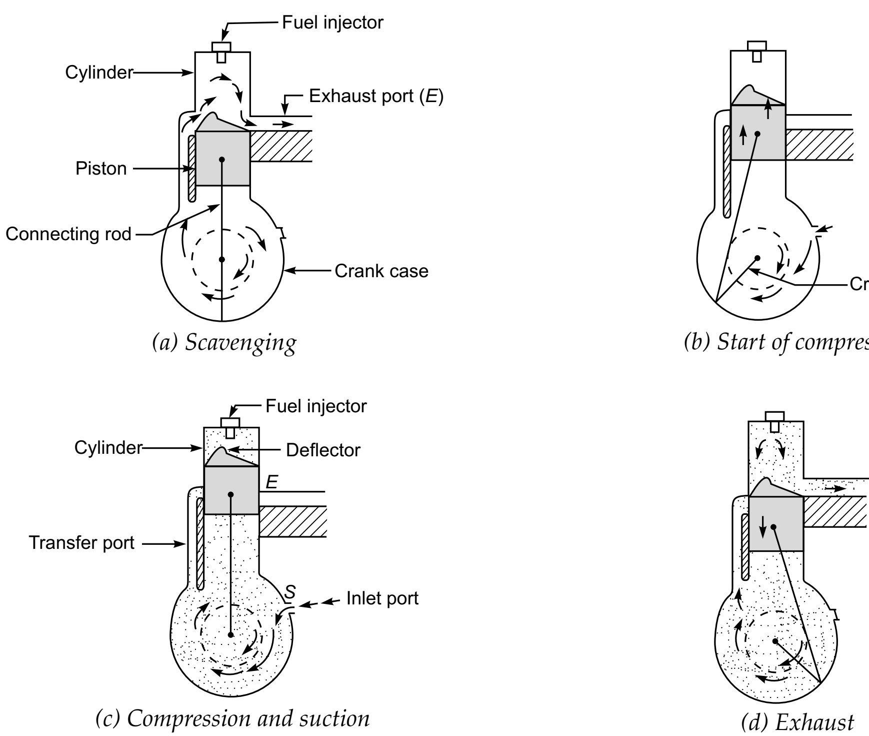


Fig. 1.14 *Operations of a two-stroke Diesel engine*

(*v*) *Charging* The slightly compressed air in the crank case passes through the transfer port and enters the cylinder as soon as it is uncovered by the descending piston and when it approaches the bottom dead centres, the cycle is completed.

Theoretical p–V Diagram

The theoretical *p–V* digaram for a two-stroke Diesel engine is shown in Fig. 1.15. The valve timing diagram for a two-stroke diesel engine is very similar to that of a two-stroke petrol engine as shown in Fig. 1.12 except, only air is inducted in the crank case and fuel is injected at the end of the compression stroke, instead of spark from the spark plug.

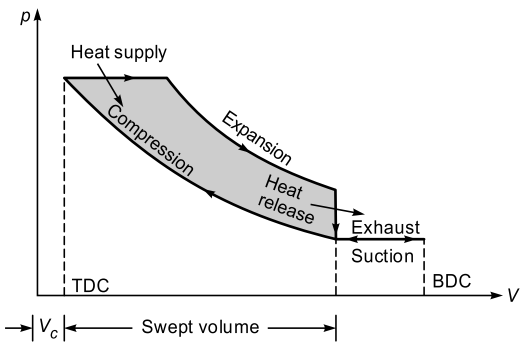


Fig. 1.15 *p–V diagram for two-stroke Diesel engine*

1.5.2 Four-stroke Diesel Engine

A four-stroke diesel engine contains a fuel injector, fuel pump, cylinder, cylinder head, inlet and exhaust valves, piston attached with piston rings, connecting rod, crank shaft, cams, camshaft, etc. shown in Fig. 1.16. One cycle of a four-stroke diesel engine is completed in four strokes of the piston or two revolutions of the crank shaft.

(*a*) *Working of Engine*

The four-stroke diesel engine operates in a similar manner as a four-stroke petrol engine. A schematic of a four-stroke Diesel engine is shown in Fig. 1.16. The details of operations are discussed below.

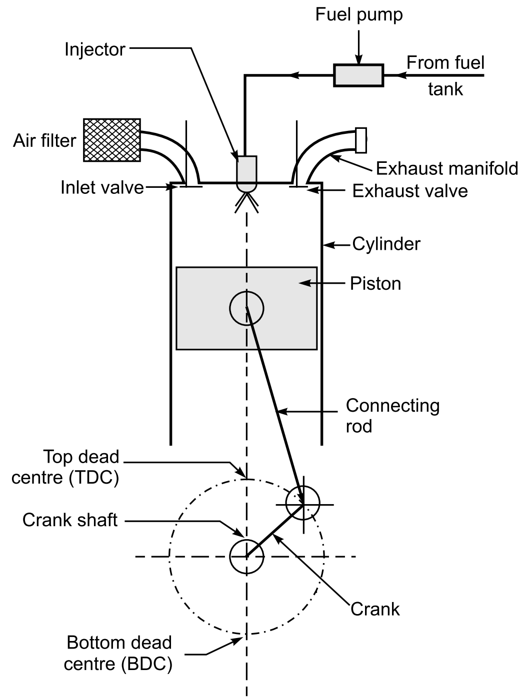


Fig. 1.16 *Schematic of a four-stroke Diesel engine*

1. Suction Stroke The inlet (suction) valve opens, the exhaust valve remains closed, only air is drawn into the cylinder as the piston moves from the top dead centre to the bottom dead centre. This stroke ends as the piston approaches the bottom dead centre. Fig. 1.17(a)

2. Compression Stroke As the piston moves from the bottom dead centre to the top dead centre, the inlet valve closes, exhaust valve remains closed as shown in Fig. 1.17(b). The air trapped into the cylinder is compressed in the cylinder till the piston approaches the top dead centre. The air temperature reaches about 800°C by compression. At the end of the compression stroke, the fuel is injected at very high pressure into the compressed hot air. The temperature of hot compressed air is sufficient to ignite the injected fuel. Thus, ignition takes place inside the cylinder.

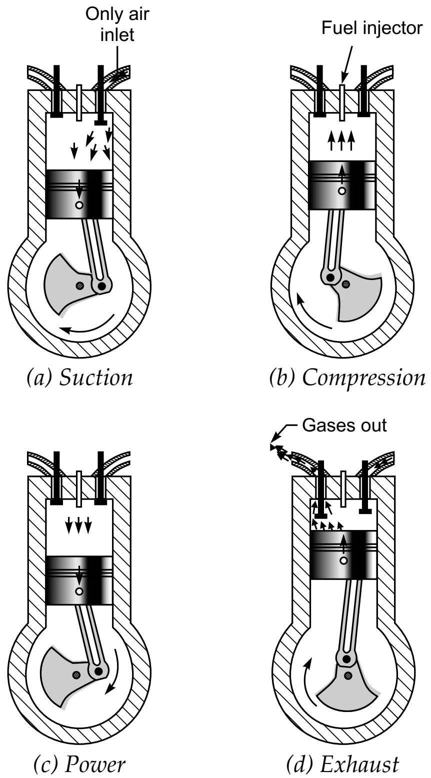


Fig. 1.17 *Operations of four-stroke diesel engine*

3. Expansion Stroke During this stroke, both valves remain closed as shown in Fig. 1.17(c). The piston at the top dead centre is pushed by expansion of burning gases. Actual work is obtained during this stroke due to the force obtained by high pressure burning gases. Therefore, this stroke is called *power stroke or working stroke*.

4. Exhaust Stroke During this stroke, the piston moves from the bottom dead centre to the top dead centre, exhaust valve opens and the inlet valve remains closed. Burnt gases of the previous stroke are expelled out from the cylinder by upward movement of the piston.

The theoretical *p–V* diagram is shown in Fig. 1.18 for a four-stroke diesel engine operation.

(*b*) *Valve-Timing Diagram*

Theoretically, the inlet and exhaust valves open at dead centres as shown in Fig. 1.12(a). A typical valve-timing diagram for a four-stroke Diesel engine is shown in Fig. 1.19.

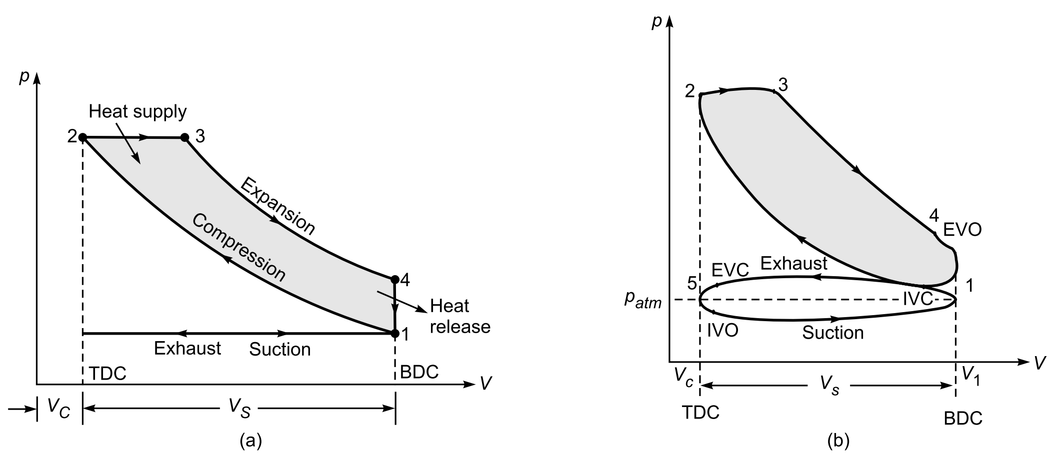


Fig. 1.18 *(a) Theoretical p-V diagram (b) Actual p-V diagram engine*

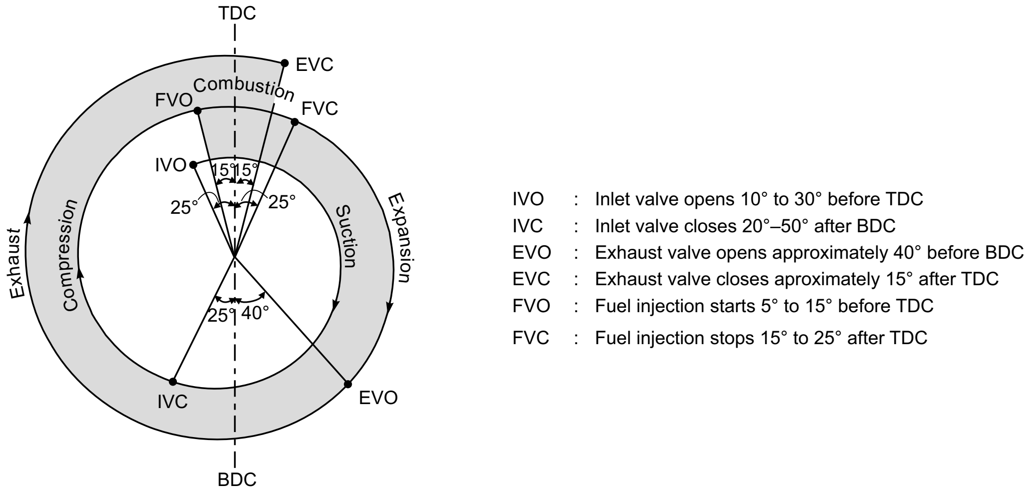


Fig. 1.19 *Typical valve-timing diagram for a four-stroke Diesel engine*

(*c*) *Applications*

The four-stroke diesel engine is one of the most popular prime movers. It is manufactured from 50 mm to 1000 mm cylinder bore with speeds ranging from 100 rpm to 4500 rpm. It has wide applications. Some of these are

1. Small pumping sets for agriculture,
2. Construction machinery,
3. Air compressor and drilling jigs,
4. Tractors, jeeps, cars, taxies, buses, trucks,
5. Diesel-electric locomotives,
6. Small power plants, mobile electric generating plants,
7. Boats and ships,
8. Power saws, Bulldozers, tanks, etc.

1.6 COMPARISON BETWEEN PETROL AND DIESEL ENGINES

According to	*Petrol Engine*	*Diesel Engine*
1. Basic cycle	It operates on constant-volume cycle.	It operates on constant-pressure cycle.
2. Fuel used	It uses gasolene or petrol as fuel.	It uses diesel and oils as a fuel.
3. Fuel induction	The air–fuel mixture is prepared in the carburettor and inducted into the engine cylinder during the suction stroke.	The Diesel engine takes in only air during the suction stroke, and it is compressed. At the end of the compression stroke the fuel is injected under the high pressure by a fuel injector.
4. Ignition of charge	The charge (air–fuel mixture) is ignited by a high-intensity spark produced at the spark plug.	Fuel is injected in very hot air, therefore, it is self-ignited.
5. Compression ratio	It uses less compression ratio, usually range of 4 to 10.	It uses high compression ratio, range of 14 to 21.

Contd.

Contd.

6. Pressure rise	Lower and controlled rate of pressure rise; therefore, operation is salient and smooth.	High rate of pressure variation, so engine operation is rough, and noisier.
7. Efficiency of cycle	Due to lower compression ratio, the efficiency of petrol engine is poor.	It has better thermal efficiency due to high compression ratio.
8. Pollution	Comparatively lower pollution for same power output.	Higher pollution for same power output.
9. Weight	It has comparatively less number of parts, thus is less in weight.	It uses large number of sturdier parts, thus engine is heavy.
10. Cost	Engines are cheaper.	Costlier engine due to complicated parts.
11. Maintenance	It requires less and cheaper maintenance.	It requires costlier and large maintainance.
12. Starting	Very easy to start due to lower compression ratio.	Very difficult to start due to higher compression ratio.
13. Engine Speed	Petrol engines run at high speed	Disel engines run at slow speed
14. Knocking	Knocking in SI engines occur at the end of combustion.	Knocking in CI engines occur at the start of combustion.

1.7 COMPARISON BETWEEN TWO-STROKE AND FOUR-STROKE ENGINES

According to	*Two-Stroke Engine*	*Four-Stroke Engine*
1. Working stroke	There is one working stroke in each revolution. Hence engine has more even torque and reduced vibration.	There is one working stroke in two revolutions. Hence engine has uneven torque and large vibration.
2. Engine design	It uses ports and hence engine design is simple.	It uses valves, therefore, mechanism involved is complex.
3. Mechanical efficiency	The working cycle completes in one revolution and hence it has high mechanical efficiency.	Working cycle completes in two revolution, hence, it has more friction, thus less mechanical efficiency.
4. Scavenging	The burnt gases are not completely driven out. It results in dilution of fresh charge.	It has separate stroke for explusion of burnt gases, thus ideally no dilution of fresh charge.
5. Thermal efficiency	Poor thermal efficiency due to poor scavenging and escaping of charge with exhaust gases.	Very good thermal efficiency.
6. Cost	Less cost due to less parts in engine.	More cost due to large number of parts.
7. Maintenance	Cheaper and simple.	Costlier and slightly complex.
8. Weight	Lighter engine body.	Heavier engine body.

1.8 ADVANTAGES AND DISADVANTAGES OF TWO-STROKE CYCLE ENGINES

Advantages The two-stroke cycle engines have the following advantages over four-stroke cycle engine:

(i) The two-stroke engines are simple in design, manufacturing and operation. Hence they are used in small power engines such as scooters, mopeds, motor cycles and autorickshas, etc.

(ii) The two-stroke cycle engine develops net work as positive because it works above atmospheric pressure.

(iii) The turning moment of the two-stroke cycle engine is more uniform, thus a smaller flywheel is required.

(iv) There are less frictional losses due to absence of valves, thus higher mechanical efficiency is obtained in two-stroke cycle engines.

(v) All the operations in two-stroke cycle engines complete in two strokes, i.e., one revolution of the crank shaft, thus theoretically, a two-stroke cycle engine develops twice the power of the four-stroke cycle engine for the same size and same speed of the engine.

(vi) The two-stroke cycle engine is much lighter and more compact for same power output than a four-stroke cycle engine.

(vii) Due to less parts in a two-stroke cycle engine, the maintenance required is very less. Further, the maintenance cost is very low.

(viii) Actually, the gas dilution in a two-stroke cycle engine is comparatively less than the four-stroke cycle engine.

(ix) Engine cost is also less than the four-stroke cycle engine.

Disadvantages The two-stroke cycle engines have the following disadvantages over four-stroke cycle engines:

(i) The effective compression in a two-stroke cycle engine is lower for the same stroke.

(ii) Due to less effective cooling arrangement in two-stroke cycle engines, some amount of lubrication oil burns during the combustion of charge, thus more lubrication oil is consumed.

(iii) A two-stroke cycle engine is more noisy because of sudden release of exhaust gases in the port.

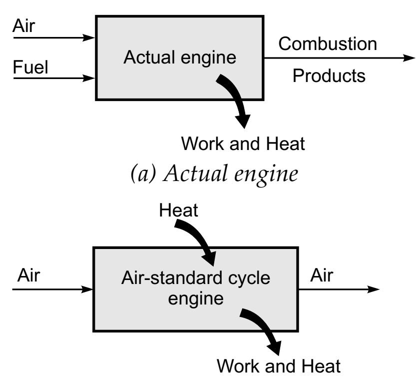
(iv) In a two-stroke cycle engine, the scavenging (gas exhange process) is poor. The fresh charge pushes the burnt gases out. In this process, some fresh charge always flows with the exhaust gases.

(v) More wear and tear take place in a two-stroke cycle engine.

(vi) Thermal efficiency of a two-stroke cycle engine is less due charge loss during scavenging.

1.9 AIR STANDARD ANALYSIS

The simplest models for both spark ignition and compression ignition engines are air standard cycles shown in Fig. 1.20. The processes of these ideal cycles are selected such that they should be similar to actual cycles, as much as possible, because the analysis of actual cycles is very much complex. The assumptions made in the analysis of air standard cycles are the following:



(a) Actual engine

(b) Air-standard cycle engine

Fig. 1.20

1. The system is closed, thus the cycle is a complete thermodynamic cycle and the same fluid is used repeatedly.
2. Air as an ideal gas is used as working fluid in the cycle.
3. The compression and expansion processes are reversible, adiabatic.
4. The combustion process is replaced by a reversible heat-addition process from an external source.
5. The exhaust process is replaced by a reversible constant-volume, heat-rejection process to the surroundings to restore the system back to its initial state.
6. In addition, in cold air standard analysis, the specific heats of air are assumed to be constant at their ambient temperature value.

$$C_p = 1.005 \text{ kJ/kg}\cdot\text{K}$$
$$C_v = 0.717 \text{ kJ/kg}\cdot\text{K}$$
$$R = 0.287 \text{ kJ/kg}\cdot\text{K}$$
$$\gamma = 1.4$$

7. Combustion takes place instantaneously at top dead center.
8. The fuel is mixed well with air.
9. Subsequent to combustion, the change is always in chemical equilibrium.

Moreover, with air standard analysis, one can predict the effect of compression ratio on cycle efficiency, while fuel air cycle does not give any information of efficiency variation with change in fuel air ratio.

1.10 FUEL AIR CYCLE

It is a theoretical cycle based on the actual properties of the cylinder contents. The fuel – air cycle takes into consideration the following facts:

1. The actual composition of the cylinder contents.
2. The variation in the specific heats of the gases in the cylinder.
3. The dissociation effect.
4. The variation in the number of moles present in the cylinder as the pressure and temperature change.
5. Compression and expansion processes are reversible adiabatic.
6. No chemical changes in either fuel or air prior to combustion.

1.10.1 Composition of Cylinder Gases

The actual composition of the cylinder contents are

1. Fuel+Air + Water vapor + residual gas.
2. The fuel air ratio changes during the engine operation.
3. The change in air-fuel ratio affects the composition of gases before and after combustion particularly the percentage of CO_2, CO, H_2O etc. in the exhaust gas.
4. The amount of exhaust gases in the clearance volume varies with speed and load on the engine.
5. The fresh charge composition varies when it enters into the cylinder and comes in contact with the burnt gases.

The composition of the working fluid, which changes during the engine operating cycle, is indicated in the following table.

Process	*SI Engine*	*CI Engine*
Intake	Air, Fuel, Recycled exhaust and Residual gas	Air, Recycled exhaust and Residual gas
Compression	Air, Fuel Vapor, Recycled exhaust and Residual gas	Air, Recycled exhaust and Residual gas

Contd.

Contd.

Expansion	Composition products (CO_2, CO, H_2, O_2, NO, OH, O, H, …)	Composition products CO_2, CO, H_2, O_2, NO, N_2, OH, H_2O, O, H, …)
Exhaust	Composition products mainly N_2, CO_2, H_2O) If $\phi < 1$ O_2 or If $\phi > 1$ CO and H_2	Composition products (mainly N_2, CO_2, H_2O and O_2)

Where ϕ is equivalence ratio and $\phi = 1$ for chemically correct air fuel ratio.

1.10.2 Variable Specific Heats

The variation of specific heats with temperature

1. All gases except mono-atomic gases, show an increase in specific heats with temperature.
2. The increase in specific heats does not follow any specific law.
3. However, between the temperature range 300 K-1500 K the specific heats curve is nearly a straight line which may be approximately expressed in form

$$C_p = A + K_1T$$
$$C_v = B + K_1T$$
$$R = C_p - C_v = A-B$$

   Where A, B and K_1 are constants.
   Above 1500 K, the increase in specific heats is more rapid, and they may be expressed in the form

$$C_p = A + K_1T + K_2T^2$$
$$C_v = B + K_1T + K_2T^2$$

4. Since the difference between C_p and C_v is constant, the value of k decreases with increase in temperature.

$$k = \frac{C_p}{C_v} = \frac{C_v + R}{C_v} = 1 + \frac{R}{C_v}$$

Physical Expression

At 300 K; $C_p = 1.005$ kJ/kg·K
At 2000 K, $C_p = 1.343$ kJ/kg·K

Similarly,

At 300 K; $C_v = 0.718$ kJ/kg·K
At 2000 K, $C_p = 1.055$ kJ/kg·K

The larger fraction of heat input is utilized to increase the motion of atoms within the system, thus specific heats increase and the temperature and pressure at the end of compression would be lower compare to the values obtained with constant specific heats.

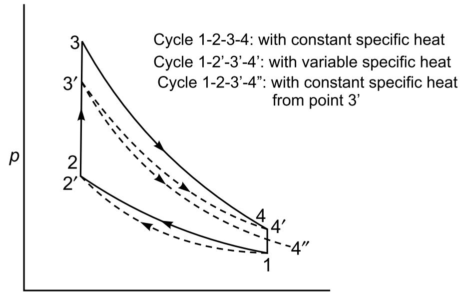


Fig. 1.21 *Effect of specific heats variation on Fuel Air Cycle*

1.10.3 Dissociation Effect

The dissociation is a process of disintegration of combustion products at high temperature. Dissociation is the reverse process to combustion. During this process considerable amount of heat is absorbed (endothermic process) while during combustion process, heat is liberated (Exothermic process).

The dissociation of CO_2 into CO and O_2 takes place above 1000°C

$$2CO_2 + \text{Heat} \Leftrightarrow 2CO + O_2$$

The dissociation of H_2O into H_2 and O_2 takes place above 1300°C as

$$2H_2O + \text{Heat} \Leftrightarrow 2H_2 + O_2$$

1. The presence of CO and O_2 in the gases tends to prevent dissociation of CO_2.
2. In a rich fuel mixture, the presence of CO, suppresses dissociation of CO_2. On the other hand, there is no dissociation of burnt gases in lean mixture due to low temperature during combustion process.
3. The maximum dissociation occurs in the burnt gases of the chemically correct fuel-air mixture when the temperature is expected to be high.

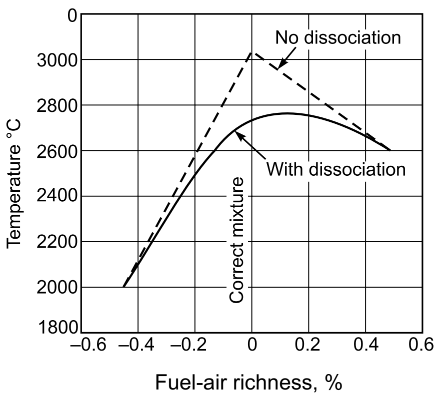


Fig. 1.22 *Effect of dissociation with temperature rise*

Figure 1.22 shows a typical curve; the reduction in temperature of exhaust gases due to dissociation with respect to degree of richness of air-fuel mixture:

1. Without dissociation, the maximum temperature occurs at chemically correct mixture.
2. With dissociation, the maximum temperature approaches with slightly rich mixture.
3. The dissociation reduces the maximum temperature by about 300°C at chemically correct air fuel ration.

The effect of dissociation in SI engines is much larger than CI engines. This is mainly due to

1. Presence of homogeneous mixture in SI engines, and
2. Excess air present in CI engines to ensure complete combustion.

1.10.4 Effect of Number of Molecules

The number of moles presented after combustion depend upon Fuel-Air ratio, type of combustion process in the cylinder. According to gas law the pressure and temperature are related as

$$pV = N R_u T$$

where N is number of moles, R_u is universal gas constant and T is absolute temperature.

The pressure varies with number of moles present in the cylinder. It puts direct impact on amount of work that gases in cylinder do on the piston.

1.10.5 Comparison of Fuel Air Cycle with Air Standard Cycle

1. In air standard cycles, the working fluid is air; a perfect gas.
2. In air standard cycles; the specific heats of air are treated constant , while variable in fuel air cycle.
3. The air standard cycles are highly simplified approximations. Therefore, engine performance analysis is much the higher than actual performance.
4. There is no dissociation loss in air standard cycles, since no chemical reaction takes place in cycle.
5. Number of moles does not vary in air standard cycles, while they vary in fuel air cycle.
6. In fuel air cycle, the heat is generated due to combustion of fuel and air, while in air standard cycles, the heat exchange takes place from external source and sink.
7. Air standard cycle analysis does not consider the effect to Fuel-Air ratio on the thermal efficiency because the working medium was assumed to be only air.

1.11 ACTUAL CYCLES

Actual cycle is much more complex than fuel air cycle. Compression and expansion processes are polytropic due to heat transfer through cyclinder walls during the processes. Cycle efficiency is much lower than the air standard cycle due to following facts:

1. Losses due to variation in specific heats with temperature,
2. Dissociation losses during combustion,
3. Time losses,
4. Losses due to incomplete combustion,
5. Direct heat losses,
6. Exhaust blow down losses,
7. Pumping losses or gas exchange losses, and
8. Blowby loss and friction during processes.

If we substract losses due to variable specific heats and dissociation losses from air standard cycle, resulting cycle is *fuel air cycle analysis* and we further substract other losses from fuel air cycle analysis, then resulting cycle is *actual cycle*.

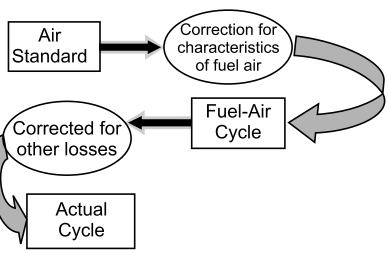


Fig. 1.23 *Air Standard and actual cycles*

The losses due to variation in specific heats and dissociation are discussed earlier in fuel- air cycle. The remaining losses are discussed below.

1.11.1 Time Losses

This loss is due to time required for mixing of fuel and air and for combustion.

(a) Incomplete Mixing of Fuel and Air

It is not possible to obtain perfect homogeneous mixture of fuel and air and residual gases in the cylinder before ignition begins. It is due to insufficient time available for mixture preparation. In one part of the cylinder there may be excessive oxygen and in other part excessive fuel may be present. Excessive fuel may not find enough oxygen for complete combustion, which may result into presence of both CO, and O_2 and unburnt fuel in the exhaust. It will lead to lower thermal efficiency of the cycle.

1.11.2 Progressive Burning

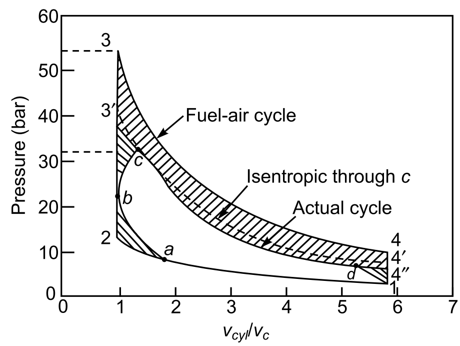


Fig. 1.24 *Effect of time loss on pressure rise*

The combustion takes place progressively rather than the instantaneous.

1. In SI engine, the crankshaft usually turns about 30 to 40 degree between the initiation of the spark and the end of combustion. It is the time loss due to progressive combustion.
2. Time loss for progressive combustion process varies with fuel composition, combustion chamber shape and size, including number and position of ignition points and engine operating conditions.
3. Due to the finite time of combustion, the peak pressure will not occur when the volume is minimum (TDC) but will occur slightly after TDC. Therefore, the pressure rises in the first part of the working stroke from b to c as shown in Fig. 1.24. This loss of work reduces the efficiency of the cycle.

1.11.3 Direct Heat Losses

Heat transfer from cylinder gases to cylinder walls during comptression stroke is negligible due to gas temperature being not high. But during combustion and early part of expansion process, the temperature of cylinder gases is too high and a significant amount of heat is tranferred from hot gases through cylinder walls and cylinder head into cooling medium. Some heat enters the piston head and flows to the piston rings and then into the cylinder walls from where it is taken up by lubricating oil, which splaces on the underside of piston. Some heat is convected and radiation from cylinder walls directly to ambient.

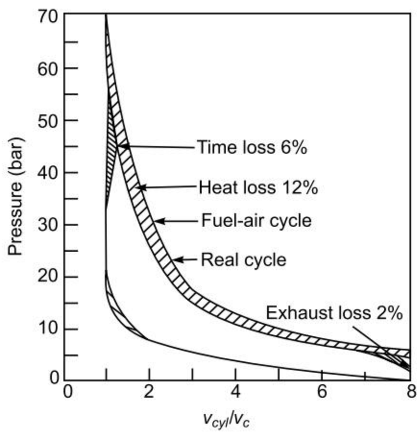


Fig. 1.25 *Time loss, Heat Loss and exhaust loss in SI engine*

Some observations are listed below:

1. Heat loss during combustion will have the maximum effect on the cycle efficiency
2. The effect of heat loss during combustion reduces the maximum temperature and therefore the specific heats are lower.
3. Out of various losses, the heat losses contribute around 12%.

1.11.4 Exhaust Blow Down Losses

The actual exhaust process consists of two phases:

(i) Blowdown of expanding gases, and
(ii) Displacement of cylinder gases.

Blow down losses are due to early opening of exhaust valve. If the exhaust valve is opened at bottom dead centre, the piston would have to do large work to displace the high pressure exhaust gases from the cylinder during the exhaust stroke. If exhaust valve is opened 40-70 degree before BDC, a part of expansion work will be lost; but it reduces cylinder pressure half way before exhaust stroke begins as shown in Fig. 1.26.

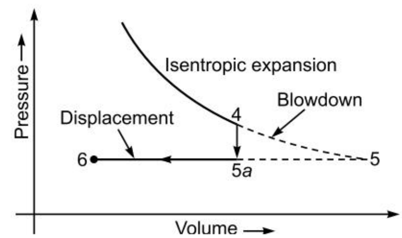


Fig. 1.26 *Illustration of Blowdown and displacement*

1.11.5 Pumping Losses

Loss of work during gas exchange process is called pumping loss. The pumping work includes displacement of exhaust gases from cylinder and suction of fresh charge into the cylinder for next cycle of operation. The pumping loss increases at part throttle, because throttling reduces the suction pressure. Pumping loss also increases with speed. Pumping loss also affects the volumetric efficiency of the engine. Fig. 1.27 shows blow down losses and pumping losses.

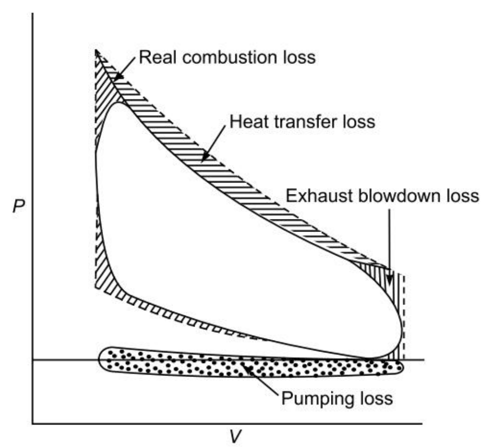


Fig. 1.27 *Blow-down and pumping losses*

1.11.6 Blowby Loss and Friction During Processes

The **blowby loss** is due to the leaking of gas flow through gaps between the piston, piston rings and cylinder walls. The gas usually leaks through them to the crankcase. Blow-by takes place when the combustion occurs in combustion chamber. Very high pressure of burning gases force past the rings into the crankcase. The causes of blowby are wear of piston rings, and soot and deposits on the cylinder walls. Fig.1.28 shows the leakage of air and exhaust gases during compression and expansion strokes respectively.

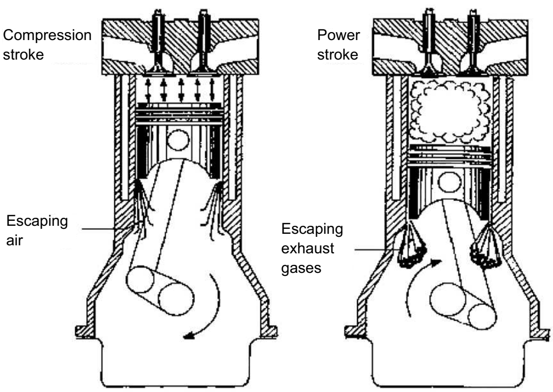


Fig. 1.28 *Blow-by losses during compression and expansion strokes*

Friction loss includes friction between piston and cylinder walls, friction in various bearings and friction in auxiliary equipments such as pumps and fans. This loss is also called mechanical loss. The piston friction, bearing friction and friction in auxiliary equipment increase with engine speed. It also increases with increase in mean effective pressure. The efficiency of an engine is maximum at full load and decreases at part load because the percentage of direct heat loss, pumping loss and mechanical loss increase at part loads.

1.12 COMPARISON BETWEEN AIR STANDARD CYCLE AND ACTUAL CYCLES

The actual cycles for internal combustion engines differ from air-standard cycles in many respects. The differences are listed below:

1. The air standard cycles are theoretical cycles, while actual cycles are practical cycles.
2. Mass of working substance in air standard cycles does not change or recirculated, while in actual cycles, fresh charge is inducted in every cycle except some amount of residual gases.
3. In air standard cycles, compression and expansion processes are isentropic, while in actual cycles, these processes are polytropic.
4. Only air as perfect gas is used as working substance in air standard cycles, while, working substance in actual cycles is a mixture of air and fuel vapour or finely atomized liquid fuel in air combined with the products of combustion left from the previous cycle.
5. The chemical composition of working substance does not change in air standard cycles, while the change in chemical composition of the working substance takes place in actual cycles.
6. In air standard cycles, the specific heats are considered constant, while in actual cycles, the specific heats increases with temperature.
7. In air standard cycles, initial state remains same at the start of each cycle, while in actual cycles, the change in composition, temperature and actual amount of fresh charge take place because of presence of residual gases from previous cycle.
8. In air standard cycles, the heat is added at TDC from external source, rather than the progressive combustion in actual cycles.

9. The heat is transferred to and from the working medium in air standard cycles, while heat is liberated by actual combustion of combustible mixture inside combustion chamber in actual cycles.
10. In air standard cycles, the mass of air considered constant, while in actual cycles, gas leakage, fluid friction etc. are involved.
11. Very less heat transfer during exhaust stroke of air standard cycles, while a substantial heat loss takes place during exhaust blow down in actual cycles.

Summary

- An internal combustion engine is a heat engine that converts chemical energy of fuel into mechanical energy.
- The two-stroke engine completes its working cycle in two strokes of the piston. The three ports; inlet, transfer, and exhaust ports are used for suction, transfer and discharge of charge, respectively. A deflector-shapped piston is used to direct the charge inside the cylinder.
- The four-stroke engine completes its working cycle in four strokes of the piston as suction, compression, expansion and exhaust stroke. They are widely used on motor cycles, cars, buses, trucks and aeroplanes. Due to good thermal efficiency of four-stroke engines, the specific fuel consumption is less.
- Petrol engines use low compression ratio in the range of 4 to 10, while Diesel engines use high compression ratio usually in the range of 14 to 21.
- The lubrication minimizes the friction between moving parts. The mist lubrication system is used in two-stroke engines, while all four-stroke engines use wet or dry sump lubrication system.
- Scavenging is a gas exchange process into the cylinder of the engine where the admitted charge pushes the combustion products out of the cylinder. Scavenging takes place in two-stroke engines at the end of the expansion stroke. Poor scavenging may lead to dilution of charge, thus less power is obtained from the engine.

Glossary

A/F Mass ratio of air to fuel

BDC Bottom dead centre

Bore Internal diameter of cylinder

CI engine Compression ignition (Diesel) engine

Clearance volume Volume left in the centre, when piston is at TDC

Compression ratio Ratio of maximum volume to minimum volume in the cylinder

IC engines Internal combustion reciprocating engines

Mean effective pressure Ratio of net work done to swept volume in the cycle

SI engine Spark ignition (petrol) engine

Scavenging Gas exchange process in two-stroke engines

Stroke Linear distance between TDC and BDC

Swept volume Piston displacement volume in cylinder

TDC Top dead centre

Thermostat An instrument which controls the temperature

Review Questions

1. What is an internal combustion engine?
2. Write the classification of internal combustion engines.
3. List the parts of an IC engine.
4. Explain the construction, working and applications of a two-stroke petrol engine.
5. Explain construction, and working of a two-stroke Diesel engine.
6. Explain the working of a four-stroke petrol engine.
7. Discuss the construction of a four-stroke petrol engine.
8. Explain the working of a four-stroke Diesel engine.
9. Why are four-stroke engines preferred over two-stroke engines?
10. Compare petrol and Diesel engines.
11. Compare two-stroke and four-stroke engines.

Objective Questions

1. Which one of the following parts does not exist in an IC engine?
   (a) Crank shaft (b) Cam shaft
   (c) Piston rod (d) Connecting rod
2. Stoichiometric air–fuel ratio of petrol is roughly
   (a) 50 : 1 (b) 25 : 1
   (c) 15 : 1 (d) 1 : 1
3. A two-stroke engine has
   (a) two ports (*b*) three ports
   (c) two valves (d) three valves
4. In a two-stroke engine, one power stroke is obtained in
   (a) one revolution of the crank shaft
   (b) two revolutions of the crank shaft
   (c) four revolutions of the crank shaft
   (d) none of the above
5. In a four-stroke engine, one power stroke is obtained in
   (a) one revolution of the crank shaft
   (b) two revolutions of the crank shaft
   (c) four revolutions of the crank shaft
   (d) none of the above
6. In a two-stroke engine, the gas exchange process is called
   (a) charging (b) scavenging
   (c) combustion (d) none of the above
7. In a four–stroke Diesel engine, during suction stroke
   (a) fuel–air mixture is inducted
   (b) only fuel is inducted
   (c) only air is inducted
   (d) none of the above
8. The function of venturi in the carbureter is
   (a) to decrease the air velocity
   (b) to increase the velocity
   (c) to decrease the fuel flow
   (d) to increase the manifold vacuum
9. Which one of the following is true for a Diesel engine?
   (a) It has high compression ratio.
   (b) It does not have a spark plug.
   (c) It has large noise and vibrations.
   (d) All of the above
10. The air–fuel ratio of a petrol engine is controlled by
    (a) fuel injector (b) fuel pump
    (c) carburetter (d) none of the above
11. In an IC engine, the exhaust pressure in the cylinder is
    (a) equal to the atmospheric pressure
    (b) below the atmospheric pressure
    (c) above the atmospheric pressure
    (d) none of the above

12. In a two-stroke IC engine, the piston top has a deflector for
    - (a) better combustion of fuel
    - (b) better scavenging of exhaust gases
    - (c) better mixing of air and fuel
    - (d) better charging of the cylinder

13. The pumping power is required in the following stroke(s):
    - (a) Suction stroke
    - (b) Exhaust stroke
    - (c) Compression stroke
    - (d) Suction and exhaust stroke

Answers

1. (c) 2. (c) 3. (b) 4. (a) 5. (b) 6. (b) 7. (c) 8. (b)
9. (d) 10. (c) 11. (c) 12. (b) 13. (d)

2 CHAPTER

Fuel and Engine Systems

Introduction

Fuel is the source of energy of the engine. Gasoline, diesel, natural gas, LPG and other oils are used as fuel for engines. Fuel mixes with air to prepare charge outside the engine and introduced in cylinder of spark ignition engines. Diesel engines receive air as charge and fuel is injected near the end of compression stroke.

Apart from fuel feeding, the engine can perform in healthier mode, if its systems like cooling and lubrication are operating in correct manner. Cooling arrangement of engine keeps the temperature of engine parts within operating limits and lubrication system reduces the friction between rubbing parts and improves mechanical efficiency.

2.1 FUELS FOR INTERNAL COMBUSTION ENGINES

A fuel is simply a combustible substance. It burns in the presence of oxygen and releases heat energy. Most commonly liquid, and gaseous fuels are used in automobile and stationary engines. An ideal fuel should have the following properties:

1. High calorific value.
2. Moderate ignition temperature.
3. Low moisture content.
4. Low NO_x emission and high combustible matter.
5. Moderate velocity of combustion.
6. Burn clean and Products of combustion should not be harmful.
7. Knock resistant.
8. Low cost.
9. Easy to transport.

2.2 LIQUID FUELS

Liquid fuels have some physical and chemical properties (e.g. density, calorific value, vapour pressure, chemical formula, etc.) However, most of the times, combustion properties are also assigned to fuels, inspite of the fact that these properties depend on the oxidiser and the actual processes. The some information of liquid fuels:

- Ultimate analysis is used for composition investigation
- Heating value is used for energy content evaluation
- Liquid fuels can be stored in tanks
- Liquid fuels can be delivered via pipelines by means of appropriate pump

Some essential properties of liquid fuel are listed and discussed below:

Point of solidification is the temperature at which the product no longer flows under the effect of the gravitational force. For transportation of fuel, point of solidification value is important. Thus fuel temperature should be kept well above to point of solidification.

Flash point is the temperature at which the vapor generates from the liquid fuel under atmospheric pressure and mixed with the ambient air. When a spark or flame approaches it flashes over the whole oil surface. This value is also used for characterization of explosion and fire protection parameters.

Firing point is the temperature at which vaporization of the liquid is of such extent, that the fuel vapour will continue to burn for at least 5 seconds after ignited by an open flame. The firing point is characteristic to the inflammability of the fuel.

Pour Point is the lowest temperature at which fuel can flow and be pumped.

Volatility is an important property of liquid fuel, which determines its suitability for its use in the engine. Volatility can be viewed as tendency of liquid to vaporize.

Viscosity is the property of the fuel, which allows the flow of fuel through the system and the strainer under the lowest operating temperature of fuel. Its kinematic value is measured in m^2/s.

Heating Value is also called **calorific value**. It is the amount of heat librated by complete combustion of unit quantity of a fuel. It is measured in kJ/kg for solid and liquid fuels or kJ/m^3 for gaseous fuels. It is classified as higher calorific value (HCV) and lower calorific value (LCV).

Higher calorific value (HCV or gross calorific value is obtained, when combustion products are cooled to the reactant's temperature, the water vapour gets condensed and the heat of its vaporisation is recovered.

The *lower calorific value (LCV)* or *net calorific value* is the amount of heat released by complete combustion of unit quantity of fuel, when the vapour carries its heat of vaporisation. It is obtained by deducting the heat necessary to form the vapour from hydrogen. That is, lower calorific value

$$(\text{LCV}) = \text{HCV} - m_v h_{fg}$$

where m_v = mass of water vapour formed per kg of fuel,

h_{fg} = the heat of vaporisation of water at 25°C and 1 atm,

= 2441.5 kJ/kg.K

Thermal Expansion

This property deals with volumetric expansion of fuel with temperature rise. When fuel oil in pipeline heats up even by a few degrees; it expands. This expansion contained within the pipe systems generates tremendous pressure. It is measured in K^{-1}.

Vapor pressure is an important physical property of volatile liquids especially of spark-ignition engine fuels. It is measure of the volatility of gasoline, volatile crude oil, and other volatile petroleum fuels. It provides an idea of how a fuel will perform under different operating conditions. It is an absolute vapor pressure exerted by a liquid at 37.8°C (100°F) as determined by the test method ASTM-D-323. It is measured in kPa.

Starting characteristics: The fuel used in the engine should help the engine to start easily. It requires high volatility of fuel to form a combustible mixture for easy ignition.

Dilution of the lubricating oil in crankcase: As the fuel is splashed in the cylinder, some lubricating oil from the crankcase is also washed away with it. This leads to overall decrease in the quantity and quality of the lubricating oil. To prevent such possibilities, it is important that the fuel used for the engine should vaporize before it undergoes combustion process.

Antiknock qualities of the fuel: During detonation large quantity of fuel burns at multiple locations and releasing a huge amount of heat inside the engine cylinder which excessively increases the temperature and pressure. The fuel should have the tendency to avoid creating the situation of detonation; this quality of the fuel is called the antiknock property of the fuel.

The antiknock property of the fuel depends actually on the self-ignition properties, the chemical composition, and chemical structure of fuel. The fuel for the SI engines must have highest antiknock property, enabling the engine to work with high compression ratios, which in turn leads to higher fuel efficiency and higher power output.

Low sulfur content: Sulfur in fuel creates corrosion on engine parts and fuel line, carburetor parts, injection pumps, etc. Sulfur in fuel also promotes knocking of engine; hence its content in the gasoline fuel should be kept to a minimum level.

2.2.1 SI Engine Fuels

Gasoline is widely used liquid fuel in SI engines. The gasoline is produced by fractional distillation of crude oil in refineries. It contains some undesirable unsaturated compounds. It has boiling temperature of 30-180°C as shown in Fig. 2.1

Liquid fuel cannot be burnt in liquid form. Its fumes ignited very first and force the residual liquid to evaporate and then burn. Gasoline is very volatile and combustible. The some properties of gasoline are listed below:

1. Heating Value HCV = 45.7 MJ/kg, LCV = 42.9 MJ/kg.
2. Density = 750 kg/m^3 (varies from 720 kg/m^3 to 760 kg/m^3 at 20ºC).
3. Vapour pressure 50-90 kPa at 20ºC, typically 70 kPa at 20ºC.
4. Thermal expansion coefficient = 900×10^{-6} K^{-1}
5. Kinematic viscosity = 0.5×10^{-6} m^2/s at 20ºC.
6. Gasoline Carbon Number = 4-12
7. Theoretical air/fuel ratio: A/F = 14.5 kg air per kg fuel.
8. *Octane number* (ON) = 92-98.
9. *Cetane number* (CN) = 5-20. This means the gasoline has a relative large time-lag between injection in hot air and for auto-ignition.

Kerosene Oil

Kerosene oil is obtained from 180°C to 260°C during fractional distillation of crude petroleum. Kerosene is always vaporized before burning. It is used in domestic appliances. It burns with smokeless blue flame in presence of sufficient oxygen. It is used for rural illumination, jet engine fuel, tractor fuel and as additives.

2.2.2 CI Engine Fuels

Diesel oil is used as fuel in CI engines. Diesel oil is obtained from 260°C to 350°C during fractional distillation of crude petroleum. Usually Diesel oil contains 85% carbon and 12% Hydrogen. Diesel is a mixture of aliphatic hydrocarbons extracted from crude petroleum. Diesel consists of longer hydrocarbons and low value of ash, sediment, water and sulphalt contents. Diesel can easily be ignited under compression temperature. The some properties of diesel are listed below:

1. Heating Value: HCV = 47 MJ/kg, LCV = 43 MJ/kg.
2. Density = 830 kg/m^3 (varies from 780 kg/m^3 to 860 kg/m^3 at 20ºC).
3. Vapour pressure. 1-10 kPa at 38ºC.
4. Thermal expansion coefficient = 800×10^{-6} K^{-1}
5. Kinematic viscosity = 3×10^{-6} m^2/s
6. *Cetane number* (CN) = 45. This is a measure of a fuel's ignition delay. It is the time period between the start of injection and start of combustion of the fuel. The larger Cetane numbers having lower ignition delays.
7. Flash-point = 50ºC typical.

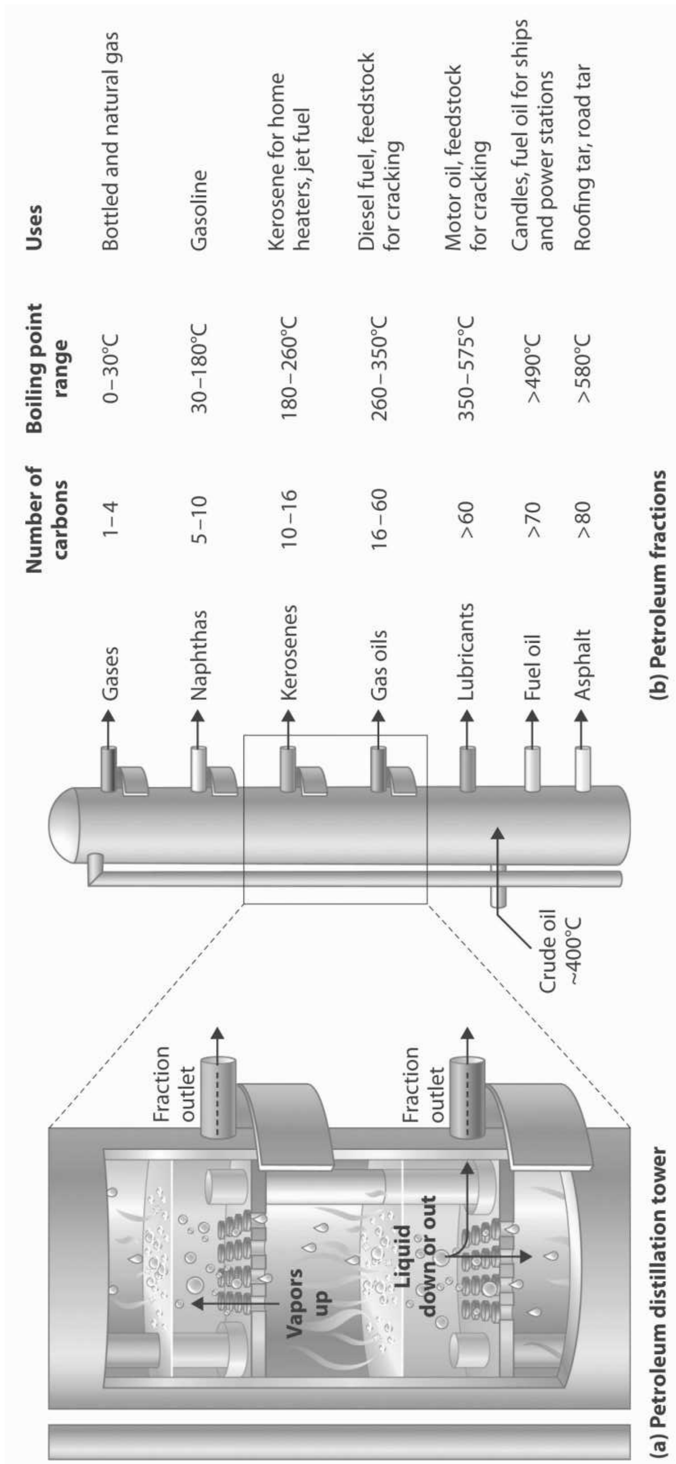


Fig. 2.1 *Petroleum distillation tower*

2.2.3 Biodiesel

Biodiesel is a biomass-derived fuel. It is very similar to conventional fossil diesel. It is the substitute of diesel oil in compression-ignition engines. It is safer, cleaner, renewable, non-toxic and biodegradable.

Biodiesel can be produced from vegetable oils, animal oils/fats, tallow and waste cooking oils.

Biodiesel is a mono-alkyl-ester mixture. It is produced by a process called *transesterification*. It is a reaction process between an alcohol and triglycerides to form esters and glycerol. The schematic of biodiesel production is shown in Fig. 2.2.

Usually 10% methanol (non-renewable) is added, and some 10% glycerol forms.. Biodiesel are longer-chain hydrocarbons than petro-diesel: $C_{13}H_{28}$, $C_{14}H_{30}$, or $C_{15}H_{32}$ etc.

Some important properties of biodiesel are

1. Heating Value: HCV = 40 MJ/kg,
2. Density = 880 kg/m^3 at 20ºC.
3. Thermal expansion coefficient = 800×10^{-6} K^{-1}
4. Kinematic viscosity = 6×10^{-6} m^2/s
5. *Cetane number* (CN) = 60-65.
6. Flash-point = 370-430 K.

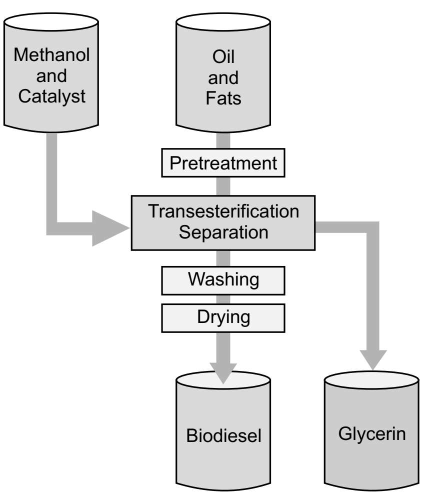


Fig. 2.2 *Production process of biodiesel*

2.3 GASEOUS FUELS

Gaseous fuels are the chemically simplest among the three groups. The two primary sources of gaseous hydrocarbon fuels are natural gas wells and certain chemical manufacturing process.

LPG (liquified petroleum gas) is mainly a mixture of propane and butane. LPG is a mixture of those hydrocarbons which are in gaseous phase at normal atmospheric pressure, but may be condensed to the liquid state at normal temperature, by the application of moderate pressures.

LPG vapour is denser than air, because butane is about twice as heavy as air and propane about 1.5 times as heavy as air. It is used as fuel in SI engines.

Natural gas contains methane as the main constituent and other gases in small amounts. Natural gas has high calorific value and it burns without smoke. It is lighter than air and disperses into air. It is compressed to high pressure and then used as CNG fuel in CI engines.

Producer gas is obtained by partial combustion of coal, coke and charcoal in a mixed air stream. Its manufacturing cost is low and it has a calorific value of about 6500 kJ/m^3. It is used with diesel in CI engines.

Coke oven gas is obtained as a by-product during the formation of bituminous coal. It is used for industrial heating and power generation application. It has a calorific value of about 17,000 kJ/m^3.

Advantages of Gaseous Fuel

1. The supply of fuel can be easily and accurately controlled.
2. The gaseous fuels are free from impurities.
3. The combustion of gaseous fuel is possible with very less excess air.
4. They produce relatively less smoke and pollution.
5. They are capable of attaining very high temperatures.
6. The gaseous fuels are light in weight.

7. Gaseous fuels can be stored in tanks either in gaseous form under high pressure or in liquefied form under medium pressure
8. Gaseous fuels can be delivered via pipelines by means of pressure difference and thus its transportation is very easy.

Disadvantages

1. They require larger storage capacity.
2. They are highly flammable and thus can cause fire hazards.

2.4 FUEL RATING

2.4.1 Octane Number or Octane Rating

Octane number of a fuel, especially for gasoline, is a measure of its anti-knock properties. The Octane number represents a fuel's ability to resist engine knocking. Higher Octane number can permit more compression ratio of engine. Fuels with high Octane numbers are used in high performance gasoline engines. Fuels with low Octane number (or high Cetane numbers) are used in diesel engines, where fuel is not compressed.

Gasoline characteristics that have an important effect on engines and emissions include its Octane number and chemical composition. Usually, gasoline has a Octane number 93 – 97.

Octane number is classified by its *research Octane number* (RON) and *motor Octane number* (MON). Another less common method is the Aviation Method, which is used for aircraft fuel and gives an Aviation Octane Number (AON).

Measurement of Octane Number

Octane numbers are based on a scale of iso-octane (C_8H_{18}) and n-heptane (C_7H_{16}). Iso-octane has minimum tendency to knock, therefore its Octane number is assigned 100. n-heptane has highest possibility to knock and its Octane number is assigned is 0. These two fuels are known as primary reference fuels.

Test engine is a single-cylinder variable compression, overhead valve engine that operates on the four-stroke Otto cycle. The compression ratio of engine can be adjusted from 3 to 30. Test conditions to measure MON and RON are given in Table 2.1.

Table 2.1 *Test conditions for Octane number measurement*

	RON	*MON*
Engine Speed (RPM):	600	900
Inlet Air Temperature (°C):	52 (125°F)	149 (300°F)
Coolant Temperature (°C):	100 (212°F)	100
Oil Temperature (°C):	57 (135°F)	57
Ignition Timing:	13° bTDC	19°-26° bTDC
Spark Plug Gap (mm):	0.508 (0.020 in.)	0.508
Inlet Air Pressure:	atmospheric pressure	
Air—Fuel Ratio:	adjusted for maximum knock	
Compression Ratio:	adjusted to get standard knock	

The Octane number of an unknown fuel is given as percentage of iso-octane in primary reference fuels, that gives same knock intensity. For instance, a fuel that has the same knock characteristics as a blend of 87% isooctane and 13% n-heptane would have an ON of 87. Octane number of a fuel can be extended beyond 100 by adding TEL (tetra ethyl-lead) to iso-octane.

The motor test is carried out under more severe conditions such as high engine speed and higher mixture temperature to determine MON, while research test to determine RON is carried under mild operating conditions that are low engine speed and low mixture temperature.

2.4.2 Cetane Number

The ignition quality of diesel fuel is defined by its **Cetane number** CN. Cetane number of a diesel engine fuel is an indication of its ignition characteristics. Cetane number affects a number of engine performance parameters like combustion, stability, drivability, white smoke, noise and emissions of CO and HC.

For low Cetane fuels the ignition delay is longer and most of the fuels are injected before auto-ignition takes place in chamber and instantaneously burning, builds abrupt pressure rise and produces an audible knocking sound referred to as "**diesel knock**".

For *high* Cetane fuels the ignition delay is short and very little fuel is injected before auto-ignition commences in chamber and hence the heat release rate is controlled by the rate of fuel injection and fuel-air mixing –smoother engine operation. The higher the CN is the better and it has the shorter ignition delay.

Measurement Cetane Number of Fuel

The method used to determine the ignition quality in terms of CN is analogous to that used for determining the antiknock quality using the ON.

The Cetane number scale is defined by blends of two pure hydrocarbon reference fuels: iso-cetane (hepta-methylnonane) and Cetane(n-hexadecane, $C_{16}H_{34}$). The CN value to iso-cetane is assigned CN= 15 and n-hexadecane is assigned as CN =100.

The method developed to determine CN uses a standard single-cylinder variable compression engine with operating conditions:

Table 2.2 *Test conditions for Cetane rating*

Engine parameters	*Standard Value*
Inlet temperature	65.6°C
Speed	900 rpm
injection advance (degree)	13 bTDC
Coolant temperature	100°C
Injection pressure	10.3 MPa

When engine is running at above test conditions on the test fuel, the compression ratio is varied until combustion starts at TDC and for ignition delay period of 13° degree of crank angle.

The above procedure is repeated using blends of Cetane and hepta-methylnonane. The blend that gives a 13 degree ignition delay with the same compression ratio is used to calculate the test fuel Cetane number as

CN = (% hexadecane) + 0.15 (% hepta-methylnonane)

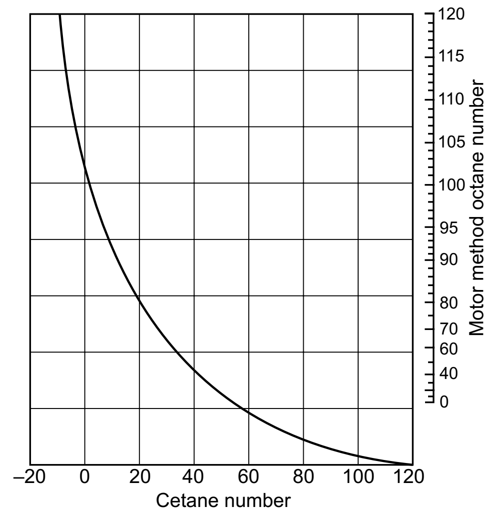


Fig. 2.3 *Cetane and Octane number variation*

The Octane number and Cetane number of a fuel are inversely correlated as shown in Fig. 2.3. Gasoline is a poor fuel in diesel engine and vice versa.

2.4.3 Fuel Additives

These are called dopes. The dopes are substances, which are added in petrol in a very small amount for increasing the Octane rating of fuel. These include benzole, ethanol, methanol, acetone, nitrobenzene and tetra-ethyl lead. These are soluble in petrol. The tetra-ethyl lead (TEL) $Pb(C_2H_5)_4$ is very useful. TEL delays auto-ignition thus leaded fuel enables the high compression ratio compared to that with normal petrol alone. TEL is a heavy liquid and its prolong use in the fuel put lead deposition on the spark plug, exhaust valves and combustion chamber. Therefore, use of leaded petrol has been banned due to its heavy pollutants in the atmosphere.

Equivalence Ratio The equivalence ratio is defined as the ratio of the actual fuel/air ratio to the stoichiometric fuel/air ratio and it is denoted by ϕ. Stoichiometric combustion occurs when all the oxygen is consumed in the reaction, and there is no molecule of oxygen (O_2) present in the exhaust gases.

1. If the equivalence ratio is equal to one ($\phi = 1$), the combustion is stoichiometric.
2. If $\phi < 1$, the combustion is lean with excess air, and
3. If $\phi > 1$, the combustion is rich with incomplete combustion.

2.5 AIR–FUEL MIXTURE

The *air–fuel ratio* is defined as the *ratio of mass of air to the mass of fuel*. The engines operating on different loads and speeds require different air–fuel ratios.

2.5.1 Fuel Feeding System

The fuel system provides the correct air–fuel mixture in the cylinder for efficient burning. There are basically two methods available, one is called *carburetion* and is particularly used for petrol engines, and the other is *fuel injection* used in SI and CI engines. There are various designs of each system, but only the basic principle will be discussed in the chapter.

2.5.2 Types of Mixtures

The fuel and air are mixed to form the three types of mixtures.

(i) Chemically correct mixture (stoichiometric mixture)
(ii) Rich mixture
(iii) Lean mixture

Chemically correct or *stoichiometric air–fuel mixture* is one, which has just sufficient air for complete combustion of fuel. The mass of air required for complete combustion of 1 kg of particular fuel (A/F ratio) is computed from a chemical equation. This computed value for most of the hydrocarbons is usually approximated to be 15.

A mixture which contains more air than the stoichiometric requirement is called a *lean mixture* (example A/F ratio of 17 : 1, 19 : 1, etc.)

A mixture which contains less air than the stoichiometric required air is called a *rich mixture*, example A/F ratio of 12 : 1, 10 : 1, etc.

The air–fuel ratio used in the engine has a considerable influence on the engine performance. The mixture corresponding to maximum power is called the *best power mixture* and its A/F ratio is approximately 13 : 1. The mixture corresponding to minimum fuel consumption per kWh of power is called the *best economy mixture* and its A/F ratio is approximately 16 : 1. Figure 2.4 shows a typical A/F mixture requirement by a carbureted engine at constant speed and full throttle opening.

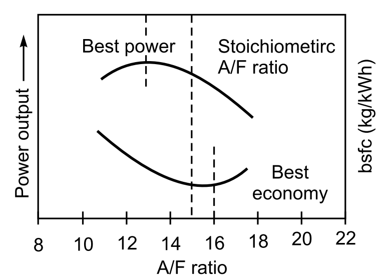


Fig. 2.4 *Variation of power output and bsfc with A/F ratio for an SI engine*

A petrol engine requires a rich mixture for starting and idling (no load condition). When an engine operates at part loads of up to 75% of the designed load, the mixture required is slightly weak. But on acceleration and maximum load conditions, the engines requires a rich mixture (such as A/F ratio of 13).

2.5.3 Mixture Requirement at Various Loads and Speed

Gasoline is used as a fuel in SI engine. Gasoline is mainly iso-octane (C_8H_{18}) for which the chemically correct or stoichiometric mixture of air-fuel ratio is 15 : 1 by mass approximately. This mixture gives most rapid combustion in the engine, almost the great power and reasonable economy of fuel.

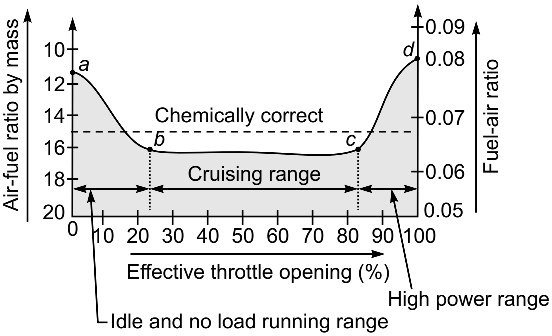


Fig. 2.5 *Mixture requirement at various loads and effective throttle opening*

Rich mixtures give more power in the ratio of 11 : 1 to 14.5 : 1 of air and fuel and or lean mixtures of about 16 to 18 : 1 gives better fuel economy. Rich mixtures having A/F ratio below 11:1 and lean mixtures above 20 : 1 cannot sustain flame and therefore, cannot burn effectively.

There are three general range of throttle operations and their mixture requirements as shown in Fig. 2.6.

(i) Idling Range (a-b) During idling phase, there is no load on engine, throttle valve is almost closed. The pressure in intake manifold is below atmospheric pressure. Very less air enters into the engine. When suction valve opens, the pressure difference between combustion chamber and intake manifold results into backward flow of exhaust gases. Therefore, the fresh charge is diluted with exhaust gases during suction.

Presence of exhaust gases, obstructs the reaction between fuel and oxygen molecules and results into poor combustion. Therefore, during idling range, engine requires rich mixture to ensure contact between fuel and air.

(ii) Cruising Range (b-c) In this range, the exhaust dilution is minimized due to partial opening of throttle valve. The prime object is to obtain fuel economy by loading engine upto 80% of its capacity. Therefore, carburetor provides a mixture with best economy in this range.

(iii) High Power Range During this range of operation, the throttle is wide opened, a rich mixture is required. Due to more fuel burning, more heat is produced and more power is delivered by the engine. Further, enriching mixture also prevent the overheating of exhaust valve and suppress detonation.

2.6 CARBURETION

The process of preparation of combustible mixture by mixing the proper amount of fuel with air before admission to the engine cylinder is called *carburetion*. A device which atomises the fuel and mixes it with air and prepares the charge for Otto cycle engine is called carburetor.

When the piston descends in the cylinder, it creates vacuum, the charge is drawn through the carburetor into the cylinder through intake valve and induction manifold as shown in Fig. 2.6.

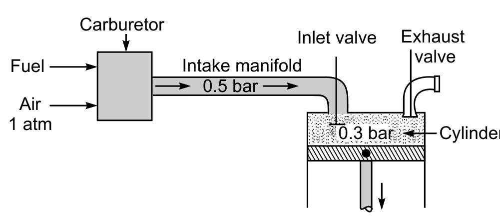


Fig. 2.6 *Schematic of carburation system*

2.6.1 Factors Affecting Carburetion

The factors affecting the carburetion process are

(i) the engine speed
(ii) the vapourisation characteristics of fuel
(iii) the temperature of the incoming air, and
(iv) the carburetor design.

2.6.2 Simple Carburetor

The carburetor are used on most SI engines for preparation of combustible air-fuel mixture as charge. Figure 2.7 shows a simple carburetor, which provides an air-fuel mixture for normal range at a single speed. It consists of a float chamber, fuel discharge nozzle and metering orifice, a venturi, a throttle valve and a chock valve.

How it Works The basic carburetor as shown in Fig. 2.7 is built around a hollow tube called a *throat venturi*. The downward motion of the piston creates a partial vacuum inside the cylinder that draws air into the carburetor's throat and past a nozzle that sprays fuel. The mixture of air and fuel produced inside the carburetor is delivered to the cylinder for combustion.

Throttle Valve A throttle valve at the base of the carburetor controls the amount of charge sucked through the engine by the partial vacuum in the cylinder. The driver opens the throttle valve by pressing down the accelerator. When throttle valve opens wider, more air flows through the carburetor, delivering larger amounts of fuel to the engine. The driver can regulate or close the opening of the throttle valve by decreasing pressure on the accelerator.

Venturi The *venturi* is a gradually decreasing cross-sectional hollow tube. The venturi is the narrowing passage of the carburetor's throat. Air rushing through the narrow part speeds up. At the same time, air pressure against the sides of the passage-way decreases, creating a partial vacuum inside the throat. This partial vacuum draws fuel through the nozzle that is injected into the air.

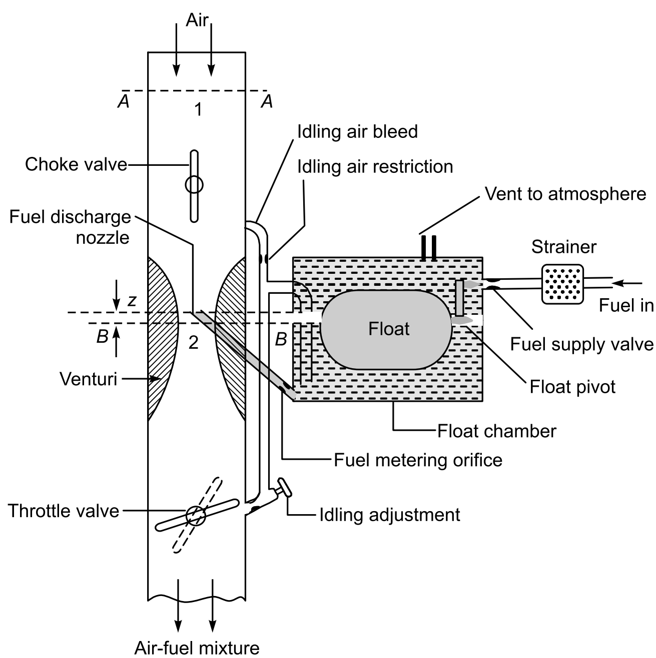


Fig. 2.7 *Simple Carburetor*

Float Chamber The fuel that enters the carburetor is stored in a reservoir called a *float chamber* or *float bowl*. A float and needle valve system maintains a constant level of gasolene in the float chamber. The float floats on the gasolene surface and closes the needle valve. If the level of fuel in the float chamber falls below the designed level, the float goes down, thereby opening the fuel supply valve, and fuel enters the float chamber. As fuel reaches the designed level, the needle closes the fuel supply valve. The fuel level in the float chamber is maintained slightly below the tip of the discharge jet of the nozzle in order to avoid its overflow through the jet, when the carburetor is not in operation.

Idle and Transfer Ports In addition to the main nozzle in the venturi portion of the carburetor, two other nozzles, or ports, deliver fuel to the engine. The idle port is located below the venturi and allows the engine to get fuel when air flow through the carburetor is minimum, such as when the engine is idling at a low speed. Fuel from the idle port is drawn into the cylinder by the engine vacuum. The idle port supplies enough fuel to keep the engine running at slow speeds. Fuel from the main nozzle is necessary to run the engine at normal operating speeds.

Choke and Cold Starts The choke is a device that can partially block air from entering into the carburetor. The throttle valve is almost closed and when the choke is applied (closed), the vacuum from the engine is strong enough inside the carburetor to draw more fuel from all the nozzles. This added fuel produces a rich air-fuel mixture to help a cold engine get started. Once the engine warms-up, the choke is shut off.

Air–fuel Ratio A carburetor can be adjusted to mix larger or smaller amounts of air with the fuel. An idling engine at normal operating temperature requires an air-to-fuel ratio of about 15 : 1 (by weight) to completely burn the fuel. Raising or lowering the air flow makes the mixture either lean (containing less fuel) or rich (containing more fuel). A lean mixture produces a cleaner, hotter combustion for normal speeds, but not enough fuel for starting the engine efficiently or allowing it to produce more power. A rich mixture burns easily in the engine but produces more pollutants as by-products.

The carburetor is adjusted to provide a rich mixture for cold engine starts because the rich mixture burns easier and longer. As the engine warms up, the carburetor alters the air–fuel ratio for a leaner mixture.

2.6.3 Limitations of a Simple Carburetor

A simple carburetor discussed above suffers from the following drawbacks.

1. It can only provide required A/F ratio at one throttle position at constant speed. As speed or throttle position changes, the mixture is either richer or leaner.
2. At a wider open throttle, the air flow rate increases at the venturi throat, the air density and the pressure of air decrease, while the fuel density remains constant. Thus, a simple carburetor provides a progressively rich mixture with increasing throttle opening or speed of engine.
3. It fails to supply a richer mixture at the time of engine start.
4. It is not able to supply a rich mixture during idling.
5. It fails to supply a rich mixture during acceleration and overload.

2.7 COMPENSATING DEVICES IN CARBURETORS

An internal combustion engine runs on different loads and speeds. During such conditions, the carburetor must be able to supply nearly constant air-fuel ratio mixture that is economical. However, the tendency of a simple carburetor is to produce progressively richer mixture as the throttle starts opening. The main metering system alone will not be sufficient to take care of the load on the engine. Therefore, certain compensating devices are usually added in the carburetor along with the main metering system to supply a mixture with the required air-fuel ratio. These are

(i) Air-bleed jet
(ii) Compensating jet
(iii) Emulsion tube
(iv) Back suction control mechanism
(v) Acceleration pump system
(vi) Economizer and power enrichment system

In all modern carburetors, the automatic compensating devices are provided to maintain the desired air-fuel mixture at all speed ranges. The type of compensation mechanism used determines the metering system of the carburetor. The principle of operation of the compensating devices is discussed briefly in the following sections.

2.7.1 Air-bleed Jet

Figure 2.8 illustrates a principle of an air-bleed system in a typical modern downdraught carburetor. Air passage is built into the main nozzle. An orifice restricts the flow of air through this bleed and therefore, it is called restricted air-bleed jet. When the engine is not in operation, the main jet and the air bleed jet will be filled with fuel. When the engine starts, initially the fuel starts coming through the main jet as well as the air bleed jet. As the engine picks up, only air starts coming through the air bleed jet and mixes with fuel resulting into air fuel

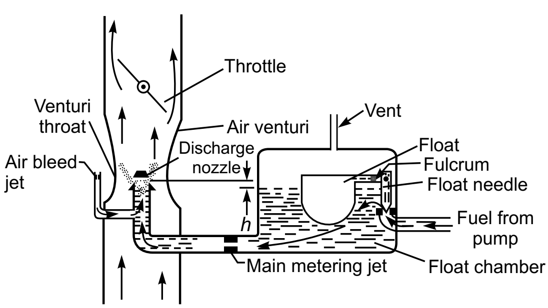


Fig. 2.8 *Air bleed jet system in a carburetor*

emulsion. Thus the rate of fuel flow is augmented and more fuel is sucked at low suctions. By proper design of air bleed jet, it is possible to maintain a constant air-fuel ratio for the entire power range of the operation of an engine.

2.7.2 Compensating Jet

The provision of compensating jet is shown in Fig. 2.9. The main Jet A is fed from the float chamber in the normal way and it supplies its own potion of air-fuel mixture that becomes increasingly rich with increasing engine suction (choke pressure depression). The compensating Jet C on the other hand is fed from the fuel well D through a restricted orifice, which is in direct communication with the atmosphere.

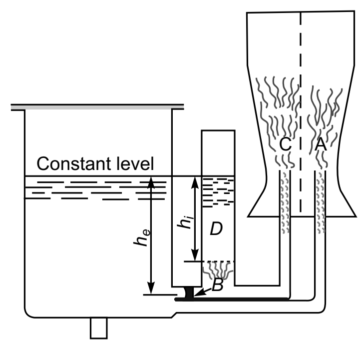


Fig. 2.9 *Compensating jet in carburetor*

With the increase in airflow rate, there is decrease of fuel level in the compensating well, that results into decrease in fuel supply through the compensating jet. The compensating jet thus progressively makes the mixture leaner as the main jet progressively makes the mixture richer. The main jet fuel supply rate and the compensating jet fuel supply rate are more or less reciprocals of each other.

2.7.3 Emulsion Tube

Usually modern carburetors consist of submerged jet. The main metering jet is kept at a level of about 25 mm below the fuel level in the float chamber. The sides holes in the submerged jet are in communication with the atmosphere. At the start of engine, the level of fuel in the float chamber and the well is the same. When the throttle is opened, the pressure at the venturi throat decreases and fuel is drawn into the air stream. This results in progressively uncovering the holes in the emulsion tube leading to increasing air-fuel ratios. When all holes have been uncovered, the normal flow takes place from the main jet. The air is drawn through these holes in the well, and the fuel is emulsified and the pressure difference across the column of fuel is not as high as that in simple carburetor. Figure 2.10 shows the principle of emulsion of fuel.

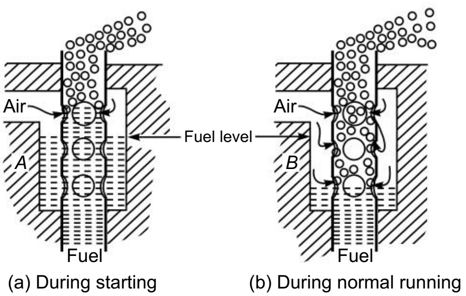


Fig. 2.10 *Emulsion tube in a carburetor*

2.7.4 Acceleration Pump System

The purpose of the accelerator pump system is to provide the momentary extra fuel to the engine under a rapid acceleration or overloading. This extra fuel is instantaneously need of engine because when the throttle is suddenly opened wide, there is corresponding increase in air flow into venturi. But due to inertia of liquid fuel, fuel flow does not increase in same proportion as that of air flow.

The pump comprises of spring loaded plunger and necessary linkages as shown in Fig. 2.11. When the operator of the motor vehicle presses the accelerating pedal down, the rod which connects the throttle lever to the accelerator pump is forced upward causing the accelerator pump plunger to be moved downward through the fulcrum into the pump well. The pump plunger is in turn forced an additional jet of fuel at the venturi throat.

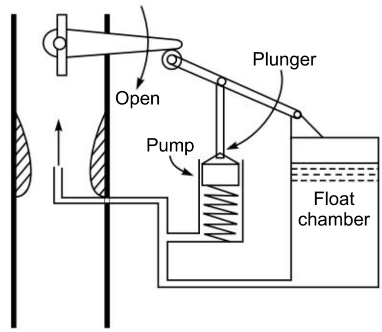


Fig. 2.11 *Accelerating pump system*

2.7.5 Economizer and Power Enrichment System

The provision to supply leaner and rich mixture is provided in all modern carburetors as shown in Fig. 2.12. An economizer system consists of needle valve that fits in orifice hole. The needle valve remains closed during normal running conditions. Economizer valve monitors the additional fuel supply during full throttle operation. The economizer system adjusts the air fuel mixture from minimum economy to maximum power. The needle valve allows an additional fuel to supply rich mixture, when throttle is opened beyond a specified limit. The rich mixture is required, when engine is operated nearly by full load (80% to 100%).

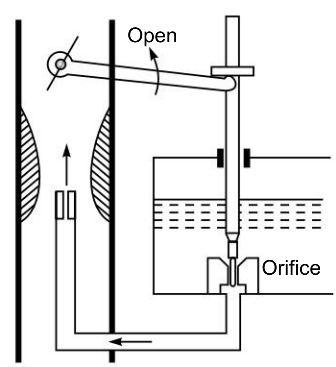


Fig. 2.12 *Economizer and power enrichment system*

2.8 TYPES OF CARBURETORS

Various types of carburetors are used to match fuel flow to engine requirements and are classified as

1. On the basis of direction of flow
   (i) Up-draught,
   (ii) Downdraught, and
   (iii) Cross draught
2. On the basis of venturi type.
   (i) Constant choke carburetor, and
   (ii) Constant vacuum carburetor.
3. Multiple Venturi Carburetor

2.8.1 Up-draught Carburetor

Figure 2.13(a) shows up-draught carburetor in which the air enters at the bottom and leaves at the top so that the direction of its flow is always upwards. It lifts the sprayed fuel droplet upward by air flow. Its design should lift and carry the fuel particles along with air even at low engine speeds. Otherwise, there may be possibility to separate out fuel droplets from air providing only a lean mixture to the engine.

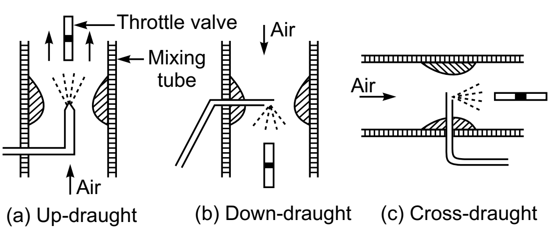


Fig. 2.13 *Types of carburetors*

2.8.2 Down-draught Carburetor

Figure 2.13(b) shows down-draught carburetor. Carburetor is placed at a level higher than the inlet manifold and in which the air and mixture usually follow a downward flow. In this carburetor, the fuel flows under the gravity effect and does not have to be lifted as in the up-draught carburetors. Mixture strength remains uniform even at low speed. Hence, the mixing tube and throat can be made large which permits high engine speeds and high specific engine outputs.

2.8.3 Cross-draught Carburetor

A cross-draught carburetor consists of a horizontal mixing tube with a float chamber on one side as shown in Fig. 2.13(c)]. In the cross-draught carburetor, right-angled turn in the inlet passage is made smooth in order to reduce the resistance to fuel flow.

2.8.4 Constant Choke Carburetor

The air and fuel flow areas are always kept same, but vacuum or depression in the throat is being varied as per the demand on the engine to meet the fuel-air mixture. When air is passed through a choke of fixed size, its velocity and the depression over the fuel jet will vary and mixture is prepared according to demands of the engine. Solex and Zenith carburetors have constant choke.

2.8.5 Constant Vacuum Carburetor

In the constant vacuum carburetor, air and fuel flow areas are varied along the flow according to load on the engine, while the vacuum is kept constant. The S.U. and Carter carburetors have constant vacuum in the venturi section.

2.8.6 Multiple Venturi Carburetor

The multiple venturi system uses two or three boost venturi in a carburetor. The arrangement of double and triple venturi is shown in Fig. 2.14.

The boost venturi is positioned concentrically and upstream within the larger main venturi. The discharge edge of the boost venturi is positioned at the throat of the main venturi. A portion of air can pass through boost venturi. The pressure at the exit of the boost venturi is equal to the pressure at the main venturi. The fuel nozzle is positioned at the throat of boost venturi. This arrangement results into following effects:

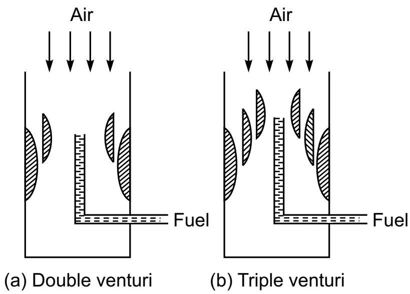


Fig. 2.14 *Double and triple venturi arrangement*

(a) High depression is created at the region of fuel nozzle. Hence better control over the fuel flow rate and improved fuel atomization.
(b) Fuel jet is covered by annular layer of air that prevents the fuel drops away from walls of induction system.
(c) At boost venturi throat velocity of air is always high as 200 m/s.
(d) Excellent low speed full throttle operation is possible.
(e) More efficient mixing of the air and fuel is obtained without significant reduction in volumetric efficiency.

2.9 AUTOMOBILE CARBURETORS

Requirements of air fuel mixture of an automobile engine are discussed in earlier sections. Some popular brand carburetors are discussed below.

2.9.1 Zenith Carburetor

Zenith carburetor is a down draught, constant choke type carburetor as shown in Fig. 2.15. This is the oldest type of carburetor. It was used in all automobile engines before 25 years.

It consists of a compensating Jet C, main jet M, pump well W and emulsion block E. In normal course, the compensating jet C is not delivering fuel into an intermediate chamber formed in the Emulsion block, E.

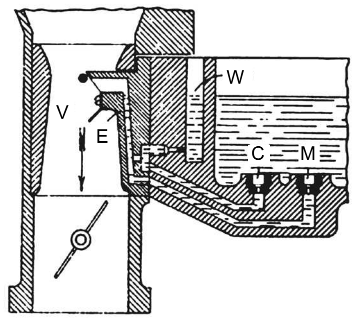


Fig. 2.15 *Zenith Carburetor*

The fuel pressure at an emulsion block E is an intermediate pressure between pump well and choke pressures and it depends on the size of the communicating holes, choke depression and the resistance of the communicating passage in the emulsion block. Therefore, the amount of compensation fuel depends largely on engine suction. The secondary suction effect is intentionally introduced into the choke by chamfering the emulsion outlet.

Zenith carburetor is capable to meet engine requirements at all speeds and load conditions.

2.9.2 Solex Carburetor

Solex carburetor is very reliable and gives excellent performance. It is a constant choke type carburetor. It is down draught carburetor. This carburetor is capable to supply richer mixture for starting and idling, leaner mixture during normal running conditions and richer mixture during accelerating and overloading. Accordingly, it consists of various fuel circuits such as starting, idling, normal running and acceleration etc. The systems of Solex carburetor are shown in Fig. 2.16.

It consists of following features:

(a) Cold Starting and Warming Mode Bi-starter provided in the Solex carburetor is an unique feature that is very useful to start engine in cold conditions. It consists of flat disc (9) with holes of different sizes, petrol jet (10) and starter air jet starting passage (11) just below the throttle valve and starting lever (12) is operated by flexible cable from the dashboard control.

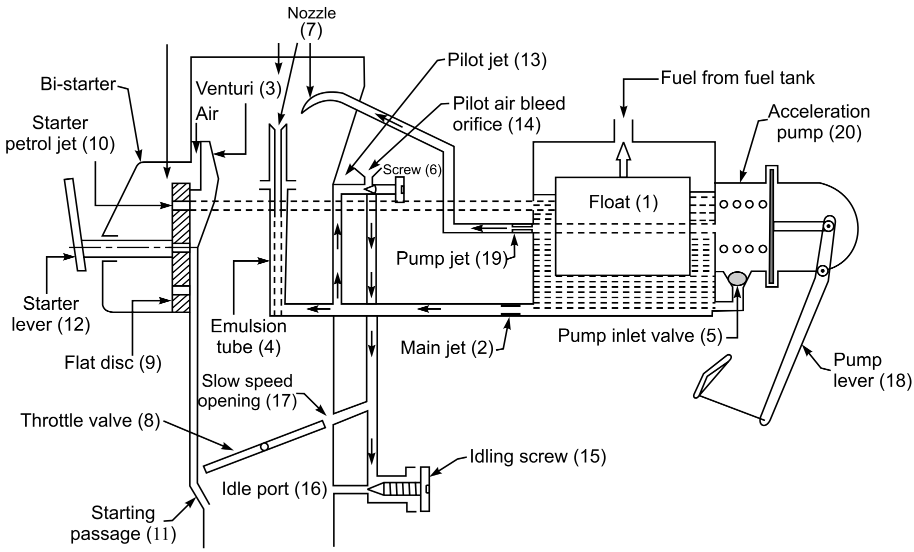


Fig. 2.16 *Solex carburetor*

During starting of engine, the throttle valve (8) is almost in closed position; whole of engine suction is applied on starting passage (11). The fuel jet connects the bigger hole on the disc (9) and rich mixture is prepared with air and comes through air passage and supplied to engine just below the throttle valve.

After start of engine, starting lever is released and its holes do not match with fuel jet and mixture supply through starting passage is stopped.

(b) Idling and Slow Running Mode During idling condition, the throttle valve (8) is almost closed. Vacuum is created by engine during suction stroke acting on idle port (16). The pilot jet (13) and pilot air bleed orifice (14) come in operation and idling mixture is supplied to engine through idle port (16) below the throttle valve. Idle running mixture regulation is possible by idle adjustment screw (15) and pilot air bleed screw (6).

For slow speed operation, throttle is partially opened and partial idle mixture through slow speed opening (17) and remaining mixture is supplied through emulsion tube (4) and venturi (3).

(c) Acceleration Mode A diaphragm type acceleration pump (20) is incorporated to supply rich mixture during acceleration of engine. When pedal is pressed by foot, the pump lever (18) moves towards left and presses the pump diaphragm. It forces fuel through pump jet (19) and injector nozzle (7). This pump supplies extra fuel through nozzle in addition to fuel supplied by emulsion tube. On releasing the pressure on the pedal, the diaphragm is pulled back and it opens pump inlet valve (5) by creating vacuum and fuel from fuel float chamber enters the pump well.

2.9.3 Carter Carburetor

Carter Carburetor is a multi-jet, down-draught, plain tube type carburetor. It is normally used in jeeps. It is an American make down-draught carburetor.

The fuel enters the float chamber and air enters the carburetor from top. The carter carburetor has triple venturi diffusing type. Three venturies designated as 8, 9 and 10 are arranged one above the other as shown in Fig. 2.17. The choke valve (12) in the air passage remains open during normal running condition.

The multiple venturies pass results into better formation of air fuel mixture and when this mixture

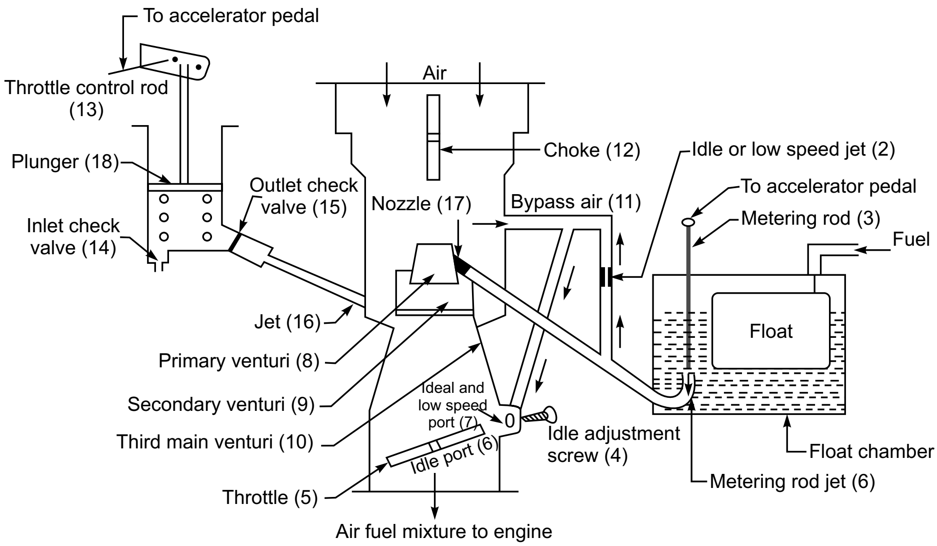


Fig. 2.17 *Carter carburetor*

enters the engine, it gives steady and smooth operation at low speeds.

This carburetor also employs mechanical metering system. The metering rod (3) actuated by mechanism connected with the main throttle. Metering rod is stepper with two more steps of diameter. At top speed the metering rod is completely lifted and maximum amount of fuel passes to mix with the air.

Starting Circuit During start choke valve (12) comes in closed position and whole of the engine suction acts on main nozzle, which then delivers the fuel. Air leaks from the small passage, therefore, quantity of air is small, the mixture prepared is very rich and sufficient to start engine. After start of engine, choke valve is half open due to its spring control and richness of fuel decreases during warming up period of engine.

Idle and Low Speed Circuit During idling operation of engine, rich mixture is required in small quantity. In idling, throttle valve (5) is almost closed. Whole of engine suction acts on idle port (6). Therefore, fuel is drawn through the idle feed jet (2) and the air is bypassed through (11) and rich idle mixture is supplied.

At very low speed, the suction in primary venturi (8) is sufficient to draw the fuel from float chamber. The nozzle (17) is located in primary venturi at an angle, so it delivers the fuel upwards against the air stream ensuring an even flow of fine atomized fuel. The air fuel mixture from primary venturi passes to secondary venturi (9), where it is covered by air blanket and finally enters to third (main) venturi (10), where fresh air is again surrounding the mixture stream coming from second venturi.

Acceleration Pump Circuit Acceleration is required during speeding or overloading of engine. During acceleration throttle (5) is wide open, air enters at high speed. Due to inertia of fuel, the required quantity cannot match the mixture requirement. During acceleration and overloading of engine, a slight rich mixture is required which is supplied by acceleration pump circuit.

The pump consists of a plunger (18) moving inside a cylinder with an inlet check valve (14) and outlet check valve (15). The pump plunger is connected to accelerator pedal by throttle control rod (13).

When throttle is suddenly opened wide by pressing accelerating pedal, pump is also actuated and small quantity of fuel is injected through jet (16). On releasing of accelerating pedal, plunger moves up and allows suction of fuel from float chamber through inlet check valve (14) for next acceleration operation.

2.9.4 SU Carburetor

The SU carburetor is one of the simplest carburetor. It has horizontal venturi and a variable choke orifice type. It has a minimal number of moving parts and easy to tune and maintain its performance. Schematic of a SU carburetor is shown in Fig. 2.18.

A variable choke orifice is obtained in the SU carburetor by the vertical movement of a close-fitting piston (1) positioned above the fuel jet (9) in the centre of the body casting. The tapered metering needle (7) fits into jet. The suction disc (12) is integral with the piston and it moves in a concentric chamber bolted to the top of the body casting.

The choke orifice is varied over wide limits by the movement of the piston throughout the speed range, the fuel jet orifice must also be varied. This is achieved by means of a tapered needle (7) attached to the piston and projecting into the jet. Correct discharge areas are obtained by the accurate dimensioning of this needle.

The variation in the throttle valve (6) opening allows the change in manifold depression and that is to be communicated to the body of the carburetor and also to the suction chamber located above suction disc. The piston moves up, allowing a mixture of air and fuel to pass under it to relieve the depression. The piston moves continue to rise until the depression has reached a value which is just sufficient to balance the weight of the piston, together with the load exerted by the piston spring.

It will be appreciated that approximately the same depression can be obtained whatever the demand and that the piston height will be governed by the mass of mixture flowing beneath it. This depression is arranged to be of sufficient value to ensure that good atomization is obtained, but small enough to ensure adequate engine filling at high speeds.

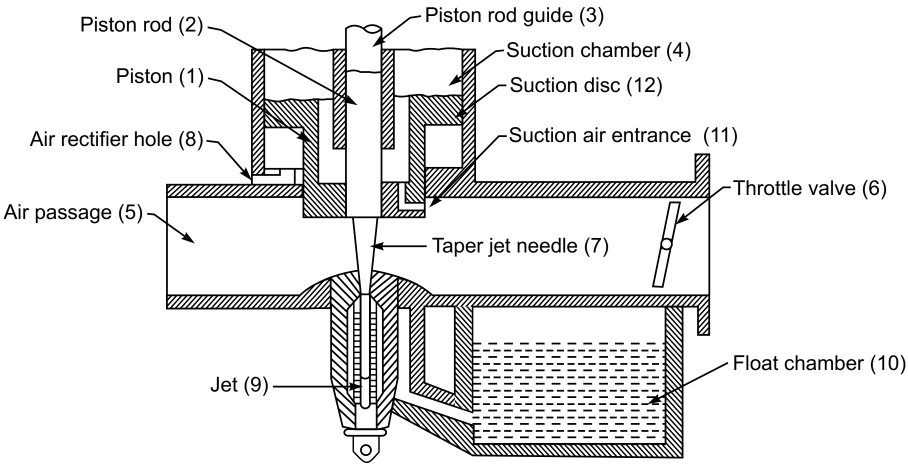


Fig. 2.18 *SU carburettor*

2.10 ADVANTAGES AND DISADVANTAGES OF CARBURETION SYSTEM

Advantages

1. Carburetor is cheaply available and its parts are not expensive.
2. Modern carburetors can supply air and fuel mixture required for all range of operation of engine.
3. In terms of road test, carburetors are more suitable for more power and precision.
4. Carburetors can easily tuned for required air fuel ratio.
5. Carburetors require negligible maintenance.

Disadvantages

1. The mixture supplied by a carburetor at a very low speed is very weak and its combustion becomes erratical.
2. The mixture strength prepared by carburetor is affected by changes of atmospheric pressure.
3. It gives the proper mixture at only one engine speed and load, therefore, suitable only for engines running at constant speed.
4. The fuel consumption in carburetors is more than fuel injectors.
5. Use of carburetors lead to more air emissions than fuel injectors.
6. In cold climate there may be possibility of *ice formation* in the carburetor and it is very difficult to start engine, since there is no heating arrangement.
7. Sometimes, there is problem of *vapour lock* in carburetors due to high volatile fuel used.
8. Occasionally, there is possibility of *backfiring* or popping in the carburetor.
9. There is an uneven distribution of mixture quantity and quality in multi-cylinder engine.
10. There is possibility of loss of volumetric efficiency due to restriction in free flow passage for mixture through venturi tube.
11. In tilted position of carburetor, surging of fuel is caused in float chamber.

2.11 FUEL INJECTION SYSTEM IN SI ENGINE

Apart from the several disadvantages of carburetion system, the fuel injection in SI engine is an alternative. Fuel injection is getting popularity on modern vehicle with multi-cylinder engine. Fuel injection is a system for mixing fuel with air in an internal combustion engine. An overview of fuel injection system is shown in Fig. 2.19.

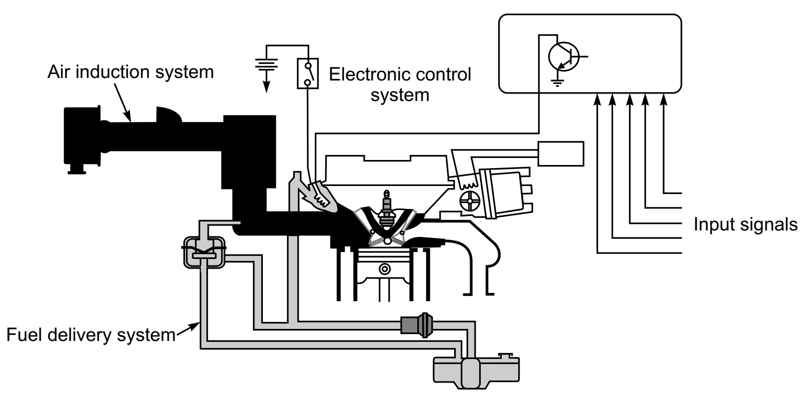


Fig. 2.19 *Overview of electronic fuel injection system*

Advantages of Fuel Injection

Fuel injection system has better control on combustion due to

(i) The fuel speed at the point of delivery is greater than the air speed, therefore, fuel is properly atomized into very fine droplets.

(ii) Proper spray pattern to ensure rapid mixing of fuel and air.

(iii) Accurate metering of the fuel injected per cycle: The quantity of the fuel metered should vary to meet changing speed and load requirements of the engine.

(iv) Timing of fuel injection can precisely monitored in the cycle to obtain maximum power ensuring fuel economy and clean burning.

(v) Proper control of rate of injection: The amount of fuel delivered into the air stream going to the engine is controlled by a pump which forces the fuel under pressure.

(vi) Precise fuel distribution between cylinders

(vii) The desired heat - release pattern is achieved during combustion.

(viii) Uniform distribution of fuel droplets throughout the combustion chamber.

(ix) To supply equal quantities of metered fuel to all cylinders of multi-cylinder engines.

(x) No lag during beginning and end of injection i.e., to eliminate dribbling of fuel droplets into the cylinder.

(xii) Low initial cost and low maintenance cost.

(xiii) Use of sensors to monitor operating parameters gives accurate matching of air/fuel requirements: It improves power output and reduces fuel consumption and emissions.

(xiv) Fuel transportation in manifold is not required so no wall wetting.

(xv) Fuel surge during fast cornering or heavy braking eliminated.

(xvi) Adaptable and suitable for supercharging (SPI and MPI).

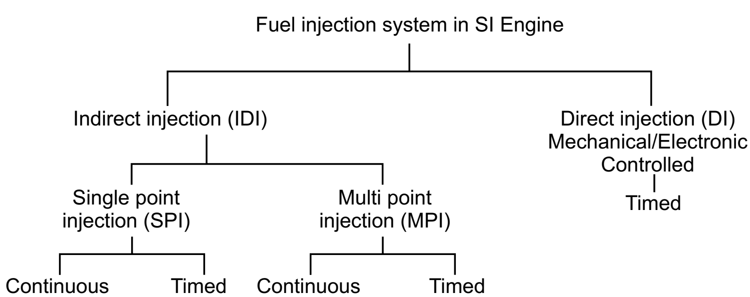
(xvii) Increased power and torque outputs.

Therefore, the several competing objectives are achieved with fuel injection system, such as:

1. Better power output from engine;
2. Fuel economy;
3. Low emissions from engine;
4. Reliability of uniform mixture in the cylinder;
5. Smooth operation of engine;

2.12 CLASSIFICATION OF FUEL INJECTION SYSTEMS IN SI ENGINE

The fuel injection systems in SI engines are classified as



2.12.1 Single Point Fuel Injection System

It is also called single point throttle body injection (TBI). Very first, the carburetors were replaced with **throttle body fuel injection systems.** It incorporates electrically controlled fuel-injector valves into the throttle body.

The single point fuel injection system can easily retrofit as replacement of carburetor in the engine without any drastic changes to design of the engines.

It is an electronic fuel injection system that uses a single injector or pair of injectors mounted in a centrally located throttle body. The throttle unit resembles a carburetor except that there is no float chamber and metering jets. Fuel is sprayed directly into the throttle bore by the injector. Figure 2.20 shows single point fuel injection system. It consists of

1 – Fuel Entrance
2 – Air intake
3 – Throttle valve
4 – Admission
5 – Injector
6 – Engine

The single point fuel injection system uses sensors and actuators.

Sensors

(i) Inductive pickup-position of the crankshaft and speed of the engine;
(ii) Lambda probe;
(iii) Temperature sensors for cooling water system and air intake system;
(iv) Throttle potentiometer.

Actuators

(i) Fuel injector, and
(ii) Throttle valve control

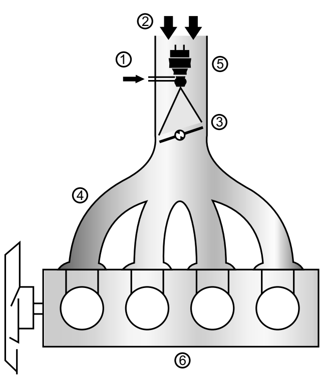


Fig. 2.20 *Single point fuel injection system*

Its salient features are

- Only one fuel injector;
- The system pressure is not dependent on the intake air pressure;
- Reduced fuel consumption - precise adaptation of engine changing conditions;
- Improved performance through greater latitude of the intake region;
- The large distance of heat-stressed parts leads to fewer steam bubbles and a cheaper delivery pump.

2.12.2 Multipoint Injection System

It is also called multi-point port fuel injection or indirect multi-point injection (IMPI). In the Multipoint Injection System, one injector per cylinder is assigned to inject the fuel into the admission port which admits the fuel and air into the cylinder as shown in Fig. 2.21. This gives an individual control on this cylinder, improving the fuel consumption in relation of the single point injection. The advantages of multi-point fuel injection system are

(i) Increased power and torque,
(ii) Improved volumetric efficiency,
(iii) More uniform fuel distribution to each cylinder,
(iv) More rapid engine response to change in throttle position
(v) More precise control of equivalence ratio during the cold starting and warming up of engine.

The multi-point fuel injection system consists of

1 – Fuel rail
2 – Air
3 – Throttle
4 – Induction manifold

5 – Injectors
6 – Engine

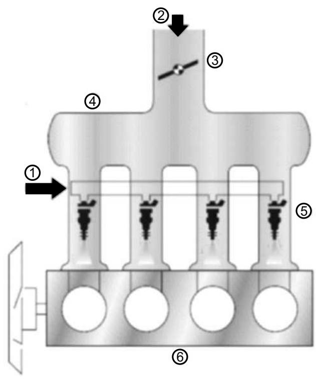


Fig. 2.21 *Multi-point fuel injection system*

2.12.3 Continuous Injection System

In continuous injection system, the injection nozzle and its valve is permanently open during engine operation. Fuel is injected in the air stream continuously before entering in the combustion chamber. Fuel spray may be delivered from single point injector or multi-point injector. In the continuous injection system, amount of fuel delivered to the engine is not varied by pulsing the injector on and off. The amount of fuel can either be varied by regulating the metering orifice or fuel discharge pressure or combination of both.

2.12.4 Timed Injection System

Timed injection system in SI engine injects a measured quantity of fuel in certain time. Timing is synchronized with suction stroke. The injector opens at specified time, injects the fuel into the intake manifold or in cylinder head under the pressure. This system can be used with direct and indirect fuel injection system. Input signal decides the length of time, for which injector is kept open.

Modern electronic fuel injection (EFI) systems use multi-point fuel injection system, in which injection time is synchronized with each cylinder suction stroke.

2.12.5 Direct Fuel Injection System

In the direct fuel injection system, fuel is directly injected into combustion chamber. Modern petrol engines are using this system. It is the next generation fuel injection system in SI engines.

2.13 AIR FUEL MIXTURE IN CI ENGINES

In CI engines, irrespective of load condition at a given speed, the engine sucks almost a constant quantity of air. According to load condition, the quantity of fuel is injected into the cylinder, thus A/F ratio varies from cycle to cycle. The overall A/F ratio that is used in a CI engine ranges from 80 : 1 at no load and 20 : 1 at full load. Actually, in a CI engine, the mixture is heterogeneous with different A/F ratios in different areas of the combustion chamber. There may be some pockets where there is only air or fuel, having rich or lean mixture. But there are always certain areas where the A/F ratio is within the combustible limit and combustion starts from such areas and spreads over the combustion chamber.

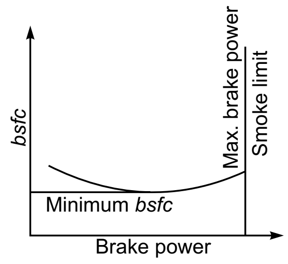


Fig. 2.22 *The bsfc against BP for CI engine*

For a CI engine, the amount of fuel required varies directly with load on the engine, thus the curve of *bsfc* remains almost horizontal with brake power as shown in Fig. 2.22. Figure 2.23 shows the gross fuel consumption against brake power. It indicates that the fuel consumption of a CI engine increases as load increases.

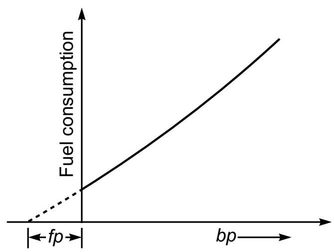


Fig. 2.23 *Fuel consumption against brake power for a CI engine*

2.14 FUEL-INJECTION SYSTEMS IN CI ENGINES

The fuel-injection system is the most important component in the working of a CI engine. The engine performance, i.e., power output, fuel economy, etc., depend on the effectiveness of a fuel injection system. A typical fuel injection system functions in the following ways.

1. It measures the correct quantity of fuel to be injected per cycle.
2. It injects fuel in the cylinder at the correct time.
3. It controls the rate of fuel injection.
4. It atomizes the fuel into very fine droplet-like spray.
5. It gives a proper spray pattern to the fuel droplets in order to mix it into air in a short period.
6. It supplies equal quantity of fuel to all cylinders in case of a multicylinder engine.

2.14.1 Classification of Fuel-injection Systems

In CI engines, two methods of fuel injection are used:

(a) Air injection system, and
(b) Solid injection system.

(*a*) *Air-injection System* This system is also called *indirect injection* (IDI) system. In this method, the fuel is injected into the cylinder by means of compressed air. The air is compressed in a compressor to a pressure higher than that at the end of the compression stroke. Then air and fuel are injected through the fuel nozzle into the engine cylinder. The advantages and disadvantages of air injection systems are given below.

Advantages

1. It provides better atomisation and distribution of fuel.
2. Inferior-quality fuel can also burn efficiently.
3. The charge burns completely and the engine gives better thermal efficiency.
4. The chance of choking of the fuel valve is negligible due to compressed air supply.

Disadvantages

The air-injection system is little used nowadays due to the following reasons:

1. It requires multistage compression for high pressure of air. The large number of parts, the intercoolers, etc., make the system heavy, complicated and expensive.
2. Separate mechanical linkage is required for operation of fuel valve and compressor.
3. Due to large number of moving parts, the mechanical efficiency of the engine reduces.
4. More space is required for the engine.
5. Charge burns in the combustion chamber very near to the injection nozzle, which may lead to overheating and burning of the valve and its seat.

(*b*) *Solid-injection System* It is also called *airless injection system or direct injection* (DI) system. In this system, the liquid fuel is directly injected into the combustion chamber without use of compressed air. It is also called *mechanical injection*.

Advantages

1. It is compact and simple in construction.
2. It does not require compressed air.
3. It has better control on the quantity of fuel to be injected.

Disadvantages

1. It requires very high accuracy in the fuel injector and fuel pump.
2. With this system, inferior quality of fuel cannot be injected.
3. The prepared charge is more heterogenous.

2.14.2 Main Components of Solid-injection Systems

The fuel-injection system consists of mainly the following components.

(i) Fuel tank
(ii) Fuel feed pump to supply fuel from the fuel tank to the injector
(iii) Fuel filter to separate dust and dirt in the fuel
(iv) Fuel-injection pump to meter and pressurise the fuel for injection
(v) Governor to ensure the correct quantity of fuel according to load and speed
(vi) Fuel piping and injectors to take the fuel from the pump and distribute in the combustion chamber by atomizing it into fine droplets

A typical arrangement of various components for a solid fuel-injection system used in diesel engines is shown in Fig. 2.24.

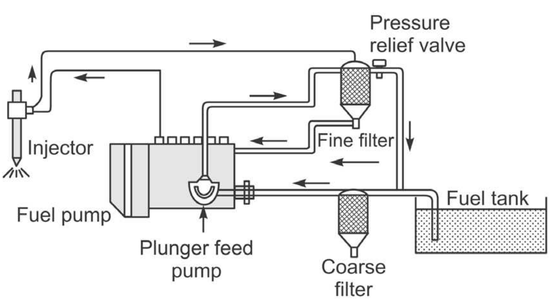


Fig. 2.24 *Typical solid fuel injection system for a CI engine*

The fuel-injection systems for diesel engines employ a high-pressure fuel pump, which increases the pressure of the fuel to about 120 to 200 bar and this fuel is injected through the nozzle(s) into the hot air present into the combustion chamber at the end of the compression stroke.

2.15 SIMPLE FUEL-INJECTION PUMP

The basic principle of a fuel-injection system can be understood with the help of Fig. 2.25. It consists of a spring loaded, plunger-type pump. The plunger is activated through a push rod from the cam shaft.

When the follower on the push rod is at the minimum lift position of the cam, the spring forces the plunger for its lowest position. Thus a suction is created in the barrel and the fuel from the main tank flows into the barrel through the fuel filter. When the cam rotates and reaches its maximum lift, the plunger is lifted upwards, the inlet valve closes and the fuel is forced through the delivery valve.

When the fuel-operating pressure is reached, the fuel from the injector is injected into the cylinder. The spring pressure above the valve rod in the injector is used to set the fuel injection pressure.

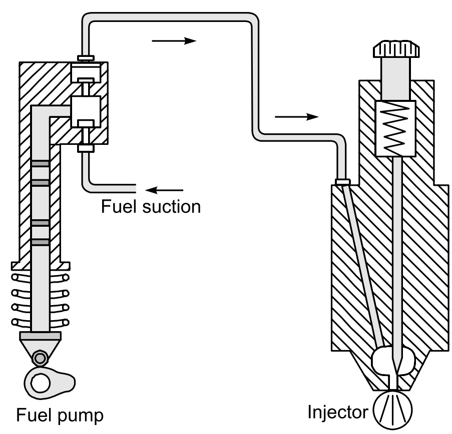


Fig. 2.25 *Schematic diagram of a fuel feed pump*

2.16 JERK-TYPE FUEL-INJECTION PUMP

Nowadays, the '*jerk pump*' fuel-injection system is universally used over the whole range of CI engines. The jerk pump as shown in Fig. 2.26, is a precision equipment and consists of plunger in a barrel, a very close fit. A cam gives vertical movement to the

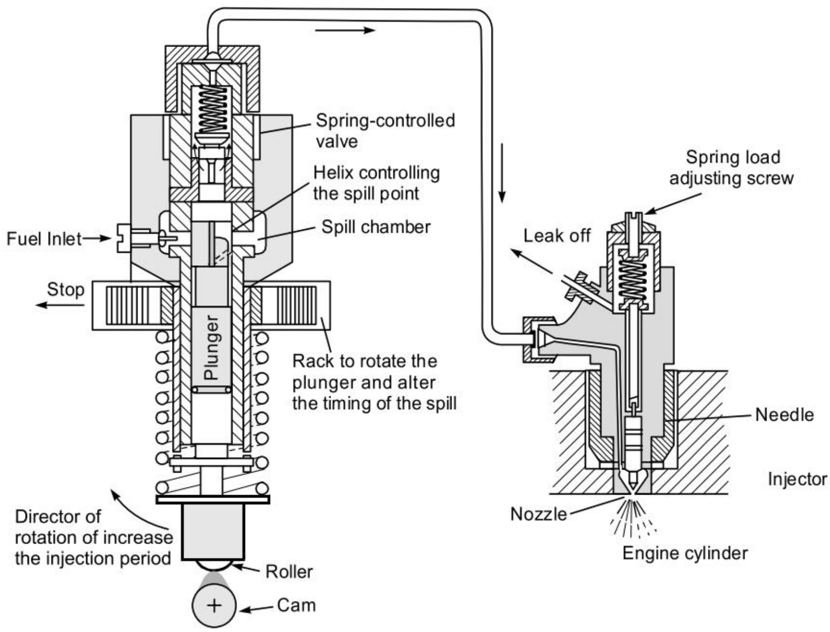


Fig. 2.26 *Fuel injector pump and fuel injector for a CI engine*

plunger, while a rack controls its angular movement in the barrel. The plunger is provided with a helical groove and the barrel has a supply port, spill port and a spring-loaded delivery valve.

This system functions in the following way:

1. It uses a variable stroke of the plunger.
2. It measures the correct quantity of fuel at the beginning of the plunger stroke and spilling back the excess fuel.
3. The axial distance travelled by the plunger in each stroke is the same. The angular rotation of the plunger by rack decides the length of effective stroke for fuel injection.
4. Using the plunger stroke, the rack brings the end of the fuel delivery by suddenly spilling off the fuel from the cylinder.

2.16.1 Construction

The upper part of the helical groove in the plunger controls the uncovering of the spill port. The timing of the opening of spill port by the helix is thus decided by the angular movement of the plunger. The plunger is rotated by the rack, which is moved in or out by the governor. By changing the angular position of the helical groove in the plunger, the length of the stroke, during which fuel is delivered, can be varied and thereby the quantity of fuel to be delivered to the cylinder is also varied accordingly.

The fuel at high pressure passes to the spring loaded injector, where the needle is set to lift at a predetermined pressure in the delivery line.

2.16.2 Operation of Fuel Pump

The operation of a fuel pump as plunger undergoes a stroke is illustrated with the help of Fig. 2.27.

Fig. 2.27 (a) and (b)

1. The plunger is at its down stroke.
2. The fuel inlet port *F* is uncovered and the fuel enters the barrel above the plunger and into the recesses in the plunger, Fig. 2.27(a).
3. As the plunger rises up, it closes the port *F*

4. Fuel flows out through the spill port *S*, because recesses in the plunger align with the port *S* as shown in Fig. 2.27(b).
5. A spring force acts on the check valve and it does not open.
6. The effective stroke ends and no fuel is delivered to the injector.

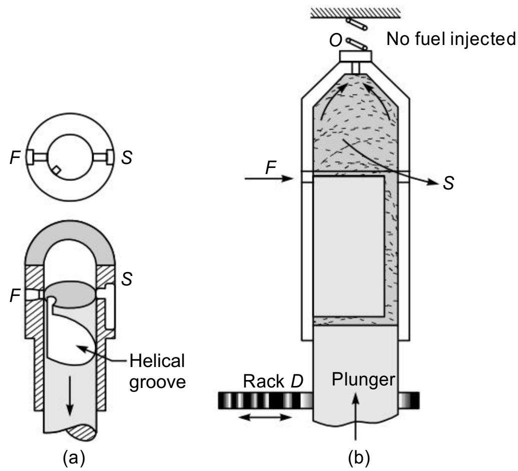


Fig. 2.27 *(a) and (b) fuel pump operation for no load*

Fig. 2.27 (c) and (d), For full load on the engine

1. During the upward stroke of the plunger, the rack rotates the plunger in such a way that the fuel inlet port and spill port, both are closed.
2. The trapped fuel in the barrel is now forced to pass the check valve through the orifice *O* to the injector as shown in Fig. 2.27(c).

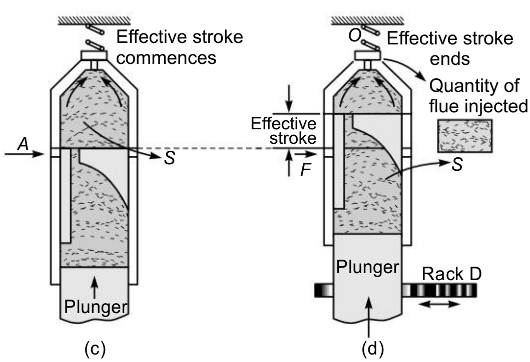


Fig. 2.27 *(c) and (d) for load on the engine*

3. The injection of fuel starts.
4. The injection continues during the upward stroke of the plunger till the helical recess on the plunger uncovers the spill port *S*.
5. The check valve closes the orifice, the fuel injection stops. The effective stroke of the plunger ends as shown in Fig. 2.27(d).
6. The remaining fuel escapes through the spill port.

2.16.3 Fuel Pump Operation for Partial Load

1. During upward stroke, the plunger is rotated by the rack to the position shown in Fig. 2.27(e).
2. The trapped fuel is forced to pass the check valve through the orifice *O*.
3. The fuel injection starts.
4. But for partial load on the engine, the spill port *S* is uncovered sooner as shown in Fig. 2.27(f).
5. The effective stroke is shortened.
6. The quantity of fuel injected is small.

It is to remember that the axial distance travelled by the plunger in each stroke is the same. The angular rotation of the plunger by the rack decides the length of the effective stoke.

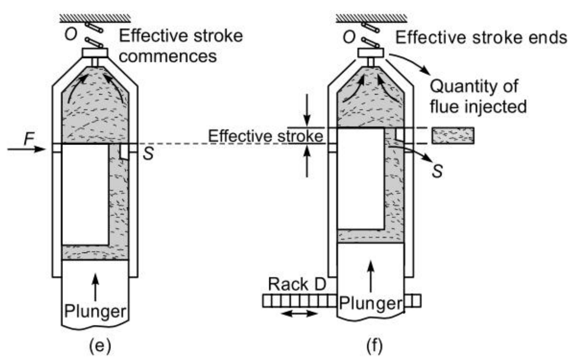


Fig. 2.27 *(e) and (f) fuel pump operation for partial load*

2.17 FUEL INJECTOR

The fuel pump in its effective stroke supplies the fuel to the fuel injector. The fuel injector delivers the fuel under pressure into the combustion chamber. The fuel injector serves to fulfil the following tasks:

1. It atomizes the fuel into fine droplets.
2. It distributes the fuel uniformly by proper spray pattern.
3. It prevents the injection of fuel on the cylinder walls and piston top.
4. It controls the start and stop of fuel instantaneorusly.

A cross-sectional view of a typical fuel injector is shown in Fig. 2.28. The injector assembly consists of

(i) a needle valve
(ii) compression spring
(iii) a nozzle
(iv) adjusting screw with lock nut
(v) an injector body

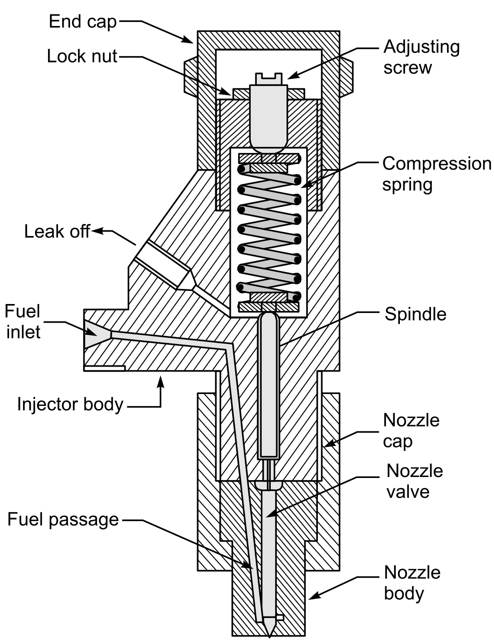


Fig. 2.28 *Fuel injector (Bosch)*

The compression spring exerts the force on the nozzle valve through the spindle to close it. When the fuel is supplied under high pressure by a fuel pump, as it overcomes the spring force, the nozzle valve lifts and the fuel is sprayed into the combustion chamber in finely atomized particles. As the fuel pressure falls, the nozzle valve is pushed on its seat by a spring force. The amount of fuel injected is regulated by the duration of open period of the nozzle valve. The pressure of fuel injection can be adjusted by spring tension by means of an adjusting screw.

2.18 IGNITION SYSTEM

The SI engines use lower compression ratio and self-ignition temperature of gasoline is higher. For initiation of combustion of charge, an ignition system is required. The ignition system provides the high intensity spark needed to ignite the fuel. The ignition system produces, distributes, and regulates electric sparking that ignites fuel vapour in the combustion chambers.

2.18.1 Types of Ignitions Systems

Two broad categories of ignition systems are defined as

1. Battery ignition system
2. Magneto ignition system

The whole ignition system can be divided into *primary* and *secondary* circuits. The primary circuit consists of a battery or *magneto* as a source of primary current, ignition switch, ballast resistance, primary windings, contact breaker points and capacitor. One end of the capacitor is connected to the contact breaker and the other is grounded (to the engine itself).

The secondary circuit consists of a secondary winding (of large number turns of fine wires), distributor and spark plugs.

2.18.2 Components of an Ignition System

An ignition system consists of the following components:

1. Source of electric current—battery or magneto

2. Ignition switch
3. Ballast resistance
4. Ignition coil
5. Contact breaker
6. Capacitor
7. Distributor
8. Spark plug

1. Battery The battery is the source of electrical energy. It produces electrical energy from chemical energy stored in it. A 6–12 V battery is used to supply the voltage to the primary circuit of an ignition system. It is charged by an engine-driven dynamo.

2. Ignition Switch It is an electrical switch which is used to allow and stop the flow of electrical energy from source to primary circuit by turning it on or off.

3. Induction Coil The induction coil is an iron core, wrapped with primary and secondary windings of differently sized wires. The primary winding has 100 to 200 turns of relatively thick wire, whereas the secondary winding has approximately 20,000 turns of fine wire.

4. Ballast Resistance The ballast resistance is an electrical resistance which is provided in series of primary winding to regulate the current in a primary circuit.

5. Capacitor A capacitor is a device that temporarily stores electric charge. In the ignition system, a capacitor helps to produce a sharply defined cut-off current when the breaker points open. The capacitor also absorbs the surge of high-voltage electricity as it moves from the coil to the points. In doing so, the capacitor minimizes arcing across the breaker points when they open, thus increasing their service life.

6. Distributor The distributor serves two primary functions. It routes high-voltage pulses to individual cylinders in the correct sequence and with precise timing. It also houses a mechanical switching system involving breaker points—two contact points that open to interrupt the flow of electric current.

A wire conductor carries the pulses of current from the coil to the distributor, which routes them through other wires to individual spark plugs. The *spark plugs* deliver sparks that ignite the fuel.

2.19 BATTERY IGNITION SYSTEM

The battery ignition system is shown in Fig. 2.29. The battery is used as a source of electrical energy.

When the ignition switch is turned on and the rotating cam makes the contact on breaker points, a 12-V current flows from the battery in *primary winding* through the ignition switch, and the primary circuit is completed through the ground. The magnetic field is set up around the secondary winding.

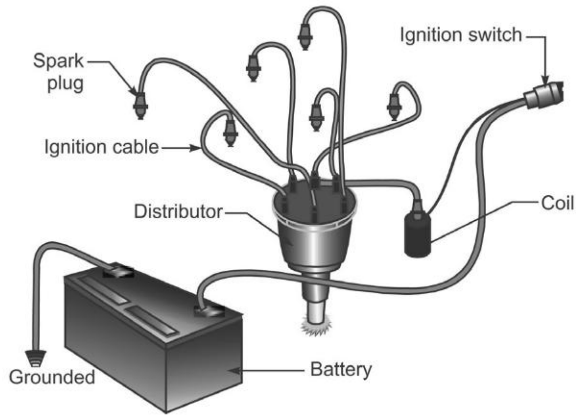
When the rotating cam opens the breaker points, the flow of low-voltage current stops and the magnetic field collapses, inducing a high-voltage surge of about 20,000 volts in the *secondary winding.* This high voltage current passes to the distributor which connects the spark plug in correct sequence, depending upon the firing order of the engine.

Advantages of Battery Ignitions System

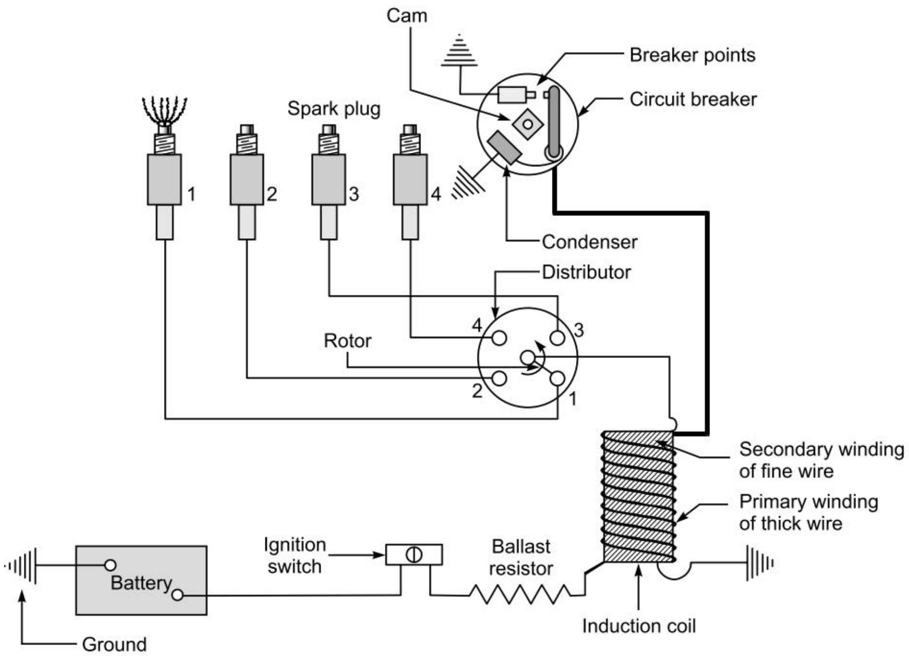
1. It gives constant voltage irrespective of speed of the engine.
2. It gives better spark at low speed and starting of engine.
3. It is reliable and requires very less maintenance, except battery and contact points.
4. The battery is charged by dynamo run by engine.

Disadvantages of Battery Ignition System

1. The system is bulky and occupies more space.



(a) Pictorial view of Battery Ignitions system



(b) Schematic of battery ignitions system

Fig. 2.29

2. At higher speed, the sparking voltage decreases.
3. If battery discharges, the engine cannot be started as induction coil fails to operate.

2.20 MAGNETO IGNITION SYSTEM

A magneto is an electric generator that provides the current for ignition in systems that do not have batteries. It is mounted to the engine. When engine is started, it supplies kinetic energy to the magneto. The magneto then converts this energy to electrical energy. The schematic of magneto ignition system is shown in Fig. 2.30.

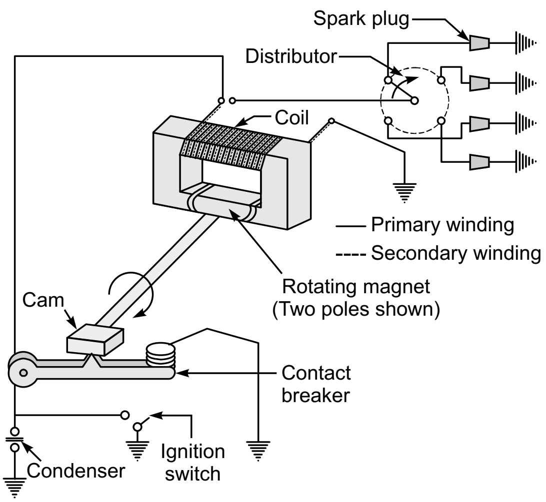


Fig. 2.30 *Magneto ignition system*

The difference between battery and magneto ignition systems is in the source of electrical energy, all other components being the same. A magneto system is popular on motorcycles, racing cars, and a variety of small engines.

The advantages and disadvantages of magneto ignition systems are listed below.

Advantages

1. The system is more reliable due to absence of battery and connecting cables.
2. This system generates secondary voltage according to engine speed, thus it is more suitable for medium and high speed engines.
3. The system is very light and compact, and requires very less space.
4. The spark intensity improves as engine speed increases.
5. The system is much cheaper.
6. It require very less and cheaper maintenance.

Disadvantages

1. The system is not useful at low speed and starting of the engine, because at low speed, it produces poor spark.
2. The powerful spark at high speed may cause burning of electrodes of the spark plug.

2.21 COMPARISON BETWEEN BATTERY IGNITION SYSTEM AND MAGNETO IGNITION SYSTEM

The battery ignition system uses a battery which converts the stored chemical energy into electrical energy. A magneto is an electrical generator which generates low voltage. The graphical comparison of current produced in two systems with engine speed is shown in Fig. 2.31.

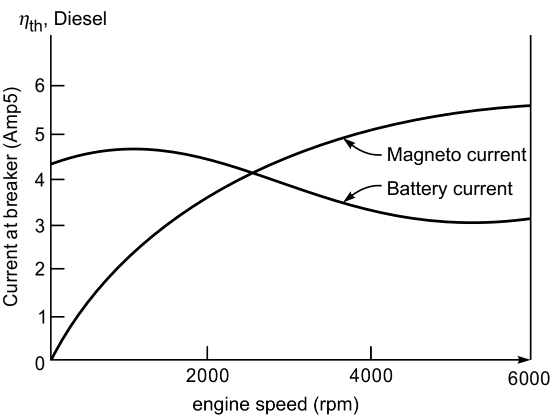


Fig. 2.31 *Speed vs Breaker current for Battery and magneto ignition systems*

It is evident from Fig. 2.31 a magneto ignition system has poor starting current, but as speed increases, the current also increases. For a battery ignition system, the starting current is excellent and it gives good spark at low speeds, but current intensity does not match with speed particularly at high speeds. Therefore, for low speed engines, the battery ignition system is more suitable.

The diffrences between battery ignition systems and magneto ignition systems are given in the table as follows.

Sl. No.	*Aspect*	*Battery Ignition system*	*Magneto Ignition system*
1.	Source of energy	Conversion of chemical energy into electrical energy.	Conversion of kinetic energy into electrical energy.
2.	Primary current	Obtained from battery	Obtained from magneto.
3.	Starting	Easy start, because battery gives good spark.	Poor spark at start and low speed of engine.
4.	Low speed	Good spark, thus no problem.	Poor spark at low speed, thus engine runs eratically.
5.	High Speed	Current for spark decreases as engine speed increases.	Current for spark increases with speed of engine, thus excellent spark at high speed.
6.	Space required	System is bulky and requires more space.	System is light weight and compact, thus less space is required.
7.	Maintenance	System requires more maintenance and it is difficult to start engine when battery is in discharge condition.	Very less and cheap maintenance, due to absence of battery.
8.	Applications	In cars, light commercial vehicles.	On two wheelers and racing cars, aircrafts, etc.

2.22 ELECTRONIC IGNITION

Electronic ignition systems use semiconductors and other solid-state electronic components to switch current flow on and off in the coil, eliminating the need for breaker points. Automobile manufacturers began installing electronic ignition systems in the 1970s and 1980s in an effort to produce cleaner, more efficient combustion in the engines. A few electronic ignition systems are discussed below.

2.22.1 Transistor Ignition System

The breaker-point type of transistor ignition system was developed to replace the conventional ignition system. A conventional ignition system does not perform well at high speed and heavy loads. At high speeds, the breaker points of a conventional ignition system cannot handle the increased current flowing across them without pitting too much. Further, the dwell angle of the breaker points is too small for complete saturation of the ignition coil. The transistor ignition system takes care of both drawbacks. The transistor ignition system is shown in Fig. 2.32

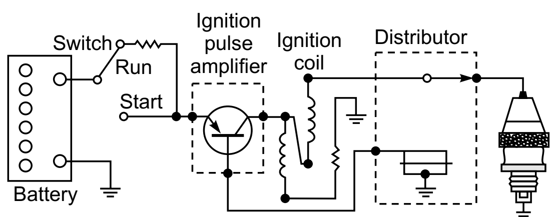


Fig. 2.32 *Transistor ignition system*

The features of transistor ignition system are

1. An electronic ignition system is efficient in operation and allows maximum power and speed and fuel economy to the engine.
2. An electronic type of ignition systems is the best choice, which provides a more uniform spark at a more precise interval.
3. An electronic type of ignition systems promotes more efficient burning of the air/fuel mixture in the combustion chamber, producing less exhaust emissions.
4. It also results into better engine performance and an increased mileage.

5. The increased reliability of electronic ignition allows less frequent maintenance by increasing parts of life.

2.22.2 CDI Ignition Electronic Ignition System

CDI ("Capacitive Discharge Ignition) ignition system is an electronic ignition device. Figure 2.33 shows a CDI ignition system.

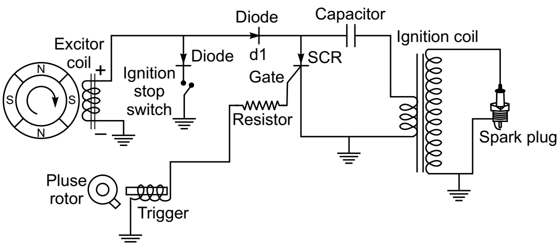


Fig. 2.33 *CDI ignition system*

A CDI ignition system has "capacitor" storage of its own charge and then discharges it through an ignition coil in order to produce a powerful spark from the spark plugs in a petrol engine. Here the ignition is provided by the capacitor charge. The capacitor charges and discharges within a fraction of time making it possible to create sparks. The capacitor charge sends a short high voltage pulse through the coil. The coil acts like a transformer and multiplies the voltage even higher.

Modern CDI coils step up the voltage about 100 : 1. So, a typical 250 V CDI module output is stepped up to over 25,000 V output from the coil. The potential output of CDI coils can be over 40,000 volts. CDI systems are most widely used today on motorbikes, scooters and marine engines.

A CDI system works by passing an electric current over a capacitor. This type of ignition builds up a charge quickly. A CDI ignition starts by generating a charge and storing it up before sending it out to the spark plug in order to ignite the engine.

The big advantage of CDI system is the higher coil output and "hotter" spark. The spark duration is much shorter (about 10-12 microseconds) and very precise. This system performs better at high speed of engine but it can be troublesome at starting with a lean mixture.

2.22.3 Transistor Coil Ignition (TCI) Ignition System

Transistor Coil Ignition (TCI) ignition system is a type of an Inductive Discharge Ignition (IDI). It is an electronic ignition system used in spark ignition internal combustion engines. An ignition system provides a high-voltage spark in each cylinder of the engine to ignite the air-fuel mixture. TCI uses the charge stored in an inductor to fire the spark plug.

An IDI or TCI system builds up a charge in the primary circuit of the ignition coil, that is released at the right moment to spark plug to ignite the charge in the cylinder. The TCI system provides a slightly longer spark duration than in a Capacitive Discharge Ignition (CDI) system. The basic block diagram of a TCI system is shown in Fig. 2.34.

The primary ignition coil is connected to a 12V battery source and charged for a particular dwell time. The circuit is switched ON and OFF by an IGBT switch. When the switch is ON, the current flows from the primary windings in the ignition coil through the switch and it is grounded. The flow of current in the primary winding creates a magnetic field to be formed around the ignition coil's secondary windings. When the switch is turned OFF, the current cannot flow to ground and the magnetic field collapses around the secondary windings of ignition coil. This results into a very high-voltage current in range of 30,000 to 40,000 volts to be induced in the secondary windings of ignition coil. This voltage is strong enough to jump the spark plug gap and generate the high intensity spark.

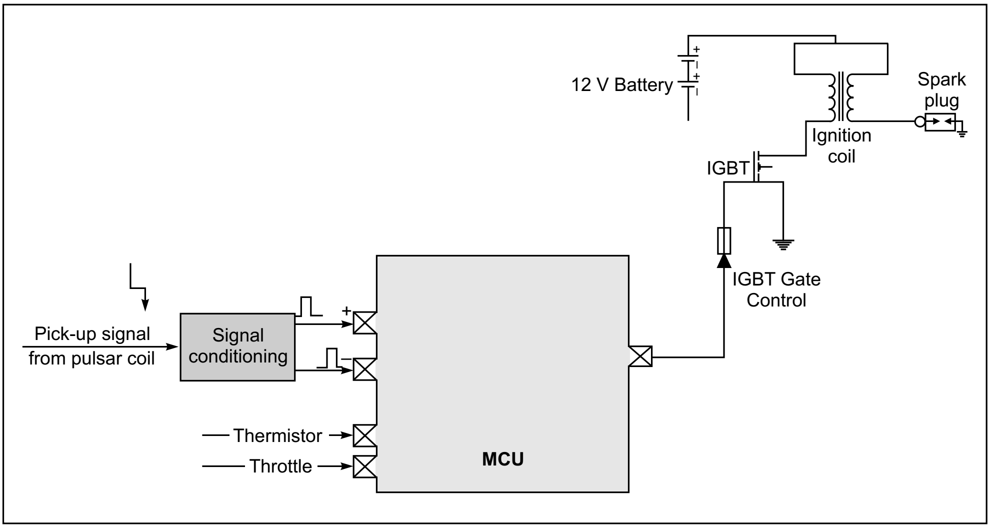


Fig. 2.34 *Basic TCI block diagram*

2.23 SPARK PLUG

The spark plug is a device which conducts a high-voltage current from ignition system and produces spark to ignite the compressed charge into the combustion chamber. The spark plug is shown in Fig. 2.35, mainly consists of

- Central electrode,
- Ceramic (insulated) body, and
- Outer or ground electrode.

The central electrode is surrounded by a ceramic body. It has a threaded base, which is screwed at the top of the engine cylinder. Two electrodes on the base of the spark plug project into the combustion chamber. The upper end of the central electrode is connected to a high tension cable from the distributor. The other electrode is located at a small distance from the central electrode and is welded to a steel shell of the plug and is grounded with the engine body. The gap between the two electrodes is known as *spark gap.* The high-tension current passes through the central electrode and jumps over the spark gap to grounded electrode and produces the high intensity spark to ignite the fuel vapour in the combustion chamber. The intensity of spark is greatly affected by the spark gap.

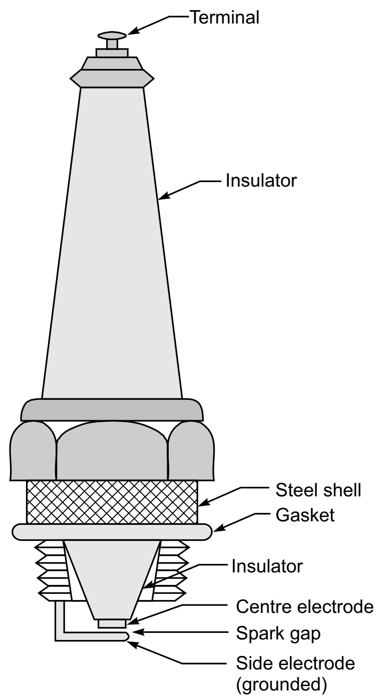


Fig. 2.35 *Spark-plug*

1. If the gap is too large, there may be possibility of no spark due to insufficient high voltage.

2. If the gap is too small, there may be possibility of very less spark, which is insufficient to ignite the charge.

Requirements of a Good Spark Plug

1. It must withstand high temperature attained in the combustion chamber.
2. It must produce a good spark under all working conditions.
3. It must provide suitable insulation between two electrodes to prevent short circuiting.
4. It must maintain proper gap between the two electrodes.
5. It must offer very high resistance to current leakage.
6. It must offer maximum resistance to erosion, burning away the spark points.
7. It must be heat resistant not to offer pre-ignition of charge within cylinder, when it is hot.

2.24 FIRING ORDER

The firing order is a sequence in which the firing takes place in various cylinders of a multicylinder engine. The firing order of a multicylinder engine is arranged to obtain an uniform torque and power by burning of charge, reduced vibrations and balancing of the complete engine. The firing order depends on the number of cylinders in the engine and it may differ from engine to engine. The firing order for some mullicylinder engines are given below:

Number of cylinders	*Firing order*
2-cylinder engine	1, 2
3-cylinder engine	1, 3, 2
4-cylinder engine	1, 3, 4, 2 or 1, 2, 4, 3
6-cylinder engine	1, 5, 3, 6, 2, 4 or 1, 4, 2, 6, 3, 5
8-cylinder engine	1, 6, 2, 5, 8, 3, 7, 4 or 1, 8, 7, 3, 6, 5, 4, 2

2.25 ENGINE-COOLING SYSTEM

2.25.1 Necessity of Cooling

The temperature of gases inside the engine cylinder, due to combustion, reaches a quite high value (more than 2500°C) during a cycle. If an engine is allowed to run without external cooling, the cylinder walls, cylinder liner, and piston will tend to attain the average temperature of the hot gases to which they are exposed, which may be of the order of 1000 to 1500°C. At such high temperatures, obviously, the metal will lose its strength and piston will expand considerably and sieze the liner. Of course, theoretically, the thermal efficiency of the engine will be better without cooling, but in actual practice, this engine will not run longer. If the cylinder wall temperatrue exceeds 265°C, the lubricating oil starts losing its viscosity, starts evaporating, thus tending towards lubrication failure. Also, the high temperature may cause excessive stresses in some parts, making them useless for further operations. Therefore, the internal combustion engines are provided with cooling arrangement, which keeps the engine temperature well within the safe working temperature limits. A typical heat balance for a reciprocating internal combustion engine is shown in Fig. 2.36.

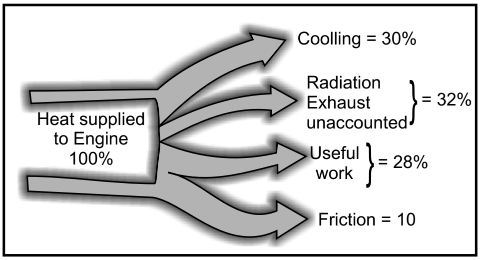


Fig. 2.36 *A typical heat distribution for an internal combustion engine*

2.25.2 Effects of Engine Overheating

The following are the harmful effects of engine overheating:

1. High temperature reduces the strength of the piston and cylinder liner.
2. Overheating may lead to burning of lubricants, thus there may be possibility of lubrication failure and metal to metal contact, thus more heat generation due to friction.
3. The overheating may cause uneven expansion of the piston and cylinder that may lead to piston seizure.
4. The overheated cylinder or piston may lead to pre-ignition of charge in SI engines.
5. As temperature of the cylinder increases, the volumetric efficiency decreases, and hence the power output of the engine is reduced.

2.25.3 Effects of Engine Undercooling

The cooling system in an internal combustion engine should provide adequate cooling but not excessive cooling. However, the excessive cooling is not as harmful as overheating. But undercooling is undesirable due to the following reasons.

1. At low temperature, starting of the engine becomes difficult.
2. At low temperature, there is poor vaporisation of fuel, the combustion is not proper and the engine runs eratically.
3. At low temperature, the viscosity of lubricating oil increases, it offers more frictional resistance, and thus the output of the engine decreases.
4. Undercooling of the engine may change the valve clearance and settings.
5. Overcooling may reduce engine life due to corrosion and carbon deposits.

In general, undercooling affects the economy and life of the engine.

There are two basic types of cooling systems used in reciprocating engines to absorb and dissipate the heat from hot cylinders.

1. Air-cooling system, and
2. Liquid-cooling system

2.25.4 Air-Cooling System

In an air-cooling system, the outer surface of the cylinder and cylinder head is cooled by air flowing over them. To increase the heat transfer rate from the surface, the metallic fins are cast on the cylinder and cylinder head as shown in Fig. 2.37. These fins increase the heat transfer area, and thereby heat transfer rate.

Air cooling system is a very simple, reliable and maintenance-free cooling system, with no operating cost. It is very suitable for small engines of automobiles.

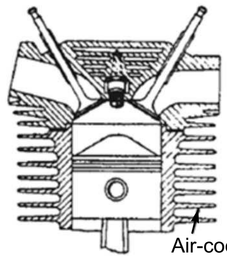


Fig. 2.37 *Air cooling system*

Applications

1. It is used in small engines, i.e., motor cycles, scooters, mopeds, aeroplanes, and combat tanks, where speed of the vehicle gives a good velocity to the air to cool the engine.
2. It is also used in small stationary engines employed for agriculture and industries.

Advantages

1. The design of the engine becomes simpler with an air-cooling system.
2. There is no cooling pipe radiator, fan pump and liquid cooling jacket, and hence the engine has less weight.
3. In an air-cooled engine, the cylinder wall temperature is relatively higher. Thus there is more output from the engine.
4. No danger of coolant leakage, coolant freezing, etc.

5. Installation, assembly, dismantling of the engine are quick and simple.
6. The weight per kW of an air-cooled engine is less than that of a water-cooled engine.
7. The engine is almost maintenance free.

Disadvantages

1. Non-uniform cooling of engine.
2. The compression ratio of the engine is limited due to high wall temperature.
3. The volumetric efficiency of the engine is less than a water-cooled engine.
4. It produces aerodynamic noise.
5. It can be used for only small-sized engines due to its small capacity of heat dissipation.

2.25.5 Liquid-Cooling System

The liquid cooling system is explained with the help of block diagram Fig. 2.38 and operational diagram Fig. 2.39. The water-cooling system mainly consists of a radiator, fan water pump and thermostat.

The water or other coolant solutions flow through the water jacket around the cylinder and cylinder head to absorb the heat. The hot liquid coming out of the water jacket is cooled in the radiator, where circulated air absorbs the heat from the radiator. The cold liquid coming out of the radiator is again pumped to the water jacket for absorbing heat.

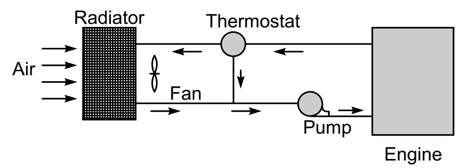


Fig. 2.38 *Schematic of the forced circulation water-cooling system*

1. Pump The pump maintains the water circulation through the water jacket around the engine. The bottom side of the radiator is connected to the suction side of the pump. The pump is mounted on the engine chassis and driven by a crank shaft with a fan belt.

2. Fan The fan is mounted in front of the radiator and is driven by a belt-pulley arrangement. The fan draws air through the spaces between the radiator tube and fins, thus bringing down the temperature of the water flowing in the tubes.

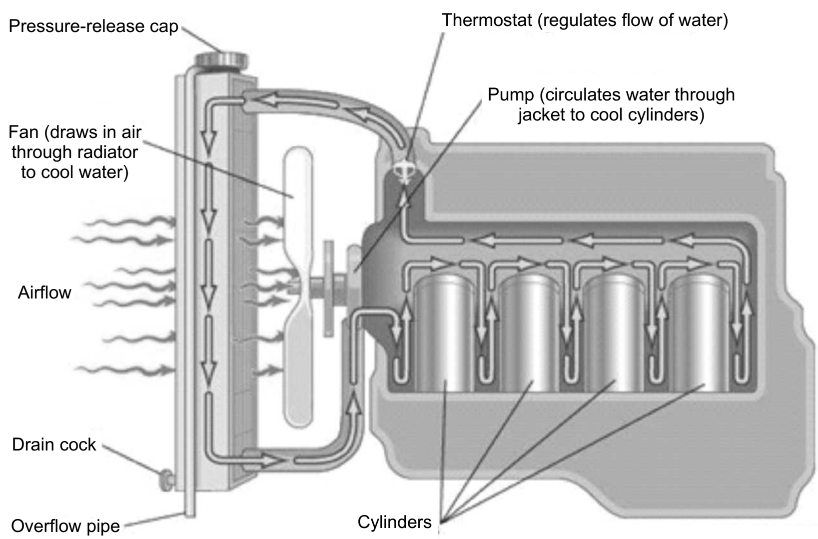


Fig. 2.39 *Components of water cooling system*

3. Water Jacket The water passages between the double walls of the cylinder and cylinder head are called the water jacket. The water jacket is usually cast as an integral part of the cylinder block and head. The jacket covers the entire length of the stroke in order to avoid unequal thermal expansion of cylinder and to prevent the breakdown of lubricating oil film by excessive temperature.

4. Radiator The radiator is basically a compact heat exchanger. It is provided with a large surface area for effective heat transfer. It consists of an upper header and lower header. Between these headers, there is a core of the radiator, which consists of a large number of elliptical or circular brass tubes, pressed into a large number of brass fins. As hot water flows from top to down in the radiator core, it transfers its heat to the radiator fins from where the heat is picked up by circulating air.

5. Thermostat The lower cylinder temperature results into poor performance and rough operation of the engine. In cold starting, if water is circulated through the radiator, as the engine starts, the circulation of cold water in the water jacket brings down the cylinder temperature continuously and the engine will take a long time to reach the safe operating temperature.

The thermostat is an instrument which automatically maintains the preset minimum temperature and permits a quick warm up of the engine after starting. The thermostat is located in the upper hose connection and its opening and closing is controlled by the water temperature in the cooling system. During the warm-up period, the thermostat is closed and the water pump circulates the water through the water jacket only. When the preset operating temperature is reached, the thermostatic valve opens and allows the water to circulate through the radiator. The preset operating temperature range varies from 60°C to 76°C.

6. Pressure Cap In order to avail the advantages of higher boiling temperature of water, the pressurised water is used in the radiator. The pressure is built up within the system due to continuous pumping. A pressure cap is fitted with two valves, a safety valve loaded by compression spring, and a vacuum valve. When the coolant is cold, both valves are closed. But as the engine warms up, the coolant temperature rises and reaches the desired preset pressure. If pressure in the system exceeds the preset value, the safety valve opens and releases some of the gases and liquid to maintain the desired pressure.

2.25.6 Advantages and Disadvantages of Liquid Cooling System

Advantages

1. Efficient cooling as compared to air-cooling system.
2. Fuel consumption of liquid-cooled engines is less than that of air-cooled engines.
3. Liquid-cooled engines require less frontal area.
4. For water-cooled engines, the cooling system can be located conveniently anywhere on the automobile. Some vehicles have it at the rear, while in air cooled engines, it is not possible.
5. Size does not pose a serious concern in the water cooled engine, while in a high-output engine, it is difficult to circulate the correct quantity of air in an air-cooled engine.

Limitations

1. It requires pure water or costly coolant supply for proper functioning.
2. The pump absorbs considerable power and it reduces the output of the engine.
3. In case of failure of cooling system, the engine may get a serious demage.
4. Cost of the system is considerably high.
5. The system requires continuous maintenance of its parts.

2.25.7 Evaporative Cooling System

An evaporative cooling system of an internal combustion engine is mostly used for stationary engines. It uses the principle of evaporation of water by absorbing latent heat from cylinder block of engine.

In this system, water is introduced in liquid state through an inlet port of cooling jacket and it is allowed for evaporation by extracting latent heat from engine cylinder and is discharged in steam through an outlet port formed in the engine as shown in Fig. 2.40.

A condenser is used to convert the steam coming from the coolant jacket of the engine into water. The lower tank collects the water and supplies back to engine jacket by an electric pump.

This system uses the advantage of high latent heat of evaporation of water by allowing it to evaporate in cylinder jacket. It also allows the engine to operate at higher temperature by keeping the evaporation of water at a pressure that is above atmospheric pressure.

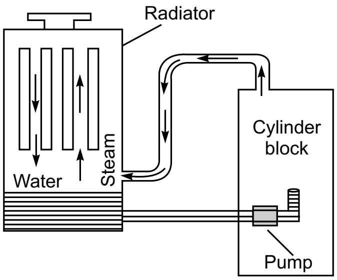


Fig. 2.40 *Schematic diagram of an evaporative cooling system*

2.26 ANTI-FREEZE SOLUTIONS

In order to prevent the water in the cooling jacket of an engine from freezing during very cold climates, some chemical solutions are added to water to avoid it freezing. These solutions are known as anti-freeze solutions like ethylene glycol or propylene glycol. In cold locations, if the engine is kept in off condition without these solutions for a period the water may freeze in water jacket and expand, leading to fractures of the cylinder block, cylinder head, pipes and radiators. An antifreeze solution is also called coolant and it is available in bright yellow and green liquid mixes with water. An antifreeze also raises boiling temperature of coolant to prevent overheating. It also prevents engine from corrosion and scale formation in the path of coolant circulation.

Desirable properties of antifreeze solution

1. An ideal anti-freeze mixture should easily dissolve in water.
2. It should be reasonably cheap and
3. It should not deposit any foreign matter in the jacket, pipes and radiator.

2.27 ENGINE LUBRICATION

The lubrication is the supply of oil between two surfaces having relative movement. The objectives of lubrication are

1. To minimize the friction between the parts having relative motion.
2. To reduce the wear and tear of moving parts.
3. To cool the surfaces by carrying away the heat generated due to friction.
4. To seal the space between piston rings and cylinder liner.
5. To absorb the shocks between bearings and other parts and consequently, reduce noise.
6. To act as cleaning agent and remove dirt, grit and any deposits that might be present between the moving parts.

Properties of Lubricants

A good lubricating oil should have the following characteristics:

1. Constant Viscosity The viscosity of the oil should not be changed with temperature rise.

2. Oilness It ensures the adherence to the bearings and spread over the surface. This property makes the oil smooth and very important in boundary lubrication.

3. Strength The oil must have high strength to avoid metal to metal contact and seizure under heavy loads.

4. Chemical Stability The oil should not react with surfaces and leave any deposit in the cylinder.

5. Pour Point It should be low to allow the flow of lubricant to the oil pump.

6. Flash Point and Fire Point The lubricating oil should not burn inside the cylinder, otherwise it will leave heavy deposit and poisonous exhaust. Therefore, the flash point and fire point of the lubricating oil must be high enough.

7. Neutralisation The oil should not have a tendency to form deposits by reacting with air, water, fuel or the products of combustion.

8. Cleaning The oil should act as cleaning agent inside the engine and should carry any deposits with it. It should also have non-foaming characteristics, low cost, and be non-toxic.

2.28 LUBRICATION SYSTEMS

The various lubrication systems used for lubricating the above parts of an internal combustion engine are classified as

(i) Mist lubrication system,
(ii) Wet sump lubrication system, and
(iii) Dry sump lubrication system

2.28.1 Mist Lubrication System

It is a very simple system of lubrication. In this system, the small quantity of lubricating oil (usually 2 to 3%) is mixed with the fuel (preferably gasoline). The oil and fuel mixture is introduced through the carburetor. The gasoline is vaporised and oil in the form of mist enters the cylinder via the crank case. The droplets of oil strike the crank case, lubricate the main and connecting rod bearings and the rest of the oil lubricates the piston, piston rings and cylinder.

The system is preferred in two-stroke engines where crank case lubrication is not required. In a two-stroke engine, the charge is partially compressed in a crank case, so it is not possible to have the oil in crank case.

This system is simple, low cost and maintenance-free because it does not require any oil pump, filter, etc. However, it has certain serious disadvantages. Therefore, it is not popular among the lubrication system. Its *disadvantages* are the following:

1. During combustion in the engine, some lubricating oil also burnt and it causes heavy exhaust and forms deposits on the piston crown, exhaust port and exhaust system.
2. Since the lubricating oil comes in contact of acidic vapours produced during the combustion, it gets contaminated and may result in the corrosion of the bearings surface.
3. When the vehicle is moving downhill, the throttle is almost closed, and the engine suffers lack of lubrication as supply of fuel is less. It is a very serious drawback of this system.
4. There is no control over the supply of lubricating oil to the engine. In normal operating conditions, the two-stroke engines are always over-oiled. Thus consumption of oil is also more.
5. This system requires thorough mixing of oil and fuel prior to admission into the engine.

2.28.2 Wet-Sump Lubrication System

In the wet-sump lubrication system, the bottom of the crank case contains an oil pan or sump that serves as oil supply, oil storage tank and oil cooler. The oil dripping from the cylinders, bearings and other parts, falls under gravity back into the sump, from where it is sucked by pump and recirculated through the engine lubrication system. There are three types of wet-sump lubrication system.

(i) Splash system,
(ii) splash and pressure system, and
(iii) full-pressure system.

(*i*) *Splash Lubrication System* It is used on small, stationary four-stroke engines. In this system, the cap of the big end bearing on the connecting rod is provided with a scoop which strikes and dips into the oil-filled troughs at every revolution of the crank shaft and oil is splashed all over the interior of crank case into the piston and over the exposed portion of the cylinder as shown in Fig. 2.41. A hole is drilled through the connecting rod cap through which the oil passes to the bearing surface. Oil pockets are provided to catch the splashed oil over all the main bearings and also the cam shaft bearing. From these pockets, oil passes to the bearings through drilled hole. The surplus oil dripping from the cylinder flows back to the oil sump in the crank case.

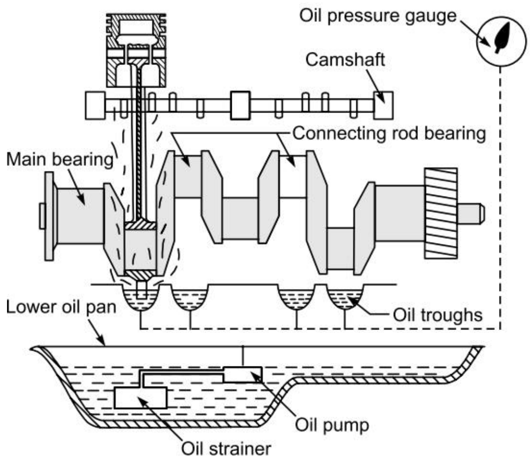


Fig. 2.41 *Splash lubrication system*

(*ii*) *Splash and Pressure System* This system is a combination of splash and pressure system as shown in Fig. 2.42. In this system, the lubricating oil is supplied under the pressure to main and cam shaft bearings. The oil is also directed in the form of spray from nozzle or splashed by a scoop or dipper on the big end bearings to lubricate the connecting rod, crank pin, gudgen pin, piston rings and cylinder.

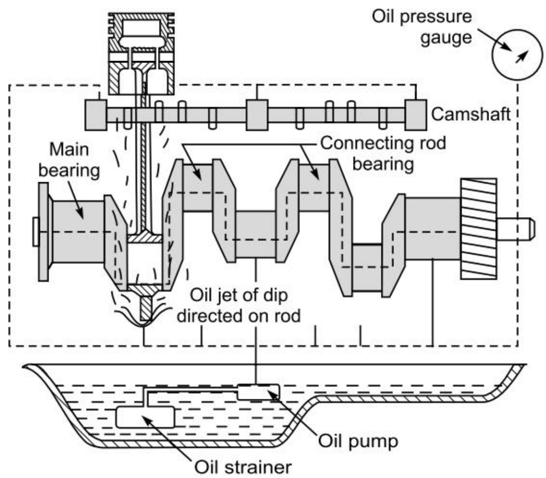


Fig. 2.42 *Splash and pressure lubrication system*

(*iii*) *Pressurized Lubrication System* In this system, the lubricating oil is supplied by a pump under pressure to all parts requiring lubrication as shown in Fig. 2.43. The oil under the pressure is supplied to main bearings of the crank shaft and camshaft. Holes drilled through the main crank shaft bearing journals, communicate oil to big end bearing and small end bearings through a hole drilled in the connecting rod. A pressure gauge is provided to confirm the circulation of oil to various parts.

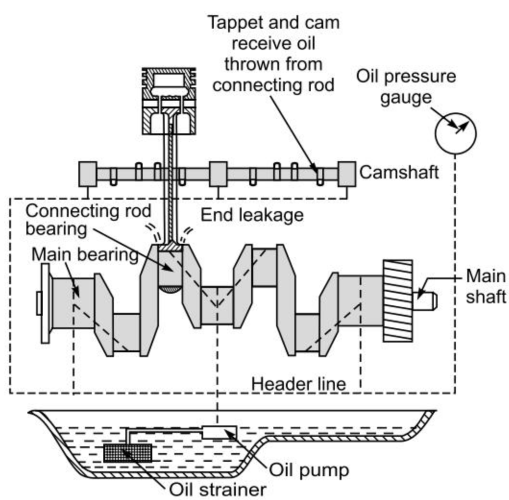


Fig. 2.43 *Pressurized lubrication system*

This system provides sufficient lubrication to all parts and is favoured by most of the engine manufacturers. Thus, it is used in most heavy-duty and high-speed engines.

2.28.3 Dry-Sump Lubrication System

In this system, the oil supply is carried from an external tank. The oil from the sump is pumped by means of a scavenging pump through filters to the external storage tank as shown in Fig. 2.44. The oil from the storage tank is pumped to the engine cylinder through an oil cooler. The oil pressure may vary from 3 to 8 bar.

The dry-sump lubrication system is generally used for heavy-duty engines.

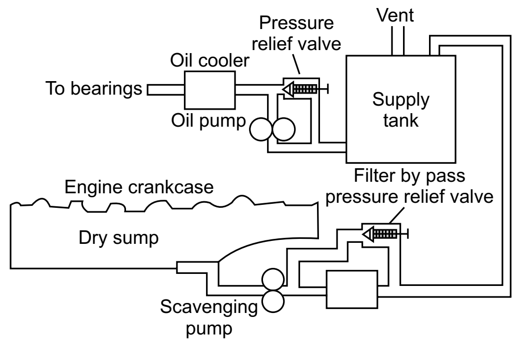


Fig. 2.44 *Dry sump lubrication system*

2.29 MAIN PARTS OF ENGINE TO BE LUBRICATED

The main parts of the engine which need lubrication are shown in Fig. 2.45 and listed below.

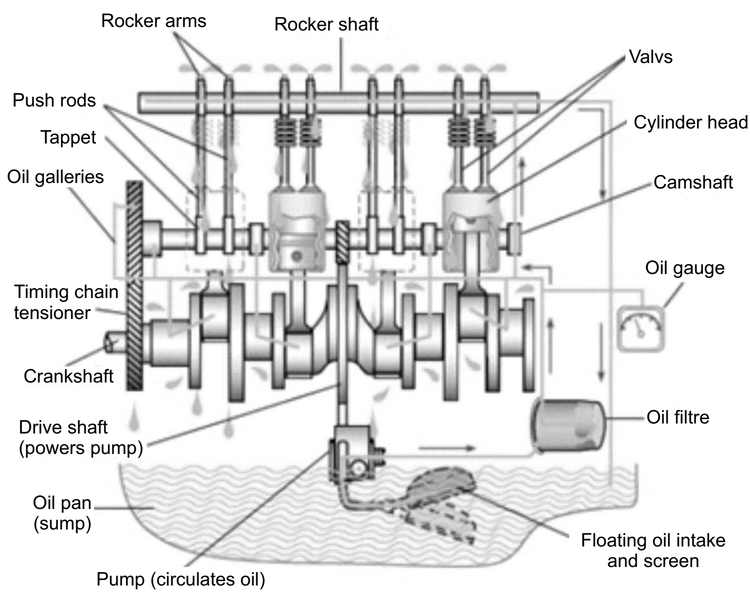


Fig. 2.45 *Parts of engine, which requires lubrication in internal combustion engine*

1. Main crankshaft bearings.
2. Big end bearings or crank pin.
3. Gudgeon pin in bearings.
4. Piston rings and cylindrical walls.
5. Timing gears.
6. Camshaft and camshaft bearings.
7. Valve mechanism.
8. Valve guide, valve tappets, push rods and rocker arms.
9. Water pump bearing.
10. Governor.

Summary

- A fuel is simply a combustible substance. It burns in the presence of oxygen and releases heat energy.
- **Calorific value** is the amount of heat librated by complete combustion of unit quantity of a fuel. It is measured in kJ/kg for solid and liquid fuels or kJ/m^3 for gaseous fuels.
- **Octane number** is a measure of auto-ignition resistance of gasoline in a SI engine. It is the volume percentage of iso-octane in a iso-octane/n-heptane mixture.
- **Cetane number** is the measure of combustion quality of ***diesel oil*** or it is the measure of the ignition delay. The higher the ***Cetane number of diesel*** offers, shorter the ignition delay, and the greater the fuel quality, more complete combustion is charge in the combustion chamber.
- Petrol or Gasoline is widely used liquid fuel in SI engines. The process of preparation of combustible mixture by mixing the proper amount of fuel with air before admission to the engine cylinder is called *carburation*.
- The fuel-injection system is the most important component in the working of a CI engines. The engine performance, i.e., power output, fuel economy, etc., depend on the effectiveness of a fuel injection system.
- The petrol engines induct carbureted homogeneous air–fuel mixture as charge into the cylinder while Diesel engines induct only air during suction, and diesel is injected at the end of the compression stroke. The fuel burns in the presence of hot air. Therefore, the Diesel engines are also called compression ignition (CI) engines.
- Biodiesel is a biomass-derived fuel. It is very similar to conventional fossil diesel. It is the substitute of diesel oil in compression-ignition engines. It is safer, cleaner, renewable, non-toxic and biodegradable.
- Gaseous fuels are the chemically simplest among the three groups. The two primary sources of gaseous hydrocarbon fuels are natural gas wells and certain chemical manufacturing process.
- The ignition system is used to produce a high intensity spark for initiation of combustion in the petrol engine. The battery ignition system is used on heavy-duty engines, while magneto ignition system is used on two wheelers, racing cars and aircrafts.
- The internal combustion engines are subjected to very high temperature during combustion of charge. Due to overheating of engine, there may be uneven expansion in some parts, burning of lubricant, valve seats, etc. Therefore, the engine should be provided with adequate cooling arrangement.
- The air-cooling system is simple, cheaper and maintenance-free and is widely itscd on two wheelers and light-duty engines.
- The heavy-duty engines release a large amount of heat during combustion, and therefore, they are water cooled. The water-cooling arrangement consists of a pump, a fan, a water jacket around the engine and a radiator.

- The lubrication minimizes the friction between moving parts. The mist lubrication system is used in two-stroke engines, while all four-stroke engines use wet or dry sump lubrication system.
- The supercharging refers to a process which supplies the charge to the cylinder above atmospheric pressure. Thus, the volumetric efficiency of the engine improves and the engine produces more power.

Glossary

Carburation Preparation of combustible mixture for petrol engine

Cetane number Measure of combustion quality of ***diesel oil***

Fuel injector Device to inject the pressurized fuel into cylinder

Fuel pump Device to pressurize the fuel for injection

IC engines Internal combustion reciprocating engines

Magneto An electric generator, which converts kinetic energy into electrical energy

Octane number Measure of auto-ignition resistance of gasoline

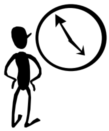
SI engine Spark ignition (petrol) engine

Stoichiometric air Amount of air just sufficient for complete combustion

Thermostat An instrument which controls the temperature

Review Questions

1. What are three general ranges of throttle operations for an SI engine and also specify the type mixture required for each of them?
2. How can you rate the fuels?
3. What is the Octane number? Explain how SI engine fuels are rated?
4. What is meant by Cetane number?
5. What is the Cetane number? Explain how CI engine fuels are rated?
6. Describe the three desirable properties of CI engine fuels.
7. What is meant by equivalence ratio and give its significance?
8. List the factors which affect the carburetion process.
9. What is the purpose of venturi in SI engine fuel supply system?
10. How does the Zenith carburetor fulfill the requirements of a good carburetor?
11. Explain what is meant by cruising range?
12. Draw the neat sketch and explain the working of carter carburetor.
13. Compare and contrast dry sump lubrication and crank case ventilation.
14. Describe the working of Solex carburetor with a neat sketch.
15. What are the functional requirements of an injection system? Discuss them.
16. Explain the working of solid injection system with neat sketch.
17. Draw the line diagram and explain typical fuel feed system for a CI engine.
18. What is the firing order for a multi-cylinder petrol engine? Mention the commonly used firing order for a four cylinder and six cylinder engines.
19. What is air cooling system and in which type of engine is it normally used?
20. Describe the evaporative cooling system.
21. What are various components to be lubricated in an engine?
22. Describe the working of pressure feed lubrication system with a neat diagram.
23. Explain the working of battery ignition system with the neat sketch.
24. Explain the working of splash lubricating system with neat sketch.
25. Briefly discuss the various factors which affect the ignition timing in SI engine.
26. Explain TCI ignition system with a neat sketch.



Objective Questions

1. The gaseous fuel is considered best fuel, because it
   (a) can be pressurized
   (b) can be easily transported
   (c) easily starts engine
   (d) All of above
2. Heptane is
   (a) paraffin, (b) Olefin
   (c) Napthalene (d) aromatic
3. Crank case dilution means
   (a) dilution of lubricating oil with water
   (b) dilution of lubricating oil with fuel
   (c) dilution of lubrication oil with gases
   (d) All of above
4. Heating value of diesel in kJ/kg is
   (a) 23000 (b) 30000
   (c) 37000 (d) 47000
5. Heating value of Bio-diesel in kJ/kg is
   (a) 28000 (b) 32000
   (c) 35000 (d) 40000
6. Net Calorific value of gasoline is
   (a) 43000 (b) 30000
   (c) 35000 (d) 23000
7. Octane number of iso-octane is
   (a) 100 (b) 80
   (c) 50 (d) 0
8. Octane number of Indian lead free petrol is
   (a) less than Octane number of leaded petrol
   (b) equal to Octane number of leaded petrol
   (c) greater than Octane number of leaded petrol
   (d) Not specified
9. Carburetion is to suuply
   (a) diesel + air
   (b) petrol + air
   (c) petrol + lubrication oil
   (d) air + lubrication oil
10. The choke is applied in an automobile carburetor to obtain
    (a) lean mixture
    (b) chemically correct mixture
    (c) rich mixture
    (d) weak mixture
11. In an automobile, the choke is applied for
    (a) acceleration
    (b) starting of engine in cold weather
    (c) fuel economy
    (d) reduction of speed
12. Mixture formation in carburetor is based on principle of
    (a) Newton's law
    (b) Pascal's law
    (c) Venturi principle
    (d) Buoyancy principle
13. In an automobile carburetor, the throttle valve controls the supply of
    (a) air only (b) air fuel mixture,
    (c) fuel only (d) none of above
14. The example of variable choke orifice type carburetor is
    (a) Carter carburetor (b) SU Carburetor
    (c) Solex carburetor (d) Zenith Carburetor
15. The most accurate petrol injection system is
    (a) direct injection
    (b) manifold injection
    (c) port injection
    (d) throttle body injection
16. Advantage of air injection system in a CI engine is
    (a) good atomization
    (b) complete combustion
    (c) ability to use cheaper fuel
    (d) all of above
17. A single cylinder diesel engine may have fuel distribution system comprising of
    (a) common rail system
    (b) distribution system
    (c) individual pump and injector
    (d) single injector system

18. Compression ratio in a CI engine is
    (a) 20 : 1 (b) 12 : 1
    (c) 10 : 1 (d) 7 : 1
19. Ideally ignition in a SI engine should occur
    (a) at the start of compression stroke
    (b) at the end of compression stroke
    (c) at the start of expansion stroke
    (d) at the end of suction stroke
20. Capacitor is used in coil ignition system to
    (a) transform voltage
    (b) act as mechanical switch
    (c) prevent arcing at the contact breaker
    (d) direct the current to appropriate spark plug
21. Contact breaker points are usually made of
    (a) copper (b) plastic
    (c) steel (d) tunguston
22. Firing of six cylinder engine is
    (a) 1-6-2-4-3-5 (b) 1-5-3-6-2-4
    (c) 1-2-5-4-6-3 (d) 1-3-2-6-5-4
23. By use of lubrication, which efficiency of an IC engine improves
    (a) volumetric efficiency
    (b) mechanical efficiency
    (c) charging efficiency
    (d) indicated thermal efficiency
24. By use of cooling, which efficiency of an IC engine decreases?
    (a) volumetric efficiency
    (b) mechanical efficiency
    (c) charging efficiency
    (d) thermal efficiency
25. The primary objective to use lubrication in an engine is to
    (a) reduce friction and wear
    (b) provide sealing
    (c) provide cleaning
    (d) increase cooling effect

Answers

1. (d)	2. (a)	3. (b)	4. (d)	5. (d)	6. (a)	7. (a)	8. (b)
9. (b)	10. (c)	11. (b)	12. (c)	13. (b)	14. (b)	15. (c)	16. (d)
17. (d)	18. (a)	19. (b)	20. (c)	21. (a)	22. (b)	23. (b)	24. (d)
25. (a)							

3 CHAPTER

Combustion and Combustion Chambers

Introduction

Combustion is a chemical reaction of certain elements present in the fuel with oxygen. The chemical energy of fuel is transformed into heat energy during combustion and thus increasing temperature and pressure of combustion products. In conventional Spark ignition engines, charge preparation as mixture of air and gasoline takes place in intake system. This charge is inducted into the cylinder during suction stroke and where mixing with residual gases take place. This mixture undergoes compression and then combustion with the help of spark plug. In compression ignition engine, only air is inducted as charge and it is compressed alone during compression stroke, then fuel is injected in the engine. With the auto-ignition of fuel combustion occurs. This chapter deals with normal and abnormal combustion, factors affecting the combustion in SI and CI engines.

3.1 NORMAL COMBUSTION IN SI ENGINES

The homogeneous mixture of fuel and air prepared in carburetor as charge is inducted in the spark ignition engine. This charge is compressed in the cylinder during compression stroke. At the end of the compression stroke, the combustion is initiated by a high intensity spark produced by an electric discharge. The typical stages of combustion are shown in Fig. 3.1.

First Stage A–B It is referred as the preparation phase (*ignition lag*) in which fuel elements become ready to react chemically with the oxygen present in compressed air. The chemical process depends on temperature and pressure and nature of charge. The growth and development of the propagating nucleus of flame takes place in this phase.

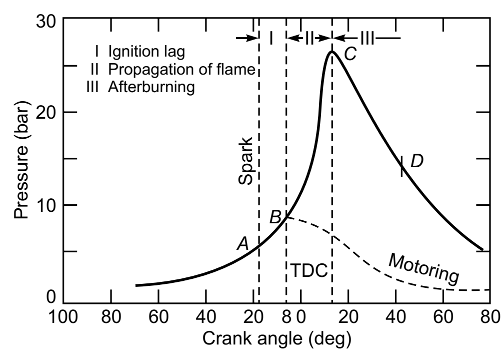


Fig. 3.1 *Stages of combustion in S.I. engine*

Second Stage B–C The second stage occurs when the piston approaches the top dead centre and the flame is propagated at a faster rate throughout the combustion chamber and it reaches to the farthest end of the cylinder. The rate of heat release in this stage depends on turbulence intensity and reaction rate. The rate of pressure rise is proportional to the rate of heat release. The initial gas force is exerted on the piston for its power stroke.

Third Stage C–D This stage starts from the moment at which maximum pressure is reached in the cylinder. This stage occurs during earlier part of the expansion stroke and the flame velocity decreases, and the rate of combustion becomes slow. There is no pressure rise during this stage.

3.2 FLAME FRONT PROPAGATION

Flame development and subsequent propagation within the cylinder during the combustion process is quite critical. Theoretically, as the flame propagation becomes faster, more efficient engine operation can be obtained. Faster burning is essential as it can reduce engine knock and results into higher thermal efficiency, higher power output. Two important factors decide the propagation of flame front across the combustion chamber and these are reaction rate and transposition rate.

The *reaction rate* purely depends on the chemical combination process, in which flame itself enlarges on its way into unburned charge. The *transposition rate* is the result of physical movement of flame front relative to cylinder wall and it also depends on pressure difference between burning gases and unburned charge.

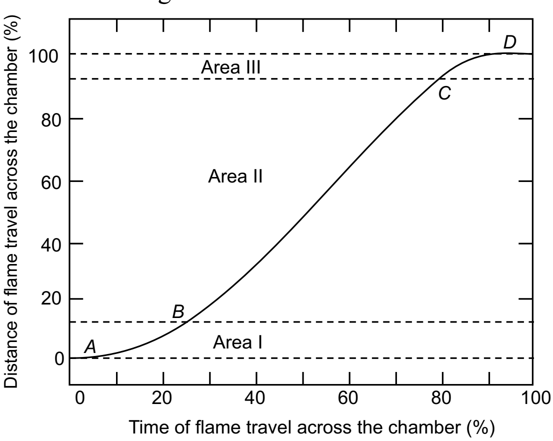


Fig. 3.2 *Flame travel in combustion chamber*

During the combustion a turbulent flame propagates across the combustion chamber and burns the premixed fuel-air mixture with a pattern as shown in Fig. 3.2. The curve is drawn between relative distance travelled by flame and relative time taken by flame to travel. The slope of the curve indicates speed of the flame. The flame travel can be divided into three district stages.

In area I, the spark is initiated and combustion begins very slowly. The low reaction rate results into slower flame propagation. It is due to

1. Very little mass of charge burns at the beginning.
2. Flame front progress slowly due low transposition rate and low turbulence.
3. Further spark from spark plug is created near cylinder wall in quiescent mixture and thus lack of turbulence reduces reaction rate and consequently flame speed.
4. Heat librated by burning portion heats the adjacent unburned charge and prepares for reaction

In area II, the flame leaves the quiescent zone and propagates in unburned charge at the faster rate. It is due to

1. Intensity of turbulence is more and thus flame proceeds with faster speed.
2. The combustion rate of charge is also faster.
3. The flame speed is almost constant in this stage.

The area III occurs at the end of combustion, the flame speed is further reduced. It is due to

1. The volume of unburned charge is very less.
2. The reaction rate is also slow because flame entered in low turbulence zone.
3. The combustion occurs near to combustion chamber wall, thus the transposition rate is also negligible once again.

3.3 FACTORS AFFECTING THE IGNITION LAG

The ignition lag is the first phase of combustion in SI engines, in which combustible mixture becomes ready to burn. Its duration is very small approximately 0.0015 seconds that is 10 to 20 degree of crank rotation. The duration of ignition lag depends on following factors:

1. **Fuel self ignition temperature** Fuel with higher self ignition temperature has longer ignition delay.
2. **Mixture strength** Slight rich air-fuel mixture offers maximum temperature thus reduces the ignition lag
3. **Initial temperature and pressure** The rate of chemical reaction is faster at higher temperature. The rate of chemical reaction also depends on pressure to some extent, since at higher pressure fuel and oxygen molecules are closure. The ignition lag therefore, decreases with increase in pressure and temperature of mixture at the time of spark. Thus the ignition lag can be reduced by supercharging of mixture, higher compression ratio, and retarding the spark.
4. **Electrode gap** Electrode gap also affects the ignition lag. If gap is too small, the spark intensity is also weak thus difficult to initiate flame nucleus. At low compression ratio, large electrode gap with high voltage is required to produce strong spark.

3.4 EFFECT OF ENGINE VARIABLES ON FLAME PROPAGATION

Flame propagation is an important parameter in combustion process of SI engines. Rate of flame propagation affects the combustion process. Higher flame propagation velocities result into higher combustion efficiency and better fuel economy. The factors which affect the flame propagation are

1. Air fuel ratio
2. Compression ratio
3. Load on engine
4. Turbulence and engine speed
5. Other factors

3.4.1 Air/Fuel ratio

The mixture strength influences the rate of combustion and amount of heat generated in the engine. The maximum flame occurs at nearly 10-15% rich mixture. Flame speed is reduced both for lean and as well as for very rich mixture as shown in Fig. 3.3. Lean mixture releases less heat resulting lower flame temperature and lower flame speed. Very rich mixture results into incomplete combustion with less heat liberation and flame speed remains low.

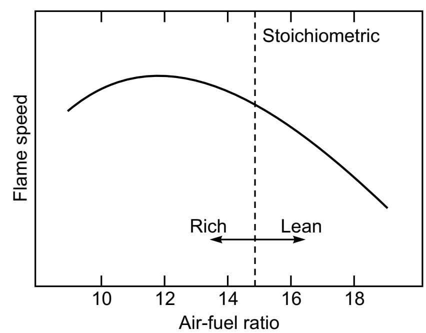


Fig. 3.3 *Average flame speed in the combustion chamber of an SI engine as a function of the air-fuel ratio for gasoline-type fuels.*

3.4.2 Turbulence

Turbulence plays very crucial role in combustion phenomenon as the flame speed is directly proportional to the turbulence of the mixture. The turbulence increases the mixing of burned and unburned charge and heat transfer rate between the burned and unburned mixture. The turbulence of the mixture can be increased at the end of compression by suitable design of the combustion chamber.

Insufficient turbulence provides low flame velocity that results into incomplete combustion and reduced the power output. Further, the too much turbulence is also not suitable as it increases the combustion rate and leads to detonation.

Excessive turbulence also causes to cool the flame generated and flame propagation is reduced. Moderate turbulence is always desirable as it accelerates the chemical reaction, reduces ignition lag, increases flame propagation and even allows weak mixture to burn efficiently.

3.4.3 Engine Speed

The turbulence of the mixture increases with an increase in engine speed. The flame speed almost increases linearly with engine speed as shown in Fig. 3.4. If the engine speed is doubled, flame propagation speed into the combustion chamber is halved. Double the original speed and half the original time give the same number of crank degrees for flame propagation. The crank angle required for the flame propagation, will remain almost constant at all speeds. This is an important characteristic of all petrol engines.

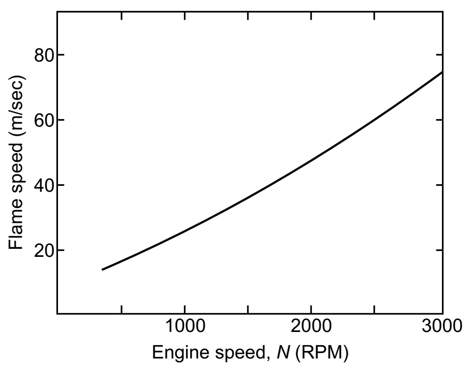


Fig. 3.4 *Average combustion chamber flame speed as a function of engine speed for a typical SI engine.*

3.4.4 Compression Ratio

The higher compression ratio increases the pressure and temperature of the mixture and decreases the concentration of residual gases. All these factors reduce the ignition lag and help to speed up the second phase of combustion. Higher compression ratio increases the surface to volume ratio and thereby increases the part of the mixture which after-burns in the third phase.

3.4.5 Load on Engine

With increase in load on the engine, the mean effective pressure increases and the flame speed also increases. In SI engine, the power developed by an engine is controlled by throttling. At part load and wider throttle, the initial and final compression pressure of the mixture decease and the mixture dilution with residual gases increases. This reduces the flame propagation and prolongs the ignition lag. This difficulty can be partly overcome by providing rich mixture at part loads but this increases the chances of after burning. The poor combustion at part loads and need of rich mixture are the main disadvantages of SI engines which causes wastage of fuel and discharge of large amount of CO with exhaust gases.

3.4.6 Engine Size

Engines of similar design generally run at the same piston speed. This is possible by using small engines having larger rpm and larger engines having smaller rpm. Due to same piston speed, the inlet velocity, degree of turbulence and flame speed are nearly same in similar engines regardless of the size. However, in small engines the flame travel is small and in large engines large. Therefore, if the engine size is doubled the time required for propagation of flame through combustion space is also doubled. But with lower rpm of large engines the time for flame propagation in terms of crank would be nearly same as in small engines. In other words, the number of crank degrees required for flame travel will be about the same irrespective of engine size provided the engines are similar.

3.4.7 Other Factors

Other factors which affect the flame speed are supercharging of the engine, spark advance and residual gases left in the engine at the end of exhaust stroke. The moisture present in air also affects the flame velocity.

3.5 ABNORMAL COMBUSTION

The combustion phenomenon discussed in Section 3.1 is normal combustion in SI engine. In normal combustion, the flame starts from spark plug and travels across combustion chamber in fairly even way. Under certain operating conditions, the abnormal combustion may take place, which is undesirable for engine life and engine performance. The causes of abnormal combustion are

1. Detonation or knock
2. Pre-ignition of charge
3. Run-out.

Detonation is risky and puts restriction on compression ratio of engine, which in turn control the thermal efficiency and power output of the engine.

3.6 DETONATION IN SI ENGINE

The engine knock occurs when some of the unburnt gases ahead of the flame in SI engine are ignited spontaneously. The unburnt gas ahead of the flame is compressed as flame propagates through the mixture, and the temperature and pressure of the unburnt mixture rise. If the temperature of the end unburnt mixture at some moment exceeds the self-ignition temperature of fuel, the auto-ignition of unburnt mixture occurs. This phenomenon is called *detonation*, or knocking.

Detonation in the combustion chamber generates a shock wave which traverses from the end gas region and an expansion wave which traverses into the end gas region. The two waves collide at the boundary of the combustion chamber and interact to produce high-amplitude severe pressure pulses.

The phenomenon of detonation may be illustrated with the help of Fig. 3.5. Figure 3.5(a) shows normal combustion when the flame travels from the spark plug to the farthest end of the combustion chamber and pressure wave travels in one direction only.

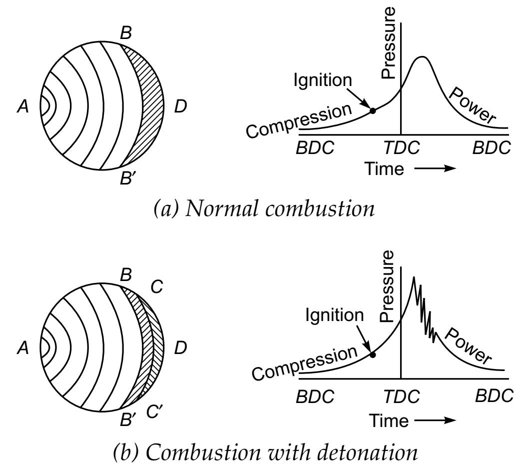


Fig. 3.5 *Normal combustion and detonation in SI engine*

Figure 3.5(b) shows the combustion with detonation. The advancing flame front compresses the unburned charge *BB'D* in the end zone of the combustion chamber, thus raising its pressure and temperature abruptly as shown in Fig. 3.6(c). The temperature of this charge is also increased due to heat transfer from the burning charge. If the temperature of the end charge reaches the self-ignition temperature of the fuel, and remains for some duration, the auto-ignition of charge takes place leading to knocking combustion.

During auto-ignition, another pressure wave starts traveling in the opposite direction to the main pressure wave. When the two pressure waves collide, a severe pressure pulse is generated. The gas in the combustion chamber is subjected to vibration along the pressure pulse until the pressure pulse is subsidized to an equilibrium state. The gas vibration can force the combustion chamber to vibrate. An objectionable audible sound can be heard from the engine.

Detonation during combustion can cause total engine failure. The gas vibration can scrub the chamber walls causing increased heat loss.

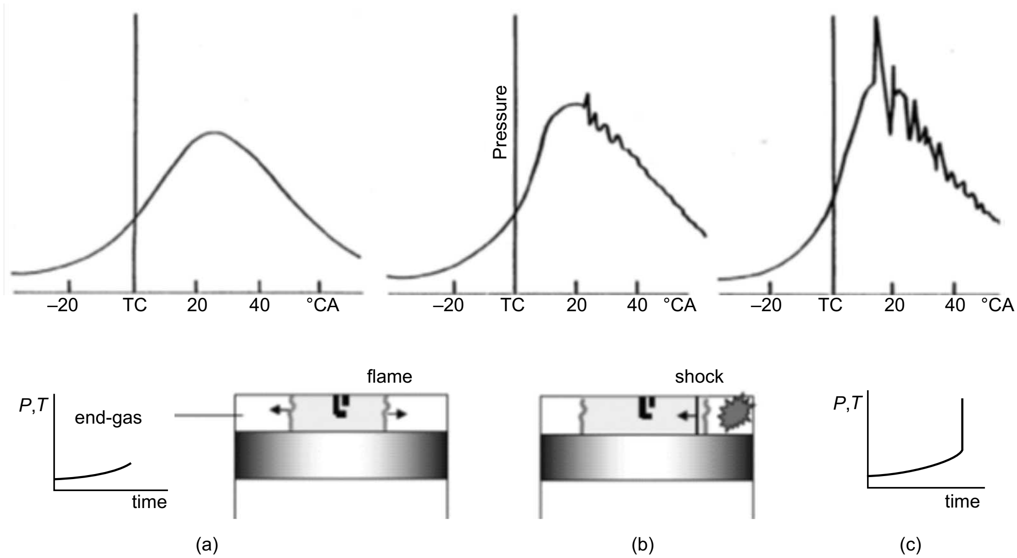


Fig. 3.6 *Normal combustion, light knock Combustion and heavy knock combustion*

The pressure variation in the cylinder during normal, light knock and heavy knock operation is shown in Fig. 3.6.

3.7 EFFECT OF DETONATION

The harmful effect of detonation or knock in SI engine are discussed below.

1. **Noise and roughness** The mild knock is almost audible and not harmful to the engine. If intensity of knock increases, it leads to a loud pulsating noise, due development of pressure wave, which vibrates back and forth in the combustion chamber. This imbalance force causes crankshaft vibration and an additional vibration to the engine.
2. **Mechanical damage** During detonation, very rapid local pressure rise offers increased wear rate of piston crown, cylinder and inlet and exhaust valves and sometimes complete wreckage of piston in large engines.
3. **Carbon deposits** Detonation in the engine offers an increased rate of carbon deposits in the combustion chamber, spark electrodes, piston crown and valves.
4. **Increase in heat transfer** During detonation, the average temperature of combustion chamber is about 150°C higher than in normal course. Further, due to vibration during knock, the protective lubricant and inactive gas layers on cylinder walls are scoured away. These two factors are responsible for increased heat transfer.
5. **Decreased in efficiency and power output** Due to high rate of heat transfer, some of the available energy is converted into heat and thus the thermal efficiency and power output from the engine are reduced.
6. **Pre-ignition** Scouring of cylinder walls and increased temperature put another effect to create hot spots due to local heating in the cylinder. These local hot spots can ignite the charge before commencement of spark. It is called *pre-ignition*. An engine detonating for long period can lead to pre-ignition and it is further responsible for high intensity detonation. Effects of pre-ignition are
    1. It increase the tendency of denotation in the engine

2. It increases heat transfer to cylinder walls because high temperature gas remains in contact for a longer time.
3. Pre-ignition in a single cylinder will reduce the speed and power output.
4. Pre-ignition may cause seizer in the multi-cylinder engine, if only one cylinder has pre-ignition.

3.8 ENGINE VARIABLES AFFECTING DETONATION

From above discussion, it can be concluded that four major factors are responsible for detonation in SI engine. These are density, temperature, pressure of unburned charge and time factor. The density, pressure and temperature are closely related with each other, thus they are kept in one group and time factor in other group.

3.8.1 Density, Pressure and Temperature

As the temperature of charge increases, the flame propagation speed increases and the possibility of end unburned charge to reach its self-ignition temperature also increases. Therefore, any factor during combustion process that reduces the temperature of unburned charge will reduce possibility of detonation. The factors, which are responsible to increase the temperature of unburned charge, are discussed below.

1. Compression Ratio Compression ratio of an engine is very important factor, which decides the pressure and temperature at the beginning of combustion. An increase in the compression ratio boosts the pressure and temperature at the compression stroke. High compression also increases the density of charge and pre-flame reaction in the end charge. Therefore, increase in compression ratio increases tendency for detonation. For given engine design and fuel, there is critical compression ratio for safe engine operation. This compression ratio is called *highest useful compression ratio* (HUCR) or *knock limited compression ratio*.

2. Mass of Inducted Charge The decrease in mass of inducted charge into the cylinder during suction stroke will reduce the charge density. It will reduce pressure and temperature at the end of compression stroke, thus tendency of detonation at the end of combustion. The supercharged engines have more tendency of detonation.

3. Inlet Temperature of Mixture Decrease in the intake temperature of the charge decreases temperature at the end of compression, which in turn lowers the temperature of end unburned charge and therefore, tendency of detonation. Further, the volumetric efficiency is also improved with low intake temperature of charge.

4. Coolant Temperature An increase in temperature of coolant leads to higher combustion chamber temperature and hot spot may present in combustion chamber. Therefore, increase in coolant temperature will increase tendency of knocking.

5. Spark Timing By retarding the spark timing, from the optimize timing, the peak pressure will reach further down to expansion stroke and thus the compression of end unburned charge will reduce and therefore, tendency of knocking will also be reduced.

6. Power Output of the Engine An increase in power output of the engine requires more fuel to burn and higher temperature during combustion, which leads to an increase pressure and temperature of end charge and tendency to engine knock.

7. Exhaust Gas Recirculation (EGR) The dilution of fresh charge with exhaust gas from previous cycle will reduce the maximum pressure in the cycle, and therefore, decreases the tendency of knocking.

8. Carbon Deposits Incomplete combustion of charge offers the carbon deposits on the combustion chamber wall, cylinder head, exhaust valve and spark plug. A part of heat of combustion is absorbed by these deposits due to their poor thermal conductivity. This heat is transferred back to fresh charge of next cycle and causing its temperature rise. Thus the temperature and pressure at the end of compression will be higher and will offer tendency to knock.

3.8.2 Time Factor

Any time alteration that will increase the normal flame propagation in the cylinder and increase in ignition lag of end unburned charge will reduce the tendency of knocking. The following factors reduce the possibility of knocking in SI engines.

1. Turbulence The intensity of turbulence depends on the design of combustion chamber and engine speed. An increase in the turbulence increases the flame propagation and reduces the time available for end unburned charge to attain auto-ignition and therefore decreasing the tendency to knock.

2. Engine Speed An increase in engine speed increases the turbulence of the mixture. It will increase the flame propagation and combustion completes in short duration. Therefore, tendency to knocking will be diminished at higher speed of the engine.

3. Flame Travel Distance By reducing the flame travel distance, the time required for flame front travel will be shortened and the knocking tendency will be reduced. Combustion chamber size and spark plug location are two important factors to decide the flame travel distance.

4. Engine Size The smaller engines have small combustion chamber and shorter flame travel distance and bare minimum combustion duration and rare possibility to knock. Therefore SI engine is smaller in size of about 150 mm bore.

5. Combustion Chamber Shape Compact combustion chamber has shorter flame travel distance and least combustion time and thus better antiknock characteristics. Spherical combustion chambers have shorter flame travel distance and better turbulence for given volume and therefore, low knocking tendency.

6. Location of Spark Plug A combustion chamber with centrally located spark in cylinder head will have shorter flame travel distance and minimum tendency to knock.

7. Location of Exhaust Valve The exhaust valve is a hottest part in the combustion chamber and it may initiate ignition, if located in the region of end unburned charge. Therefore, the exhaust valve should be located close to spark plug in order to reduce tendency to knock.

3.8.3 Composition Factors

The air fuel ratio and properties of the fuel also play important role in controlling engine knock. The following composition factors influence the tendency of engine knocking.

1. Fuel- Air Ratio The flame speed is influenced by the fuel-air ratio of charge. The slightly rich mixture gives best power and highest flame temperature and maximum flame speed, but shorter delay period may lead to engine knock. The maximum flame speed is obtained for equivalence ratio $(\phi) \cong 1.1\text{-}1.2$ and the length of delay period can be increased by using mixture richer or leaner than $(\phi) \cong 1.1\text{-}1.2$ and reduced tendency of knocking.

2. Octane Rating of Fuel The tendency of engine knocking also depends on the properties of fuel used. The higher self-ignition temperature of fuel and lower pre-flame reactivity would reduce tendency of engine knocking. The Octane rating determines the possibility of knocking of fuel in the engine under the given operating conditions. Higher the Octane number, better the resistance to knock.

Fuels of Paraffin series of hydrocarbon have the maximum and that of aromatic series have the minimum tendency to knock. The naphthene series comes in between the two.

3. Mal-distribution of Charge Uneven distribution of air and fuel mixture between various cylinders of multi-cylinder engine is called mal-distribution, in which air fuel ratio changes to different cylinders. Mal-distribution of charge leads to different knocking tendency in different cylinders. The better charge distribution with constant air fuel ratio can give better control over tendency to knock.

Table 3.1 shows a summary of variables affecting knock in SI engine and gives remedial corrective action to reduce the knock tendency.

Table 3.1 *Summary of variables affecting knock in an SI engine*

Increase in variable	*Major effect on unburned charge*	*Action to be taken to minimize knocking*	*Can operator usually control?*
Compression ratio	Increases temperature and pressure	Reduce	No
Mass of charge inducted	Increases pressure	Reduce	Yes
Inlet temperature	Increases temperature	Reduce	In some cases
Chamber wall temperature	Increases temperature	Reduce	Yes with better cooling arrange-ment
Spark advance	Increases temperature and pressure	Retard	In some cases
A/F mixture strength	Increases temperature and pressure	Make very rich or lean	In some cases
Turbulence	Decreases time factor	Increase	Somewhat (through engine speed)
Engine speed	Decreases time factor	Increase	Yes
Distance of flame travel	Increases time factor	Reduce	No

3.9 COMBUSTION CHAMBERS FOR SI ENGINES

The design of the combustion chamber for an SI engine has an important influence on the *engine performance and its knocking tendencies.*

The design of a combustion chamber involves

(a) the shape of the combustion chamber,
(b) the location of spark plug, and
(c) the location of inlet and exhaust valves.

The *important requirements of an SI engine* combustion chamber are

(a) to provide high power output with minimum octane requirement,
(b) high thermal efficiency, and
(c) Smooth engine operation.

Requirements of an SI engine combustion chamber

1. Smooth Engine Operation The smooth engine operation can be achieved by

(a) Moderate Pressure Rise Limiting the rate of pressure rise as well as the position of the peak pressure with respect to TDC affect smooth engine operation.

(b) Reducing the Tendency of Knock Decrease in the possibility of knocking in an engine can be achieved by

- ***Reducing the distance of the flame travel*** by centrally locating the spark plug and also by avoiding pockets of stagnant charge.
- ***Satisfactory cooling of the spark plug and of exhaust valve area*** which are the source of hot spots in the combustion chambers.
- ***Reducing the temperature of the last portion of the charge***, through application

of a high surface to volume ratio in that part where the end portion unburned charge burns.

2. High Power Output and Thermal Efficiency High power and better thermal efficiency of the engine can be achieved by considering the following factors:

(a) A High Degree of Turbulence is Required to Achieve a High Flame Front Velocity

- Turbulence is induced by *inlet flow configuration or squish*
- Squish is the rapid radial movement of the gas trapped between the piston and the cylinder head into the bowl or the dome.
- Squish can be induced in spark-ignition engines by having a bowl in piston or with a dome shaped cylinder head.

(b) High Volumetric Efficiency

- Induction of more charge during the suction stroke, results in an increased power output.
- This can be achieved by providing ample clearance around the valve heads,
- Large diameter valves and straight passages with minimum pressure drop during charge induction.

(c) Improved Anti-knock Characteristics Improved anti-knock characteristics permit the use of a higher compression ratio that results into increased power output and better thermal efficiency.

(d) A Compact Combustion Chamber It reduces heat loss during combustion process and increases the thermal efficiency.

3.10 TYPES OF COMBUSTION CHAMBERS FOR SI ENGINES

The different types of combustion chambers have been developed over a period of time, some of them are discussed below.

3.10.1 T-Head Type Combustion Chamber

These combustion chambers were used in the early stage of engine development. A typical T-head combustion chamber is shown in Fig. 3.7.

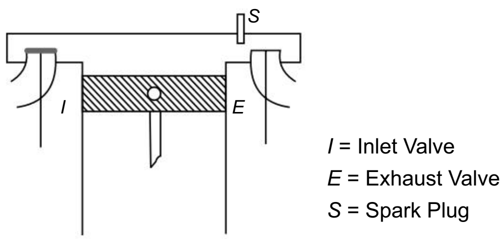


Fig. 3.7 *T-head combustion chamber*

They have large flame travel distance across the combustion chamber thus the knocking *tendency is high.*

The T-head combustion chamber is provided with two valves on either side of the cylinder, thus requiring two cam-shafts. The use of two cam shafts offers complexity and mechanical losses.

3.10.2 L-Head Type

A modification of the T-head combustion chamber is the L-head type which *provides the two valves on the same side of the cylinder* shown in Fig. 3.8. The two valves are operated by a single camshaft.

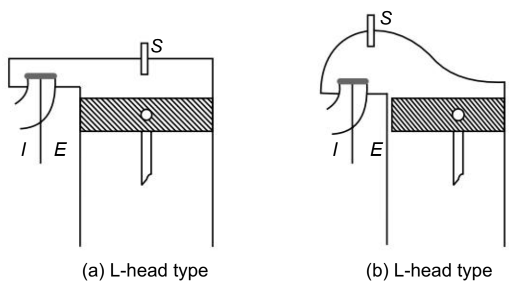


Fig. 3.8 *L- Head combustion chambers*

The main objectives of the Ricardo's turbulent head design, Fig. 3.8.(b), to obtain fast flame speed and reduced knock. The lubrication of valve mechanism is also easy in this design.

3.10.3 I-Head Type or Overhead Valve

I-head combustion chamber is also called over head combustion chamber. Both the valves are located on the cylinder head. A typical over head combustion chamber is shown in Fig. 3.9.

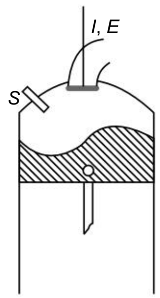


Fig. 3.9 I-head type

The over head valve engine is superior to a side valve or an L-head engine at high compression ratios. Some of the important characteristics of this type of valve arrangement are:

- *less surface to volume ratio and therefore less heat loss*
- *less flame travel distance and hence greater freedom from knock*
- *higher volumetric efficiency from larger valves or valve lifts, thus better power output*

3.10.4 F-Head Type

The F-head type chamber with valve arrangement is a compromise between L-head and I-head types as shown in Fig. 3.10.

- *In F-head* combustion chambers one valve is located in the cylinder head and other is located in the cylinder block.
- Modern F-head engines have exhaust valve in the head and inlet valve in the cylinder block.
- The main disadvantage of this type of combustion chamber is that the inlet valve and the exhaust valve are separately actuated by two cams mounted on to camshafts driven by the crankshaft through gears. This added complexity in mechanism.

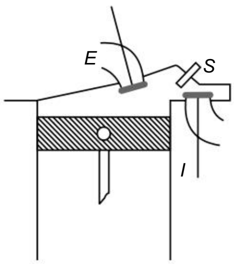


Fig. 3.10 F-head type

3.10.5 Hemispherical Combustion Chamber

The hemispherical combustion chamber is called Pent-roof combustion chamber. It is shown in Fig. 3.11. Effectively, a hemispherical combustion chamber is one half of a sphere cast into the bottom of the cylinder head. The valves are located at the outside of the bore area and at a specific angle from the crankshaft centerline.

1. It offers better air flow and high degree turbulence.
2. Hemispherical chambers generally have a central spark plug, which offers excellent octane tolerance.
3. A hemispherical chamber offers the better combustion and thermal efficiency.
4. An additional benefit is the distance between the intake and exhaust valves, which further limits heat transfer.

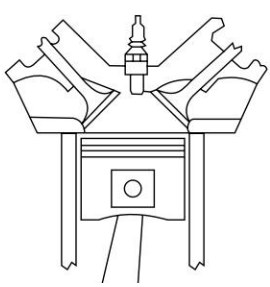


Fig. 3.11 *Hemispherical combustion chamber*

3.10.6 Wedge-Shaped Chambers

This type of chamber resembles an inclined basin recessed into the deck of the head as shown in Fig. 3.12. Inline valves are normally inclined to accommodate the sloping roof of this design. The spark plug is located on the thick side of the wedge and is usually positioned midway between the valves.

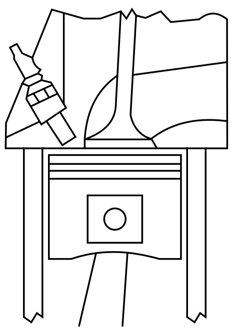


Fig. 3.12 *Wedge shaped combustion chamber*

The relatively steep walls in the combustion chamber design force the charge flow through a deflected path and force it to move in a downward spiral around the cylinder axis. During the compression stroke, the compressed charge area reduces to such an extent that the trapped mixture is violently thrust from the thin to the thick end of the chamber. This builds up significant kinetic energy, which when ignited contributes to overall power.

3.10.7 Bath-tub Combustion Chamber

This is the somewhat oval-shaped combustion chamber with the valves side by side as shown in Fig. 3.13. The name has been derived from its shape, which is similar to an inverted bath tub. The spark plug is located on one side. This arrangement provides a short flame path from the spark-plug. The valves are usually vertical in cylinder head and are in-line.

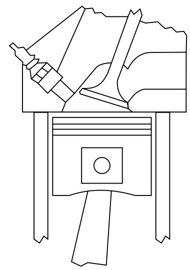


Fig. 3.13 *Bath-tub type combustion chamber*

3.11 COMBUSTION IN COMPRESSION IGNITION ENGINE

Combustion phenomenon in CI engine is completely behaved differently. In SI engine, a homogeneous mixture of air and fuel in some proportion as charge is inducted during suction stroke, it undergoes compression process (compression ratio 6:1 to 10:1) and ignited with the help of spark created between two electrodes of spark plug. A single flame propagates in the combustion chamber after ignition in normal course.

In CI engine, only air as charge is inducted during the suction stroke. It undergoes high compression ratio usually 16 : 1 to 20 : 1. The temperature of pressure of air increase to very high value, then fuel is injected through one or more jets into highly compressed air in the combustion chamber. After injection, the fuel jet is disintegrated into a core of fuel surrounded by a spray envelope of air and fuel particles as shown in Fig. 3.14.

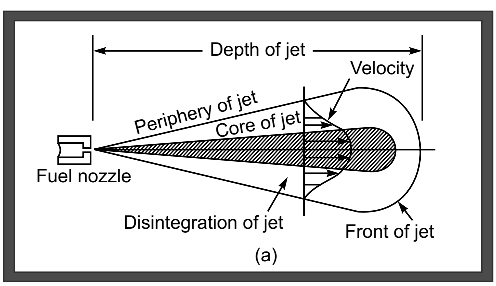


Fig. 3.14 *Schematic of representation of fuel jet*

In addition to the swirl and turbulence of the air in the combustion chamber, a high injection velocity is required to spread the fuel throughout the cylinder and cause it to mix with the air in short time. After injection, the fuel must go through a series of events to assure the proper combustion process:

1. Atomization Fuel enters into combustion chamber in the form of a fine spray. The tiny drops in fuel jet undergo quickly and more efficiently atomization process.

2. Vaporization The small droplets of liquid fuel evaporate to vapor by absorbing latent heat from surrounding air. This occurs very quickly due to the high air temperatures present in combustion chamber at the end of compression stroke. About 90% of the fuel injected into the cylinder is vaporized within 0.001 second after injection. As the first fuel evaporates, the immediate thin layer of surrounding air is cooled by evaporative cooling. This greatly affects subsequent evaporation.

3. Mixing After vaporization, the fuel vapor is mixed with air to form a combustible mixture. This mixing results into the high fuel injection velocity and swirl and turbulence in the cylinder. Combustion can occur within air-fuel ratio from 18:1 at full load and 80:1 at no load.

4. Self-Ignition Self-ignition of combustible mixture starts at about 8 degree before TDC and continues 6-8 degree after the start of injection. Actual combustion is preceded by secondary reaction including breakdown of large fuel molecules into smaller species and some oxidation. These reactions caused by the high-temperature air are exothermic and therefore, the air temperature is further increased in the immediate local vicinity.

5. Combustion Combustion starts from self-ignition of fuel vapour simultaneously at many locations in the slightly rich zone. Up to this time about 70-90% of the fuel in the combustion chamber is vaporized. When combustion starts with multiple flame fronts spreading from the many self-ignition sites and rapidly consumes all combustible air-fuel mixture, even at sites where self-ignition wouldn't occur. This gives a very quick rise in temperature and pressure within the cylinder as shown in Fig. 3.15. Point A is start of fuel injection, A to B is ignition delay, and point C is the end of fuel injection.

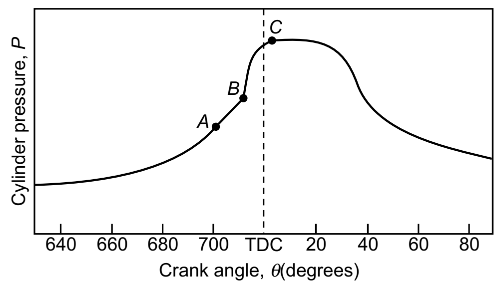


Fig. 3.15 *Cylinder pressure rise as a function of crank angle for a CI engine.*

The higher temperature and pressure reduce the vaporization time and ignition delay time for additional fuel particles and cause more self-ignition points to further increase the combustion process. Liquid fuel is still being injected into the cylinder after the first fuel is already burning. The rest of the combustion process is controlled by injection rate of fuel.

3.12 STAGES OF COMBUSTION IN CI ENGINES

In compression ignition engines, the fuel is injected into the highly compressed air in the combustion chamber. The jet of fuel disintegrates into tiny droplets evaporate and a mixture of air and fuel vapour formed at some locations. As soon as the temperature of this mixture attains self-ignition temperature, the auto-ignition takes place.

The combustion in CI engines can be considered to take place in four stages as shown in Fig. 3.16:

1. Ignition delay period,
2. Period of uncontrolled combustion,

3. Period of controlled combustion, and
4. Period of after burning.

1. Ignition Delay Period This phase is also called *preparatory phase*. In this phase, the injected fuel droplets first heated and evaporated by absorbing heat from the surrounding compressed air. It reduces the temperature of the thin layer of surrounding air and some time elapses before the temperature of the mixture again reaches the self ignition temperature.

The time period required to start actual burning of mixture after injection of the first fuel droplets into the combustion chamber is called *ignition delay period*. The ignition delay period is the combination of physical and chemical delays.

The *physical delay* is the time lapse between the beginning of fuel injection and start of chemical reaction. During this period, fuel jet is atomized, evaporated, mixed with compressed air and raises its self-ignition temperature.

The *chemical delay* is the time lapse for development of inflammation after the beginning of reaction. The chemical delay depends on temperature of compressed air. It is analogous to ignition lag in an SI engine.

The ignition delay period influences the engine performance, combustion rate, exhaust quality and engine knock.

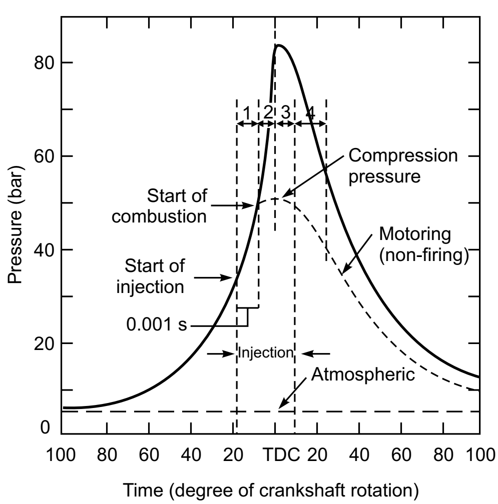


Fig. 3.16 *Stages of combustion in CI engines*

2. Period of Uncontrolled Combustion During the ignition delay period, most of the fuel admitted would have been evaporated and formed a combustible mixture with compressed air. By that time the pre-flame reactions would have been also completed. The auto-ignition of the combustible mixture starts from different sites of the combustion chamber simultaneously. It leads to uncontrolled combustion and rapid pressure rise.

The pressure rise during uncontrolled combustion depends on the length of the delay period. Longer the delay period, rapid the pressure rise, since more fuel would have been accumulated in the combustion chamber.

3. Period of Controlled Combustion The uncontrolled combustion is followed by controlled combustion. The temperature and pressure in the second stage are already quite high. Hence, the ignition delay reduces and fuel injected in the combustion chamber burns faster as it gets oxygen. During this stage, the pressure rise can be controlled by controlling the fuel-injection rate.

4. Period of after Burning Combustion does not come to an end with the end of the fuel-injection process. The burning of unburnt and partially burnt fuel particles left in the combustion chamber continue as they come in contact of oxygen. The duration of this process is called the period of after burning.

3.13 HEAT RELEASE RATE

A typical heat release rate pattern in direct injection compression ignition engine is shown in Fig. 3.17.

Figure 3.17 can be obtained by plotting the measured pressure at every crank angle of engine rotation. The first phase is ignition delay, in which fuel is injected, but combustion does not take place. Second phase is rapid premix uncontrolled combustion. In this phase, the combustion starts with multiple flame fronts spreading from the many self-ignition sites and rapidly consumes all combustible premixed air-fuel mixture that is prepared during ignition delay. Therefore, heat release rate is uncontrolled and independent of load on the engine. The first peak occurs in this phase.

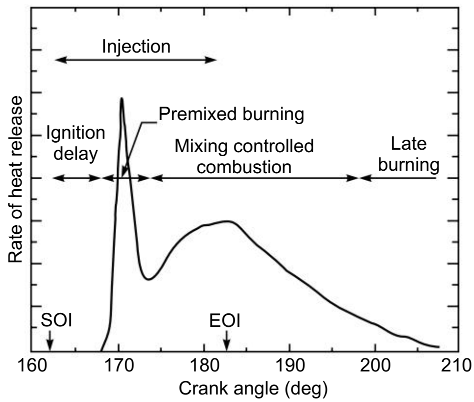


Fig. 3.17 *Heat release pattern in direction injection CI engine*

The third phase is controlled combustion. During this period heat release rate depends on the fuel injection rate. At high load, the injection period is longer; the amount of fuel injected is more, thus increasing magnitude and duration of mixing controlled combustion A second peak is observed in this phase. In this phase heat release rate decreases as phase progresses. Last phase is late combustion phase. It is result of burning of fuel after end of fuel injection, that portion could not burn during controlled combustion phase.

3.14 FACTORS AFFECTING THE IGNITION DELAY PERIOD

The ignition delay in a diesel engine is defined as the time interval between the start of injection and the start of combustion. This delay period consists of two parts: (a) physical delay, wherein atomization, vaporization and mixing of air fuel occur and (b) of chemical delay attributed to pre-combustion reactions. Physical and chemical delays occur simultaneously. Factors affecting the delay period are discussed below.

1. Compression Ratio With an increase in compression ratio, the temperature and pressure increase at the time of fuel injection, therefore, ignition lag reduces. A higher pressure increases density resulting in closer contact of the molecules which reduce the time of action when fuel is injected.

2. Inlet Air Temperature With an increase in inlet temperature of air, the temperature at the end of compression increases and hence the ignition delay decreases.

4. Engine Speed With an increase in engine speed, cylinder air temperature increases and the turbulence in cylinder also increase and therefore, ignition delay decreases.

5. Coolant Temperature With an increase in jacket water temperature, compressed air temperature also increases and hence delay period is reduced.

6. Fuel Temperature With increase in fuel temperature, both physical and chemical delay period decrease.

7. Intake Pressure (Supercharging) With an increase in intake pressure or supercharging, the compressed air pressure will be higher. Higher will pressure will increase density of charge and hence delay period decreases. Further with supercharging, the power output from the engine will be more, and hence more fuel can be injected per stroke.

8. Engine Output (Load) With an increase in load on the engine, air-fuel ratio decreases, and operating temperature increase due to more fuel burning in the cylinder and hence delay period decreases.

9. Air-Fuel Ratio With an increase in air-fuel ratio (leaner mixture), the combustion temperatures decreases and cylinder walls temperature decreases and hence the delay period increases.

10. Injection Timing At normal engine conditions the minimum delay occurs with the start of injection at about 15-20 degree before TDC. An increase in the delay time is observed with earlier or later injection timing because of lower air temperature and pressure at the beginning of injection.

Table 3.2 *Effect of engine variable on ignition delay period*

Increase in Variable	*Effect on Delay Period*	*Reason*
Increase in cetane number of fuel	Reduces	Reduces the self-ignition temperature
Increase in injection pressure	Reduces	Reduces physical delay due to greater surface volume ratio.
Injection timing advance	Reduces	Reduced pressure and temperature when the ignition begins
Increase in compression ratio	Reduces	Higher air temperature and pressure and reduces auto-ignition temperature
Increase in intake temperature	Reduces	Higher air temperature
Increase in coolant temperature	Reduces	Increases wall and hence air temperature
Increase in fuel temperature	Reduces	Increases chemical reaction due to better vaporization
Intake pressure (supercharging)	Reduces	Increases density and reduces auto-ignition
Increase in engine speed	Reduces in the order of milliseconds	Reduces loss of heat
Increase in engine load	Reduces	Operating temperature increase, delay period

11. Injection Quantity For a CI engine the air is not throttled so the load is varied by changing the amount of fuel injected. Increasing the load (bmep) increases the residual gas and wall temperature which results in a higher air temperature at time of fuel injection which results into a decrease in the ignition delay period.

12. Engine Size The engine size has little effect on the delay period in milliseconds. Usually, the large engines operate at low revolutions per minute (rpm) because of inertia stress limitations, the delay period in terms of crank angle is smaller and hence less fuel enters the cylinder during the period, thus controlled combustion phase is longer, which gives smooth operation of large slow speed CI engines.

3.15 PHENOMENON OF KNOCKING IN CI ENGINES

In CI engines, injection process takes place over a period of time. As fuel droplets enter the cylinder, they undergo the ignition delay period. During ignition delay, an addition amount of fuel is also injected. If the *ignition delay period is short*, very small amount fuel undergoes combustion process at the beginning and relatively a small amount of fuel will be accumulated in the chamber at the time of actual burning starts. This combustion releases heat rate in such a way that the pressure rise in the chamber will exert smooth force on the piston. It is the normal combustion of CI engine.

If *ignition delay period is quite long* then relatively more amount of fuel is accumulated in the combustion chamber. When actual burning commences, a large amount of fuel is burnt spontaneously from many sites in the chamber. It causes an instantaneous and abrupt pressure rise that creates large pressure differences in the chamber and resulting into jamming forces on the piston crown and results into rough engine operation. The knocking in CI engine is shown on the pressure time graph in Fig. 3.18.

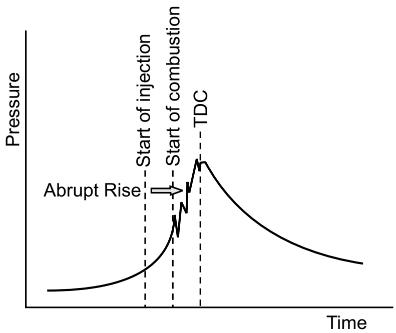


Fig. 3.18 *Pressure rise with respect to time*

Knocking in CI engine is the phenomenon which occurs due to the instantaneous abrupt pressure rise in the combustion chamber as burning of a large amount of fuel takes place that is accumulated during the long delay period.

3.16 COMPARISON OF ABNORMAL COMBUSTION IN SI AND CI ENGINES

Figure 3.19 shows the comparison of knocking in SI and CI engines on the pressure – time diagram and comparison of knocking process in SI and CI engines is tabulated below in Table 3.3 criterion wise.

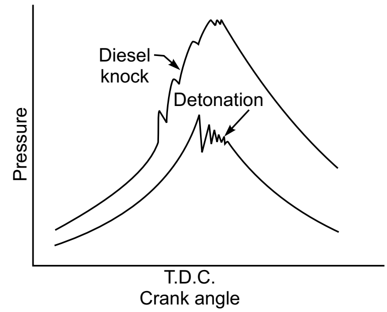


Fig. 3.19 *Comparison of Detonation in SI engines and Diesel knock in CI engines*

Table 3.3 *Comparison of knocking process in SI and CI*

Sr. No.	*Criterion*	*SI Engines*	*CI Engines*
1	Time of knock	Knocking occurs near the end of the combustion	Knocking occurs near the beginning of the combustion
2	Reason of knock	The knocking is caused by the auto-ignition of end unburned charge towards the end of combustion.	The knocking is due to accumulation of large amount of fuel that auto-ignites instantaneously as first charge at the start of combustion.
3	Intensity of knock	The air fuel mixture that self ignites in SI engines is homogeneous and thus the rate of pressure rise and hence intensity of detonation is substantial high.	The heterogeneous air-fuel mixture present at the time of self-ignition, thus the knocking is not severe.
4	Compression ratio	Compression ratio has to be limited in SI engines beyond which detonation would occur.	For CI engines, higher the compression ratio, the ignition delay will be shorter and thus lesser probability of knocking.
5	Size of engine	Larger size of cylinder promotes detonation in SI engines, therefore, the engines cylinder bore is limited to 150 mm.	The knocking in CI engines is reduced with increase in size of cylinder, therefore, cylinder bore larger than 250 mm is always for CI engines
6	Fuel rating	Fuels with high Octane rating and longer ignition lag time are good to prevent detonation in SI engines.	The fuels with higher Cetane rating are proved to have minimum possibility of knocking in CI engines. Further, fuels with higher octane rating have poor Cetane rating and vice-versa.
7	Remedial action	To avoid detonation, the possibility of auto-ignition of end unburned charge has to be minimized.	The auto-ignition of combustible mixture should occur at the earliest.

3.17 SWIRL AND SQUISH IN DIESEL ENGINES

3.17.1 Swirl

Swirl is defined as the organized rotation of charge about the cylinder axis. Swirl can also be interpreted as orderly turbulent motion of charge, which is obtained from different piston head design, combustion chamber geometry and air fuel injection methods.

Swirl ratio is defined as ratio of the air rotation speed about cylinder axis to crankshaft rotational speed. Engines are designed with a specific swirl ratio, typically 2.5.

In diesel engines, the swirl is generated during the intake process through the intake port by passing the charge into the cylinder with an angular momentum and subsequently by combustion chamber geometry during the compression stroke. The intake generated swirl is usually carried on through the compression, combustion and expansion processes.

The swirl intensity increases the tangential component of the velocity of air inside the cylinder, which promotes the rapid mixing of compressed air and injected fuel thoroughly within the combustion chamber. Swirl also affects the combustion and emission characteristics of diesel engines significantly. Action of swirl during induction of charge into the cylinder is shown in Fig. 3.20.

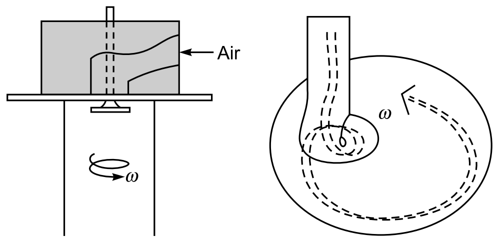


Fig. 3.20 *Air intake deflected and swirled as it enters in cylinder*

Types of Swirl

1. Induction Swirl During suction stroke, swirl can be created as shown in Fig. 3.21.

- forcing air for rotational movement (a)
- by masking one side of inlet valve (b)
- by lip over one side of inlet valve (c)

2. Compression Swirl Compression swirl can be created by

- Forcing air from periphery to the centre cavity in the piston during compression stroke.

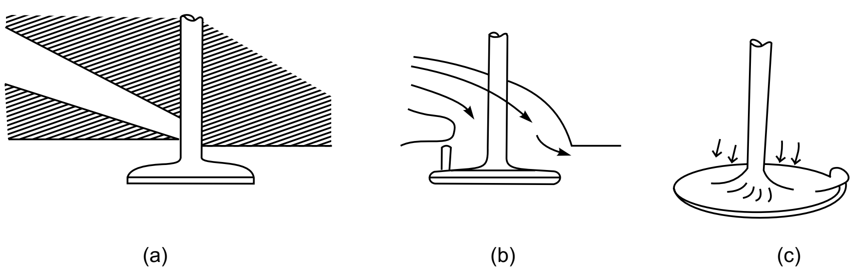


Fig. 3.21 *Types of swirl*

- The squishing of air is created and forced to enter tangentially in the piston cavity when the piston reach to TDC as shown in Fig. 3.22. Comparison between induction and compression swirl is given in Table 3.4.

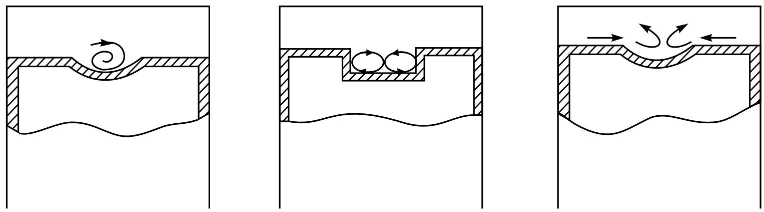


Fig. 3.22 *Compression swirl*

Table 3.4 *Comparison of induction and compression swirl*

Induction Swirl	*Compression Swirl*
Advantages	***Disadvantages***
1. High excess air (low temperature), low turbulence (less heat loss). Therefore indicated high thermal efficiency.	1. Less excess air, lower indicated thermal efficiency 5 to 8% more fuel consumption. Decreased exhaust valve life.
2. Easier starting due to low intensity of swirl.	2. Cold starting trouble due to high heat loss due to strong swirl, greater surface to volume ratio.
3. No additional work for producing swirl. High mechanical efficiency and hence high brake thermal efficiency.	3. Work absorbed in producing compression swirl, thus mechanical efficiency decreases.
4. Used with low speeds. Therefore low quality of fuel can be used.	4. Cylinder more expensive in construction.
Disadvantages	***Advantages***
1. Weak swirl, multiorifice nozzle, high injection pressure, clogging of holes. High maintenance.	1. Single injector, pintle type (self cleaning), less maintenance.
2. Influences minimum quantity of fuel. Complication at high loads and idling.	2. Large valves, higher volumetric efficiency.
3. Shrouded valves, smaller valves, low volumetric efficiency.	3. Greater air utilization due to strong swirl. Smaller (cheaper) engine.
4. Weak swirl, low air utilization (60%), lower mcp, large size (costly) engine.	4. Swirl proportional to speed, suitable for variable speed operation.
5. Swirl not proportional to speed-efficiency not maintained in a variable speed engine.	5. Smooth engine operation.

3. Combustion Swirl Swirl during combustion is created in indirect combustion engine (Fig. 3.23). It is

- Created due to partial combustion so called as a **Combustion Induced Swirl**.

- Only for pre-combustion chamber.
- Combustion during delay period in pre-combustion chamber so A/F mixture becomes rich and forces the gases with high velocity into the main combustion chamber.
- Creating high temperature and provides better combustion.

3.17.2 Squish

Squish is an effect in internal combustion **engines** which creates radially inward or transverse gas motion in the chamber towards the end of compression stroke. In an engine**,** the **squish** effect is realized, when piston crown comes very close to top dead centre near the cylinder head. Near the end of compression stroke, air is squeezed in the piston crown as shown in Fig. 3.23. Squish causes a flow of air from the periphery of the cylinder to its center and into the recess in the piston crown.

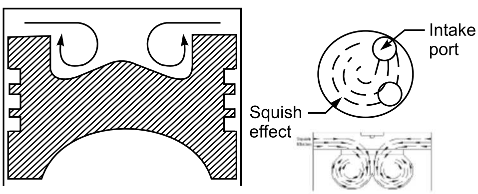


Fig. 3.23 *Illustration of squish effect in cavity in piston crown*

Squish effect is generally observed in engines with overhead valves and overhead camshafts.

3.18 COMBUSTION CHAMBERS IN DIESEL ENGINES

The shape of the combustion chamber is one of the important factors. It decides the quality of combustion, engine performance and exhaust characteristics. Diesel engine combustion is greatly influenced by air turbulence, created by the shape of combustion chamber. Each combustion chamber shape creates its own unique turbulence pattern that is required for some specific applications while not suitable for others.

3.18.1 Objective of Good Combustion Chamber Design

1. To optimize the induction and discharge of the cylinder with fresh and burnt charges respectively over the engine's operating range.
2. To facilitate thoroughly mixing of air and fuel in the cylinder into a highly turbulent state.
3. To complete the burning of the charge in the shortest possible time.

3.18.2 Factors Considered in Combustion Chamber Design

The following factors are crucial when designing a combustion chamber for a diesel engine

- Heat loss to combustion chamber walls
- Injection pressure
- Nozzle design: Number, size, and number of holes in the nozzle
- Effortless starting
- Fuel requirement: Ability to burn less expensive and poor quality fuels
- Utilization of air: Ability to use maximum amount of air in cylinder
- Weight to power output ratio of engine
- Capacity to operate at wide range of speed
- Smoothness with which force created by expanding gases on to the piston
- Ease of maintenance.

3.19 DIRECT INJECTION (DI) COMBUSTION CHAMBER

This type of combustion chamber is also called an *open combustion chamber*. In DI combustion chambers, the entire volume of the combustion chamber is located in the main cylinder and the fuel is injected into its volume. The Direct Injection combustion chamber is shown in Fig. 3.24. Its use is increasing due to their more economical fuel consumption (up to 20% savings).

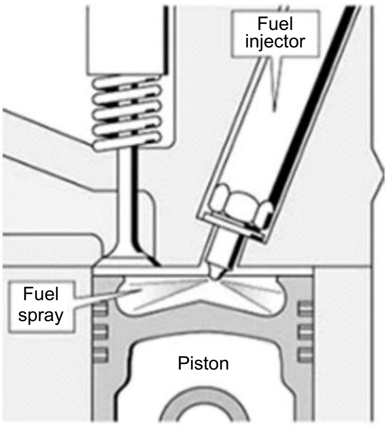


Fig. 3.24 *DI combustion chamber*

3.19.1 Features of Direct Injection Combustion Chambers

- For DI engine, ω piston crown recess is most widely used. In this design, the fuel is injected directly into the cylinder chamber.
- Lower combustion surface wall area compared to combustion volume in comparison with IDI.
- More combustion takes place in and on the piston surface and less contact with coolant.
- DI chamber has highest fuel efficiency rating compared to other chamber design.
- Smaller engines tend to be of the high-swirl type, while bigger engines tend to be of the quiescent type.

3.19.2 Advantages and Disadvantages

Advantages The main advantages of this type of chambers are:

(1) Minimum heat loss takes place during compression of air due to lower surface area to volume ratio and hence, it has better efficiency.

(2) Engine can easily be started in cold conditions.

(3) Multi-hole nozzle is used to ensure fine atomization.

Disadvantages The main drawbacks of these combustion chambers are:

(1) Engine requires high fuel injection pressure and hence complex design of fuel injection system.

(2) Accurate metering of fuel is required by the injection system, particularly for small engines.

3.19.3 Classification of Direct Injection Combustion Chambers

DI combustion chambers can be classified as

(a) Shallow depth chamber,
(b) Hemispherical chamber,
(c) Cylindrical chamber, and
(d) Toroidal chamber

(a) Shallow Depth Chamber The shallow depth combustion chamber has large diameter cavity in the piston. The depth of the cavity is moderately small. This chamber is normally used for large engines running at low speeds, since the squish is negligible. It shown in Fig. 3.25(a).

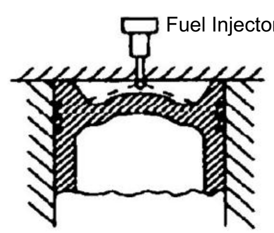


Fig. 3.25(a) *Shallow depth combustion chamber*

(b) Hemispherical Chamber The hemispherical combustion chamber provides small squish and swirl. However, in this chamber, the depth to diameter ratio can be varied to give any desired squish for better engine performance. It shown in Fig. 3.25(b).

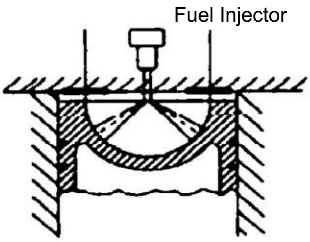


Fig. 3.25(b) *Hemispherical combustion chamber*

(c) Cylindrical Chamber Cylindrical combustion chamber is a modified version of the hemispherical chamber. It has cavity in the form of a truncated cone with a base angle of 30 degree. The swirl was produced by masking the valve for nearly 180 degree to the circumference. Squish can also be varied by varying the depth. It shown in Fig.3.25 (c).

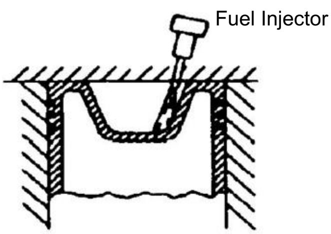


Fig. 3.25(c) *Cylindrical combustion chamber*

(d) Toroidal Chamber The toroidal combustion chamber provides a powerful squish along with the swirl in form of smoke ring within the chamber. Due to powerful squish the mask needed on inlet valve is small and there is better utilization of oxygen. The cone angle of spray for this type of chamber is 150 to 160 degree. It shown in Fig. 2(d).

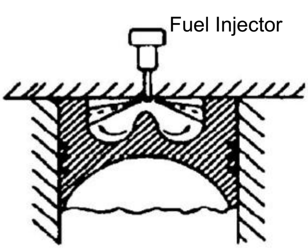


Fig. 3.25(d) *Toroidal combustion chamber*

3.20 SWIRL IN COMBUSTION CHAMBER

According to swirl creation, the combustion chambers are also classified as

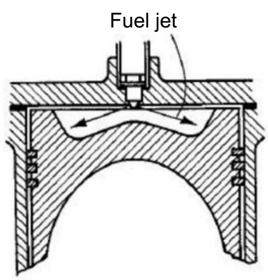
(a) Low-swirl or quiescent engines
(b) Semi-swirl combustion chamber, and
(c) High-swirl design combustion chamber.

3.20.1 Low-swirl or Quiescent Engines

These are characterized by having a **shallow bowl** in the piston, a **large number of holes** in the injector and higher injection pressures. The air movement is almost quiescent and the mixing of the fuel and air is purely achieved by the intensity and distribution of the spray and atomizing fuel particles, therefore known as *quiescent open chambers*.

The main features of low-swirl or quiescent combustion chambers are

- They are suitable for large, slow and medium speed engines running up to 1500 rev/min.
- There is sufficient time for the fuel to be injected into the cylinder and for it to be distributed and thoroughly mixed with the air charge so that combustion takes place over the most effective crank angle movement just before and after TDC, without induction swirl and compression squish.



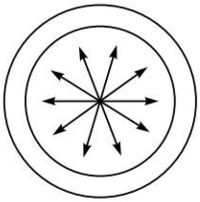


Fig. 3.26 *Shallow depth combustion chamber with quiescent air*

- Further, they have very low ratio of surface area to volume, and no significant velocity of hot gases. Therefore, heat loss from the engine is the least compared with all other semi-open or divided combustion chambers, thus its thermal efficiency is the highest.
- The injector is located in the center of a four valve cylinder head. The injector nozzle has 8 to 12 holes all equally spaced and pointing radially outwards so that they are directed towards the shallow wall of the combustion chamber as shown in Fig. 3.26.
- Usually, open quiescent combustion chambers provide good cold starting and the lowest specific fuel consumption relative to semi-open and divided combustion chambers.

3.20.2 Semi-swirl Combustion Chamber

It has a slightly offset bowl in the piston combustion chamber surrounded by a large annular squish zone formed between the piston crown and flat cylinder head. The combustion phenomenon in semi-swirl combustion chamber is discussed below with the help of Fig. 3.27.

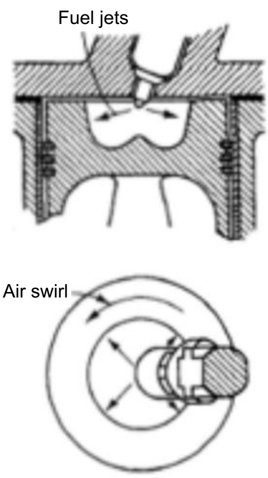


Fig. 3.27 *Semi-swirl combustion chamber with multi-hole nozzle*

The induction of swirl in charge during induction, compression, injection, ignition and expansion in semi-swirl combustion chamber is illustrated with the help of Fig. 3.28.

- The incoming air enters in a tangentially downward direction due to the valve port and seat being positioned to one side of the cylinder axis. Air is thus forced in spiral way down in the cylinder as shown in Fig. 3.28(a).

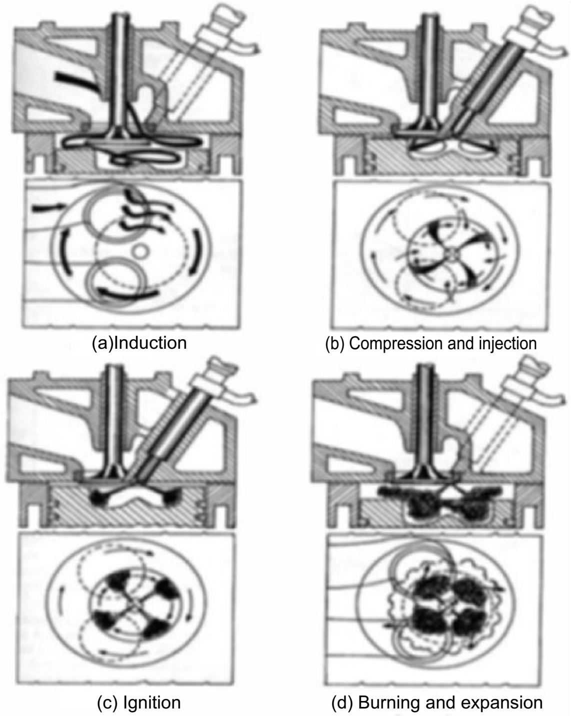


Fig. 3.28 *Induction, compression, injection, ignition and expansion in semi-swirl combustion chamber*

- Towards the end of the compression stroke the bump-clearance between the flat annular piston crown and the cylinder head quickly decreases causing to squeeze the swirling air inward towards the inner chamber bowl. The air stream from all sides of the annual squish zone flows radially inward meeting at the center bottom of bowl from where it is then deflected radially outward. The upward moving air experiences inward moving compression squish which again pushes the air towards the center and down as shown in Fig. 3.28(b).

- The fuel is injected radially outwards until it strikes the chamber wall. Some of this fuel bounces back off the wall while the remainder stays and spreads over the wall. The fuel jets first become finely atomized and then vaporized and heated to get its self-ignition, causing ignition to occur as shown in Fig. 3.28 (c).
- The nuclei of flames, established randomly around the fuel vapors, propagate rapidly towards the bulk of the mixture concentration near the chamber walls, the flames are then distributed and spread throughout the bowl due to the general air movement within the chamber as shown in Fig. 3.28(d).
- During expansion stroke, the outward movement of the piston enables mixing of air and fuel efficiently by the combined effect of air swirl and reversed squish.

3.20.3 High-swirl Design Combustion Chambers

The high-swirl design combustion chamber is a spherical cavity in the piston crown with a small secondary recess on one side, which aligns with the injector in the cylinder head to provide access for the fuel spray discharge as shown in Fig. 3.29. The injector has a **low number of holes** and moderate injection pressures. It uses two valves in cylinder head with a high swirl or vortex type induction port with an inclined injector, which is located to one side of the cylinder axis.

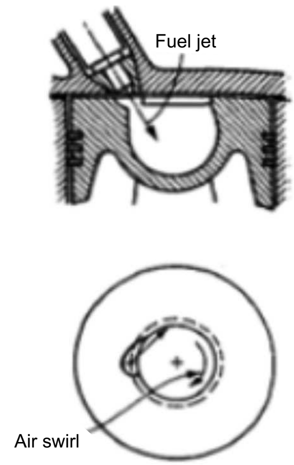


Fig. 3.29 *High-swirl combustion chamber*

Salient features and formation of swirl of the combustion chambers are discussed below with the help of Fig. 3.30.

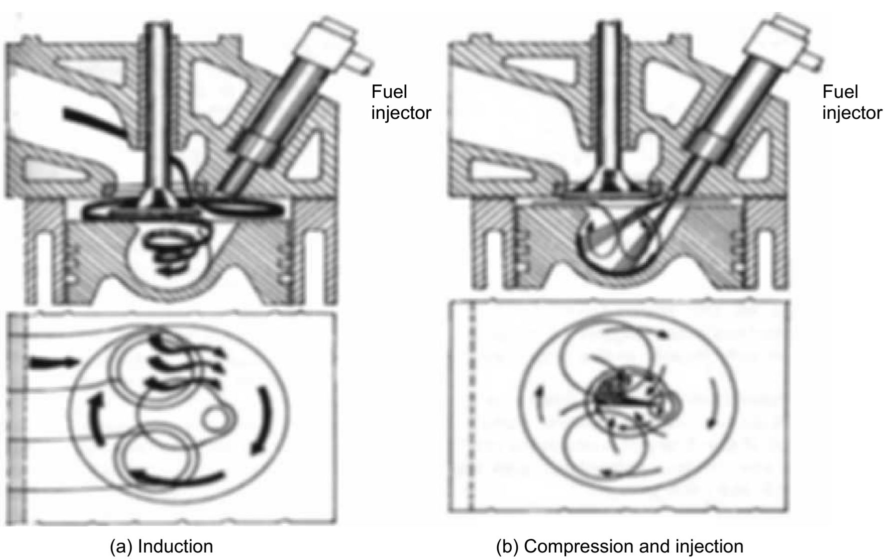


Fig. 3.30 *Induction and injection in semi-open M Type DI combustion chamber*

- A high degree of swirl is generated in air within the curved induction port passage before it enters into the cylinder and then air is forced to rotate about the cylinder axis in a progressive spiral fashion as the piston moves away from the cylinder head on its induction stroke as shown in Fig. 3.30 (a).
- After the cylinder has been filled with air having a high intensity of swirl, the inlet valve closes and the air is compressed between the cylinder head and the inwardly moving piston crown.
- As the piston rapidly approaches TDC, air from the annular squish area surrounding the chamber recess is squeezed towards the center of the chamber; it is then forced downward to follow the contour of the spherical chamber wall. The direction of air is changed from the annular squish area to the inner chamber causes the rotational movement of the air around the cylinder to be considerably increased as it moves into the much smaller spherical chamber.
- Near to the end of the compression stroke, fuel is injected into the cylinder from two nozzle hole set at acute angles to the chamber walls so that after the spray penetrates the swirling air reaches the cylinder wall, it is not reflected but spreads over the surface in the form of a thin film as shown in Fig. 3.30(b).
- 5 to 10% of the total quantity of fuel discharged per cycle burns in the spray stream near the injection nozzle with the minimum delay period. The vaporized fuel is carried away by the air stream and burns in the flame front spreading from the initial ignition zone to the center of the chamber. The energy released due to combustion, causes a rapid pressure rise and simultaneously an expansion of the burning charge as shown in Fig. 3.30(c).

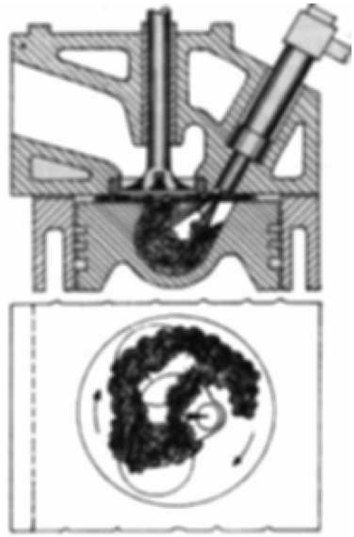


Fig. 3.30(c) *Combustion process in semi-open M-Type DI combustion chamber*

3.21 INDIRECT INJECTION (IDI) COMBUSTION CHAMBERS

Indirection injection (IDI) combustion chambers have the combustion space divided into two or more compartments connected by restricted passage(s). During combustion, there is a substantial pressure difference between them that creates a high degree turbulence and swirl.

3.21.1 Features of IDI Combustion Chambers

The good and bad features of IDI combustion chambers are

Good Features

- Excellent mixing of burnt and unburned charge is possible due to very high degree turbulence characteristics of the chamber.
- In IDI combustion chamber, low quality fuel can easily be burnt.
- They require lower injection pressure
- The chances of knock are minimum in these combustion chambers.

- They create low noise and complete combustion is possible with low exhaust emissions.

Bad Features

- IDI combustion chamber has very high temperature and pressure in injection chamber
- Due to high temperature during combustion, there is possibility of formation NOx in emission
- Engine with IDI combustion chamber require more power to start and glow plugs
- They are less efficient.

3.21.2 IDI Combustion Chamber Types

- Turbulent pre-combustion chamber type
- Swirl pre-combustion chamber type
- Air-cell pre-combustion chamber type.

3.21.3 Pre-combustion Chamber

Figure 3.31 shows an indirect fuel injection pre-combustion chamber. The combustion space is divided into pre-chamber and a spherical combustion space. The pre-combustion chamber is mounted in a heat resisting alloy in the cylinder head slightly to one side of the single inlet and exhaust valve seats. The glow plug helps in initiation of nuclei of flame at the start of cold engine conditions.

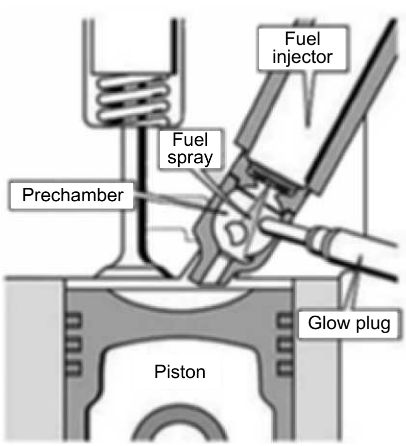


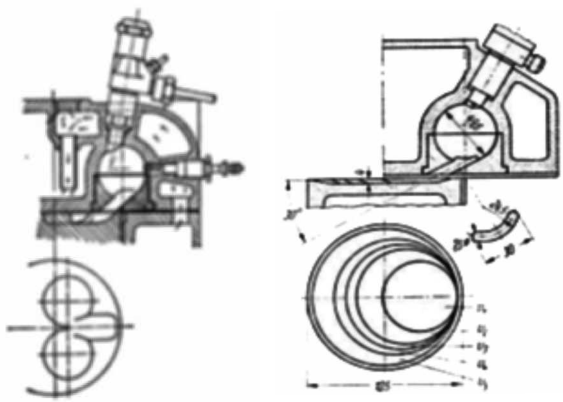
Fig. 3.31 *Indirect injection pre-combustion chambers*

During the compression stroke, as piston approaches TDC, 40% to 50% of compressed air enters into pre-chamber through throat with a vigorous and high turbulence. The fuel is injected through a pintle nozzle. It has a specially shaped baffle in the centre of the chamber that diffuses the jet of fuel and mixes it thoroughly with the air.

The resulting pressure from burning of charge forces burnt and unburnt charge through the throat into the piston crown. The thrust of combustion of gases project the directional jet of flame-fronts towards the cylinder walls and in doing so, sweeps the burnt gases and soot to one side while exposing the remaining fuel vapour to fresh oxygen.

3.21.4 Swirl Combustion Chamber

The swirl indirect injection combustion chamber is divided into two chambers. The upper half is a sphere swirl chamber, cast directly in the cylinder head, and the lower half is a separate twin disc shaped recesses in the piston crown as shown in Fig. 3.32. The combustion is initiated in the swirl chamber that has approximately 60% of the compression volume. As soon as combustion starts in swirl chamber, the air-fuel mixture is forced under pressure through the throat into the main cylinder chamber, where it is turbulently mixed with the remaining compressed air as shown in Fig. 3.32 (b).



(a) (b)

Fig. 3.32 *Swirl Indirect injection combustion chamber.*

3.21.5 Air-cell chamber

This chamber is divided into the main combustion chamber and energy cell. The energy cell is divided into two parts, major and minor, which are separated from each other and from the main chamber by narrow orifices as shown in Fig. 3.33. The high degree turbulence is created by an energy cell.

During the compression stroke, the piston forces a small amount of compressed air into the energy cell. Near the end of the compression stroke, a pintle type of nozzle injects the fuel, a small quantity of fuel is directed into the cell and remaining injected into main combustion chamber as shown in Fig. 3.33 (a).

While the fuel charge is traveling across the center of the main chamber, the fuel mixes with the hot air and burns at once. The remainder of the fuel enters the energy cell and starts to burn. At this point, the cell pressure rises sharply, causing the products of combustion to flow at high velocity back into the main combustion chamber as shown in Fig. 3.33(b). This sets up a rapid swirling movement of fuel and air in each lobe of the main chamber, promoting the final fuel-air mixing and ensuring complete combustion as shown in Fig. 3.33(d).

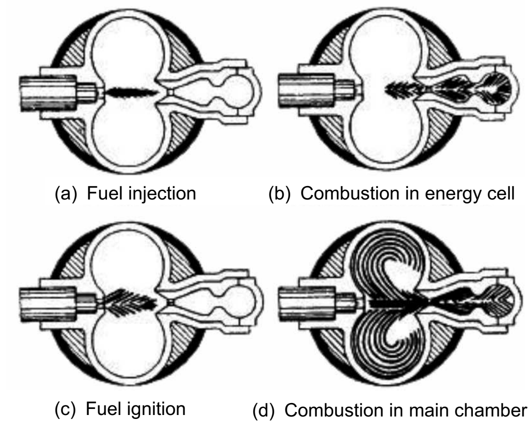


Fig. 3.33 *Different operations in Lanova air cell indirect injection combustion chamber*

The two restricted openings of the energy cell control the time and rate of expulsion of the turbulence-creating blast from the energy cell into the main combustion chamber. Therefore, the rate of pressure rise on the piston is gradual, resulting in smooth engine operation. The air cell combustion chamber gives a cleaner more complete burning in the cylinder. This system is very efficient and provides lots of medium rpm torque.

Summary

- Combustion is a chemical reaction of certain elements present in the fuel with oxygen in the atmosphere air. The engines operating on different loads and speeds require different air–fuel ratios.
- The homogeneous mixture of fuel and air prepared in carburetor as charge is inducted in the spark ignition engine. This charge is compressed in the cylinder during compression stroke. At the end of the compression stroke, the combustion is initiated by a high intensity spark produced by an electric discharge. ***First phase of combustion*** is referred as the preparation phase (*ignition lag*) in which fuel elements become ready to react chemically with the oxygen present in compressed air.
- The engine knock occurs when some of the unburnt gases ahead of the flame in SI engine are ignited spontaneously. If the temperature of the end unburnt mixture at some instant exceeds the self-ignition temperature of fuel, the auto-ignition of unburnt mixture occurs. During auto-ignition, another pressure wave starts traveling in the opposite direction to the main pressure wave. When the two pressure waves collide, a severe pressure pulse is generated. This phenomenon is also called *detonation*.

- Knock in SI engine results into rough and noisy operation of engine, increased heat transfer, decreased thermal efficiency, pre-ignition and sometimes complete failure of engine by mechanical damage.
- The design of combustion chamber in SI engine should offer smooth operation and high output and thermal efficiency.
- Combustion phenomenon in CI engine is completely behaved differently than in SI engine. In CI engine, only air as charge is inducted during the suction stroke. It undergoes high compression ratio. The temperature of pressure of air increase to very high value, then fuel is injected through one or more jets into highly compressed air in the combustion chamber. After injection, the auto-ignition of fuel air mixture from multiple location initiate burning.
- Ignition delay in the diesel engine is the time period required to start actual burning of mixture after injection of the first fuel droplets into the combustion chamber. In diesel engine, the ignition delay period is the combination of physical and chemical delays.
- If ignition delay in diesel engine is quite long, a large amount of fuel is accumulated in the combustion chamber before actual burning commences. It causes instantaneous abrupt pressure rise in combustion chamber resulting into rough and noisy operation of engine. This situation is called diesel knock.
- Knocking in CI engines occurs at the beginning of combustion process, while knock in SI engine occurs near end of combustion process.
- The shape of the combustion chamber is one of the important factors. It decides the quality of combustion, engine performance and exhaust characteristics.
- In DI combustion chambers, the entire volume of the combustion chamber is located in the main cylinder and the fuel is injected into this volume.
- Indirection injection (IDI) combustion chambers have the combustion space divided into two or more compartments connected by restricted passage(s). During combustion, there is a substantial pressure difference between them during combustion, that creates a high degree turbulence and swirl.

Glossary

Atomization Disintegration of fine spray jet of fuel

Chemical Delay Time lapse for development of inflammation after the beginning of reaction.

Combustion Exothermic chemical reaction between fuel and oxygen

Detonation Auto-ignition of end unburnt charge before reaching of flame to it.

DI Combustion Chamber Entire volume of combustion space is located in main cylinder.

IDI Combustion Chamber Combustion space is divided into two or more parts.

Ignition Delay The time period required to start actual burning of mixture after injection of the first fuel droplets.

Ignition Lag Preparation time in which fuel elements become ready to react with oxygen.

Physical Delay Time lapse between the beginning of fuel injection and start of chemical reaction.

Pre-ignition Ignition of charge before commencement of spark due to local heating in the cylinder.

Squish Inward or transverse gas motion in the chamber near end of compression stroke.

Swirl Organized rotation of charge of gas about the cylinder axis.

Swirl ratio Ratio of the air rotation speed about cylinder axis to crankshaft rotational speed.

Review Questions

1. Explain the abnormal combustion in SI engines.
2. Discuss then engine variable affect the knocking in SI engines.
3. Define ignition delay. How it differs from ignition lag?
4. Indicate whether the following parameter increase or decrease the knock in SI and CI engine respectively.
   (a) speed,
   (b) cylinder size, and
   (c) ignition delay.
5. Explain the stages of combustion in SI engine.
6. Why compression ratio is restricted to maximum 12 in case of petrol engines?
7. Explain the phenomenon of knocking in SI engine.
8. Discuss the various factors responsible for knocking in SI engine.
9. Discuss the stages of flame propagation in combustion chamber in SI engine.
10. Discuss the factors affecting the ignition lag.
11. Enumerate the effect engine variables on flame propagation in SI engine.
12. Explain the various factors that influence the flame speed.
13. What is meant by flame speed and how to measure it?
14. Briefly explain the stages of combustion in SI engines elaborating the flame front propagation.
15. What is meant by ignition delay in CI engines and explain it with p-θ diagram.
16. What is meant by knock in SI engines and what are the parameters causing their effect on it?
17. How will you define abnormal combustion in SI engine?
18. Discuss the effect of density factor on knocking in SI engine.
19. Enumerate the effects of time factor on knocking in SI engine.
20. Discuss the effect of compression ratio, inlet temperature, power output and coolant temperature on knocking in SI engine.
21. What are requirements of a good combustion chamber in SI engine?
22. Draw and explain T-head and L-head combustion chamber of SI engine.
23. What are the advantages of use of overhead valves combustion chamber in SI engine?
24. What is physical delay? Discuss the factors affecting the physical delay period.
25. Discuss the stages of combustion in CI engine.
26. Explain the heat release pattern in CI engine.
27. Discuss the factors affecting ignition delay.
28. What is delay period and what are the various factors that affect the delay period?
29. Draw p-θ diagram for CI engine combustion and indicate the stages.
30. Discuss then engine variable affect the knocking in CI engines.
31. Explain the phenomenon of knocking in CI engines.
32. Compare abnormal combustion in SI and CI engines.
33. Define swirl and squish.
34. What are the different types of swirl and how are these created?
35. Distinguish between suction, compression and combustion induced turbulence for CI engine.
36. Compare induction swirl and compression swirl.
37. What are the objectives of a good combustion chamber design in CI engine?
38. What are the factors considered during design of combustion chamber in CI engine?
39. Draw a direct injection combustion chamber for a CI engine and explain its operation.
40. State the features of direct injection combustion chamber of a CI engine.
41. What are advantages and disadvantages of direct injection combustion chamber of a CI engine.
42. At least two combustion chambers required in CI engines represent by line diagram and explain its working.
43. Discuss the features of IDI combustion chamber of a CI engine.

44. Name three combustion chamber of IDI type of CI engine.
45. Discuss the swirl IDI combustion chamber of a CI engine.
46. What is advantage of using energy cell in Lanova air cell combustion chamber?
47. Bring out clearly the process of combustion in CI engines and also explain the various stages of combustion.

Objective Questions

1. In SI engines, the flame speed increases with
   (a) turbulence, (b) fuel air ratio,
   (c) both (a) and (b), (d) none of above
2. In SI engines, with increase in compression ratio, the flame speed
   (a) increases, (b) decreases,
   (c) remains the same, (d) none of above
3. Knocking tendency in SI engine with increase in engine speed
   (a) increases, (b) decreases,
   (c) not affected, (d) none of above
4. Detonation in SI engine occurs due to
   (a) Pre-ignition of charge before spark,
   (b) Sudden ignition of charge before spark,
   (c) Auto-ignition of charge after spark struck,
   (d) None of above
5. Which one of the following is not an anti knock agent
   (a) Tetraethyl lead
   (b) Aminonium hydroxide
   (c) Benzene
   (d) ethyl alcohol
6. For minimizing knocking tendency in SI engines the spark plug should located
   (a) Near inlet valve
   (b) Away from inlet valve
   (c) Near exhaust valve
   (d) between inlet & exhaust valve
7. For high thermal efficiency, SI engine should have
   (a) High compression ratio
   (b) Excess air
   (c) No turbulence
   (d) Maximum heat loss
8. If the ignition of a charge inside the cylinder occurs before the passage of spark , it is knock as
   (a) Detonation (b) ping
   (c) Pre-ignition (d) afterburning
9. In order to get high volumetric efficiency in SI engines
   (a) Residual exhaust gases should be high
   (b) Diameter of inlet valve should be large
   (c) Exhaust valve diameter should be large
   (d) Location of inlet valve directly above piston centre
10. In SI engines for smooth engine operation
    (a) Pressure rise during combustion should be moderate
    (b) Flaming travel should be short
    (c) Combustion space should be compact
    (d) All of the above
11. Thermal efficiency of a SI engines at load is usually in the range
    (a) 10-25 per cent
    (b) 30-35 percent
    (c) 50-60 per cent
    (d) 75-90 percent
12. In CI engines, with increase in compression ratio the delay period
    (a) increases, (b) decreases
    (c) not affected (d) none of above
13. In CI engine, the delay period is affected by
    (a) Compression ratio (b) engine speed
    (c) power output (d) All of above

14. Thermal efficiency of a spark ignition engine increase with
    (a) Speed
    (b) Lubrication
    (c) Compression ratio
    (d) Cylinder dimensions
15. Knocking tendency of a CI engine increases with
    (a) Increase in compression ratio
    (b) Increase in inlet air temperature
    (c) Decrease in compression ratio
    (d) increase in jacket water temperature
16. The direct injection combustion chambers in CI engines require
    (a) high injection pressure
    (b) increased compression ratio
    (c) high air inlet temperature
    (d) high water temperature in jacket
17. Indirect combustion chamber has advantage of
    (a) low compression ratio
    (b) accurate metering of fuel by injection system
    (c) high fuel injection pressure
    (d) high water temperature in jacket
17. Indirect combustion chamber has advantage of
    (a) low compression ratio
    (b) accurate metering of fuel by injection system
    (c) low fuel injection pressure
    (d) high water temperature in jacket
18. Lanova air cell indirect injection combustion chamber has advantage of
    (a) low fuel injection pressure
    (b) gradual pressure rise
    (c) insignificant direction of spray
    (d) all of above

Answers

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

4 CHAPTER

Testing and Performance of IC Engines

Introduction

Internal combustion engines usually operate within certain range of speeds and loads. An engine is supposed to give its better performance at rated speed within certain loading conditions. The specific fuel consumption of an engine is a very important parameter, that varies with change in load and speed. The performance of an engine is an indication of the degree of success of the engine for its assigned duty, i.e. the inter-relationship between mechanical power developed, speed and the specific fuel consumption at different operating conditions. The performance of an engine is evaluated with following objectives:

1. To justify rating of an engine,
2. To get information which cannot be obtained by calculations
3. To validate the data used in design.

An engine is selected for a particular application on the basis of its power output and rated speed. Other factors include capital cost and operational cost. Engine performance is expressed in terms of certain parameters. Therefore, certain measurements and calculations are required to judge the performance of an engine. These are enumerated below.

1. Indicated power
2. Brake power
3. Frictional power
4. Fuel consumption
5. Air consumption
6. Brake thermal efficiency
7. Indicated thermal efficiency
8. Mechanical Efficiency
9. Volumetric efficiency
10. Air–fuel ratio.

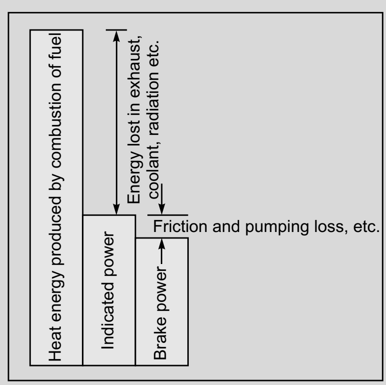


Fig. 4.1 *Heat Energy distribution in IC engine*

The distribution of energy produced in the engine by the combustion of fuel is shown in Fig. 4.1.

4.1 INDICATED POWER (*IP*)

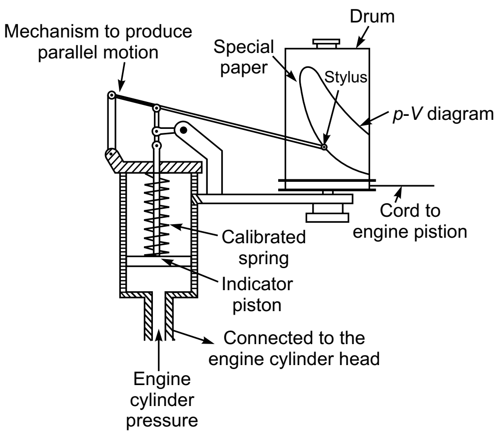
It is defined as the *rate of work done* on the piston by burning of charge inside the cylinder. It is evaluated from an indicated diagram obtained from the engine. Indicated power of an engine gives an indication about the health of the engine and conversion of heat energy of fuel into power. Indicated power of an engine can be estimated by

(a) Using indicator diagram, and
(b) Addition of brake power and friction power

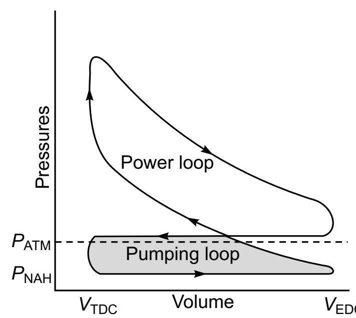
4.1.1 Measurement of Indicated Power

Engine indicator includes the measurement of instantaneous in-cylinder pressure against the instantaneous crank angle. The indicated power of an engine is normally measured by making an indicator diagram with a device, called engine indicator as shown in Fig. 4.2(*a*). Indicator diagram is a graphical representation of the pressure and volume changes during the movement of piston in a cycle on a paper, Fig. 4.2(*b*).

The engine indicator is connected to cylinder head. It consists of piston cylinder assembly, which communicates with gas inside the engine. The indicator piston slides in the cylinder and the indicator piston rod is connected to the linkage mechanism through the calibrated spring. The linkage mechanism is produced parallel notion. The spring controls the piston movement, which is caused by pressure variation inside the cylinder. The movement of indicator piston is transferred to a stylus (pencil) by a linkage.



(a) Engine indicator



(b) Indicator diagram

Fig. 4.2 *Engine indicator and indicator diagram*

The stylus is moved in vertical direction in proportion to record the gas pressure corresponds to the piston movement. The drum rotates about its axis by pulley and cord connected to engine piston through the reducing mechanism. The movement of the cord is proportion to the piston stroke. The vertical movement of the stylus and the horizontal movement of the cord together generate a close figure. This figure is known as indicator diagram. The net area of the diagram represents the net work developed by engine and length of the diagram represents swept volume. The area of the indicator diagram can be measured by planimeter.

$$\text{Net work done by engine} = \text{Area of positive loop} - \text{Area of negative loop} \quad \ldots(4.1)$$

The indicated mean effective pressure (p_{mi}) is given by

$$p_{mi} = \frac{\text{Area of indicator diagram}}{\text{length of the indicator diagram}} \times \text{spring constant} \quad \ldots(4.2)$$

The indicator diagram is capable to supply genuine information of process of combustion, ignition lag, and actual pressure rise.

The indicated power is also called gross power of engine. It is calculated as

IP = Indicated mean effective pressure × Swept volume rate

$$= \frac{p_{mi}\, L\, A\, n\, k}{60}\ (\text{kW}) \quad \ldots(4.3)$$

where p_{mi} = Indicated mean effective pressure, (kPa or kN/m^2)

L = Stroke length, (m)

A = $(\pi/4)\, d^2$, cross-sectional area of cylinder of bore, d, (m)

n = Number of power strokes per minute, when engine has a speed of N rotations per minute

= N for two-stroke engines

= $\frac{N}{2}$ for four-stroke engines

k = Number of cylinders

4.1.2 Measurements of Cylinder Pressure and Engine

The pressure in the cylinder of an internal combustion engines is measured by pressure transducers. High-speed electronic transducers are capable to convert the deflection of a low inertia diaphragm into an electrical signal. They have piezoelectric crystals as measuring elements exhibit superior characteristics to thermal solicitations than those based on strain gauges. The piezoelectric transducers are popular to measure the in-cylinder pressure. Piezoelectric transducers are capable of maintaining high characteristics in frequency response and linearity over a wide range of pressures. Although, they have drawbacks to their use include instability of the baseline and low intensity of its output signal.

Operating Principle of the Piezoelectric Pressure Transducer

Figure 4.3 illustrates the principle of operation of a piezoelectric pressure transducer. The pressure

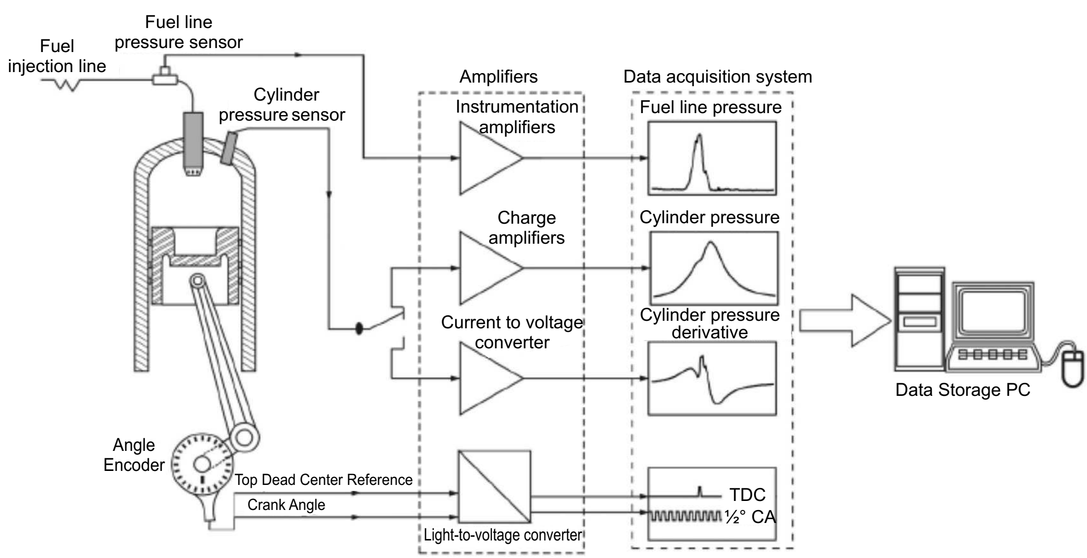


Fig. 4.3 *A typical engine indicating measurement system*

change rate of (dp/dt) experienced by the transducer diaphragm is transmitted to a piezoelectric crystal through intermediate elements, causing its deformation at a rate of $d\varepsilon/dt$. Due to piezoelectric effect, this deformation polarizes charge q in the transducer electrode originating an electric current i, which generates the transducer output signal:

$$i = -\frac{dq}{dt} = -G_s \frac{dP}{dt} \quad \ldots(4.4)$$

where G_s is the transducer sensitivity (gain).

During the measurement of in-cylinder pressure, the transducer is exposed to a transient heat flow that causes continuous changes in its temperature. These temperature changes modify the sensitivity of the piezoelectric element and impose thermal stresses in the diaphragm and in the sensor housing, generating artificial forces that act on the quartz element and cause additional distortion of the signal provided by the transducer.

4.2 MEASUREMENT OF BRAKE POWER: BRAKE DYNAMOMETERS

4.2.1 Brake Power (*BP*)

It is the *net power* available at the engine shaft for external use. However, it is less than the power developed in the cylinder due to various *frictional losses.* The brake power (BP) is measured by dynamometers The measurement of brake power involves determination of torque and angular speed of the engine shaft. Torque measuring device is called dynamometer. It is calculated as

Brake power = brake load (F) × velocity of brake drum $\left(\frac{2\pi RN}{60}\right)$

$$\text{or} \quad BP = \frac{2\pi N(FR)}{60{,}000} = \frac{2\pi NT}{60{,}000} \text{ (kW)} \quad \ldots(4.5)$$

where, F = Braking force, (N)

T = FR, the torque is the product of force F and effective radius R of the brake drum

N = Speed of the engine in rpm

Brake power can also be obtained in terms of brake mean effective pressure (p_{mb}) as

BP = Brake mean effective pressure × Swept volume rate

$$= \frac{p_{mb}\, L\, A\, n\, k}{60} \text{ (kW)} \quad \ldots(4.6)$$

where, p_{mb} = Brake mean effective pressure, (kPa or kN/m^2)

L = Stroke length, (m)

A = $(\pi/4)\, d^2$, cross-sectional area of cylinder of bore, d, (m)

n = Number of power strokes per minute, when engine has a speed of N rotation per minute

= N for two-stroke engine

= $\frac{N}{2}$ for a four-stroke engine

k = Number of cylinders

Dynamometers can be broadly classified as absorption type and transmission type.

(i) ***Absorption Dynamometers*** These dynamometers are coupled to engine output shaft and they absorb engine power by frictional resistance. The prony brake, rope brake, hydraulic and eddy current dynamometers are some example of absorption dynamometers.

(ii) ***Transmission Dynamometers*** These dynamometers are also coupled to the engine and they transmit engine power to the load and it is indicated on some scale.

4.2.2 Prony Brake

It is the simplest method to measure engine brake power by attempting to stop the engine by absorbing its power with an application of brakes on flywheel. Prony brake consists of frame with two wooden brake shoes gripping the flywheel as shown in Fig. 4.4. The rim pressure can be adjusted by means of nuts operating through compression springs on tie bars. A torque arm extends from the top of the upper brake and to the end of the load bar. The weight can be hung on the weight hanger.

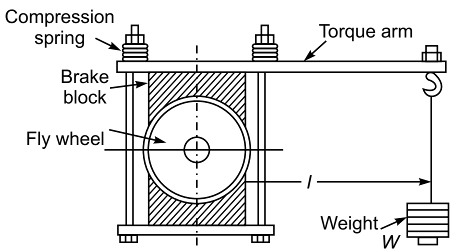


Fig. 4.4 *Prony Brake Dynamometer*

When the wooden block is pressed against the rotating flywheel, it tries to stop flywheel by opposing engine torque. Engine power absorbed by brakes is converted into heat through frictional resistance. Hence this type of dynamometers requires cooling arrangement.

The brake power is calculated by

$$BP = \left(\frac{2\pi NT}{60000}\right)$$

Where

T = torque (N-m) = Weight applied (mg) × distance (L) and

N = speed of flywheel, rpm

4.2.3 Rope Brake Dynamometer

It is also an absorbing type dynamometer. A schematic of a rope brake dynamometer is shown in Fig. 4.5. A rope wrapped over the fly wheel (brake drum) applies frictional resistance to the rotation of the brake drum, mounted on the engine shaft.

Let W = Weight applied on rope, N (= mg)
S = Spring balance reading, N
N = Speed of crank shaft, *rpm*
R = Effective radius of the brake drum (m)
= 1/2 (Dia. of brake drum + Dia. of the rope)

Then $BP = \text{Force} \times \dfrac{\text{displacement}}{\text{time}}$

$$= \frac{(W - S)}{1000} \times 2\pi R \frac{N}{60} \text{ (kW)} \qquad ...(4.7)$$

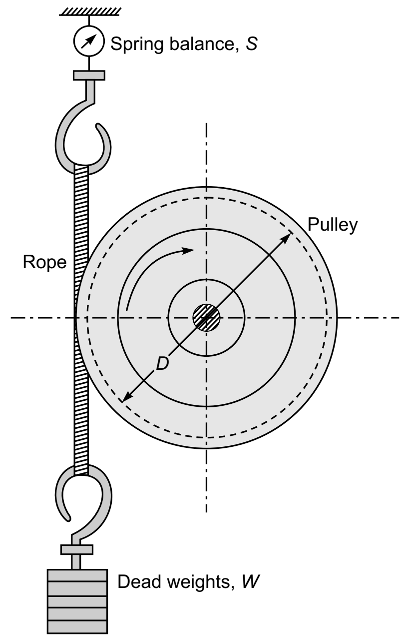


Fig. 4.5 *Rope brake dynamometer*

4.2.4 Hydraulic Dynamometer

A hydraulic dynamometer is also an absorbing type dynamometer as shown in the Fig. 4.6. It works on the principle of dissipating the power in fluid friction instead in dry friction.

The hydraulic dynamometer operates like a hydraulic turbine/pump. The working medium, usually water, is circulated within the housing creating frictional resistance to impeller rotation.

The hydraulic dynamometer consists of an impeller an inner rotating member coupled with engine output shaft. The impeller rotates in a casing filled with a fluid. The stator (outer casing), which is free to rotate with the input shaft, is connected to a torque arm supporting the balance weight. The output power can be controlled by regulating the sluice gates which can be moved in and out, partially or wholly to obstruct the flow of water between the casing and the impeller.

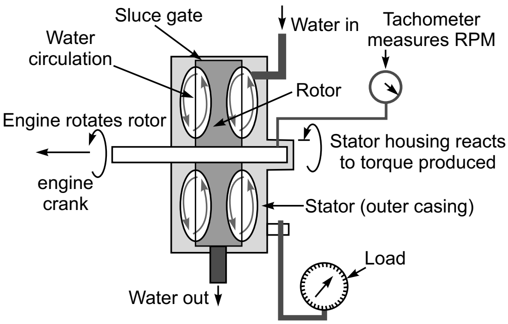


Fig. 4.6 *Hydraulic dynamometer*

The outer casing tends to revolve with the impeller due to centrifugal force developed, but its movement is resisted by fluid friction. The generated frictional force between the impeller and the fluid is measured by the spring balance fitted on the casing.

The heat developed due to the dissipation of power in hydraulic dynamometer is carried away by a continuous supply of the working fluid.

The brake power is calculated as

$$BP = \frac{\text{Load (N)} \times \text{Speed } (N \text{ rpm})}{\text{Dynamometer Constant}} \qquad \ldots(4.8)$$

With the exception of bearing friction, the load measured is proportional to that power supplied to the dynamometer. Its main features are

- Small size, easy to install
- Simple structure, easy operation and maintenance
- Large braking torque
- High measurement accuracy
- Stable and reliable
- Using magnetic speed sensor to achieve high-precision instantaneous speed measurement

4.2.5 Eddy Current Dynamometer

It is also an absorbing type dynamometer. The working principle of **eddy current dynamometer** is shown in the Fig. 4.7. The principle of operation of eddy-current dynamometer is based on Fleming's right hand law. It consists of a rotor, which is driven by a prime mover and stator fitted with a number of electromagnets. The field coil which excites the magnetic pole is wound in radial direction.

When the rotor rotates, eddy currents are produced in the stator due to magnetic flux set up by the passage of coil current in the electromagnets. These eddy currents are converted into heat energy; therefore, this dynamometer requires some cooling arrangement. The torque is measured with the help of a moment arm. The load on the engine can be controlled by regulating the current in the electromagnets during the test.

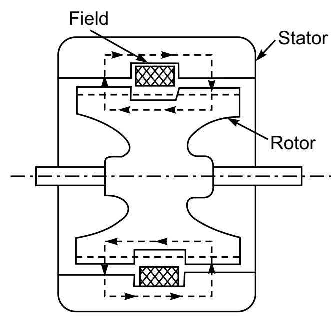


Fig. 4.7 *Eddy current dynamometer*

The main advantages of eddy current dynamometers are:

1. Its weight per unit brake power is low.
2. They offer the highest ratio of constant power speed range (up to 5 : 1).
3. The variation in field excitation is below 1% of total power of the engine. Thus, they are easy to control and operate.
4. Eddy current development is smooth hence the torque is also smooth and continuous under all conditions.
5. Relatively higher torque under low-speed conditions.
6. It has no intricate rotating parts except shaft bearing.

4.2.6 Transmission Dynamometer

It is also called torque meter. It consists of a beam and a pair of strain-gauges fixed on the rotating shaft as shown in Fig. 4.8. The torque is measured by the angular deformation of the shaft which is indicated as the strain on the strain gauge. A four arm bridge is used to reduce the effect of temperature.

Transmission dynamometers measure brake power very accurately and are used where continuous transmission of the load is necessary. These are mainly used in automatic units.

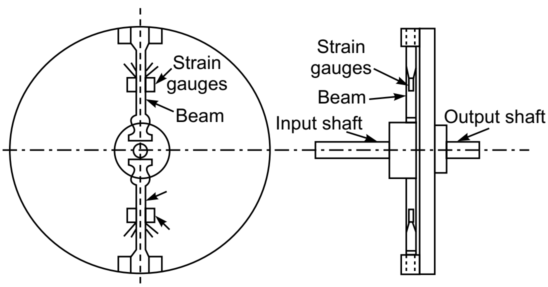


Fig. 4.8 *Transmission dynamometer*

4.3 MEASUREMENT OF FRICTION POWER

4.3.1 Friction Power

Friction power is the part of the indicated power which is used to *overcome the frictional effects* within the engine. The friction power also includes power required to operate the fuel pump, lubrication pump, valves, etc. Therefore, it is given as the difference between the indicated power and brake power.

$$FP = IP - BP \qquad \ldots(4.9)$$

4.3.2 Willan's Line Method

Willan's line method is used to find the friction power of an engine. In willan's line method, fuel consumption rate is plotted against brake power as shown in Fig. 4.9.

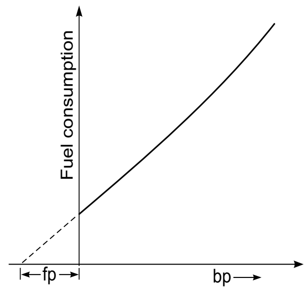


Fig. 4.9 *Fuel consumption against brake power for a CI engine*

This method is also known as fuel rate extrapolation method. It indicates that the fuel consumption of a CI engine against increase in brake power. This test is conducted at constant speed on diesel engines.

- In most of the power range, the fuel consumption varies linearly with increases in brake power.
- Extrapolation of the Willan's line to zero fuel consumption gives a measure of friction power of the engine
- When engine does not develop power ($BP = 0$). It consumes certain amount of fuel in overcoming combined losses due to mechanical friction, pumping and blowby etc..
- Since the fuel consumption in SI engine is not linear, therefore, the use of this method is limited to diesel engines only.

4.3.3 Morse Test

Morse test is applicable to multi-cylinder engine only. In this test indicated power of an engine is determined without any special equipment. In this test, the pumping and friction losses are assumed constant.

In this test, the engine is run at a constant speed with all cylinder firing and its maximum output is measured as *BP*. The IP of *n* number of cylinders is given by

$$IP = BP + FP \qquad \ldots(\text{i})$$

Then power development from first cylinder is cut out by shorting the spark plug of cylinder of SI engine or cutting off fuel supply in case of diesel engine, while remaining $n - 1$ cylinders remain operative. The engine brake power, BP_{n-1} is measured by keeping the speed same. The difference of brake powers in two conditions is the indicated power of inoperative cylinder. The IP for $(n - 1)$ cylinders is given by

$$IP_{n-1} = BP_{n-1} + FP \qquad \text{...(ii)}$$

Since, the engine runs at the same speed with $n - 1$ cylinders, hence its friction power FP also remains constant. Therefore, the indicator power of first cylinder is obtained by subtracting Eq. (ii) from Eq. (i)

$$IP_1 = BP - BP_{n-1}$$

Now make the first cylinder operative and cut out the power from second, third and next cylinder in turn. The indicated power of second cylinder is obtained as

$$IP_2 = BP - BP_{n-2}$$

Similarly, the indicated power of third cylinder

$$IP_3 = BP - BP_{n-3}$$

and so on.

The addition of indicated power of all cylinders becomes total indicated power IP of engine.

$$IP = IP_1 + IP_2 + IP_3 \ldots = \sum_{k=1}^{k=n} IP_k = BP + FP \qquad \text{...(4.10)}$$

Where IP, BP and FP represent indicated power, brake power and friction power, respectively, and n is number of cylinders. The friction power can be obtained by subtracting total brake power from total indicated power.

$$FP = IP - BP$$

4.3.4 Motoring Test

In the motoring test, the engine is first run up to a desired speed by its own power and allowed to remain at the same speed and load conditions for sufficient time so that oil, water, and engine parts temperatures can attain a steady state conditions. The power of the engine during this period is absorbed by a swinging field type electric dynamometer, which is most suitable for this test.

The fuel supply to the engine is then cut-off and by suitable electric-switching devices the swinging field type electric dynamometer is converted to run as an electric motor to drive for the engine at the same speed.

The power supply to the motor is measured which is a measure of the FP of the engine. During the motoring test the water supply is also cut-off so that the actual operating temperatures are maintained.

Although, this method approximates the FP at temperature conditions very near to the actual operating temperatures at the test speed and load, but it does, not give the true losses occurring under firing conditions.

4.3.5 From the difference of Indicated Power and Brake Power

In this method, the friction power is determined by computing the difference between the indicated power and brake power. The indicated power can be obtained from an indicator diagram, and brake power from a brake dynamometer.

4.4 FUEL CONSUMPTION

A calibrated burette can be used for measurement of fuel consumption as shown in Fig. 4.10. A quantity of fuel is taken from the fuel tank through a valve and then the valve is closed. The fuel flows into the running engine only from graduated burette. The time, Δt is recorded for a known volume of fuel-consumption. The fuel consumption rate is then given as

$$\dot{m}_f = \frac{V_{fuel} \times \rho_{fuel} \times 3600}{\Delta t} \text{ (kg/h)} \qquad \text{...(4.11)}$$

where, V_{fuel} = Volume of fuel in m^3 used in time Δt,
ρ_{fuel} = Density of fuel, and
Δt = Time in seconds.

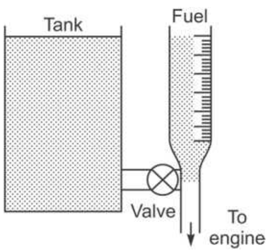


Fig. 4.10 *Fuel consumption measurement*

4.4.1 Specific Fuel Consumption (*sfc*)

It is defined as the ratio of the mass of fuel consumed per hour per unit power output (*BP*). It is also designated as *Bsfc* (*brake specific fuel consumption*). It is a parameter which decides the economics of power production from an engine.

$$Bsfc \text{ or } sfc = \frac{\dot{m}_f \text{ (kg/h)}}{BP \text{ (kW)}} \text{ (kg/kWh)} \quad ...(4.12)$$

The specific fuel consumption in kg/kWh based on the indicated power (*IP*) is called the *Isfc* (*indicated specific fuel consumption*) and is expressed as

$$Isfc = \frac{\dot{m}_f \text{ (kg/h)}}{IP \text{ (kW)}} \text{ (kg/kWh)} \quad ...(4.13)$$

4.5 AIR CONSUMPTION

Air-consumption rate of an engine can be effectively calculated by means of an orifice meter installed in an air box as shown in Fig. 4.11.

An orifice is fitted to an air-tight air box. Inlet manifold is connected to the air box through a flexible pipe. A U-tube manometer is installed on the air box to measure the pressure depression in the water column (h_w), when the engine sucks the air. A rubber diaphragm is also installed to minimize the pressure pulsation.

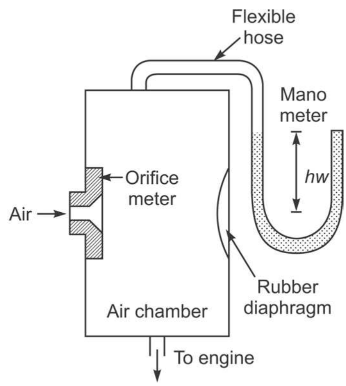


Fig. 4.11 *Air-consumption measurement*

The volume flow rate of air passing through the orifice, in m^3/s, can be calculated by using the relation

$$\dot{V}_a = A_{orifice} \times \text{velocity of air}$$

which can be expressed as

$$\dot{V}_a = \frac{\pi}{4} d_o^2 C_d \sqrt{2 g h_a} \quad ...(4.14)$$

where, $\dot{V}_a$ = Volume flow rate of air, m^3/s
d_o = Diameter of orifice, m
C_d = Coefficient of discharge of orifice
g = 9.81 m/s^2, acceleration due to gravity
h_a = Head of air = $\frac{\rho_w h_w}{\rho_{air}}$ (m),
h_w = Water column in m and
ρ_w = density of water,
ρ_{air} = density of air.

The mass-flow rate of air through the orifice can be calculated as

$$\dot{m}_a = \rho_{air} \times \dot{V}_a \quad ...(4.15)$$

4.6 AIR-FUEL RATIO (A/F)

It is the ratio between the mass of the air and mass of the fuel supplied to the engine. It is expressed as

$$\text{A/F} = \frac{\dot{m}_a \text{ (mass flow rate of air)}}{\dot{m}_f \text{ (mass flow rate of fuel)}} \quad ...(4.16)$$

Theoretically, the correct (stoichiometric) air–fuel ratio is 15 : 1. But the combustion of air–fuel mixture can take place in A/F ratio ranges from 12 to 19 for petrol engines and 20 to 80 in Diesel engines.

It is also possible to calculate the air-fuel ratio and the mass of exhaust gases per kg of fuel from the volumetric analysis of exhaust gases by Orsat apparatus and the ultimate analysis of fuel on mass basis.

Mass of air supplied per kg of fuel

$$m_a = \frac{N \times C}{33 \times (C_1 + C_2)} \text{ (kg)} \qquad \ldots(4.17)$$

where, N, C_1 and C_2 are percentages of nitrogen, carbon dioxide and carbon monoxide by volume in exhaust gases and C is percentage of carbon in fuel on mass basis.

Mass of exhaust gases per kg of fuel = $(m_a + 1)$ kg.
Mass of exhaust gases per minute (m_g)

$$= (m_a + 1) \times m_f \text{ (Mass of fuel in kg per min)} \qquad \ldots(4.18)$$

The mass/volume flow rate of air and percentage excess air can be calculated as illustrated below.

4.6.1 Minimum Air Required for Complete Combustion of Solid Fuel

Minimum mass of air required for complete combustion of fuel,

$$m_{th} = \frac{\mathbf{100}}{\mathbf{23}}\left[\frac{\mathbf{8}}{\mathbf{3}}\mathbf{C} + \mathbf{8}\left(\mathbf{H} - \frac{\mathbf{O}}{\mathbf{8}}\right) + \mathbf{S}\right] \qquad \ldots(4.19)$$

since air contains 23% of oxygen by ***mass.***

4.6.2 Minimum Air Required for Complete Combustion of Gaseous Fuel

The atmospheric air contains 21% of oxygen by volume. Hence, the minimum volume of air required for complete combustion of fuel

$$V_{th} = \frac{100}{21} \times [0.5\,CO + 0.5\,H_2 + 2\,CH_4 + 3\,C_2H_4 - O_2] \text{ (m}^3\text{)} \qquad \ldots(4.20)$$

4.6.3 Excess Air

The actual amount of air required for complete combustion of fuel is always more than the stoichiometric air required. The amount of air in excess, supplied to burn fuel completely is called the *excess air*. This can be expressed as a per cent excess air in terms of stoichiometric or theoretical amount of air as

Percent excess air

$$= \frac{\text{Actual air used} - \text{Stoichiometric air required}}{\text{Stoichiometric air required}} \qquad \ldots(4.21)$$

4.7 EFFICIENCIES OF IC ENGINES

4.7.1 Brake Thermal Efficiency (η_{bth})

The power output of an engine is obtained from the combustion of charge. Thus the *overall efficiency* of an engine is given by brake thermal efficiency, i.e.,

$$\eta_{bth} = \frac{\text{Brake power}}{\text{Energy supply rate}} = \frac{BP}{\dot{m}_f \times CV} \qquad \ldots(4.22)$$

where, $\dot{m}_f$ = mass flow rate of the fuel (kg/s)
CV = Calorific value of fuel, (kJ/kg)

4.7.2 Indicated Thermal Efficiency (η_{ith})

The indicated thermal efficiency is defined as the ratio of the indicated power to the heat supply rate, i.e.,

$$\eta_{ith} = \frac{\text{Indicated power}}{\text{Heat supply rate}} = \frac{IP}{\dot{m}_f \times CV} \qquad \ldots(4.23)$$

4.7.3 Mechanical Efficiency (η_{mech})

It is the ratio of the brake power and indicated power.

$$\eta_{mech} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{BP}{IP} \qquad \ldots(4.24)$$

It can also be expressed as

$$\eta_{mech} = \frac{\text{Brake thermal efficiency}}{\text{Indicated thermal efficiency}} = \frac{\eta_{bth}}{\eta_{ith}} \qquad \ldots(4.25)$$

and $$\eta_{mech} = \frac{\text{Brake mean effective pressure}}{\text{Indicated mean effective pressure}}$$

$$= \frac{p_{mb}}{p_{mi}} \quad \text{...(4.26)}$$

4.7.4 Relative Efficiency

It is the ratio of actual thermal efficiency to air standard efficiency of the engine. It is sometimes referred as *efficiency ratio*. It is expressed as

$$\eta_{Relative} = \frac{\text{Brake thermal efficiency}}{\text{Air standard efficiency}} \quad \text{...(4.27)}$$

Relative efficiency for most of the engines varies from 75 to 95% with air standard efficiency.

4.7.5 Volumetric Efficiency (η_{vol})

It is defined as the ratio of the mass of the actual charge inducted into the cylinder to the mass of the charge corresponding to the swept volume, or

$$\eta_{vol} = \frac{\text{Actual mass flow rate of the charge}}{\text{Density} \times \text{Swept volume per second}}$$

$$= \frac{\dot{m}_a \text{ (kg/s)}}{\rho\left(\frac{\pi}{4} d^2 L\right)\frac{n}{60}} \quad \text{...(4.28)}$$

where, ρ = Density of inlet charge,

d = bore, L = stroke

n = Number of effective suction strokes per cycle per minute

n = N for a two-stroke engine

= $\frac{N}{2}$ for a four-stroke engine

The volumetric efficiency can also be defined as the ratio of the volume of the charge inducted in the cylinder, measured at NTP to the swept volume of the cylinder.

$$\eta_{vol} = \frac{V_{act}}{V_s} \quad \text{...(4.29)}$$

4.8 HEAT BALANCE SHEET OR ENERGY AUDIT

A heat balance sheet is an account of heat supplied (credit) and heat utilized (debit) in various ways in the engine systems. Heat balance sheet of an engine provides necessary information related with the performance. A Sankey diagram for an internal combustion engine is shown in Fig. 4.12. It indicates distribution of energy supplied to various systems of engine.

Energy balance is often taken over a period of time, therefore, it is normally prepared on minute basis, but it can also be done on second basis or hour basis. The various energy transactions are expressed in quantity as well in percentage of energy supplied by fuel.

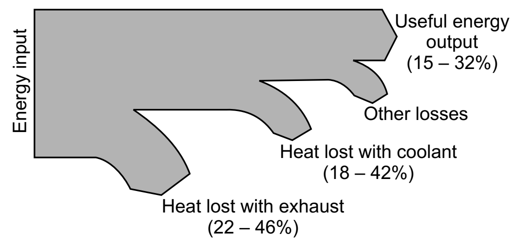


Fig. 4.12 *Sankey diagram for heat distribution in an internal combustion engine*

4.8.1 Credit

The heat energy supplied to the engine by burning of fuel

$$\dot{Q}_{in} = \dot{m}_f \times CV \quad \text{...(4.30)}$$

Where, $\dot{m}_f$ = mass of fuel used in kg/min

CV = Calorific value of fuel in kJ/kg

4.8.2 Debit

The various operations in which the heat utilized in the engine are

1. Heat equivalent to useful work or brake power of the engine.
2. Heat carried away by the cooling water
3. Heat carried away by the exhaust gases, and
4. Unaccounted heat losses.

1. Heat Equivalent to Useful Work or Brake Power of the Engine

$$\dot{Q}_1 = BP \times 60 \text{ (kJ/min)} \quad \text{...(4.31)}$$

2. Heat Carried Away by the Cooling Water

$$\dot{Q}_2 = \dot{m}_w C_{pw} (\Delta T)_w \text{ (kJ/min)} \quad \ldots(4.32)$$

where, m_w = mass flow rate of cooling water in kg/min,

C_{pw} = Specific heat of cooling water = 4.186 kJ/kg.°C

$(\Delta T)_w$ = Temperature rise of water of cooling water in °C.

3. Heat Carried Away by the Exhaust Gases

$$\dot{Q}_3 = \dot{m}_g C_{pg} (T_g - T_a)_w \text{ (kJ/min)} \quad \ldots(4.33)$$

where, $\dot{m}_g$ = mass flow rate of exhaust gases in kg/min,

C_{pg} = Mean specific heat of exhaust gases in kJ/kg.°C

T_g = Temperature of exhaust gases in °C.

T_a = Temperature of ambient air in °C.

The mass of exhaust gases can be calculated as sum of fuel rate and air consumption rate in kJ/min.

4. Unaccounted Heat Losses

It includes heat loss by convection, radiation, friction losses, internal pumping work etc. This loss cannot be measured directly. It is calculated by subtracting quantities, $\dot{Q}_1$, $\dot{Q}_2$ and $\dot{Q}_3$ from $\dot{Q}_{in}$ as

$$\dot{Q}_4 = \dot{Q}_{in} - (\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_g) \text{ (kJ/min)} \quad \ldots(4.34)$$

Heat Balance Sheet in kJ per minute

Heat supplied	*kJ/min*	*%*	*Heat expendilure*	*kJ/min*	*%*
Heat supplied by combustion of fuel	$\dot{Q}_{in}$	100	(1) Heat equivalent of brake power	$\dot{Q}_1$	–
			(2) Heat lost to jacket cooling water	$\dot{Q}_2$	–
			(3) Heat lost to exhaust gases (wet)	$\dot{Q}_3$	–
			(4) Heat lost to radiation errors of observation, etc. (by difference)	$\dot{Q}_4$	–
Total	–	100	Total		100

4.9 ENGINE PERFORMANCE CURVES

The performance of an engine varies with load and speed. Therefore, in following section, performance variation of engine variables with speed and load discussed.

4.9.1 Torque

Torque is also called moment of a force or twisting force that tends to cause rotation. It can be defined as product of force and distance measured from centre of rotation. It is usually measured in N-m.

When speed of engine increases, torque increases initially, reaches a maximum value and then starts decreasing as shown in Fig. 4.13.

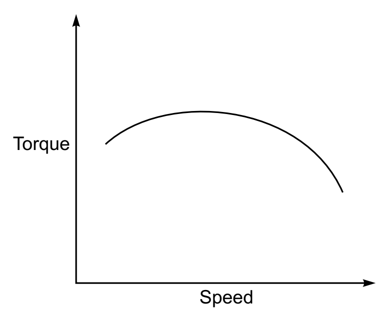


Fig. 4.13 *Variation of torque with engine speed*

When an engine operates at constant speed, then, increase in torque is in proportion to *BP* (load) increase and torque-*BP* graph results into a straight line as shown in Fig. 4.14.

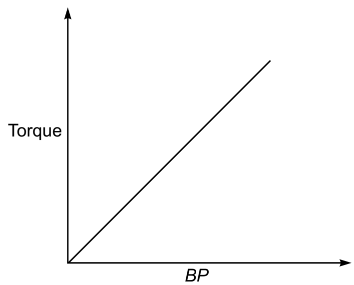


Fig. 4.14 *Variation of torque with BP*

4.9.2 Brake Power and Indicated Power

The brake power and indicated power both increase with increase in engine speed. Both graphs are slightly curved towards the top. Friction power also increase with increase in engine speed. At any speed, the difference between IP and BP values gives corresponding friction power as shown in Fig. 4.15.

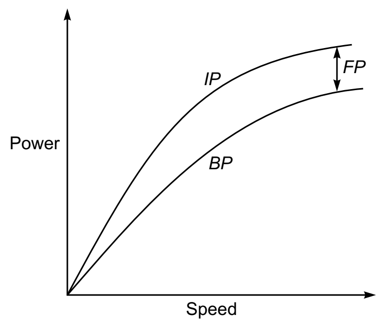


Fig. 4.15 *Variation of Power with engine speed*

At constant engine speed, the friction power remains almost constant, and hence IP increase with increase in BP and corresponding curve is generally straight line as shown in Fig. 4.16.

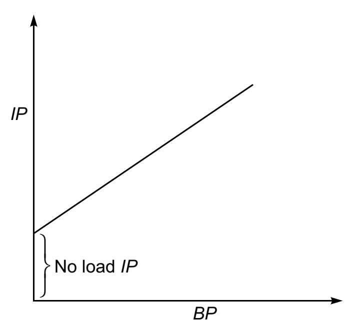


Fig. 4.16 *Variation of IP with BP*

4.9.3 Fuel Consumption

The fuel consumption of a diesel engine increases linearly with an increase in speed as shown in Fig. 4.17. When engine speed is kept constant, the fuel consumption also increases linearly with increase in load (BP) as shown in Fig. 4.18. This curve also shows fuel consumption at no load condition.

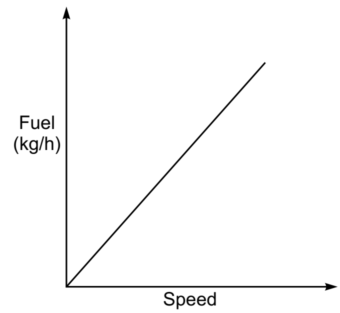


Fig. 4.17 *Variation of fuel consumption with speed*

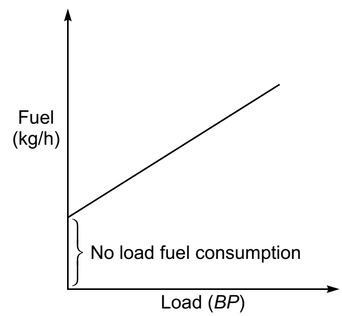


Fig. 4.18 *Variation of fuel consumption with load*

4.9.4 Brake Specific Fuel Consumption (*bsfc*)

Either engine speed increases at constant load or brake power increase at constant speed, the brake (or indicated) specific fuel consumption follow the same pattern as shown in Fig. 4.19. For variable speed engine *Bsfc* (kg/kWh) is plotted against speed and for constant speed engine, the *bsfc* is plotted against *BP* (kW).

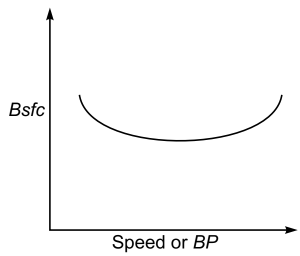


Fig. 4.19 *Variation of Bsfc with speed and BP*

4.9.5 Mechanical Efficiency (η_{mech})

Either engine speed increases at constant load or brake power increase at constant speed, the mechanical efficiency follow the same pattern in both cases. It increase from zero to maximum and then begins to decrease in a smooth curve as shown in Fig. 4.20. It show that the maximum efficiency occurs at 75% load on an engine, after that, the friction power increases at faster rate and thus mechanical efficiency starts to decrease.

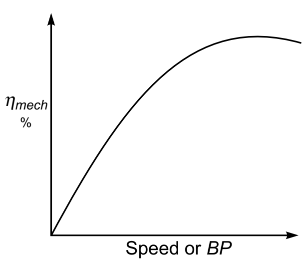


Fig. 4.20 *Variation of mechanical efficiency with speed and BP*

4.9.6 Thermal Efficiency

Thermal efficiency of an engine increases with increase in load when speed is kept constant or with increase in speed when load is kept constant. Its curve also increases to maximum value and then decreases with speed or load as shown in Fig. 4.21. The condition of maximum efficiency reveals that engine running condition for its maximum fuel economy.

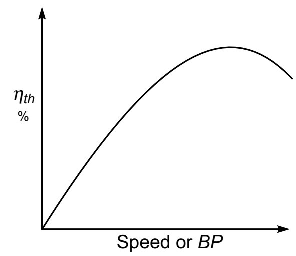


Fig. 4.21 *Variation of thermal efficiency with speed and BP*

4.10 ENGINE EMISSION

The emissions from an engine pollute the environment and contribute to the global warming, acid rain, smog, odors, respiratory system problems and other human health problems. The major causes of the emissions are non-stoichiometric combustion, reaction of nitrogen with oxygen at high temperature, and impurities in the fuel and air. The emissions of concern are hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NOx), sulfur dioxides, and solid carbon particulates.

The pollution control has put the limits on emission from the internal combustion engines. The exhaust gases from the engine contain (a) Invisible emission and (b) Visible emission.

(a) Invisible emission are
- Carbon dioxide,
- Carbon monoxide
- Oxides of nitrogen,
- Water vapours,
- Unburnt hydrocarbons, and
- Aldehydes

(b) Visible emission are
- Smoke, and
- Particulate matter.

The gasoline engines run with homogeneous air fuel mixture, ratio ranging from 12 : 1 to 18 : 1. The most serious pollutants from gasoline engine are carbon monoxide (CO), unburnt hydrocarbon

(HC), and oxides of nitrogen (NO_x). The volume of emissions per vehicle of HC, CO, and NOx can be reduced by about 95% making engines more fuel efficient, and with the use of exhaust gas recirculation.

Diesel engines run with very lean air fuel ratio, therefore, the concentration of CO and HC is comparatively lower in exhaust gases. Although, NO_x emission is comparable to those from SI engines. In addition, the particulate matter in exhaust gases from diesel engine is a serious concern.

Table 4.1 shows the composition of engine emission.

Table 4.1 *Constituents of internal combustion engine exhaust gases*

Major Constituents (greater than 1%)	*Minor Constituents (less than 1%)*
Water, H_2O Carbon dioxide, CO_2 Nitrogen, N_2 Oxygen, O_2 Carbon monoxide, CO[(a)] Hydrogen, H_2O[(a)]	Oxides of sulfur, SO_2, SO_3 Oxides of nitrogen, NO, NO_2 Aldehydes, HCHO, etc. Organic acids, HCOOH, etc. Alcohols, CH_2OH etc. Hydrocarbons C_nH_m Carbon monoxide, CO[(b)] Hydrogen, H_2O[(b)] Smoke

(a) Spark Ignition engine
(b) Diesel engine

4.10.1 Carbon Monoxide

Carbon monoxide is an invisible emission and is result of incomplete combustion of fuel due to insufficient availability of oxygen in air-fuel mixture. It may also the result of insufficient time in the cycle for complete combustion. Emission of CO is matter of concern in SI engines. Typically, the CO presence in exhaust from a SI engine is 0.2 to 5%. The CO in exhaust gases is only undesirable, but it also represents loss of chemical energy, that is not fully utilized during combustion.

The concentration of CO is high, when engine runs in idle condition or slow speed or during power mode operation, since during above mode of operations, a rich mixture is required to keep engine running. However, the percentage of CO fraction decreases with increase in speed, during deceleration and at steady speed.

4.10.2 Oxides of Nitrogen

Oxides of nitrogen in exhaust gases are also invisible emissions and these are the result of chemical reaction between nitrogen and oxygen at relatively high temperature. The oxides of nitrogen formed are Nitric oxide (NO) and Nitrogen dioxide (NO_2). When excess O_2 is available at higher combustion temperature, Nitrogen reacts with O_2 and Nitric oxide (NO) is formed.

There are a several possible chemical reactions that can form NO during the combustion process and immediately after. These are

$$O + N_2 \rightarrow NO + N \qquad \ldots(4.35)$$

$$N + O_2 \rightarrow NO + O \qquad \ldots(4.36)$$

$$N + OH \rightarrow NO + H \qquad \ldots(4.37)$$

NO, in turn, can then further react to form NO_2, by various means, including

$$NO + H_2O \rightarrow NO_2 + H_2 \qquad \ldots(4.38)$$

$$NO + O_2 \rightarrow NO_2 + O \qquad \ldots(4.39)$$

NO_X concentration in exhaust gases depend on engine design, air-fuel ratio, mode of vehicle operation etc. NOx is one of the major causes of photochemical smog, which has become now a major problem in many large cities of the world. Smog is formed by the photochemical reaction of exhaust gases and atmospheric air in the presence of sunlight. NOx decomposes into NO and monatomic oxygen (O):

$$NOx + \text{energy from sunlight} \cong NO + O + \text{smog} \qquad \ldots(4.40)$$

4.10.3 Unburned Hydrocarbon (HC)

Exhaust gases leaving the SI engine contain unburned hydrocarbon 1-1.5 % of the fuel and within this, there is about 40% unburned gasoline fuel components and remaining partially reacted components.

The composition of HC emissions is different for each gasoline blend, depending on the original fuel components, induction system design, combustion chamber geometry, air-fuel ratio, speed, load, and mode of engine operation.

When hydrocarbon emissions enter into the atmosphere, they act as irritants and odorants; some of these are carcinogenic. Fig. 4.22 shows different emission levels against equivalence ratio. With a rich mixture there is not enough oxygen to complete combustion and results in high levels of HC and CO in the exhaust.

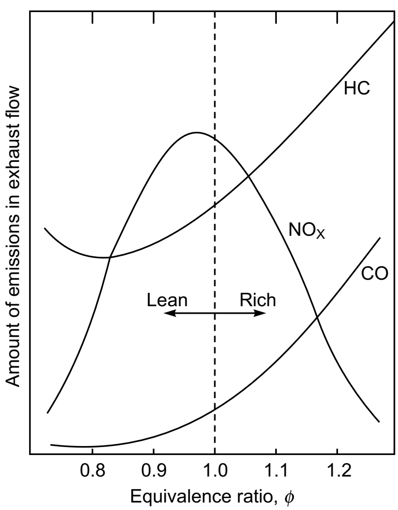


Fig. 4.22 *Emissions as a function of equivalence ratio in SI engines*

The possible causes of HC emission in an engine are

1. Incomplete mixing of the air and fuel results in some fuel particles not finding oxygen for complete combustion.
2. Quenching of flame at the walls that may leave a small volume of unburned mixture.
3. Use of rich air fuel mixture in the engine.
4. High dilution of residual exhaust causes poor combustion and a greater possibility of quenching of flame. This is experienced at low load and idle conditions.
5. High levels of EGR may also cause incomplete combustion.
6. During valve overlapping period, both the exhaust and intake valves are open, inducted air-fuel mixture can flow directly into the exhaust port, especially at idle and low speeds, when real time of overlap is greatest.

4.10.4 Aldehydes

When alcohol based fuel is used, then higher level oxygenated hydrocarbons are emitted from the engine. The oxygenated hydrocarbons are called aldehydes, invisible emissions. These aldehydes are responsible for the pungent smell from the engine and these are usually carcinogenic, eye and respiratory irritants.

4.10.5 Smoke

The smoke coming out from an engine is direct visible indicator of status of combustion process. Smoke is the result of incomplete combustion. Normally three types of smoke emitted from diesel engines: blue, white and black.

Blue Smoke When lubricating oil burns with the fuel in the combustion chamber, the resulting exhaust appears bluish.

White Smoke Vaporization of water vapours or use of antifreeze can cause exhaust as white smoke. Normally, when engine is cooled below 10°C, some of water vapour present in residual gases get condensed. During start and warming up period, white smoke usually comes out the engine.

Black Smoke The black smoke from the exhaust pipe indicates the fault in fuel system and is a result of incomplete combustion and high fuel rate of consumption. This smoke is usually clearly visible on a light background. It contains the soot particles.

In gasoline engines, the black smoke is generally result of overflow of fuel from the float chamber needle valve due to a defect or due to choking of air jets. In diesel engines, the black smoke is result of fault in the high pressure pump, injectors, and at a high angle of injection.

Example 4.1 *A rope-brake dynamometer was used to measure the brake power of a single cylinder, four-stroke cycle petrol engine. It was found that the torque due to brake load was 175 Nm and the engine makes 500 rpm. Determine the brake power developed by the engine.*

Solution

Given A single-cylinder, four-stroke petrol engine with a rope-brake dynamometer

$k = 1$

$T = 175$ Nm $\qquad N = 500$

To find Brake power

Analysis The brake power is given by

$$BP = \frac{2\pi\, N\, T}{60{,}000} = \frac{2\pi \times 500 \times 175}{60{,}000} = \mathbf{9.16\ kW}$$

Example 4.2 *A four-cylinder, four-stroke petrol engine develops indicated power of 14.7 kW at 1000 rpm. The mean effective pressure is 5.5 bar. Calculate the bore and stroke of the engine, if the stroke is 1.5 times the bore.*

Solution

Given A four-cylinder, four-stroke cycle petrol engine

$k = 4 \qquad n = \dfrac{N}{2}$

$IP = 14.7$ kW $\qquad N = 1000$ rpm

$p_m = 5.5$ bar $= 550$ kPa

$L = 1.5\, d$

To find (i) Bore (ii) Stroke

Analysis The power of an engine is given by

$$IP = p_m \frac{L\, A\, n\, k}{60}$$

$$14.7 = 550 \times \frac{(1.5\, d)}{60} \times \left(\frac{\pi}{4} d^2\right) \times \frac{1000}{2} \times 4$$

$$d^3 = 6.806 \times 10^{-4}\ \text{m}^3$$

Bore, $\quad d = 0.08796$ m or **87.96 mm**

Stroke $\quad L = 1.5\,d =$ **131.94 mm.**

Example 4.3 *A four-cylinder, two-stroke cycle petrol engine develops 30 kW at 2500 rpm. The mean effective pressure on each piston is 8 bar and mechanical efficiency is 80%. Calculate the diameter and stroke of each cylinder, if the stroke to bore ratio is 1.5. Also calculate the fuel consumption of the engine, if the brake thermal efficiency is 28%. The calorific value of the fuel is 43900 kJ/kg.*

Solution

Given A four-cylinder, two-stroke cycle petrol engine

$k = 4 \qquad BP = 30$ kW

$N = 2500$ rpm $\qquad n = N$

$\eta_{mech} = 0.8 \qquad L = 1.5\,d$

$\eta_{bth} = 0.28 \qquad CV = 43900$ kJ/kg

$p_m = 8$ bar $= 800$ kPa

To find

(i) Bore of cylinder,
(ii) Stroke of pistion, and
(iii) Fuel consumption rate (*Bsfc*).

Analysis The mechanical efficiency is given as

$$\eta_{mech} = \frac{BP}{IP}$$

$$\therefore \quad IP = \frac{30\ \text{kW}}{0.8} = 37.5\ \text{kW}$$

(i) The indicated power is expressed as

$$IP = \frac{p_m\, L\, A\, nk}{60}$$

or $$37.5 = \frac{800 \times (1.5d) \times \left(\frac{\pi}{4} d^2\right) \times 2500 \times 4}{60}$$

or $\quad d^3 = 0.000238\ \text{m}^3$

Bore $\quad d = 0.062$ m or **62 mm**

(ii) Stroke $\quad L = 1.5\,d =$ **93 mm**

(iii) The brake thermal efficiency is given by

$$\eta_{bth} = \frac{BP}{\dot{m}_f\, CV}$$

or $$\dot{m}_f = \frac{30\ \text{kW}}{0.28 \times (43900\ \text{kJ/kg})}$$

$$= 0.00244\ \text{kg/s} \quad \text{or} \quad 8.78\ \text{kg/h}$$

The brake specific fuel consumption

$$Bsfc = \frac{\dot{m}_f\,(\text{kg/h})}{BP\,(\text{kW})} = \frac{8.78}{30}$$

$$= \mathbf{0.293\ kg/kWh}$$

Example 4.4 *The following results were obtained from a test on a single-cylinder, four-stroke Diesel engine. Diameter of the cylinder is 30 cm, stroke of the piston is 45 cm, indicated mean effective pressure is 540 kPa and engine speed is 2400 rpm. Calculate the indicated power of the engine.*

Solution

Given A single-cylinder, four-stroke Diesel engine

d = 30 cm = 0.3 m $\qquad$ L = 45 cm = 0.45 m

p_{mi} = 540 kPa $\qquad$ N = 2400 rpm

k = 1

$$n = \frac{N}{2} = 1200 \text{ working stroke per minute}$$

To find Indicated power

Analysis The cross-sectional area of the cylinder

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} \times (0.3)^2 = 0.07068 \text{ m}^2$$

Indicated power (IP)

$$IP = \frac{p_{mi}\ L\ A\ n\ k}{60}$$

$$= \frac{(540 \text{ kPa}) \times (0.45 \text{ m}) \times (0.07068 \text{ m}^2) \times 1200 \times 1}{60}$$

$$= \mathbf{343.53\ kW}$$

Example 4.5 *In a test of a single-cylinder, four-stroke Diesel engine, the following data were recorded.*

Indicated mean effective pressure = 755 kPa
cylinder diameter = 10 cm
piston stroke = 15 cm
engine speed = 480 rpm
brake wheel diameter = 62.5 cm
net load on the brake wheel = 170 N

Calculate

(a) indicated power,
(b) brake power, and
(c) the mechanical efficiency of the engine.

Solution

Given A single-cylinder, four-stroke Diesel engine

p_{mi} = 755 kPa $\qquad$ N = 480 rpm

$$n = \frac{N}{2} = 240 \text{ working stroke/min.}$$

k = 1 $\qquad$ d = 10 cm = 0.1 m

L = 15 cm = 0.15 m $\qquad$ D_{brake} = 62.5 cm

or R_{brake} = 31.25 cm = 0.3125 m

F = 170 N

To find

(i) Indicated power,
(ii) brake power, and
(iii) mechanical efficiency

Analysis The cross-sectional area of the cylinder

$$A = \frac{\pi}{4} \times (0.1 \text{ m})^2 = 7.854 \times 10^{-3} \text{ m}^2$$

(i) Indicated power (IP)

$$IP = \frac{p_{mi}\ L\ A\ n\ k}{60}$$

$$= \frac{755 \times 0.15 \times 7.854 \times 10^{-3} \times 240 \times 1}{60}$$

$$= \mathbf{3.557\ kW}$$

(ii) Brake power,

$$BP = \frac{2\pi\ N\ T}{60{,}000} = \frac{2\pi\ N\ (F\ R_{brake})}{60{,}000}$$

$$= \frac{2\pi \times (480 \text{ rpm}) \times (170 \text{ N} \times 0.3125 \text{ m})}{60{,}000}$$

$$= \mathbf{2.67\ kW}$$

(iii) Mechanical efficiency (η_{mech})

$$\eta_{mech} = \frac{BP}{IP} = \frac{2.67}{3.557} = 0.750 = \mathbf{75\%.}$$

Example 4.6 *The following results refer to a test on a petrol engine:*

Indicated power = 30 kW
brake power = 26 kW
Engine speed = 1000 rpm
Bsfc = 0.35 kg/kWh
CV of fuel used = 43900 kJ/kg

Calculate

(a) Indicated thermal efficiency,
(b) Brake thermal efficiency, and
(c) Mechanical efficiency.

Solution

Given A petrol engine

$IP = 30$ kW $BP = 26$ kW

$N = 1000$ rpm $Bsfc = 0.35$ kg/kWh

$CV = 43900$ kJ/kg

To find

(i) Indicated thermal efficiency, η_{ith}

(ii) Brake thermal efficiency, η_{bth}

(iii) Mechanical efficiency, η_{mech}

Analysis Fuel consumption rate,

$$\dot{m}_f = Bsfc \times BP$$
$$= 0.35 \times 26 = 9.1 \text{ kg/h} = 2.53 \times 10^{-3} \text{ kg/s}$$

(i) Indicated thermal efficiency (η_{ith}),

$$\eta_{ith} = \frac{IP}{\dot{m}_f \times CV}$$
$$= \frac{30}{2.53 \times 10^{-3} \times 43900}$$
$$= 0.27 = \mathbf{27\%.}$$

(ii) Brake thermal efficiency (η_{bth}),

$$\eta_{bth} = \frac{BP}{\dot{m}_f CV}$$
$$= \frac{26}{2.53 \times 10^{-3} \times 43900}$$
$$= 0.234 = \mathbf{23.4\%.}$$

(iii) Mechanical efficiency (η_{mech}),

$$\eta_{mech} = \frac{BP}{IP} = \frac{26}{30} = 0.867 = \mathbf{86.7\%.}$$

Example 4.7 *The mechanical efficiency of a single-cylinder, four-stroke engine is 80%. The friction power is estimated to be 26 kW. Calculate the indicated power and brake power developed by the engine.*

Solution

Given A single-cylinder, four-stroke engine

$\eta_{mech} = 0.80$ $FP = 26$ kW

To find

(i) Indicated power, and

(ii) Brake power.

Analysis The mechanical efficiency of an engine is given by

$$\eta_{mech} = \frac{BP}{IP} \quad \text{or} \quad BP = 0.8\, IP$$

The friction power of the engine is given as

$$FP = IP - BP$$
$$26 \text{ kW} = IP - 0.8\, IP = 0.2\, IP$$

or
$$IP = \frac{26}{0.2} = \mathbf{130\ kW}$$

Then
$$BP = IP - FP = 130 - 26 = \mathbf{104\ kW.}$$

Example 4.8 *A Diesel engine has a brake thermal efficiency of 30%, if the calorific value of the fuel is 42000 kJ/kg. Calculate the brake specific fuel consumption.*

Solution

Given A Diesel engine with

$\eta_{bth} = 0.3$

$CV = 42000$ kJ/kg

To find Brake specific fuel consumption.

Analysis The brake thermal efficiency of an engine is expressed as

$$\eta_{bth} = \frac{BP}{\dot{m}_f \times CV}$$

or
$$\frac{BP}{\dot{m}_f} = \eta_{bth} CV$$
$$= 0.3 \times 42000 = 12600 \text{ kJ/kg}$$

or
$$\frac{\dot{m}_f}{BP} = 7.936 \times 10^{-5} \text{ kg/kJ}$$

and
$$Bsfc = \frac{\dot{m}_f}{BP} \times 3600 = \mathbf{0.287\ kg/kWh}$$

Example 4.9 *A two-stroke, Diesel engine develops a brake power of 420 kW. The engine consumes 195 kg/h of fuel and air–fuel ratio is 22:1. Calorific value of the fuel is 42000 kJ/kg. If 76 kW of power is required to overcome the frictional losses, calculate*

(a) Mechanical efficiency,

(b) Air consumption,

(c) Brake thermal efficiency.

Solution

Given A two-stroke Diesel engine with

$BP = 420$ kW $FP = 76$ kW

A/F $= 22 : 1$ $CV = 42000$ kJ/kg

$\dot{m}_f = 195$ kg/h

To find

(i) Mechanical efficiency, η_{mech}

(ii) Air consumption, and

(iii) Brake thermal efficiency, η_{bth}

Analysis

(i) Mechanical efficiency (η_{mech})

Indicated power,

$$IP = BP + FP = 420 + 76 = 496 \text{ kW}$$

$$\eta_{mech} = \frac{BP}{IP} = \frac{420}{496} = 0.8467$$

$$= \mathbf{84.67\%.}$$

(ii) Air consumption rate,

$$\dot{m}_a = \dot{m}_f \times \frac{A}{F} = (195 \text{ kg/h}) \times \frac{22}{1}$$

$$= 4290 \text{ kg/h} \quad \text{or} \quad \mathbf{71.5\ kg/min}$$

(iii) The brake thermal efficiency of an engine is expressed as

$$\eta_{bth} = \frac{BP}{\dot{m}_f \times CV}$$

$$= \frac{420 \text{ kW}}{\left(\frac{195}{3600} \text{ kg/s}\right) \times (42000 \text{ kJ/kg})}$$

$$= 0.1846 \quad \text{or} \quad \mathbf{18.46\%}$$

Example 4.10 *An engine is used in a process which requires 100 kW of brake power with a mechanical efficiency of 78%. The engine uses 1 kg of fuel per minute. If a simple modification in design reduces the engine friction by 8 kW then what will be the percentage saving in fuel consumption? Assume indicated thermal efficiency remains same.*

Solution

Given An engine with

$BP = 100$ kW $\qquad \eta_{mech} = 0.78$

$\dot{m}_{f_1} = 1$ kg/min. $\qquad \eta_{ith} =$ constant

Change in $FP = 8$ kW

To find Percentage saving in fuel consumtion.

Analysis

(i) Mechanical efficiency (η_{mech}),

$$\eta_{mech} = \frac{BP}{IP}$$

or
$$IP_1 = \frac{BP}{\eta_{mech}} = \frac{100}{0.78} = 128.21 \text{ kW}$$

Friction power

$$FP_1 = IP - BP = 128.21 - 100$$

$$= 28.21 \text{ kW}$$

After modified design of engine, friction power

$$FP_2 = 28.21 - 8 = 20.21 \text{ kW}$$

$$IP_2 = BP + FP_2 = 100 + 20.21 = 120.21$$

For same indicated thermal efficiency

$$\eta_{ith_1} = \eta_{ith_2}$$

or
$$\frac{IP_1}{\dot{m}_{f_1} \times CV} = \frac{IP_2}{\dot{m}_{f_2} \times CV}$$

or
$$\frac{128.21}{1 \times CV} = \frac{120.21}{\dot{m}_{f_2} \times CV}$$

or
$$\dot{m}_{f_2} = 0.9376 \text{ kg/min}$$

Percentage saving in fuel consumption

$$= \frac{\dot{m}_{f_1} - \dot{m}_{f_2}}{\dot{m}_{f_1}} \times 100 = \frac{1 - 0.9376}{1} \times 100$$

$$= \mathbf{6.24\%}$$

Example 4.11 *Calculate the brake mean effective pressure of a four-cylinder, four -stroke Diesel engine having a 100 mm bore and 120 mm stroke which develops a power of 42 kW at 1200 rpm.*

Solution

Given A four-cylinder, four-stroke Diesel engine with

$k = 4 \qquad d = 100 \text{ mm} = 0.10 \text{ m}$

$N = 1200 \text{ rpm} \qquad L = 120 \text{ mm} = 0.12 \text{ m}$

$BP = 42 \text{ kW} \qquad n = \frac{N}{2}$

To find The brake mean effective pressure.

Analysis The brake mean effective pressure relates brake power as

$$BP = \frac{p_{mb}\, L\, A\, n\, k}{60}$$

where, p_{mb} is brake mean effective pressure

$$\therefore \quad 42 \text{ kW} = \frac{p_{mb} \times 0.12 \times \left(\frac{\pi}{4}\right) \times (0.1)^2 \times 1200 \times 4}{60 \times 2}$$

or
$$p_{mb} = 1114.08 \text{ kPa} = \mathbf{11.14\ bar}$$

Example 4.12 *A single cylinder, 4-stroke Diesel engine running at 1800 rpm has an 85 mm bore and a 110 mm stroke. It takes 0.56 kg of air per minute and develops a brake power of 6 kW, while the air–fuel ratio* *is 20 : 1. CV of fuel is 42550 kJ/kg and the ambient air density is 1.18 kg/m³. Calculate*

(a) The volumetric efficiency,

(b) Brake specific fuel consumption.

Solution

Given A single-cylinder, four-stroke Diesel engine with

$k = 1$ $\quad d = 85$ mm

$L = 110$ mm $\quad n = \dfrac{N}{2} = \dfrac{1800}{2} = 900$

$\dot{m}_a = 0.56$ kg/min $\quad$ A/F $= 20:1 \quad BP = 6$ kW

$CV = 42550$ kJ/kg $\quad \rho_{air} = 1.18$ kg/m^3

To find

(i) The volumetric efficiency, and

(ii) Brake specific fuel consumption.

Analysis

(i) *Volumetric efficiency*

Swept volume rate,

$$\dot{V}_s = \frac{\pi}{4} d^2 L n k = \frac{\pi}{4} \times (0.085)^2 \times 0.11 \times 900 \times 1$$

$$= 0.5617 \text{ m}^3/\text{min}$$

Mass of air,

$$\dot{m}_a = \dot{V}_s \rho_{air} = 0.5617 \times 1.18$$

$$= 0.663 \text{ kg/min}$$

Volumetric efficiency,

$$\eta_{vol} = \frac{\text{Actual mass of air}}{\text{Mass correponds to swept volume}}$$

$$= \frac{0.56}{0.663} \times 100 = \mathbf{84.5\%}$$

(ii) *Brake specific fuel consumption:*

Mass flow rate of fuel

$$\dot{m}_f = \frac{\text{Actual mass of air}}{\text{A/F ratio}} = \frac{0.56}{20}$$

$$= 0.028 \text{ kg/min or } 1.68 \text{ kg/h}$$

$$Bsfc = \frac{\dot{m}_f\,(\text{kg/h})}{BP(\text{kW})} = \frac{0.028}{6} = \mathbf{0.28\ kg/kWh}$$

Example 4.13 *Calculate the brake mean effective pressure of a four-cylinder, two-stroke engine of 100 mm bore, 125 mm stroke, when it develops a torque of 490 Nm.*

Solution

Given A four-cylinder, two-stroke engine with

$k = 4$ $\quad d = 100$ mm

$n = N$ $\quad L = 125$ mm

$T = 490$ Nm

To find The brake mean effective pressure of the engine

Analysis The brake power is given by

$$BP = \frac{2\pi N T}{60{,}000} = \frac{p_{mb} L A n k}{60}$$

Here $n = N$ for two-stroke engine

$$p_{mb} = \frac{2\pi T}{1000 L A k}$$

$$= \frac{2\pi \times 490}{1000 \times 0.125 \times \left(\frac{\pi}{4}\right) \times (0.1)^2 \times 4}$$

$$= 784 \text{ kPa}$$

Thus, the break mean effective pressure is **7.84 bar**.

Example 4.14 *A single-cylinder, C1 engine with a brake thermal efficiency of 30% uses diesel oil having a calorific value of 42000 kJ/kg. If its mechanical efficiency is 80%, calculate (a) Bsfc, (b) Isfc, and (c) η_{ith}.*

Solution

Given A single-cylinder *CI* engine with

$\eta_{bth} = 0.3$ $\quad CV = 42000$ kJ/kg

$\eta_{mech} = 0.8$

To find

(i) *Bsfc* (ii) *Isfc* (iii) η_{ith}

Analysis

(i) The brake thermal efficiency relates brake power as

$$\eta_{bth} = \frac{BP}{\dot{m}_f CV}$$

or $$\frac{\dot{m}_f}{BP} = \frac{1}{\eta_{bth} CV} = \frac{1}{0.3 \times 42000} = \frac{1}{12600} \text{ kg/kJ}$$

and $$Bsfc = \frac{\dot{m}_f}{BP} \times 3600 = \frac{1}{12600} \times 3600$$

$$= \mathbf{0.286\ kg/kWh}$$

(ii) $$\eta_{mech} = \frac{Isfc}{Bsfc}$$

Thus $Isfc = 0.8 \times 0.286 = \mathbf{0.229\ kg/kWh}$

(iii) The mechanical efficiency is expressed as

$$\eta_{mech} = \frac{BP}{IP} = \frac{\eta_{bth}}{\eta_{ith}}$$

or $$\eta_{ith} = \frac{\eta_{bth}}{\eta_{mech}} = \frac{0.3}{0.8} = 0.375 = \mathbf{37.5\%}$$

Example 4.15 *A four-cylinder, four-stroke petrol engine has a 10 cm bore, 15 cm stroke and uses a compression ratio of 6. The engine develops 25 kW indicated power at 2000 rpm. Find the indicated mean effective pressure and air standard efficiency.*

Also calculate the fuel consumption per hour, if the indicated thermal efficiency is 30%. Take the calorific value of fuel as 42 MJ/kg.

Solution

Given A four-cylinder, four-stroke petrol engine

$k = 4$ $\quad d = 10$ cm = 0.1 m

$L = 15$ cm = 0.15 m $\quad r = 6$

$IP = 25$ kW $\quad N = 2000$ rpm

$n = \frac{N}{2}$ $\quad \eta_{ith} = 0.3$

$CV = 42$ MJ/kg

To find

(i) Air standard efficiency,
(ii) Indicated mean effective pressure, and
(iii) Fuel consumption per hour.

Assumptions

(i) The petrol engine works on air standard Otto cycle.
(ii) The ratio of specific heats $\gamma = 1.4$ for air.

Analysis

(i) *Air standard efficiency of Otto cycle:*

$$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}}$$

$$= 1 - \frac{1}{(6)^{1.4-4}} = 0.511 \text{ or } \mathbf{51.1\%}$$

(ii) *Indicated mean effective pressure*

$$IP = \frac{p_{mi} L A n k}{60}$$

$$25\text{ kW} = \frac{p_{mi} \times (0.15\text{m}) \times (\pi/4) \times (0.1\text{m})^2 \times 2000 \times 4}{60 \times 2}$$

or $p_{mi} = 318.3$ kPa = **3.183 bar**

(iii) *Fuel consumption per hour*

We have $$\eta_{ith} = \frac{IP}{\dot{m}_f \times CV}$$

or $$\dot{m}_f = \frac{IP}{\eta_{ith}\, CV} = \frac{25\text{ kW}}{0.3 \times 42 \times 10^3 \text{ kJ/kg}}$$

$$= 1.984 \times 10^{-3}\text{ kg/s} = \mathbf{7.14\ kg/h}$$

Example 4.16 *A four-cylinder, four-stroke petrol engine has a bore of 57 mm and a stroke of 90 mm. Its rated speed is 2800 rpm, torque is 55.2 Nm. The fuel consumption is 6.74 lit/h. The density of the petrol is 735 kg/m^3 and petrol has a calorific value of 44200 kJ/kg. Calculate BP, bmep, brake thermal efficiency and brake-specific fuel consumption.*

Solution

Given A four-cylinder, four-stroke petrol engine with

$d = 57$ mm = 0.057 m $\quad L = 90$ mm = 0.09 m

$T = 55.2$ Nm $\quad N = 2800$ rpm

$k = 4$ $\quad n = \frac{N}{2}$

$\dot{V}_f = 6.74$ lit/h $\quad \rho_f = 735$ kg/m^3

$CV = 44200$ kJ/kg

To find

(i) *BP,*
(ii) Brake *mep,*
(iii) η_{bth}, and
(iv) Brake-specific fuel consumption.

Analysis

(i) The brake power of the engine can be obtained as

$$BP = \frac{2\pi N T}{60{,}000} = \frac{2\pi \times 2800 \times 55.2}{60{,}000}$$

$$= \mathbf{16.18\ kW}$$

(ii) The *Bmep* (brake mean effective pressure) is given by

$$BP = p_{mb} \frac{L A n k}{60}$$

$$16.18 = p_{mb} \times 0.09 \times \frac{\pi}{4} \times \frac{(0.057)^2 \times 2800}{60 \times 2} \times 4$$

or $p_{mb} = 755.1$ kPa = **7.55 bar**

(iii) The mass flow rate of the petrol

$$\dot{m}_f = \rho_f \dot{V}_f$$
$$= (735 \text{ kg/m}^3) \times (6.74 \times 10^{-3} \text{ m}^3\text{/h})$$
$$= 4.954 \text{ kg/h} = 1.376 \times 10^{-3} \text{ kg/s}$$

The brake thermal efficiency

$$\eta_{bth} = \frac{BP}{\dot{m}_f \times CV}$$
$$= \frac{16.18}{1.376 \times 10^{-3} \times 44200} = 0.266$$
$$= \mathbf{26.6\%}$$

(iv) The brake-specific fuel consumption

$$Bsfc = \frac{\dot{m}_f \text{ (kg/h)}}{BP \text{ (kW)}} = \frac{4.954}{16.18}$$
$$= \mathbf{0.306 \text{ kg/kWh}}$$

Example 4.17 *A 4-cylinder, gasoline engine operates on four-stroke cycle. The bore of each cylinder is 90 mm and the stroke is 110 mm. The clearance volume per cylinder is 60 cc. At a speed of 3500 rpm, the fuel consumption is 18 kg/h and the torque developed is 140 N-m. Calculate:*

(i) Brake power
(ii) Bmep
(iii) Brake thermal efficiency if the calorific value of the fuel is 42,000 kJ/kg
(iv) Relative efficiency on a brake power basis assuming the engine works on the constant volume cycle.

Solution

Given Four cylinder four stroke gasoline engine

$$k = 4,$$
$$d = 90 \text{ mm} = 90 \times 10^{-3} \text{ m},$$
$$L = 110 \text{ mm} = 0.11 \text{ m}$$
$$V_c = 60 \text{ cc} = 60 \times 10^{-6} \text{ m}^3$$
$$N = 3500 \text{ rpm}$$
$$n = \frac{N}{2}$$
$$\dot{m}_f = 18 \text{ kg/h}$$
$$T = 140 \text{ N-m}$$
$$CV = 42000 \text{ kJ/kg}$$

To find

(i) Brake power
(ii) Brake mean effective pressure, p_{mb},
(iii) Brake thermal efficiency, and
(iv) Relative efficiency on brake power

Assumptions

(i) Ideal cycle for gasoline engine is Otto cycle
(ii) For air $Y = 1.4$

Analysis

(i) Brake power can be calculated as

$$BP = \frac{2\pi NT}{60000} = \frac{2\pi \times 3500 \times 140}{60000} = \mathbf{51.31 \text{ kW}}$$

(ii) Brake mean effective pressure:
BP is given by

$$BP = p_{mb} \frac{LAnk}{60}$$

$$\text{or } p_{mb} = \frac{BP \times 60}{LAnk} = \frac{51.31 \times 60 \times 2}{0.11 \times \frac{\pi}{4} \times (90 \times 10^{-3})^2 \times 3500 \times 4}$$

$$= 628.5 \text{ kPa} \quad \text{or} \quad \mathbf{6.28 \text{ bar}}$$

(iii) Heat supplied by fuel

$$\dot{Q}_{in} = \dot{m}_f \times CV = \left(\frac{18}{3600} \text{ kg/s}\right) \times (42000 \text{ kJ/kg})$$
$$= 210 \text{ kW}$$

Brake thermal efficiency

$$\eta_{bth} = \frac{BP}{\dot{Q}_{in}} = \frac{51.31}{210} = 0.244 \quad \text{or} \quad \mathbf{24.4\%}$$

(iv) Relative efficiency
Swept volume

$$V_s = \frac{\pi}{4} d^2 L = \frac{\pi}{4} \times (90 \times 10^{-3})^2 \times 0.11$$
$$= 7 \times 10^{-4} \text{ m}^3 \quad \text{or} \quad 700 \text{ cc}$$

Total volume of cylinder

$$V_1 = V_s + V_c = 700 + 60 = 760 \text{ cc}$$

$$\text{Compression ratio, } r = \frac{V_1}{V_c} = \frac{760}{60} = 12.67$$

Air standard efficiency of Otto cycle

$$\eta_{Air\ standard} = 1 - \frac{1}{r^{r-1}} = 1 - \frac{1}{(12.67)^{1.4-1}} = 63.78\%$$

Relative efficiency

$$\eta_{relative} = \frac{\eta_{actual}}{\eta_{air\ standard}} = \frac{24.4}{63.78}$$
$$= 0.3825 \quad \text{or} \quad \mathbf{38.25\%}$$

Example 4.18 *A 4 cylinder, 4 stroke gasoline engine having a bore of 80 mm and stroke of 90 mm has a compression ratio of 8. The relative efficiency is 65% when indicated fuel specific consumption is 200 gm/kWh. Estimate:*

(i) Calorific value of fuel, and

(ii) Corresponding fuel consumption, given that indicated mean effective pressure (imep) is 7.5 bar and speed is 2000 rpm.

Solution

Given 4 cylinder 4 stroke gasoline engine

$$k = 4,$$
$$d = 80 \text{ mm} = 0.08 \text{ m}$$
$$L = 90 \text{ mm} = 0.09 \text{ m}$$
$$r = 8$$
$$N = 2000 \text{ rpm}$$
$$\eta_{relative} = 0.65$$
$$isfc = 200 \text{ gm/kWh}$$
$$pmi = 7.5 \text{ bar} = 750 \text{ kPa}$$

To find

(i) Calorific value of fuel, and

(ii) Fuel consumption.

Assumptions

(i) Ideal cycle for gasoline engine is Otto cycle.

(ii) For air $\gamma = 1.4$.

(iii) Relative efficiency based on indicated thermal efficiency.

Analysis The air standard efficiency for Otto cycle is given by

$$\eta_{air\ standard} = 1 - \frac{1}{r^{r} - 1} = 1 - \frac{1}{8^{1.4-1}} = 0.564$$

Actual indicated thermal efficiency

$$\eta_{ith} = \eta_{air\ standard} \times \eta_{relative} = 0.564 \times 0.65 = 0.367$$

Further indicated power

$$IP = p_{mi} \frac{LAnk}{60}$$
$$= \frac{750 \times 0.09}{60} \times \frac{\pi}{4} \times (0.08)^2 \times \frac{2000}{2} \times 4$$
$$= 22.62 \text{ kW}$$

Heat supplied

$$\dot{Q}_{in} = \frac{IP}{\eta_{ith}} = \frac{22.62}{0.367} = 61.63 \text{ kW}$$

Further, the indicated specific fuel consumption is given by

$$isfc = \frac{\dot{m}_f \text{ (kg/h)}}{IP \text{ (kW)}}$$

$$\text{Mass of fuel, } \dot{m}_f = IP \times isfc = 22.62 \times \left(\frac{200}{1000} \text{ kg/kWh}\right)$$
$$= \mathbf{4.524 \text{ kg/h}}$$

Calorific value of fuel

$$\dot{Q}_{in} = \dot{m}_f \times CV$$
$$CV = \frac{\dot{Q}_{in}}{\dot{m}_f} = \frac{61.63 \text{ kW}}{(4.524/3600 \text{ kg/s})}$$
$$= \mathbf{49042 \text{ kJ/kg}}$$

Example 4.19 *The following data and results refer to a test on a four-cylinder, four-stroke car engine:*

cylinder bore = 7.5 cm

piston stroke = 9 cm

engine-to-rear-axle ratio = 39 : 8

wheel diameter with tyre fully inflated = 65 cm

The petrol consumption is 0.250 kg for a distance of 4 km, when the car was travelling at a speed of 60 kmph. CV of fuel is 44200 kJ/kg.

If the mean effective pressure is 5.625 bar, calculate the indicated power and indicated thermal efficiency.

Solution

Given A four-cylinder, four-stroke car engine with

$$d = 7.5 \text{ cm} = 0.075 \text{ m}$$
$$L = 9 \text{ cm} = 0.09 \text{ m}$$
$$T = 55.2 \text{ Nm}$$
$$k = 4$$
$$D_{wheel} = 65 \text{ cm} = 0.65 \text{ m}$$
$$\text{Distance} = 4 \text{ km}$$
$$\text{Engine rear axle ratio} = 39 : 8$$
$$\text{speed} = 60 \text{ km/h} = 16.667 \text{ m/s}$$
$$m_f = 0.250 \text{ kg}$$
$$CV = 44200 \text{ kJ/kg}$$
$$Bmep = 5.625 \text{ bar} = 562.5 \text{ kPa}$$
$$n = \frac{N}{2}$$

To find

(i) Indicated power, and

(ii) Indicated thermal efficiency.

Analysis

$$\text{Velocity of car} = \frac{60 \times 1000}{60} = 1000 \text{ m/min}$$
$$= \pi D_{wheel} N_{tyre}$$

Revolution of tyre,

$$N_{tyre} = \frac{1000}{\pi \times 0.65}$$
$$= 490 \text{ rotation per min.}$$

Using rear-to-axle ratio, the speed of car engine

$$N = \frac{490 \times 39}{8} = 2388 \text{ rpm}$$

Cross-sectional area of the cylinder

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} \times (0.075)^2$$
$$= 0.00441 \text{ m}^2$$

(i) Indicated power

$$IP = p_{mi} \frac{L\,A\,n\,k}{60}$$
$$= 562.5 \times \frac{0.09 \times 0.00441 \times 2388 \times \frac{1}{2} \times 4}{60}$$
$$= \mathbf{17.776\ kW}$$

(ii) Indicated thermal efficiency (η_{ith}),

Time for 4-km distance

$$= \frac{\text{Distance}}{\text{Car speed}} = \frac{4 \text{ km}}{\frac{60}{3600} \text{ km/s}}$$
$$= 240 \text{ seconds}$$

$$\text{Fuel rate } \dot{m}_f = \frac{\text{Fuel mass}}{\text{time}} = \frac{0.250 \text{ kg}}{240 \text{ s}}$$
$$= 0.00104 \text{ kg/s}$$

$$\eta_{ith} = \frac{IP}{\dot{m}_f\, CV} = \frac{17.776}{0.00104 \times 44200}$$
$$= 0.386 \quad \text{or} \quad \mathbf{38.6\%}$$

Example 4.20 *The following data and results refer to a test on a single-cylinder, two-stroke cycle engine:*

indicated mean effective pressure = 550 kPa
cylinder diameter = 21 cm
piston stroke = 28 cm
engine speed = 360 rpm
brake torque = 628 Nm
fuel consumption = 8.16 kg/h
calorific value of fuel = 42700 kJ/kg

Calculate

(a) mechanical efficiency,
(b) the indicated thermal efficiency,
(c) the brake thermal efficiency, and
(d) brake-specific fuel consumption in kg/kWh.

Solution

Given A single-cylinder, two-stroke cycle engine

$$p_{mi} = 550 \text{ kPa}$$
$$T = 628 \text{ Nm}$$
$$N = 360 \text{ rpm}$$
$$n = N = 360 \text{ working stroke/min}$$
$$d = 21 \text{ cm} = 0.21 \text{ m}$$
$$L = 28 \text{ cm} = 0.28 \text{ m}$$
$$\dot{m}_f = 8.16 \text{ kg/h}$$
$$CV = 42700 \text{ kJ/kg}$$
$$k = 1$$

To find

(i) Mechanical efficiency,
(ii) Indicated thermal efficiency,
(iii) Brake thermal efficiency, and
(iv) Brake-specific fuel consumption.

Analysis Cross-sectional area of the cylinder

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} \times (0.21)^2 = 0.0346 \text{ m}^2$$

(i) Indicated power

$$IP = p_{mi} \frac{L\,A\,n\,k}{60}$$
$$= 550 \times \frac{0.28 \times 0.0346 \times 360 \times 1}{60}$$
$$= 32.0 \text{ kW}$$

Brake power

$$BP = \frac{2\pi\, N\, T}{60{,}000} = \frac{2\pi \times 360 \times 628}{60{,}000}$$
$$= 23.675 \text{ kW}$$

Mechanical efficiency, η_{mech}:

$$\eta_{mech} = \frac{BP}{IP} = \frac{(23.675 \text{ kW})}{(32.0 \text{ kW})}$$
$$= 0.7397 \approx \mathbf{74\%}$$

(ii) Indicated thermal efficiency (η_{ith});

$$\eta_{ith} = \frac{IP}{\dot{m}_f\, CV}$$

$$= \frac{(32.0 \text{ kW})}{\left(\frac{8.16}{3600} \text{ kg/s}\right) \times (42700 \text{ kJ/kg})}$$

$= 0.330 =$ **33%**

(iii) Brake thermal efficiency (η_{bth});

$$\eta_{bth} = \frac{BP}{\dot{m}_f \, CV}$$

$$= \frac{(23.675 \text{ kW})}{\left(\frac{8.16}{3600} \text{ kg/s}\right) \times (42700 \text{ kJ/kg})}$$

$= 0.244 =$ **24.4%**

(iv) Brake-specific fuel consumption ($Bsfc$);

$$Bsfc = \frac{\dot{m}_f \text{ (kg/h)}}{BP \text{ (kW)}} = \frac{(8.16 \text{ kg/h})}{(23.675 \text{ kW})}$$

$=$ **0.3446 kg/kWh**

Example 4.21 *A single-cylinder, four-stroke Diesel engine works on the following data:*

Cylinder bore = 15 cm
Stroke = 25 cm
Speed = 250 rpm
Area of indicator diagram = 6 cm^2
Length of the indicator diagram = 9 cm
Spring constant = 7.5 bar/cm
Brake specific fuel consumption = 0.24 kg/kWh
CV of fuel = 42000 kJ/kg
Diameter of brake wheel = 70 cm
Rope diameter = 3.5 cm
Brake load = 40 kg (392.4 N)

Calculate

(a) brake power,
(b) indicated mean effective pressure,
(c) indicated power,
(d) mechanical efficiency, and
(e) indicated thermal efficiency.

Solution

Given A single-cylinder, four-stroke Diesel engine:

$n = \frac{N}{2}$ $\quad k = 1$

$d = 15 \text{ cm} = 0.15 \text{ m}$ $\quad L = 25 \text{ cm} = 0.25 \text{ m}$

$N = 250 \text{ rpm}$ $\quad a_i = 6 \text{ cm}^2$

$l = 9 \text{ cm}$ $\quad$ Spring constant $= 7.5$ bar/cm

$Bsfc = 0.24$ kg/kWh $\quad CV = 42000$ kJ/kg

$D_{brake} = 70$ cm $\quad d_{rope} = 3.5$ cm

$W_{brake} = 392.4$ N

To find

(i) BP (ii) $Imep$ (iii) IP
(iv) η_{mech} (v) η_{ith}

Analysis

Effective brake radius

$$R_{brake} = \frac{D_{brake} + d_{rope}}{2} = \frac{70 + 3.5}{2}$$
$$= 36.75 \text{ cm} = 0.3675 \text{ m}$$

$T =$ Load $\times$ Effective brake radius
$= W_{brake} R_{brake}$

$T = 392.4 \times 0.3675 = 144.2$ Nm

(i) The brake power

$$BP = \frac{2\pi N T}{60{,}000}$$

$$\therefore \quad BP = \frac{2\pi \times 250 \times 144.2}{60000} = \mathbf{3.775 \text{ kW}}$$

(ii) Indicated mean effective pressure ($Imep$)

$$p_{mi} = \frac{\text{Area of indicator diagram}}{\text{Length of the indicator diagram}} \times \text{Spring constant}$$

$$= \frac{6 \text{ cm}^2}{9 \text{ cm}} \times 7.5 \text{ bar/cm}$$

$=$ **5.0 bar** or **500kPa**

(iii) Indicated power (IP)

$$IP = p_{mi} \frac{L\,A\,n\,k}{60}$$

$$= 500 \times 0.25 \times \frac{\pi}{4} \times \frac{(0.15)^2}{60} \times \frac{250}{2} \times 1$$

$=$ **4.6 kW**

(iv) Mechanical efficiency (η_{mech})

$$\eta_{mech} = \frac{BP}{IP} = \frac{3.775}{4.6} = 0.820 = \mathbf{82.0\%}$$

(v) Indicated thermal efficiency (η_{ith});

Given $\quad Bsfc = \frac{\dot{m}_f}{BP}$

or $\quad 0.24 = \frac{\dot{m}_f}{3.775}$

or $\quad \dot{m}_f = 0.906 \text{ kg/h} = 2.516 \times 10^{-4} \text{ kg/s}$

Thus
$$\eta_{ith} = \frac{IP}{\dot{m}_f \times CV}$$
$$= \frac{4.6}{2.516 \times 10^{-4} \times 42000}$$
$= 0.435$ or **43.5%**

Example 4.22 *A full-load test was conducted on a two-stroke engine and the following results were obtained:*

Speed = 500 rpm
Brake load = 500 N
imep = 3 bar
Oil consumtion = 5 kg/h
Jacket water temperature rise = 35°C
Jacket water flow rate = 7 kg/min.
A/F ratio by mass = 30
Exhaust gas temperature = 350°C
Room temperature = 25°C
Atmospheric pressure = 1 bar
Cylinder diameter = 22 cm
Stroke = 28 cm
Brake diameter = 1.6 m
CV of fuel = 42000 kJ/kg
Proportion of H_2 by mass in fuel = 15%
Specific heat of exhaust gas = 1.0 kJ/kg·K
Specific heat of dry steam = 2.0 kJ/kg·K

Calculate
(a) indicated thermal efficiency,
(b) Specific fuel consumtion, and
(c) Volumetric efficiency based on atmospheric conditions.
Draw up a heat balance sheet for test.

Solution

Given A single-cylinder, two-stroke Diesel engine:

$n = N = 500$ rpm	A/F = 30
$d = 22$ cm = 0.22 m	$L = 28$ cm = 0.28 m
$p_{mi} = 3$ bar = 300 kPa	$\dot{m}_f = 5$ kg/h
$CV = 42000$ kJ/kg	$W_{brake} = 500$ N
$D_{brake} = 1.6$ m	$\dot{m}_w = 7$ kg/min
$T_g = 350°C$	$(\Delta T)_w = 35°C$
$T_a = 25°C = 298$ K	$p_a = 1$ bar = 100 kPa
$C_{pg} = 1.0$ kJ/kg·K	$C_{ps} = 2.0$ kJ/kg·K

H2 % by mass in fuel = 15%

To find
(i) Indicated thermal efficiency,
(ii) Specific fuel consumption,
(iii) Volumetric efficiency based on atmospheric conditions, and
(iv) Heat balance sheet.

Assumptions
(i) Single cylinder engine, i.e. $k = 1$
(ii) Specific gas constant for air, $R = 0.287$ kJ/kg·K
(iii) Specific heat of water $C_{pw} = 4.187$ kJ/kg·K
(iv) Latent heat of steam at 1 atm = 2257 kJ/kg.

Analysis The swept volume rate
$$\dot{V}_s = \frac{\pi}{4} d^2 L n k = \frac{\pi}{4} \times (0.22)^2 \times 0.28 \times \frac{500}{60} \times 1$$
$= 0.0887$ m³/min

(i) *Indicated power*
$$IP = p_{mi}\dot{V}_s = 300 \times 0.0887 = 26.6 \text{ kW}$$
Heat supplied by fuel
$$\dot{Q}_{in} = \dot{m}_f\, CV = 5 \times 42000$$
$= 210000$ kJ/h $= 58.33$ kW

Indicated thermal efficiency
$$\eta_{ith} = \frac{IP}{Q_{in}} = \frac{26.6 \text{ kW}}{(58.33 \text{ kJ/s})}$$
$= 0.456$ or **45.6%**

(ii) *Specific fuel consumption*
The brake drum radius
$$R_{brake} = \frac{D_{brake}}{2} = \frac{1.6}{2} = 0.8 \text{ m}$$
Brake torque T = Load × Effective brake radius
$= W_{brake} R_{brake}$
$= 500 \times 0.8 = 400$ N-m

Brake power
$$BP = \frac{2\pi\, N\, T}{60{,}000} = \frac{2\pi \times 500 \times 400}{60000}$$
$= 20.94$ kW

The brake-specific fuel consumtion
$$Bsfc = \frac{\dot{m}_f}{BP} = \frac{(5 \text{ kg/h})}{(20.94 \text{ kW})}$$
= **0.238 kg/kWh**

(iii) *Volumetric efficiency*
Actual mass flow rate of air into engine
$$\dot{m}_a = (\text{A/F})\ \dot{m}_f = 30 \times 5 = 150 \text{ kg/h}$$
$= 0.4167$ kg/s

Actual volume flow rate of air;

$$\dot{V}_a = \frac{\dot{m}_a R T_a}{p_a} = \frac{0.4167 \times 0.287 \times 298}{100}$$
$$= 0.03563 \text{ m}^3/\text{s}$$

The volumetric efficiency

$$\eta_{vol} = \frac{\dot{V}_a}{\dot{V}_s} = \frac{0.03563}{0.0887}$$
$$= 0.4017 \quad \text{or} \quad \mathbf{40.17\%}$$

Heat Balance sheet: On minute basis;

Heat supplied per minute by fuel

$$\dot{Q}_{in} = \dot{m}_f CV = \left(\frac{5}{60} \text{ kg/min}\right) \times 42000$$
$$= 3500 \text{ kJ/min}$$

(a) Heat equivalent to *BP*

$$\dot{Q}_1 = BP \times 60 = 20.96 \times 60$$
$$= 1257.6 \text{ kJ/min}$$

(b) Heat lost to cooling water

$$\dot{Q}_2 = \dot{m}_w C_{pw}(\Delta T) = 7 \times 4.187 \times 35$$
$$= 1025.81 \text{ kJ/min}$$

Heat lost to exhaust gases

Mass of exhaust gases formed/kg of fuel

$$m_{ex} = \dot{m}_f \times \frac{A}{F} = \left(\frac{5}{60} \text{ kg/min}\right) \times 30$$
$$= 2.5 \text{ kg/min}$$

Mass of H_2O formed during combustion

$$m_{ex} = 9H_2 \times \text{Mass of fuel used per minute}$$
$$= 9 \times 0.15 \times \frac{5}{60} = 0.1125 \text{ kg/min}$$

Mass of dry exhaust gases per minute

$$m_g = \text{Mass of wet exhaust} - \text{Mass of } H_2O \text{ formed}$$
$$= 2.5 - 0.1125 = 2.3875 \text{ kg/min}$$

(c) Heat lost to dry exhaust gases

$$\dot{Q}_3 = \dot{m}_g C_{pg}(\Delta T)_g$$
$$= 2.3875 \times 1.0 \times (350 - 25)$$
$$= 775.93 \text{ kJ/min}$$

(d) H_2O formed exit in superheated state at 350°C, thus heat carried away by steam

$$\dot{Q}_4 = \dot{m}_{H_2O}\,(C_{pw}(T_{sat} - T_a) + h_{fg} + C_{ps}(T_g - T_{sat}))$$
$$= 0.1125\,[(4.187 \times (100 - 25) + 2257 + 2.0 \times (350 - 100)]$$
$$= 345.5 \text{ kJ/min}$$

(e) Unaccounted heat loss rate

$$\dot{Q}_5 = 3500 - (1257.6 + 1025.81 + 775.93 + 345.5)$$

Heat Balance Sheet

Particulars	*Quntity*	*Percentage*
Credit (input) *Heat supplied by fuel*	*3500 kJ/min*	*100%*
Debit (output)		
Heat equivalent to BP	1257.6 kJ/min	35.93%
Heat carried by coolant	1025.81 kJ/min	29.3%
Heat carried away by dry flue gases	775.93 kJ/min	22.17%
Heat carried away by steam	345.5 kJ/min	9.87%
Unaccounted heat lost	95.16 kJ/min	2.71%

Example 4.23 *The brake power of a 6-cylinder, 4-stroke Diesel engine, absorbed by a hydraulic dynamometer is given by*

$$BP = \frac{WN}{20{,}000} \ (kW)$$

where W is the brake load in newtons and N is the rotational speed in rpm. The air consumption is measured by an air box with a sharp edge orifice. The following readings were recorded.

Bore of cylinder = 9.5 cm
Stroke of piston = 12 cm
Speed of engine = 2400 rpm
Brake load = 550 N
Ambient pressure = 75 cm of Hg
Fuel density = 0.83 kg/litre
Time for 100 cc fuel consumtion = 19.3 seconds
Room temperature = 25°C
C : H ratio in fuel mass = 83 : 17
Orific diameter = 3 cm
C_d (orific) = 0.6
Manometer head across the orifice = 14.5 cm of Hg

Calculate

(a) Brake mean effective pressure,
(b) Brake specific fuel consumtion,
(c) Percentage excess air, and
(d) Volumetric efficiency.

Assume R = 0.287 kJ/kg·K for air.

Solution

Given A six-cylinder, four-stroke Diesel engine:

$$n = \frac{N}{2}, \quad R = 0.287 \text{ kJ/kg}\cdot\text{K}$$
$$k = 6$$
$$d = 9.5 \text{ cm} = 0.095 \text{ m}$$
$$L = 12 \text{ cm} = 0.12 \text{ m}$$
$$N = 2400 \text{ rpm}$$
$$W = 550 \text{ N}$$
$$p_a = 75 \text{ cm of Hg}$$
$$\rho_f = 0.83 \text{ kg/lit.}$$
$$t = 19.3 \text{ s/100 cc consumtion}$$
$$T_a = 25°\text{C} = 298 \text{ K}$$
$$d_o = 3 \text{ cm} = 0.03 \text{ m}$$
$$C_d = 0.6 \text{ cm}$$
$$h_o = 14.5 \text{ cm of Hg}$$
$$\text{C : H ratio} = 83 : 17 \text{ by mass in fuel}$$

To find

(i) *Brake mean effective pressure,*
(ii) Brake-specific fuel consumption, (*Bsfc*),
(iii) *Percentage excess air*, and
(iv) Volumetric efficiency, η_{vol}.

Analysis

(i) *The brake mean effective pressure*

The brake power is given by

$$BP = \frac{WN}{20,000} \text{ (kW)}$$

Using numerical values,

$$BP = \frac{550 \times 2400}{20,000} = 66 \text{ kW}$$

The brake power is also given by

$$BP = \frac{p_{mb} L\,A\,n\,k}{60}$$

or $$p_{mb} = \frac{66 \times 60}{0.12 \times \frac{\pi}{4} \times (0.095)^2 \times \frac{2400}{2} \times 6}$$
$$= 646.61 \text{ kPa} \quad \text{or} \quad \mathbf{6.47 \text{ bar}}$$

(ii) *Brake specific fuel consumtion*

The mass-flow rate of fuel

$$\dot{m}_f = \frac{100 \text{ cc}}{(1000 \text{ cc/lit})} \times (0.83 \text{ kg/lit}) \times \left(\frac{3600 \text{ s/h}}{19.3 \text{ s}}\right)$$
$$= 15.48 \text{ kg/h}$$

$$Bsfc = \frac{\dot{m}_f \text{ (kg/h)}}{BP \text{ (kW)}} = \frac{15.48}{66} = \mathbf{0.2345 \text{ kg/kWh}}$$

(iii) *Prercentage excess air*

Pressure of ambient air;

$$p_a = \frac{750 \text{ mm of Hg}}{760 \text{ mm of Hg}} \times 101.325 \text{ kPa}$$
$$= 99.9917 \cong 100 \text{ kPa}$$

Density of ambient air,

$$\rho_a = \frac{p_a}{RT_a} = \frac{100}{0.287 \times 298} = 1.17 \text{ kg/m}^3$$

Since fuel contains 0.83 kg of carbon and 0.17 kg of hydrogen in 1 kg of fuel, thus the theoretical mass of air required per kg of fuel for complete combustion

$$m_{th} = \frac{100}{23}\left[\frac{8}{3} C + 8H\right]$$
$$= \frac{100}{23} \times \left[\frac{8}{3} \times 0.83 + 8 \times 0.17\right]$$
$$= 15.53 \text{ kg/kg of fuel}$$

Theoretical air supply rate to burn the fuel completely

$$\dot{m}_{th} = m_{th} \times \dot{m}_f$$
$$= 15.53 \times 15.48 = 240.4 \text{ kg/h}$$

The head of air in orifice meter

$$h_a = \frac{h_{Hg}\rho_{Hg}}{\rho_a} = \frac{(0.145 \text{ m}) \times (13600 \text{ kg/m}^3)}{1.17 \text{ kg/m}^3}$$
$$= 1685.4 \text{ m}$$

Actual volume of air supplied through orificre meter

$$\dot{V}_a = \frac{\pi}{4} d_o^2 C_d \sqrt{2g\,h_a}$$
$$= \frac{\pi}{4} \times (0.03)^2 \times 0.6 \times \sqrt{2 \times 9.81 \times 1685.4}$$
$$= 0.077 \text{ m}^3\text{/s}$$

The actual mass-flow rate of air

$$\dot{m}_a = \rho_a \dot{V}_a = 1.17 \times 0.077$$
$$= 0.0902 \text{ kg/s} \quad \text{or} \quad 324.85 \text{ kg/h}$$

Percentage excess air supplied

$$= \frac{324.85 - 240.4}{240.4} \times 100 = \mathbf{35.12\%}$$

(iv) *Volumetric efficiency* (η_{mech})

Swept volume per minute

$$\dot{V}_s = \frac{\pi}{4} \times d^2 \, Lnk$$

$$= \frac{\pi}{4} \times (0.095)^2 \times 0.12 \times \frac{2400}{2} \times 6$$

$$= 6.124 \text{ m}^3/\text{min}$$

Actual volume suction rate

$$\dot{V}_f = 0.077 \text{ m}^3/\text{s} \times (60 \text{ s/min}) = 4.62 \text{ m}^3/\text{min}$$

$$\eta_{vol} = \frac{\dot{V}_a}{\dot{V}_s} = \frac{4.62}{6.124} = 0.754 \quad \text{or} \quad \mathbf{75.4\%}$$

Example 4.24 *During a test on a diesel engine the following observations were made:*

- *The power developed by the engine is used for driving a D.C. generator. The output of the generator was 210 A at 200 V; the efficiency of generator being 82%.*
- *The quantity of fuel supplied to the engine was 11.2 kg/h; calorific value of fuel being 42,600 kJ/kg.*
- *The air-fuel ratio was 18 : 1.*

The exhaust gases were passed through a exhaust gas calorimeter for which the observations were as follows:

- *Water circulated through exhaust gas calorimeter = 580 liters/h*
- *Temperature rise of water through calorimeter = 36°C.*
- *Temperature of exhaust gases at exit from calorimeter = 98°C.*
- *Ambient temperature = 20°C*
- *Heat lost to jacket cooling water is 32% of the total heat supplied.*

If the specific heat of exhaust gases be 1.05 kJ/kg·K. Draw up the heat balance sheet on minute basis.

Solution

Given Test on a diesel engine

Generator output 210 A at 200 V

$$\eta_{generator} = 0.82$$
$$\dot{m}_f = 11.2 \text{ kg/h}$$
$$CV = 42{,}600 \text{ kJ/kg}$$
$$\text{A/F} = 18$$
$$\dot{m}_w = 580 \text{ lit/h}$$
$$(\Delta T)_w = 36°\text{C}$$
$$T_g = 98°\text{C}$$
$$C_{pg} = 1.05 \text{ kJ/kg}\cdot\text{K}$$
$$T_a = 20°\text{C}$$

Heat lost in water jacket = 32% of heat input.

To find Heat balance sheet

Assumption Specific heat of water as 4.187 kJ/kg·K

Analysis Heat supplied by fuel

$$\dot{Q}_{in} = \dot{m}_f CV = \left(\frac{11.2}{60} \text{kg/min}\right) \times (42600 \text{ kJ/kg})$$

$$= 7952 \text{ kJ/min}$$

Mass of air used

$$\dot{m}_a = \dot{m}_f \times \text{A/F} = \left(\frac{11.2}{60} \text{ kg/min}\right) \times 18 = 3.36 \text{ kg/min}$$

Mass of exhaust gases

$$m_g = \dot{m}_a + \dot{m}_f = 3.36 + \frac{11.2}{60} = 3.55 \text{ kg/min}$$

Heat lost to exhaust gases

$$\dot{Q}_3 = \text{Heat gain by water in gas calorimeter} + \text{Heat lost to atmosphere}$$

$$= \dot{m}_w C_{pw} (\Delta T)_w + m_g C_{pg} (T_g - T_a)$$

$$= \left(\frac{580}{60} \text{kg/min}\right) \times 4.187 \times 36 + 3.55 \times 1.05 \times (98 - 20)$$

or $\dot{Q}_3 = 1747.82$ kJ/min

Heat utilised to generate power

$$BP = \frac{\text{Generator output}}{\text{Generator efficiency}} = \frac{VI}{\eta_{generator}}$$

$$= \frac{200 \times 210}{0.82 \times 1000} = 51.22 \text{ kW}$$

Heat equivalent to BP, $\dot{Q}_1 = 3073.17$ kJ/min

Heat lost to coolant in jacket

$$\dot{Q}_2 = 0.32 \times \dot{Q}_{in} = 0.32 \times 7952 = 2544.64 \text{ kJ/min}$$

Unaccounted Heat loss due to radiation etc.

$$Q_4 = \dot{Q}_{in} - (\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3)$$

$$= 7952 - (3073.17 + 2544.64 + 1743.82)$$

$$= 586.37 \text{ kJ/min}$$

Heat Balance Sheet

Particulars		*Quantity*	*Percentage*
Credit, Heat input		7952 kJ/min	100%
Debit			
(i) Heat equivalent to BP,	$\dot{Q}_1$	3073.17 kJ/min	38.64%
(ii) Heat lost to coolant,	$\dot{Q}_2$	2544.64 kJ/min	32%
(iii) Heat lost with exhaust gases,	$\dot{Q}_3$	1747.82 kJ/min	21.97%
(iv) Unaccounted loss,	$\dot{Q}_4$	586.37 kJ/min	7.37%
Total		7952 kJ/min	100%

Example 4.25 *During the trial of a single-cylinder, four-stroke oil engine, the following results were obtained.*

Cylinder diameter = *20 cm*
Stroke = *40 cm*
Mean effective pressure = *6 bar*
Torque = *407 Nm*
Speed = *250 rpm*
Oil consumption = *4 kg/h*
Calorific value of fuel = *43 MJ/kg*
Cooling water flow rate = *4.5 kg/min*
Rise in cooling water temperature = *45°C*
Air used per kg of fuel = *30 kg*
Temperature of exhaust gases = *420°C*
Room temperature = *20°C*
Mean specific heat of exhaust gas = *1 kJ/kg K*
Specific heat of water = *4.18 kJ/kg K.*

Find the ip, bp, and draw up a heat balance sheet for the test in kJ/h.

Solution

Given Single cylinder four stroke oil engine

$k = 1$
$D = 20$ cm $= 0.2$ m
L = 40 cm = 0.4 m
$p_m = 6$ bar $= 600$ kPa
$T = 407$ Nm
$N = 250$ rpm,
$n = N/2$
$\dot{m}_f = 4$ kg/h
CV = 43 mJ/kg = 43000 kJ/kg
$\dot{m}_w = 4.5$ kg/min
$(\Delta T)_w = 45°C$
A/F = 30
$T_g = 420°C$
$T_a = 20°C$
$C_{pg} = 1$ kJ/kg
$C_{pw} = 4.18$ kJ/kg

To find

(i) Indicated power
(ii) Brake power, and
(iii) Heat balance sheet in kJ/h

Analysis Cross-sectional area of cylinder

$$A = \frac{\pi}{4}d^2 = \frac{\pi}{4}\times(0.2)^2 = 0.0314 \text{ m}^2$$

(i) Indicated Power,

$$IP = p_m \frac{L A n k}{60} = \frac{600\times0.4\times0.0314\times250\times1}{60\times2} = \mathbf{15.705\ kW}$$

(ii) Brake Power,

$$BP = \frac{2\pi NT}{60,000} = \frac{2\pi\times250\times407}{60,000} = \mathbf{10.655\ kW}$$

(iii) Heat balance sheet

(a) Heat supplied by fuel

$$Q_{in} = \dot{m}_f \times CV = (4 \text{ kg/h})\times 43000 = 1,72,000 \text{ kJ/h}$$

Heat carried away by cooling water

$$Q_2 = \dot{m}_w C_{pw} (\Delta T)_w$$
$$= (4.5 \times 60 \text{ kg/h}) \times 4.18 \times 45 = 50{,}787 \text{ kJ/h}$$

Mass of exhaust gases

$$\dot{m}_g = m_f \left(1 + \frac{\text{A}}{\text{F}}\right) = 4 \text{ kg/h} \times (1 + 30)$$
$$= 124 \text{ kg/h}$$

Heat carried away by flue gases

$$Q_3 = \dot{m}_g C_{pg} (T_g - T_a)$$
$$= 124 \times 1 \times (420 - 20) = 49{,}600 \text{ kJ/h}$$

Heat equivalent to Brake power

$$Q_1 = BP \times 3600 = 38{,}358 \text{ kJ/h}$$

Heat Balance Sheet

Particulars	*Quantity, kJ/h*	*Percentage*
Credit-Heat input, Q_{in}	1,72,000	100%
Debit-output Heat equivalent to BP, Q_1	38,358	22.3%
Heat carried away by coolant Q_2	50,787	29.53%
Heat carried away by flue gases Q_3	49,600	28.83%
Unaccounted Heat loss	33255	19.33%

Example 4.26 *The following results were obtained in a test on a gas engine:*

Gas used = 0.16 m³/min at NTP,
Calorific value of gas at NTP = 14 MJ/m³,
Density of gas at NTP = 0.65 kg/m³,
Air used = 1.50 kg/min,
Temperature of exhaust gas = 400°C,
Room temperature = 20°C,
Cooling water per minute = 6 kg,
Rise in temperature of cooling water = 30°C,
Specified heat of exhaust gas = 1.0 kJ/kg·K,
Specific heat of water = 4.18 kJ/kg·K,
IP = 12.5 kW,
BP = 10.5 kW.

Draw a heat balance sheet for the test on per hour basis in kJ.

Solution

Given Gas engine

$\dot{V}_f = 0.16 \text{ m}^3/\text{min}$	$CV = 14 \text{ MJ/m}^3$
$\rho_f = 0.65 \text{ kg/m}^3$	$\dot{m}_a = 1.5 \text{ kg/min}$
$C_{pg} = 1 \text{ kJ/kg} \cdot \text{K}$	$T_g = 400°\text{C}$
$T_a = 20°\text{C}$	$\dot{m}_w = 6 \text{ kg/min}$
$C_{pw} = 4.18 \text{ kJ/kg} \cdot \text{K}$	$(\Delta T)_w = 30°\text{C}$
$IP = 12.5 \text{ kW}$	$BP = 10.5 \text{ kW}$

To find

(i) Mechanical efficiency, and

(ii) Heat balance sheet on kJ/h basis.

Analysis

(i) Mechanical efficiency

$$\eta_{mech} = \frac{BP}{IP} = \frac{10.5}{12.5} = 0.84 \text{ or } \mathbf{84\%}$$

Mass of gas used per hour

$$\dot{m}_f = \dot{V}_f \times \rho_f$$
$$= (0.16 \text{ m}^3/\text{min}) \times (0.65 \text{ kg/m}^3)$$
$$= 0.104 \text{ kg/min}$$

Heat supplied by gas

$$Q_s = \dot{V}_f \times CV$$
$$= (0.16 \times 60 \text{ m}^3/\text{h}) \times (14000 \text{ kJ/m}^3)$$
$$= 1{,}34{,}400 \text{ kJ/h}$$

Heat carried away by cooling water

$$Q_2 = \dot{m}_w C_{pw} \times (\Delta T)_w$$
$$= (6 \times 60 \text{ kg/h}) \times (4.18 \text{ kJ/kg·K}) \times (30°\text{C})$$
$$= 45{,}144 \text{ kJ/h}$$

Mass of flue gas,

$$\dot{m}_g = \dot{m}_a + \dot{m}_f$$
$$= 1.5 + 0.104 = 1.604 \text{ kg/min}$$
$$= 96.24 \text{ kg/h}$$

Heat carried away by flue gases

$$Q_3 = \dot{m}_g C_{pg} \times (T_g - T_a)$$
$$= 96.24 \times 1 \times (400 - 20)$$
$$= 36{,}571 \text{ kJ/h}$$

Heat equivalent to brake power

$$Q_1 = 10.5 \text{ kW} \times (3600 \text{ s/h}) = 37{,}800 \text{ kJ/h}$$

Unaccounted heat losses

$$\dot{Q}_4 = \dot{Q}_{in} - \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3$$

$= 134400 - (37800 + 45144 + 36571)$

$= 14885$ kJ/h

Heat Balance Sheet

Particulars	*Quantity, kJ/h*	*Percentage*
(i) Credit Heat supplied by fuel Q_s	1,34,400	100%
(ii) Debit: Output (a) Heat equivalent to BP, Q_1	37800	28.12%
(b) Heat carreid by cooling water, Q_2	45,144	33.59%
(c) Heat carried by flue gases, Q_3	36,571	27.21%
(d) Unaccounted heat loss, Q_4	14885	11.07%

Example 4.27 *A four-stroke gas engine has a cylinder diameter of 25 cm and stroke 45 cm. The effective diameter of the brake is 1.6 m. The observations made in a test of the engine were as follows:*

Duration of test = 40 min,
Total number of revolutions = 8080,
Total number of explosions = 3230,
Net load on the brake = 90 kg,
Mean effective pressure = 5.8 bar,
Volume of gas used = 7.5 m³,
Pressure of gas indicated in meter = 136 mm of water gauge,
Atmospheric temperature = 17°C,
Calorific value of gas = 19 MJ/m³ at NTP,
Rise in temperature of jacket cooling water = 45°C,
Cooling water supplied = 180 kg.

Draw up a heat balance sheet and estimate the indicated thermal efficiency, brake thermal efficiency and mechanical efficiency. Assume atmospheric pressure at 760 mm of Hg.

Solution

Given Four stroke gas engine

$d = 25 \text{ cm} = 0.25 \text{ m}$

$L = 45 \text{ cm} = 0.45 \text{ m}$

$D_{brake} = 1.6 \text{ m}$

$\Delta t = 40 \text{ min}$ duration test

$N = 8080$

No of explosions $= 3230$

$W = 90 \text{ kg}$

$p_m = 5.8 \text{ bar} = 580 \text{ kPa}$

$V = 7.5 \text{ m}^3$

$p_g = 136$ mm of water

$T_a = 17°\text{C}$

$CV = 19 \text{ MJ/m}^3$

$\Delta T_w = 45°\text{C}$

$\dot{m}_w = 180 \text{ kg}$

$p_{\text{atm}} = 760$ mm of Hg

To find

(i) Indicated thermal efficiency,
(ii) Brake thermal efficiency,
(iii) Mechanical efficiency, and
(iv) Heat balance sheet.

Assumption Single cylinder engine

Analysis The cross-sectional area of cylinder

$$A = \frac{\pi}{4}d^2 = \frac{\pi}{4}\times(0.25)^2 = 0.049 \text{ m}^2$$

Indicated power for 3230 revolution in 40 min

$$IP = p_m \frac{LAnk}{\Delta t}$$

$$= 580 \times 0.45 \times (0.049) \times \left(\frac{3230 \text{ rev}}{40 \times 60 \text{ sec}}\right)$$

$$= 17.21 \text{ kW}$$

Brake power for $N = 8080$ revolution in 40 min

$$BP = \frac{2\pi NT}{60000} = \frac{\pi N(WD_{brake})}{60000}$$

$$= \frac{\pi(8080 \text{ rev})}{(40\times 60 \text{sec})} \times \frac{90\times 9.81\times 1.6}{1000} = 14.94 \text{ kW}$$

Absolute pressure of supplied gas

$$p_{abs} = p_{\text{atm}} + p_{\text{gang}} = 760 + \frac{136}{13.6} = 770 \text{ mm of Hg}$$

Volume of gas used at NTP

$$\left(\frac{pV}{T}\right)_{NTP} = \left(\frac{pV}{T}\right)_{supplied\ contions}$$

$$V_{NTP} = \frac{770}{760}\times(7.5 \text{ m}^3)\times\frac{273}{290} = 7.153 \text{ m}^3$$

Heat supplied by gas at NTP per second

$$\dot{Q}_{in} = \frac{\dot{V}_{NTP}\, CV}{\Delta t} = \frac{7.153 \times 19000}{(40 \times 60 \text{ sec})} = 56.63 \text{ kW}$$

(i) Indicated thermal efficiency

$$\eta_{ith} = \frac{IP}{\dot{Q}_{in}} \times 100 = \frac{17.21}{56.63} \times 100 = \mathbf{30.39\%}$$

(ii) Brake thermal efficiency

$$\eta_{bth} = \frac{BP}{\dot{Q}_{in}} \times 100 = \frac{14.94}{56.63} \times 100 = \mathbf{26.38\%}$$

(iii) Mechanical efficiency

$$\eta_{mech} = \frac{BP}{IP} \times 100$$

$$= \frac{14.94}{17.21} \times 100 = \mathbf{86.8\%}$$

(iv) Heat balance on minute basis

(a) Heat supplied by gas

$$\dot{Q}_{in} = 56.63 \times 60 = 3397.8 \text{ kJ/min}$$

(b) Heat equivalent to BP,

$$Q_1 = 14.94 \times 60 = 896.4 \text{ kJ/min}$$

(c) Heat lost to cooling water,

$$Q_2 = \dot{m}_w \times C_{pw} (\Delta T)_w$$

$$= \frac{180 \times 4.187}{40} \times 45 = 847.86 \text{ kJ/min}$$

(d) Unaccounted Heat loss

$$Q_3 = Q_{in} - Q_1 - Q_2$$

$$= 3397.8 - 896.4 - 847.86$$

$$= 1653.53 \text{ kJ/min}$$

Heat Balance Sheet

Particulars	*Quantity*	*Percentage*
Heat input (credit) Heat supplied by fuel	3397.8 kJ/min	100%
Debit (output) (i) Heat equivalent to BP	896.4 kJ/min	26.38%
(ii) Heat lost to coolant	847.86 kJ/min	24.95%
(iii) Unaccounted heat loss	1653.53 kJ/min	48.66%
Total	3397.8	100%

Example 4.28 *The following observations were made during a trial of a single-cylinder, four-stroke cycle gas engine having cylinder diameter of 18 cm and stroke 24 cm:*

Duration of trial = 30 min,
Total number of revolution = 9000,
Total number of explosion = 4450,
Mean effective pressure = 5 bar,
Net load on the brake wheel = 40 kg,
Effective diameter of brake wheel = 1 m,
Total gas used at NTP = 2.4 m^3,
Calorific value of gas at NTP = 19 MJ/m^3,
Total air used = 36 m^3,
Pressure of air = 720 mm Hg,
Temperature of air = 17°C,
Density of air at NTP = 1.29 kg/m^3,
Temperature of exhaust gas = 350°C,
Room temperature = 17°C,
Specific heat of exhaust gas = 1 kg/kg · K.
Cooling water circulated = 80 kg,
Rise in temperature of cooling water = 30°C.

Draw up a heat balance sheet and estimate the mechancial and indicated thermal efficiencies of the engine. Take R = 287 J/kg · K.

Solution

Given Trial of a single cylinder four stroke gas engine

$$k = 1$$
$$d = 18 \text{ cm} = 0.18 \text{ m},$$
$$L = 24 \text{ cm} = 0.24 \text{ m}$$
$$\Delta t = 30 \text{ min}$$
$$N = 9000 \text{ rev}$$
$$n = 4450$$
$$p_{mi} = 5 \text{ bar} = 500 \text{ kPa}$$
$$W = 40 \text{ kg}$$
$$D_{brake} = 1 \text{ m}$$
$$V_{NTP} = 2.4 \text{ m}^3$$
$$CV = 19 \text{ mJ/m}^3 = 19000 \text{ kJ/m}^3$$
$$V_a = 36 \text{ m}^3$$
$$p_a = 720 \text{ mm of Hg}$$
$$T_{atm} = T_a = 17°\text{C}$$
$$\rho_a = 1.29 \text{ kg/m}^3$$

$$T_g = 350°\text{C}$$
$$C_{pg} = 1 \text{ kJ/kg} \cdot \text{K}$$
$$m_w = 80 \text{ kg}$$
$$(\Delta T)_w = 30°\text{C}$$
$$R = 287 \text{ J/kg} \cdot \text{K} = 0.287 \text{ kJ/kg} \cdot \text{K}$$

To find

(i) Indicated thermal efficiency
(ii) Mechanical efficiency, and
(iii) Heat balance sheet

Analysis The cross sectional area of cylinder

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} \times (0.18)^2 = 0.0254 \text{ m}^3$$

Indicated power for $n = 4450$ explosions in 30 min

$$IP = p_{mi} \, L A n K$$
$$= 500 \times 0.24 \times \frac{0.0254 \times 4450}{30 \times 60} \times 1$$
$$= 7.55 \text{ kW}$$

Brake power for $N = 9{,}000$ revolution in 30 min

$$BP = \frac{2\pi NT}{60000} = \frac{\pi D_b NW}{60000}$$
$$= \frac{\pi \times 1 \times 9000 \times 40 \times 9.81}{30 \times 60000} = 6.16 \text{ kW}$$

Heat supplied at NTP,

$$\dot{Q}_{in} = \frac{2.4}{30} \times 19000 = 1520 \text{ kJ/min}$$

Heat equivalent to BP

$$\dot{Q}_1 = 6.16 \times 60 = 396.6 \text{ kJ/min}$$

(i) Indicated thermal efficiency

$$\eta_{ith} = \frac{IP \text{ (kW)}}{\dot{Q} \text{ (kW)}} \times 100$$
$$= \frac{7.55 \times 60}{1520} \times 100 = \mathbf{29.8\%}$$

(ii) Mechanical efficiency

$$\eta_{mech} = \frac{BP}{IP} \times 100$$
$$= \frac{6.16}{7.55} \times 100 = \mathbf{81.59\%}$$

Volume of air used at NTP

$$V_{NTP} = V_a \times \frac{T_o}{T_a} \times \frac{p_{air}}{p_o}$$
$$= 36 \times \frac{273}{290} \times \frac{720}{760} = 32.1 \text{ m}^3$$

Mass of air used

$$\dot{m}_a = \frac{32.1 \times 1.29}{30} = 1.38 \text{ kg/min}$$

Mass of gas used at NTP

$$m_g = \frac{pV}{RT} = \frac{100 \times 2.4}{0.287 \times 273} = 3.06 \text{ m}^3$$

Mass of gas per min

$$\dot{m}_g = \frac{3.06}{30} = 0.102 \text{ kg/min}$$

Total mass of exhaust gas

$$= \dot{m}_a + \dot{m}_g = 1.38 + 0.102 = 1.482 \text{ kg/min}$$

Heat lost to exhaust gases

$$\dot{Q}_3 = \dot{m}_g C_{pg} (T_g - T_a)$$
$$= 1.482 \times 1 \times (350 - 17) = 493.5 \text{ kJ/min}$$

Heat lost to cooling water

$$\dot{Q}_2 = \dot{m}_w \times C_{pw} \times (\Delta T)_w$$
$$= \left(\frac{80 \text{ kg}}{30 \text{ min}}\right) \times 4.187 \times 30 = 334.4 \text{ kJ/min}$$

Unaccounted heat loss

$$\dot{Q}_4 = \dot{Q}_{in} - (\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3)$$
$$= 1520 - (396.6 + 334.4 + 493.5)$$
$$= 322.5$$

Heat Balance Sheet

Particulars	*Quantity*	*%*
Credit (input) Heat supplied by fuel	1520 kJ/min	100%
Debit (Output) (i) Heat equivalent to BP	396.6 kJ/min	24.31%
(ii) Heat lost to coolant	334.4 kJ/min	22%
(iii) Heat lost to exhaust gases	493.5 kJ/min	32.47%
(iv) Unaccounted heat loss	322.5 kJ/min	21.22%
Total	1520 kJ/min	100%

Example 4.29 *A 6-cylinder, four-stroke Diesel engine has a bore of 33.75 cm and a stroke of 37.5 cm. It is tested at half-load conditions and gave the following observations:*

Brake power = *142 kW*
Engine speed = *350 rpm*
Indicated mean effective pressure = *3.72 bar*
Fuel consumption rate = *44 kg/h*
Calorific value of fuel = *44800 kJ/kg*
Air consumption rate = *38.6 kg/min*
Water flow rate in the jacket = *60.2 kg/min*
Rise in temperature of water = *32°C*
Piston cooling oil flow rate = *34.96 kg/min*
C_p *of piston oil* = *2.1 kJ/kg·K*
Rise in cooling oil temperature = *20 °C*
Exhaust gas temperature = *210 °C*
Ambient air temperature = *25°C*
C_p *of exhaust gases* = *1.05 kJ/kg·K*
C_p *of cooling water* = *4.187 kJ/kg·K*

Fuel contains 14% H_2 *by mass.*

Prepare the heat balance sheet on minute and percentage basis . Calculate the specific fuel consumption at full load, assuming frictional power and indicated thermal efficiency do not change with load variation.

Assume partial pressure of water vapour in exhaust gases to be 0.06 bar.

Solution

Given Six-cylinder, four-stroke Diesel engine:

$$k = 6 \qquad N = 350 \text{ rpm}, n = \frac{N}{2}$$
$$d = 33.75 \text{ cm} = 0.3375 \text{ m} \qquad L = 37.5 \text{ cm} = 0.375 \text{ m}$$
$$p_{mi} = 3.72 \text{ bar} = 372 \text{ kPa} \qquad BP = 142 \text{ kW}$$
$$\text{Loading} = \text{Half} \qquad \dot{m}_f = 44 \text{ kg/h}$$
$$CV = 44800 \text{ kJ/kg} \qquad \dot{m}_a = 38.6 \text{ kg/min}$$
$$\dot{m}_w = 60.2 \text{ kg/min} \qquad \dot{m}_{oil} = 34.96 \text{ kg/min}$$
$$(\Delta T)_w = 32°\text{C} \qquad (\Delta T)_{oil} = 20°\text{C}$$
$$T_g = 210°\text{C} \qquad T_a = 25°\text{C} = 298 \text{ K}$$
$$p_v = 0.06 \text{ bar} = 6 \text{ kPa} \qquad C_{pw} = 4.187 \text{ kJ/kg·K}$$
$$C_{pg} = 1.05 \text{ kJ/kg·K} \qquad C_{p,oil} = 2.1 \text{ kJ/kg·K}$$
$$\text{mass of } H_2 = 0.14\, m_f$$

To find

(i) Heat balance sheet, and

(ii) Specific fuel consumtion at full load.

Analysis

(i) *Indicated power*

$$IP = p_{mi} L A \frac{n}{60} k$$
$$= 372 \times 0.375 \times \frac{\pi}{4} \times (0.3375)^2 \times \frac{350}{2 \times 60} \times 6$$
$$= 218.4 \text{ kW}$$

Heat supplied by fuel

$$Q_{in} = \dot{m}_f CV = \frac{44}{60} \times 44800$$
$$= 32853.3 \text{ kJ/min}$$

The heat equivalent to BP

$$\dot{Q}_1 = 142 \times 60 = 8520 \text{ kJ/min}$$

Heat carried by cooling water

$$\dot{Q}_2 = \dot{m}_w C_{pw} (\Delta T)_w$$
$$= 60.2 \times 4.187 \times 32$$
$$= 8065.83 \text{ kJ/min}$$

Heat carried by cooling oil

$$\dot{Q}_3 = \dot{m}_{oil} C_{p,\,oil} (\Delta T)_{oil}$$
$$= 34.96 \times 2.1 \times 20$$
$$= 1468.32 \text{ kJ/min}$$

Mass of exhaust gases formed per minute

$$\dot{m}_{ex} = \dot{m}_a + \dot{m}_f$$
$$= 38.6 + \frac{44}{60} = 39.33 \text{ kg/min}$$

The amount of water vapour formed;

$$\dot{m}_{H_2O} = 9\, H_2 = 9 \times 0.14 \times \frac{44}{60}$$
$$= 0.924 \text{ kg/min}$$

Mass of dry exhaust gases;

$$\dot{m}_g = \dot{m}_{ex} - \dot{m}_{H_2O} = 39.33 - 0.924$$
$$= 38.409 \text{ kg/min}$$

Heat carried by dry exhaust gases

$$\dot{Q}_4 = \dot{m}_g C_{pg} (T_g - T_a)$$
$$= 38.409 \times 1.05 \times (210 - 25)$$
$$= 7460.95 \text{ kJ/min}$$

Heat carried by superheated water vapour at 0.06 bar in exhaust gases

$$\dot{Q}_5 = \dot{m}_{H_2O} h_{sup}$$

where $h_{sup} = h_{g\,@0.06\text{ bar}} + C_{ps}(T_{sup} - T_a)$

$$= 2567.5 + 2.1 \times (210 - 25)$$
$$= 2985.5 \text{ kJ/min}$$

Then, $\dot{Q}_5 = 0.924 \times 2985.5 = 2758.6 \text{ kJ/min}$

Heat Balance Sheet

Particulars	*Quantity*	*Percentage*
Credit (input) *Heat supplied by fuel,* $\dot{Q}_{in}$	*32853.3 kJ/min*	*100%*
Debit (output)		
Heat equivalent to BP, $\dot{Q}_1$	8520 kJ/min	25.93%
Heat carried by cooling water, $\dot{Q}_2$	8065.83 kJ/min	24.55%
Heat carried by cooling oil, $\dot{Q}_3$	1468.32 kJ/min	4.47%
Heat carried away by dry flue gases, $\dot{Q}_4$	7460.95 kJ/min	22.70%
Heat carried away by steam formed, $\dot{Q}_5$	2758.6 kJ/min	8.40%
Unaccounted heat lost $= \dot{Q}_{in} - \dot{Q}_1 - \dot{Q}_2 - \dot{Q}_3 - \dot{Q}_4 - \dot{Q}_5$	4579.6 kJ/min	13.94%

(ii) *Specific fuel consumtion at full load:*

Brake power at full load conditions

$$BP_{full} = 2\,BP_{half} = 2 \times 142 = 284 \text{ kW}$$

Frictional power at half-load condition

$$FP = IP_{half} - BP_{half} = 218.4 - 142 = 76.4 \text{ kW}$$

Since, the FP remains constant, thus IP at full-load condtions

$$IP_{full} = BP_{full} + FP = 284 + 76.4 = 360.4 \text{ kW}$$

Indicated thermal efficiency at half-load conditions

$$\eta_{ith} = \frac{IP_{half}}{\dot{Q}_{in}} = \frac{(218.4 \times 60 \text{ kJ/min})}{(32853.6 \text{ kJ/min})} = 0.3988$$

The heat input rate at full load

$$\dot{Q}_{in,\,full} = \frac{IP_{full}}{\eta_{ith}} = \frac{360.4 \text{ kW}}{0.3988} = 903.71 \text{ kW}$$

Fuel consumption rate at full load

$$\dot{m}_{f,\,full} = \frac{\dot{Q}_{in,full}}{CV} = \frac{903.71 \text{ kW}}{44800 \text{ kJ/kg}} = 0.02017 \text{ kg/s or } 72.62 \text{ kg/h}$$

The brake specific fuel consumtion

$$bsfc = \frac{\dot{m}_{f,\,full}}{BP_{full}} = \frac{(72.62 \text{ kg/h})}{(284 \text{ kW})} = \mathbf{0.255 \text{ kg/kWh}}$$

Example 4.30 *A 4-cylinder, four-stroke engine has an 85 mm bore and 130 mm stroke. It develops 30 kW of power at 1500 rpm and it runs at 20% rich mixture. If the volume of air in the cylinder measured at 17°C and 762 mm of mercury is 70% of the swept volume, theoretical air–fuel ratio is 14.8, calorific value of fuel is 45000 kJ/kg and mechanical efficiency of the engine is 90%, calculate (a) indicated thermal efficiency, and (b) brake mean effective pressure.*

Assume R = 0.287 kJ/kg·K for air.

Solution

Given Four-cylinder, four-stroke engine

$$k = 4$$
$$N = 1500 \text{ rpm}, \; n = \frac{N}{2}$$
$$d = 85 \text{ mm} = 0.085 \text{ m}$$
$$L = 130 \text{ mm} = 0.13 \text{ m}$$
$$\dot{m}_{f,act} = 1.2\, \dot{m}_{f,th}$$
$$BP = 30 \text{ kW}$$
$$CV = 45000 \text{ kJ/kg}$$
$$T_a = 17°\text{C} = 290 \text{ K}$$
$$p_a = 762 \text{ mm of Hg}$$
$$V_{act} = 0.7\, V_s$$
$$A/F = 14.8$$
$$\eta_{mech} = 0.90$$
$$R = 0.287 \text{ kJ/kg}\cdot\text{K}$$

To find

(i) Indicated thermal efficiency, and
(ii) Brake mean effective pressure.

Analysis The atmospheric pressure corresponds to 762 mm of Hg

$$p_a = \frac{762}{760} \times 101.325 \text{ kPa} = 101.59 \text{ kPa}$$

Swept volume of four cylinders of engine

$$V_s = 4 \times \frac{\pi}{4} d^2 L = 4 \times \frac{\pi}{4} \times (0.085)^2 \times 0.13 = 2.95 \times 10^{-3} \text{ m}^3$$

Actual volume of air drawn

$$V_a = 0.7\,V_s = 0.7 \times 2.95 \times 10^{-3}$$
$$= 2.065 \times 10^{-3}\ \text{m}^3$$

The mass of air inducted in cylinders

$$m_a = \frac{p_a V_a}{R T_a} = \frac{101.59 \times 2.065 \times 10^{-3}}{0.287 \times 290}$$
$$= 2.52 \times 10^{-3}\ \text{kg}$$

Theoretical mass of air inducted per minute

$$\dot{m}_a = m_a \frac{N}{2} = 2.52 \times 10^{-3} \times \frac{1500}{2}$$
$$= 1.89\ \text{kg/min}$$

Theoretical fuel used per minute

$$\dot{m}_{f,th} = \frac{\dot{m}_a}{\text{A/F}} = \frac{1.89}{14.8} = 0.1277\ \text{kg/min}$$

When using 20% rich mixture, then

$$\dot{m}_{f,act} = 1.2\,\dot{m}_{f,th}$$
$$= 1.2 \times 0.1277 = 0.1532\ \text{kg/min}$$

or $$= 2.554 \times 10^{-3}\ \text{kg/min}$$

Indicated power

$$IP = \frac{BP}{\eta_{mech}} = \frac{30}{0.9} = 33.33\ \text{kW}$$

(i) *Indicated thermal efficiency*

Heat supplied by fuel

$$\dot{Q}_{in} = \dot{m}_{f,act}\,CV = 2.554 \times 10^{-3} \times 45000$$
$$= 114.93\ \text{kW}$$

Indicated thermal efficiency of the engine

$$\eta_{ith} = \frac{IP}{\dot{Q}_{in}} = \frac{33.33\ kW}{114.93\ kW} = 0.29 \text{ or } \mathbf{29\%}$$

(ii) *Brake mean effective pressure*

Brake power is given by

$$BP = p_{mb} L A \frac{n}{60} k = p_{mb} V_s \frac{n}{60}$$

or $$p_{mb} = \frac{BP}{V_s \frac{n}{60}} = \frac{30}{2.95 \times 10^{-3} \times \frac{1500}{2 \times 60}}$$
$$= \mathbf{813.56\ kPa}$$

Example 4.31 *A six cylinder, 4 stroke SI engine having a piston displacement of 700 cm³ per cylinder developed 78 kW at 3200 rpm, and consumed 27 kg of petrol per hour. The calorific value of petrol is 44 MJ/kg. Estimate:*

(i) The volumetric efficiency of the engine if the air-fuel ratio is 12 and intake air is at 0.9 bar, 32°C

(ii) The brake thermal efficiency, and

(iii) The brake torque

Take for air, R = 0.287 kJ/kg·K.

Solution

Given Six cylinder 4 stroke SI engine

$$K = 6$$
$$V_s = 700\ \text{cm}^3$$
$$BP = 78\ \text{KW}$$
$$N = 3200\ \text{rpm}$$
$$\dot{m}_f = 27\ \text{kg/h}$$
$$CV = 44\ \text{MJ/kg} = 44000\ \text{kJ/kg}$$
$$\text{A/F} = 12$$
$$p_o = 0.9\ \text{bar} = 90\ \text{kPa},$$
$$T_o = 32°\text{C} = 305\ \text{K}$$

To find

(i) Volumetric efficiency,

(ii) Brake thermal efficiency, and

(iii) Brake Torque.

Assumption

Specific gas constant for air as 0.287 kJ/ kg · K

Analysis

(i) Brake torque can be obtained from brake power.

$$BP = \frac{2\pi NT}{60000}$$

$$\text{Brake torque } T = \frac{BP \times 60000}{2\pi N} = \frac{78 \times 60000}{2\pi \times 3200}$$
$$= \mathbf{232.76\ Nm}$$

Heat supplied by fuel

$$\dot{Q}_{in} = \dot{m}_f \times CV$$
$$= \left(\frac{27}{3600}\ \text{kg/s}\right) \times 44000$$
$$= 330\ \text{kW}$$

(ii) Brake thermal efficiency

$$\eta_{bth} = \frac{BP}{\dot{Q}_{in}} \times 100 = \frac{78}{330} \times 100 = \mathbf{23.64\%}$$

From A/F ratio, amount of air used

$$A/F = 12$$

Mass of air $\dot{m}_a = 12 \times \dot{m}_f$

$$= 12 \times 27 = 324 \text{ kg/h} = 5.4 \text{ kg/m}^3$$

Density of air

$$\rho_a = \frac{p_o}{RT_o} = \frac{90}{0.287 \times 305} = 1.028 \text{ kg/m}^3$$

Volume flow rate of air

$$\dot{V}_a = \frac{\dot{m}_a}{\rho_a} = \frac{5.4}{1.028} = 5.253 \text{ m}^3\text{/min}$$

Volume of air induced per cycle per cylinder

$$V = \frac{\dot{V}}{\text{No of suction per cycle} \times N \times k}$$

$$= \frac{5.253 \ (10^6 \text{ cc/m}^3)}{\frac{1}{2} \times 3200 \times 6} = 547.2 \text{ cc}$$

Volumetric efficiency

$$\eta_V = \frac{\text{Actual Volume Sucked}}{\text{Swept Volume}} = \frac{547.2 \times 100}{700}$$

$$= \mathbf{78.2\%}$$

Example 4.32 *The following results were obtained during a Morse test on a four-stroke cycle petrol engine:*

Brake power developed with all cylinders working = 16.2 kW

Brake power developed with cylinders No. 1 cut-out = 11.5 kW

Brake power developed with cylinders No. 2 cut-out = 11.6 kW

Brake power developed with cylinders No. 3 cut-out = 11.68 kW

Brake power developed with cylinders No. 4 cut-out = 11.57 kW

Calculate the mechanical efficiency of the engine. What is the indicated thermal efficiency of the engine, if the engine uses 7 litres of petrol per hour of calorific value of 42,000 kJ/kg and the specific gravity of petrol is 0.72.

Solution

Given Morse test on four stroke petrol engine

$$BP = 16.2 \text{ kW} \qquad BP_1 = 11.5 \text{ kW}$$
$$BP_2 = 11.6 \text{ kW} \qquad BP_3 = 11.68 \text{ kW}$$
$$BP_4 = 11.57 \text{ kW} \qquad \dot{m}_f = 7 \text{ lit/h}$$
$$CV = 42000 \text{ kJ/kg} \qquad Sp \cdot gr = 0.72$$

To find

(a) Mechanical efficiency, and

(b) Indicated thermal efficiency

Analysis When all four cylinder are in operation, with $BP = 16.2$ kW, that is

$$IP_1 + IP_2 + IP_3 + IP_4 - FP = 16.2 \text{ kW} \qquad \ldots\text{(i)}$$

When first cylinder cut out, then

$$IP_2 + IP_3 + IP_4 - FP = 11.5 \text{ kW} \qquad \ldots\text{(ii)}$$

Substracting Eqn. (ii) from Eqn (i), we get

$$IP_1 = 4.7 \text{ kW}$$

Similarly,

$$IP_2 = 4.6 \text{ kW},$$
$$IP_3 = 4.52 \text{ kW},$$
$$IP_4 = 4.63 \text{ kW}$$

Total indicated power

$$IP = IP_1 + IP_2 + IP_3 + IP_4$$
$$= 4.7 + 4.6 + 4.52 + 4.63$$
$$= 18.45 \text{ kW}$$

(a) Mechanical efficiency

$$\eta_{\text{mech}} = \frac{BP}{IP} \times 100 = \frac{16.2}{18.45} \times 100 = \mathbf{87.8\%}$$

Mass of petrol used per hour

$$\dot{m}_f = 7 \text{ lit} \times sp \cdot gr \times \text{Density of water}$$
$$= (7 \times 10^{-3} \text{ m}^3) \times 0.72 \times 1000$$
$$= 5.04 \text{ kg/h} = 1.4 \times 10^{-3} \text{ kg/s}$$

Heat supply rate by fuel

$$\dot{Q}_{in} = \dot{m}_f \times CV$$
$$= (1.4 \times 10^{-3} \text{ kg/s}) \times (42000 \text{ kJ/kg})$$
$$= 58.8 \text{ kW}$$

(b) Indicated thermal efficiency

$$\eta_{ith} = \frac{IP}{\dot{Q}_{in}} \times 100 = \frac{18.45}{58.8} \times 100 = \mathbf{31.37\%}$$

Example 4.33 *A four-stroke, solid injection, diesel engine coupled to a single-phase A.C. generator gave the following data during a trial of 45 minutes duration:*

Fuel consumption = 2.9 kg,

Calorific value of fuel oil = 46900 kJ/kg

Analysis of fuel oil on mass basis:

$$C = 86\%;$$
$$H_2 = 10\%;$$
$$\text{other matter} = 4\%$$

Percentage analysis of dry exhaust gases by volume

$$CO_2 = 7.6; \quad CO = 0.4; \quad O_2 = 6; \quad N_2 = 86$$

Mass of cylinder jacket cooling water = 260 kg,

Room and cylinder jacket water inlet temperature 26°C

Temperature of water leaving cylinder jacket = 73°C,

Temperature of exhaust gases = 290°C

Engine speed = 250 rpm.

Generator voltage and current = 440 V, and 25 Amp

Efficiency of generator = 92%

Soon after the test the engine was motored by the dynamometer taking current from the mains:

Applied voltage - 440 V;

Current - 7.8 Amp.;

Speed - 250 rpm;

Efficiency of the generator as motor - 92%

Draw up a heat balance sheet on percentage basis assuming that the steam in the exhaust gases is at atmospheric pressure (1.01325 bar). Calculate the mechanical efficiency of the engine. Take C_p of dry exhaust gases as 1 kJ/kg·K and C_p of steam as 2.1 kJ/kg·K

Solution

Given Four stroke solid injection diesel engine

$$\Delta t = 45 \text{ min}$$
$$\dot{m}_f = 2.9 \text{ kg}$$
$$CV = 46900 \text{ kJ/kg}$$

Gravimetric analysis of fuel oil

$$C = 0.86\%$$
$$H_2 = 10\%$$
$$\text{other matter} = 4\%$$

Volumetric analysis of dry exhaust gases

$$CO_2 = 7.6\%$$
$$CO = 0.4\%$$
$$O_2 = 6\%,$$
$$N_2 = 86\%$$
$$\dot{m}_w = 260 \text{ kg}$$
$$T_a = 26°C,$$
$$T_{wo} = 73°C$$
$$T_g = 290°C,$$
$$N = 250 \text{ rpm}$$

Generator output: 440 V, 25 A, $\eta_{gen} = 0.92$

During motoring test, generator input

$$440 \text{ V}, \quad 7.8 \text{ A},$$
$$N = 250 \text{ rpm},$$
$$\eta = 0.92$$
$$p_a = 1.01325$$
$$\text{bar} = 101.325 \text{ kPa},$$
$$C_{pg} = 1 \text{ kJ/kg}\cdot\text{K}$$
$$C_{ps} = 2.1 \text{ kJ/kg}$$

To find Heat balance sheet

Assumptions

(i) Water inlet temperature is equal to atmospheric temperature of 26°C

(ii) Specific heat of water as 4.187 kJ/kg·K

Analysis Heat supplied by combustion of fuel

$$Q = 2.9 \times 46900 = 136010 \text{ kJ}$$

(i) Heat supply rate in kJ/min

$$\dot{Q}_{in} = \frac{Q}{45 \text{ min}} = \frac{136010}{45} = 3022.44 \text{ kJ/min}$$

(ii) From dynamometer output, the brake power

$$BP = \frac{VI}{\eta_{gen}} = \frac{440 \times 25}{0.92 \times 1000} = 11.956 \text{ kW}$$

Heat equivalent to BP

$$\dot{Q}_1 = BP \times 60 = 717.4 \text{ kJ/min}$$

(iii) Heat lost to cooling water

$$\dot{Q}_2 = \dot{m}_w C_{pw} \times (T_{wo} - T_a)$$
$$= \left(\frac{260}{45} \text{kg/min}\right) \times 4.187 \times (73 - 26)$$
$$= 1137 \text{ kJ/min}$$

(iv) Mass of air supplied per kg of fuel

$$m_a = \frac{NC}{33 \times (C_1 + C_2)}$$

where N_1, C_1, C_2 are percentage of N_2, CO_2 and CO by volume in exhaust gases and C is mass percentage of carbon in fuel oil.

Then,

$$m_a = \frac{86 \times 86}{33 \times (7.6 + 0.4)} = 28 \text{ kg}$$

O_2 used for combustion of H_2

$$m_{O_2} = 0.1 \times 8 = 0.8 \text{ kg}$$

Mass of air used for dry exhaust products

$$m_{da} = 28 - 0.8 = 27.2 \text{ kg}$$

Mass of dry products of combustion

$$m_g = m_{da} + \text{mass of carbon in fuel}$$
$$= 27.2 + 0.86 = 28.06 \text{ kg/kg of fuel}$$

Mass of exhaust gases per min

$$\dot{m}_g = \frac{2.9}{45} \times 28.06 = 1.81 \text{ kg/min}$$

(v) Heat lost to dry exhaust gases

$$\dot{Q}_3 = \dot{m}_g C_{pg} \times (T_g - T_a)$$
$$= 1.81 \times 1 \times (290 - 26) = 477.65 \text{ kJ/min}$$

(vi) Heat lost in steam formation

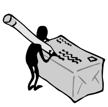
$$\dot{Q}_4 = \dot{m}_f \text{ (kg/min)} \times 9\text{H}_2 \times (h_{fg} + C_{ps}(T_g - 100) - C_{pw} \times T_a)$$
$$= \frac{2.9}{45} \times (9 \times 0.1) \times (2676 + 2.1 \times (290 - 100) - 4.187 \times 26)$$
$$= 172 \text{ kJ/min}$$

(vii) Unaccounted heat loss

$$\dot{Q}_{un} = \dot{Q}_{in} - (\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4)$$
$$= 3022.44 - (717.4 + 1137 + 477.65 + 172)$$
$$= 518.35 \text{ kJ/min}$$

Heat Balance Sheet in kJ per basis minutes basis

Heal supplied	*kJ/min*	*%*	*Heat expenditure*	*kJ/min*	*%*
Heat supplied by combustion of fuel oil	3022.44	100	(1) Heat equivalent of Brake power	717.4	23.74
			(2) Heal lost to jacket cooling water	1137	37.62
			(3) Heat lost to dry exhaust gases	477.65	15.80
			(4) Heat lost to steam in exhaust gases	172	5.69
			(5) Heat lost to radiation, errors of observation, etc. (by difference)	518.35	17.15
Total	3022.44	100	Total	3022.44	100



Summary

- *Indicated power* is defined as the *rate of work done* on the piston by burning of charge inside the cylinder. It is evaluated from an indicated diagram obtained from the engine. It is the *gross power* produced by the engine. *Brake power* is the *net power* available at the engine shaft for external use. It is measured by the brake or dynamometer. It is the part of the indicated power which is used to *overcome the frictional effects* within the engine.
- *Brake specific fuel consumption* is defined as the ratio of the mass of fuel consumed per hour per unit power output (*BP*). It is a parameter which decides the economical power production from an engine.

- *Air-consumption rate* of an engine can be effectively calculated by means of an orifice meter installed in an air box. *Air- Fuel ratio* is the ratio between the mass of the air and mass of the fuel supplied to the engine.
- *Brake thermal efficiency* is defined as ratio of brake power to heat supply rate from fuel to the engine. *Indicated thermal efficiency* is defined as the ratio of the indicated power to the heat supply rate. Mechanical efficiency is the ratio of the brake power and indicated power. The relative efficiency is the ratio of actual thermal efficiency to air standard efficiency of the engine. It is sometimes referred as *efficiency ratio*. Volumetric efficiency is defined as the ratio of the mass of the actual charge inducted into the cylinder to the mass of the charge corresponding to the swept volume.

Glossary

A/F Mass ratio of air to fuel

BDC Bottom dead centre

Bore Internal diameter of cylinder

Brake power Power available from the engine for external use

CI engine Compression ignition (Diesel) engine

Clearance volume Volume left in the centre, when piston is at TDC

Compression ratio Ratio of maximum volume to minimum volume in the cylinder

Friction power Difference between indicated power and brake power

Indicated power Power developed on the piston by combustion gases inside the cylinder

Mean effective pressure Ratio of net work done to swept volume in the cycle

Mechanical efficiency Ratio of brake power to the indicated power

Brake specific fuel consumption Mass of fuel (kg/h) consumed per unit power output (BP)

Swept volume Piston displacement volume in cylinder

TDC Top dead centre

Thermal efficiency Ratio of brake power to the heat supply rate

Review Questions

1. Develop an expression for the calculation of indicated power of an engine.
2. Explain the air box method for the measurement of air consumption in internal combustion engine.
3. Schematically explain the use of the study of heat balance of an engine?
4. What do mean by heat balance sheet. Explain its significance.
5. Name the methods that are used for the measurement of friction power of an engine.
6. Explain the method of measurement of air consumption rate by suction air box. Draw its schematic also.
7. Show the heat distribution pattern of an engine using Sankey diagram.
8. Explain the method of obtaining indicator diagram from an engine.
9. Draw an indicator diagram for a SI engine and formulate the calculation of indicated power from it.
10. Define brake power and explain its evaluation from rope brake dynamometer.
11. Explain working principle of hydraulic dynamometer.
12. Explain Morse test.

13. Why does Willan's line method not use for SI engines?
14. State the emission constituents of SI and CI engines separately.
15. What is the reason of emission of carbon monoxide from an engine? Explain.
16. What is the reason of formation of NO_X in emission of an engine? Explain.
17. What is the reason of formation of blue smoke in emission of an engine? Explain.
18. What is the reason of formation of black smoke in emission of an engine? Explain.
19. What is smog? Why does the smog formation take place from emission of an engine.
20. What are adehydes? State the reason their formation.

Problems

1. The engine of a car has four cylinders of 70 mm bore, and 75 mm stroke. The compression ratio is 8. Determine the cubic capacity of the engine and the clearance volume of each cylinder.

[1154 cc, 41.21 cc]

2. A four-cylinder, four-stroke petrol engine has a bore of 80 mm and a stroke of 80 mm. The compression ratio is 8. Calculate the cubic capacity of the engine and clearance volume of each cylinder. What type of engine is this?

[(a) 1608.4 cc, (b) 57.4 cc, (c) Square engine]

3. A four-cylinder, four-stroke Diesel engine is to develop 30 kW at 1000 rpm. The stroke is 1.4 times the bore and the indicated mean effective pressure is 6.0 bar. Determine the bore and stroke of the engine. *[176 mm, 246 mm]*

4. A 42.5-kW engine has a mechanical efficiency of 85%. Find the indicated power and friction power. If the friction power is constant with the load, what will be the mechanical efficiency at 60% of the load? *[50 kW, 7.5 kW, 77.3%]*

5. Calculate the brake mean effective pressure of a four-cylinder, four-stroke Diesel engine having a 150-mm bore and 200-mm stroke which develops a brake power of 73.6 kW at 1200 rpm.

[5.206 bar]

6. An engine is using 5.2 kg of air per minute, while operating at 1200 rpm. The engine requires 0.2256 kg of fuel per hour to produce an indicated power of 1 kW. The air–fuel ratio 15 : 1. The indicated thermal efficiency is 38% and the mechanical efficiency is 80%. Calculate (a) brake power, and (b) heating value of the fuel.

[(a) 73.7 kW, (b) 41992.9 kJ/kg]

7. An engine develops an indicated power of 125 kW and delivers a brake power of 100 kW. Calculate (a) frictional power, and (b) mechanical efficiency of the engine. *[(a) 25 kW, (b) 80%]*

8. A single-cylinder, four-stroke cycle oil engine is fitted with a rope brake dynamometer. The diameter of the brake wheel is 600 mm and the rope diameter is 26 mm. The brake load is 170 N. If the engine runs at 450 rpm, calculate the brake power of the engine. *[2.5 kW]*

9. A single-cylinder, four-stroke Diesel engine having a displacement volume of 790 cc is tested at 300 rpm. When a braking torque of 49 Nm is applied, analysis of the indicator diagram gives a mean effective pressure of 980 kPa. Calculate the brake power and mechanical efficiency of the engine. *[(a) 1.54 kW, (b) 79.4%]*

10. A four-stroke petrol engine delivers a brake power of 36.8 kW with a mechanical efficiency of 80%. The air–fuel ratio is 15 : 1 and the fuel consumption is 0.4068 kg/kWh. The calorific value of the fuel is 42000 kJ/kg. Calculate (a) indicated power, (b) friction power, (c) brake thermal efficiency, (d) indicated thermal efficiency, and (e) total fuel consumption.

[(a) 46 kW, (b) 9.2 kW, (c) 21%, (d) 26.25%, (e) 15.12 kg/h]

11. During a trial of four-stroke Diesel engine, the following observations were recorded:

Area of indicator diagram = 475 mm^2

Length of the indicator diagram = 62 mm

Spring constant = 1.1 bar/mm

Bore = 100 mm

Stroke = 150 mm

Speed = 375 rpm

Determine (a) indicated mean effective pressure, and (b) indicated power.

[(a) 8.43 bar, (b) 3.1 kW]

12. A four-stroke, gasolene engine develops a brake power of 410 kW. The engine consumes 120 kg of fuel in one hour and air consumption is 40 kg/min. The mechanical efficiency is 87% and the fuel heating value is 43000 kJ/kg. Determine the following:

(a) Air fuel ratio

(b) Indicated and brake thermal efficiencies

[(a) 20, (b) 32.875, 28.6%]

13. A twin cylinder, two-stroke internal combustion engine is operating with a speed of 4000 rpm. The fuel consumption is 10 litres per hour. The indicated mean effective pressure is 7.5 bar. Specific gravity of the fuel is 0.78. *CV* of fuel = 42 MJ/kg, *A/F* = 16, η_{vol} = 75%, η_{mech} = 80%, average piston speed = 600 m/min. Determine the dimensions of the cylinder. Also, calculate the brake thermal efficiency.

[d = 69.1 mm, L = 75 mm, η_{ith} = 30.91%]

14. A single-cylinder, four-stroke CI engine has a bore and stroke of 75 mm and 100 mm respectively. Find the *bmep*, if the torque is 25 N-m.

[35.55 bar]

15. A six cylinder four stroke engine has a bore of 80 mm and stroke of 100 mm, running at a mean speed of 12.5 m/s. It consumes the fuel at the rate of 20 kg/h and develops a torque of 150 Nm. IT has clearance volume of 75 cc per cylinder determine: (a) brake power, (b) brake mean effective pressure, (c) brake thermal efficiency, if fuel calorific value is 42.5 MJ/kg, and (d) relative efficiency based on brake thermal efficiency.

[(a) 58.875 kW, (b) 6.25 bar, (c)24.9%, (d) 44.6%]

16. The following observations were recorded during a test on a single cylinder, four-stroke oil engine: Bore 300 mm, Stroke 450 mm, Speed 300 rpm, Imep 6 bar, brake load 1.5 kN Brake drum diameter 1.8 m, Brake rope diameter 2 cm.

Calculate: Indicated power, (b) brake power, and (c) mechanical efficiency

[(a) 47.71 kW, (b) 42.88 kW, (c) 89.88%]

17. A single-cylinder, four-stroke, gas engine with explosion in every cycle, used 0.23 m^3/min. of gas during a test. The pressure and temperature of gas at the meter being 75 mm of water and 17°C respectively. The calorific value of the gas is 18.800 kJ/m^3 at N.T.P. The air consumption was 285 kg/min. The barometer reading was 743 mm of Hg. The bore of cylinder is 25 cm and stroke 48 cm. The engine is running at 240 rpm. Estimate the volumetric efficiency of the engine relative to air at N.T.P. (a) taking air and gas mixture into account, and (b) taking air only into account. Assume the volume per kg of air at N.T.P. as 0.7734 m .

[(a) 84.7%, (b) 77.9%]

18. The following readings were taken during a test on a single-cylinder, four-stroke cycle oil engine:

Cylinder bore = 20 cm;

Stroke length = 35 cm;

Engine speed = 240 rpm,

Brake torque = 450 N·m;

Indicated mean effective pressure = 700 kPa;

Fuel oil used per hour = 3.5 kg;

C_{pg} = 1 kJ/kg·K.

Calorific value of oil = 46,000 kJ/kg;

Mass of jacket cooling water = 5 kg/min;

Rise in temperature of jacket cooling water = 40°C;

Mass of air supplied = 1.35 kg/min;

Temperature of exhaust gases = 340°C;

Room temperature = 15°C;

Calculate the mechanical and indicated thermal efficiencies and brake specific fuel consumption in kg / kWh, heat loss rate to cooling water and exhaust gases.

[73.4%, 34.43%, 0.31 kg/kWh, 837.4 J/min, 659 kJ/min]

19. The following observations were made during a test on a two-stroke cycle oil engine:

Cylinder dimensions:

20 cm bore; 25 cm stroke; speed = 6 rps;

effective brake drum diameter = 1.2 metres;

net brke load = 440 newtons;

indicated mean effective pressure = 280 kPa;

fuel oil consumption = 3.6 kg/h.;

calorific value of fuel oil = 42,500 kJ/kg;

mass of jacket cooling water per hour = 468 kg;

rise in temperature of jacket cooling water = 28°C;

air used per kg of fuel oil = 34 kg;

temperature of air in test house = 30°C;

temperature of exhaust gases = 400°C;

mean specific heat of exhaust gases = 1 k.J/kg · K.

Calculate: (a) brake power, (b) indicated power, (c) the mechanical efficiency, (d) brake mean effective pressure, and (e) brake specific fuel consumption in kg/ kWh. (f) percentage heat loss to cooling water and exhaust gases. (g) Calculate also the brake thermal efficiency of the engine.

[(a) 9.95 kW, (b) 13.18 kW (c) 75.41%
(d) 211.15 kPa, (e) 0.36 kg/kWh
(f) 35.86%, and 30.47%, (g) 23.41%]

20. A six-cylinder, four-stroke Diesel engine has a bore to stroke ratio of 360 : 500 mm. During the trial, following results were obtained:

Mean area of the indicator diagram, 78 cm^2;

length of the indicator diagram, 7.5 cm;

spring constant = 700 kPa per cm of compression;

brake torque = 14,000 N-m;

speed, 480 rpm;

fuel consumption, 240 kg/h;

calorific value of fuel oil, 44,000 kJ/kg;

jacket cooling water used = 320 kg/min;

rise in temperature of the cooling water, 40°C;

piston cooling oil (specific heat, 2.1 kJ/kg K) used, 140 kg/min with a temperature rise of 28°C.

The exhaust gases give up all their heat to 300 kg/min of water circulating through the exhaust gas calorimeter and raise its temperature through 42°C.

Calculate the brake specific fuel consumption in kg/ kWh and mechanical efficiency of the engine and draw up a heat balance sheet of the engine on the basis of 1 kg of fuel oil.

[0.341 kg/kWh, 79.14%]

Heat Balance Sheet per kg of Fuel Oil

Heat supplied per kg of fuel oil	*kJ*
Heat supplied by combustion of fuel oil	44,000
Heat expenditure per kg of fuel oil	
(1) Heat equivalent of brake power	10,566
(2) Heal lost to jacket cooling water	13,398
(3) Heal lost to piston cooling oil	2,058
(4) Heat lost to exhaust gases (wet)	13,189
(5) Heat lost to radiation, errors of observation, etc. (by difference)	4,789
Total	44,000

21. A single-cylinder, four-stroke cycle gas engine of 25 cm bore and 36 cm stroke, with hit and miss governing, was tested with the following results:

Duration of trial, one hour;

Net load on the brake = 1,200 newtons;

Effective radius of the brake wheel = 0.6 metre;

Total number of revolutions = 14400;

Total number of explosions = 6,600;

Mean effective pressure from indicator diagram, 700 kPa;

Gas used, 13.7 m^3 at NTP calorific value of gas at NTP = 20,000 kJ/m^3;

Mass of cooling water passing through the jacket, 600 kg;

Temperature of jacket cooling water at inlet 15°C and at outlet 50°C;

Mass of exhaust gases, 210 kg;

Temperature of exhaust gases, 400°C;

Room temperature, 15°C;

mean specific heat of exhaust gases 1 kJ/kg K.

Calculate the thermal efficiency on indicated power and brake power basis, and draw up a heat balance sheet for the test on one minute basis in kJ and as percentages of the heat supplied to the engine.

[29.78%, 23.76%]

Heat Balance Sheet in kJ per minute

Heat supplied/min.	*kJ*	*%*
Heat supplied by com-bustion of gas	4,566.67	100
Heat expenditure/min.		
(1) Heat equivalent of brake power	1,085.16	23.77
(2) Heal lost to jacket cooling water	1,465.45	32.09
(3) Heat lost lo exhaust gases (wet)	1,347.4	29.51
(4) Heat lost to radiation, errors of observation, etc. (by difference)	668.66	14.63
Total	4,566.67	100.00

22. A single-cylinder, 4-stroke cycle gas engine of 20 cm bore and 38 cm stroke, with hit and miss governing, was tested with the following results:

Barometer, 720 mm of Hg;
Atmospheric and gas temperatures, 17°C;
Gas consumption = 0153 m^3/min at 8.8 mm of water above atmospheric pressure;
Calorific value of gas = 18,000 kJ/m^3 at N.T.P.;
Density of gas 0.61 kg/m^3 at N.T.P.;
Air used = 145 kg/ min;
Cp of dry exhaust gases, 1.05 kJ/kg.K;
Exhaust gas temperature, 400°C;
M.E.P. - Positive loop = 560 kPa at firing;
M.E.P. - Negative loop = 26.5 kPa at firing;
M.E.P. - Negative loop = 36.7 kPa at missing;
Speed, 285 r.p.m., Explosions per minute = 114;
Brake-torque = 335 N-m;

Calculate (a) the percentage of the indicated power which is used for pumping and for mechanical friction, brake power and heat lost to exhaust gases.

[6.37%, 14.92%, 9.998 kW, 616.97 kJ/min]

5 CHAPTER

Reciprocating Air Compressor

Introduction

An *air compressor* is a machine which takes in atmospheric air, compresses it with the help of some mechanical energy and delivers it at higher pressure. It is also called *air pump*. An air compressor increases the pressure of air by decreasing its specific volume using mechanical means. Thus compressed air carries an immense potential of energy. The controlled expansion of compressed air provides motive force in air motors, pneumatic hammers, air drills, sand-blasting machines and paint sprayers, etc.

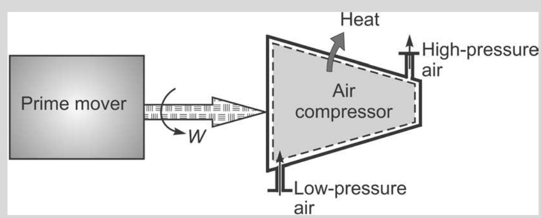


Fig. 5.1 *Schematic of an air compressor*

The schematic of an air compressor is shown in Fig. 5.1. The compressor recieves energy input from a prime mover (an engine or electric motor). Some part of this energy input is used to overcome the frictional effects, some part is lost in the form of heat and the remaining part is used to compress air to a high pressure.

5.1 USES OF COMPRESSED AIR

Compressed air has wide applications in industries as well as in commercial equipment. It is used in

1. Air refrigeration and cooling of large buildings,
2. Driving pneumatic tools in shops like drills, rivetters, screw drivers, etc.
3. Driving air motors in mines, where electric motors and IC engines cannot be used because of fire risks due to the presence of inflammable gases, etc.
4. Cleaning purposes,
5. Blast furnaces,
6. Spray painting and spraying fuel in Diesel engines,
7. Hard excavation work, tunneling, boring, mining, etc.
8. Starting of heavy-duty diesel engines,
9. Operating air brakes in buses, trucks and trains etc.
10. Inflating automobile and aircraft tyres,
11. Supercharging internal combustion engines,
12. Conveying solid and powder materials in pipelines,
13. Process industries,

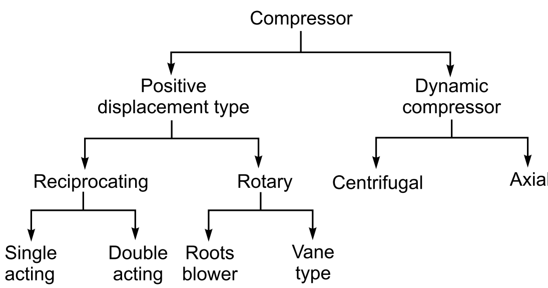
14. Operating lifts, hoists, crains and to operate pumps etc.
15. Pump sets for oil and gas transmission line,
16. Automobile suspension system.

5.2 CLASSIFICATION

The compressors are mainly classified as

(i) Reciprocating compressors, and
(ii) Rotary compressors.

The air compressors can broadly be classified as



A *reciprocating compressor* is used to produce high-pressure gas. It uses the displacement of piston in the cylinder for compression. It handles a low mass of gas and a high pressure ratio.

The *rotary compressors* are used for low and medium pressures. They usually consist of a bladed wheel or impeller that spins inside a circular housing. They handle a large mass of gas.

These compressors may be single stage or multistage to increase the pressure ratio.

5.3 RECIPROCATING COMPRESSOR TERMINOLOGY

In connection to reciprocating compressors, the following terms are defined:

1. Single-acting compressor is a compressor in which suction, compression and delivery of a gas take place only on one side of the piston during a cycle of one revolution of the crank shaft.

2. Double-acting compressor is a compressor in which suction, compression and delivery of gas take place on both sides of the piston and two cycles take place during one revolution of the crank shaft.

3. Single-stage compressor is a compressor in which the compression of gas to final delivery pressure is carried out in one cylinder only.

4. Multistage compressor is a compressor in which the compression of gas to the final pressure is carried out in more than one cylinder in series.

5. Pressure ratio is defined as the ratio of absolute discharge pressure to absolute suction pressure.

6. Free air is the air that exists under atmospheric condition.

7. Compressor displacement volume is the volume created when the piston travels a stroke. It is given as

$$V = \frac{\pi}{4} d^2 L \qquad ...(5.1)$$

where d is the bore of the cylinder and L the is stroke of the piston.

8. Induction-volume rate or volume-flow rate into the compressor is expressed in m^3/s and is given as

$\dot{V}$ = Volume inducted per cycle × No. of inductions per revolution × Number of revolutions per second

For the *single-acting reciprocating compressor*, only one cycle (thus, one induction) takes place for each revolution of the crank. Thus, for a compressor without clearance

$$\dot{V} = \frac{\pi}{4} d^2 L \frac{N}{60} \qquad ...(5.2)$$

For the *double-acting reciprocating compressor*, the induction takes place on both sides of the piston for each revolution. Thus,

$$\dot{V} = \frac{\pi}{4} d^2 L \left(\frac{2N}{60}\right) \qquad ...(5.3)$$

9. Capacity of a compressor is the actual quantity of air delivered per unit time at atmospheric conditions.

***10. Free Air delivery* (*FAD*)** It is the discharge volume of the compressor corresponding to ambient conditions.

11. Piston speed is the linear speed of the piston measured in m/min. It is expressed as

$$v_{piston} = 2LN \quad ...(5.4)$$

5.4 COMPRESSED AIR SYSTEMS

Compressed air systems consist: intake air filters, inter-stage coolers, after coolers, air dryers, moisture drain traps, receivers, piping network, control valves and lubricators.

1. Intake Air Filters They prevent dust from entering the compressor. Dust causes sticking of valves, scoured cylinders, excessive wear, etc.

2. Inter-stage Coolers These are placed between consecutive stages of multistage compressor. They reduce the temperature of compressed air, before it enters the next stage of compression.

3. After Coolers They remove heat of compression and moisture from the air by reducing the temperature in a water-cooled heat exchanger, after compression is completed.

4. Air-dryers The remaining traces of moisture, after an after-cooler are removed by using air dryers, for using compressed air in instruments and pneumatic equipment. The moisture is removed by using adsorbents like silica gel or activated carbon, or refrigerant dryers, or heat of compression dryers.

5. Moisture Drain Traps Moisture drain traps are used for removal of moisture in the compressed air. These traps are manual drain cocks, timer based/ automatic drain valves, etc.

6. Air Receivers are cylindrical tanks into which the compressed air is discharged after final stage of compression from the air compressor. Receiver acts as storage tank and it helps to reduce pulsations and pressure variations from the compressed in the discharge line.

5.5 RECIPROCATING AIR COMPRESSOR

A machine which takes in air or gas during suction stroke at low pressure and then compresses it to high pressure in a piston–cylinder arrangement is known as a reciprocating compressor. External work must be supplied to the compressor to achieve required compression. This work is used to run the compressor. A part of the work supplied to the compressor is lost to overcome the frictional resistance between rubbing surfaces of the piston and cylinder. The cylinder of air compressor is cooled to minimise the work input.

The air compressed by a reciprocating compressor cannot directly be used for an application. The reciprocating motion of the piston gives rise to pulsating flow through the discharge valve of the compressor. Thus, the compressed air is discharged from the air compressor to an air receiver.

5.5.1 Construction

Figure 5.2 shows the sectional view of a single-stage air compressor. It consists of a piston, cylinder with cooling arrangement, connecting rod, crank, inlet and delivery valves. The piston fitted with piston rings, reciprocates in the cylinder. The prime mover (an engine or electric motor) drives the crank shaft, the crank rotates and converts rotary motion into reciprocating motion of piston

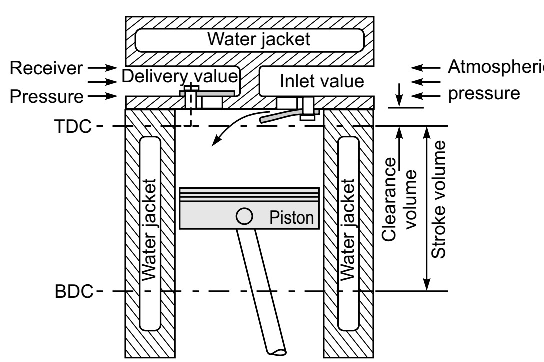


Fig. 5.2 *Sectional view of single-stage reciprocating air compressor*

with the help of a connecting rod. The cylinder head consists of spring-loaded inlet and delivery valves, which are operated by a small pressure difference across them. The light spring pressure gives a rapid closing action. The piston rings seal the gap between the piston and cylinder wall. The cylinder is surrounded by a water jacket or metallic fins for proper cooling of air during compression.

The double-acting air compressor is shown in Fig. 5.3. Its construction is very similar to that of a single-acting air compressor, except for two inlet and two delivery valves on two ends of the cylinder in order to allow air entry and delivery on two sides of the piston. When the piston compresses the air on its one side, it creates suction on the other side. Thus, the suction and compression of air take place on two sides of the piston simultaneously.

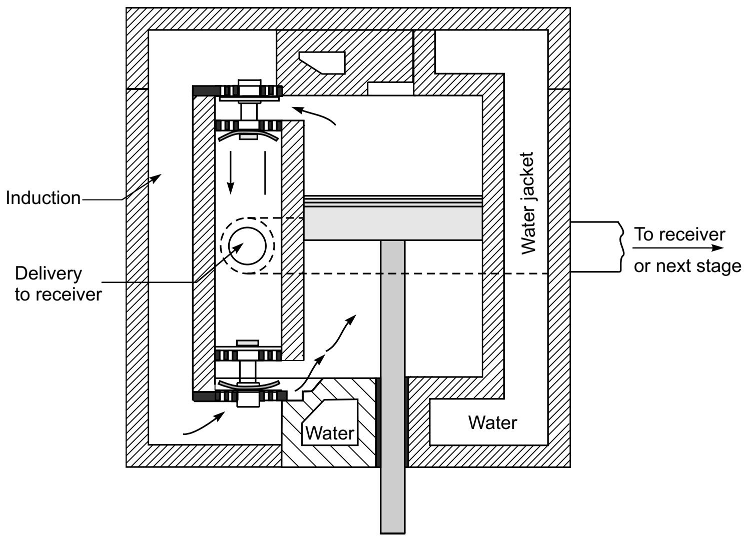


Fig. 5.3 *Double-acting reciprocating air compressor*

5.5.2 Working of a Single-Acting Air Compressor

As the piston moves in a downward stroke (from TDC to BDC), any residual compressed air left in the cylinder from the previous cycle expands first. On further movement of the piston, the pressure in the cylinder falls below the atmospheric pressure. The atmospheric air pushes the inlet valve to open and fresh air enters the cylinder as shown in Fig. 5.4. The line c-1 represents the induction stroke. During this stroke, the compressed air in the storage tank acts on the delivery valve, thus it remains closed. As the piston begins its return stroke from BDC to TDC, the pressure in the cylinder increases, and closes the inlet valve. The air in the cylinder is compressed by piston as shown by the curve 1-2.

During the compression stroke, as air pressure reaches a value, which is slightly more than the pressure of compressed air acting outside the delivery valve, the delivery valve opens and the compressed air is discharged from the cylinder to storage tank. At the end of the compression stroke, the piston once again moves downward, the pressure in the cylinder falls below the atmospheric pressure, the delivery valve closes and inlet valve opens for next cycle. The suction, compression and delivery of air take place with two strokes of the piston which is one revolution of the crank.

Figure 5.4 shows the p–V diagram for a reciprocating compressor without clearance. The processes are summarized below:

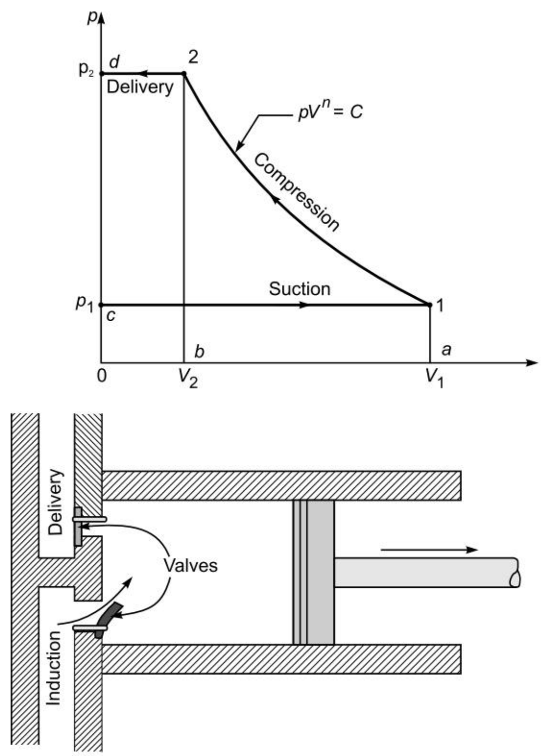


Fig. 5.4 *p–V diagram for a reciprocating compressor without clearance*

Process c–1 Suction stroke—inlet valve opens and air enters the compressor at constant pressure p_1

Process 1–2 Polytropic compression of air from pressure p_1 to pressure p_2

Process 2–d Discharge of compressed air through delivery valve at constant pressure p_2

Process d–c No air in the cylinder and return of piston for suction stroke

5.5.3 Indicated Work for a Single-acting Compressor without Clearance

The theoretical $p-V$ diagram for single-stage, single-acting reciprocating air compressor without clearance is shown in Fig. 5.4. The net work done in the cycle is equal to the area behind the curve on $p-V$ diagram and it is the work done on air.

Indicated work done on the air per cycle

= Area behind the curve, i.e., area c–1–2–d–c

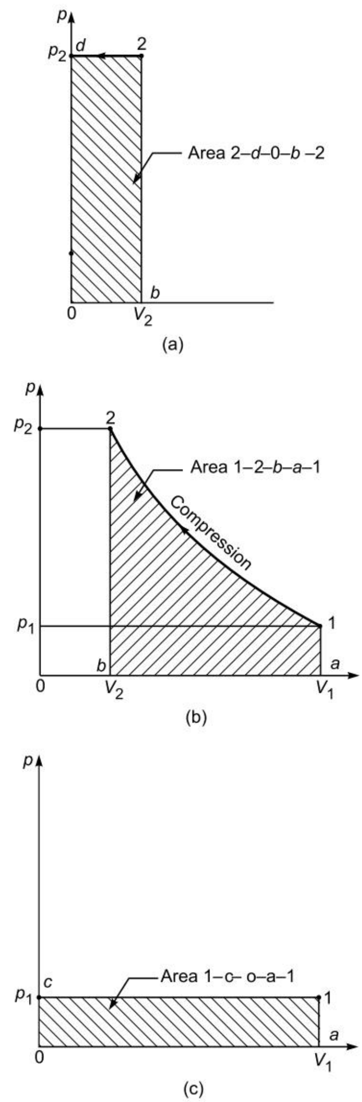


Fig. 5.5 *Three process areas on p–V diagram*

= Area 2–d –0 –b–2 + Area 1–2–b –a –1 – Area 1–c – 0 – a –1

These three areas are shown in Fig. 5.5 as (a), (b) and (c), respectively.

Area 2–d – 0 – b –2 = p_2V_2 (Flow work during discharge at constant pressure p_2)

Area 1–2–b–a–1 $= -\int pdV$ (Piston dispalcement work from p_1 to p_2, – sign is taken for compression)

Area 1–c–0–a–1 $= p_1 V_1$ (Flow work during suction at constant pressure p_1)

During compression process 1–2; the pressure and volume are related as

$$pV^n = \text{C (constant)}$$

Thus we get $-\int pdV = \dfrac{p_2V_2 - p_1V_1}{n-1}$

Therefore, the total indicated work input to compressor is

$$W_{in} = p_2V_2 + \frac{p_2V_2 - p_1V_1}{n-1} - p_1V_1 \qquad \text{...(5.5)}$$

$$= (p_2V_2 - p_1V_1)\left[\frac{1}{n-1} + 1\right]$$

$$W_{in} = \frac{n}{n-1}(p_2V_2 - p_1V_1) \text{ (kJ/cycle) ...(5.6)}$$

Using charecteristic gas equation as

$$pV = m_a RT$$

Equation (5.6) can be modified as

$$W_{in} = \frac{n}{n-1} m_a R(T_2 - T_1) \text{ (kJ/cycle)} \quad \text{...(5.7)}$$

Other expression for indicated work can be derived by arranging Eq. (5.7) as

$$W_{in} = \frac{n}{n-1} m_a RT_1\left[\frac{T_2}{T_1} - 1\right]$$

It is convenient to express the temperature ratio in terms of delivery and intake pressure ratio.

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

Then $W_{in} = \dfrac{n}{n-1} m_a RT_1\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$ (kJ/cycle) ...(5.8)

or $W_{in} = \dfrac{n}{n-1} p_1V_1\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$ (kJ/cycle) ...(5.9)

where V_1 is the volume inducted per cycle.

5.5.4 Mean Effective Pressure (p_m)

It is a hypothetical average pressure, which if acted on the piston during the entire compression stroke will require the same power input as required during the actual cycle.

Net work input in a cycle,

$$W_{in} = p_m \times \text{(Swept volume)}$$

$$= p_m \times V_s$$

Thus $p_m = \dfrac{W_{net}}{V_s} = \dfrac{\text{Work output}}{\text{Swept volume}}$ ...(5.10)

From a given indicator diagram the indicated mean effective pressure can be obtained as

$$p_m = \frac{\text{Area of indicator diagram (mm}^2\text{)}}{\text{Length of the indicator diagram (mm)}} \times \text{Spring constant (kPa/mm)} \quad \text{...(5.11)}$$

5.5.5 Power and Mechanical Efficiency

***1. Indicated Power* (*IP*)** The work done on air per unit time is called *indicated power input to the compressor*. The power required by an air compressor, running at N rpm is given as

Indicated power IP

= Work input per cycle × Number of cycles per unit time

or $IP = \dfrac{W_{in} N k}{60}$ (kW) ...(5.12)

From an indicated diagram, It is calculated as

IP = Indicated mean effective pressure × Swept volume rate

$= \dfrac{p_{mi} L A N k}{60}$ (kW) ...(5.13)

where for Eq. (5.12) and Eq. (5.13);

W_{in} = Indicated work input per cycle

p_{mi} = Indicated mean effective pressure, (kPa or kN/m^2)

L = Stroke length, (m)

A = $(\pi/4)d^2$, cross-sectional area of cylinder of bore, d, (m)

N = number of rotation per minute

k = number of suction per revolution of crank shaft

= 1 for single-acting reciprocating compressor

= 2 for double-acting reciprocating compressor

2. Brake Power (BP) The actual power (brake power or shaft power) input to the compressor is more than the indicated power because some work is required to overcome the irreversibilities and mechanical frictional effects.

Brake power;

$$BP = \text{Indicated power} + \text{Frictional power} \qquad ...(5.14)$$

3. Mechanical Efficiency η_{mech} The mechanical efficiency of the compressor is given by

$$\eta_{mech} = \frac{\text{Indicated power}}{\text{Brake power}} \qquad ...(5.15)$$

The brake power is derived from a driving motor or engine. The input of a driving motor can be expressed as

$$\text{Motor power} = \frac{\text{Shaft power (or brake power)}}{\text{Mechanical efficiency of motor and drive}} \qquad ...(5.16)$$

Example 5.1 *A single-stage reciprocating air compressor takes in 1.4 kg of air per minute at 1 bar and 17°C and delivers it at 6 bar. Assuming compression process follows the law $pV^{1.35}$ = constant, calculate indicated power input to compressor.*

Solution

Given A single-stage reciprocating air compressor

$$\dot{m}_a = 1.4 \text{ kg/min} \qquad p_1 = 1 \text{ bar}$$
$$T_1 = 17°\text{C} = 290 \text{ K} \qquad p_2 = 6 \text{ bar}$$
$$n = 1.35 \text{ bar}$$

Law $pV^{1.35} = \text{C}$

To find Indicator power input to compressor.

Assumptions

(i) Negligible clearance volume in the compressor.
(ii) No throttling effects on valve opening and closing.
(iii) Air as an ideal gas with $R = 0.287$ kJ/kg·K.

Analysis The delivery temperature of air

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (290 \text{ K}) \times \left(\frac{6}{1}\right)^{\frac{1.35-1}{1.35}}$$
$$= 461.46 \text{ K}$$

The rate of work input to compressor, Eq. (5.7)

$$W_{in} = \frac{n}{n-1}\,\dot{m}_a R(T_2 - T_1)$$
$$= \frac{1.35}{1.35-1} \times (1.4 \text{ kg/min}) \times (0.287 \text{ kJ/kg·K}) \times (461.46 - 290) \text{ (K)}$$
$$= 265.72 \text{ kJ/min}$$

Indicated power input;

$$IP = \frac{W_{in}}{60} = \frac{(265.72 \text{ kJ/min})}{(60 \text{ s/min})} = \mathbf{4.43\ kW}$$

Note: The indicated work input to compressor can also be calculated by using Eq. (5.8).

Example 5.2 *A single-acting, single-cylinder reciprocating air compressor has a cylinder diameter of 200 mm and a stroke of 300 mm. Air enters the cylinder at 1 bar; 27°C. It is then compressed polytropically to 8 bar according to the law $pV^{1.3}$ = constant. If the speed of the compressor is 250 rpm, calculate the mass of air compressed per minute, and the power required in kW for driving the compressor.*

Solution

Given A single-acting, single-cylinder reciprocating air compressor

$$d = 200 \text{ mm} = 0.2 \text{ m} \qquad L = 300 \text{ mm} = 0.3 \text{ m}$$
$$p_1 = 1 \text{ bar} = 100 \text{ kPa} \qquad p_2 = 8 \text{ bar}$$
$$N = 250 \text{ rpm} \qquad T_1 = 27°\text{C} = 300 \text{ K}$$
$$n = 1.3$$

To find

(i) The mass of air compressed in kg/min, and
(ii) Power input to compressor in kW.

Assumptions

(i) Negligible clearance volume in the cylinder.
(ii) Air as an ideal gas with $R = 0.287$ kJ/kg·K.

Analysis The swept volume of the cylinder per cycle

$$V_s = V_1 = \left(\frac{\pi}{4}\right) d^2 L$$
$$= \left(\frac{\pi}{4}\right) \times (0.2 \text{ m})^2 \times (0.3 \text{ m})$$
$$= 9.424 \times 10^{-3} \text{ m}^3$$

The mass of air, using perfect gas equation

$$m_a = \frac{p_1 V_1}{RT_1} = \frac{(100 \text{ kPa}) \times (9.424 \times 10^{-3} \text{ m}^3)}{(0.287 \text{ kJ/kg·K}) \times (300 \text{ K})}$$
$$= 0.0109 \text{ kg/cycle}$$

The mass flow rate of air;

$\dot{m}_a$ = mass of air × number of suction/min = $m_a N$

= 0.0109 × 250 = **2.74 kg/min**

Temperature of air after compression

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

$$= (300\,\text{K})\times\left(\frac{8}{1}\right)^{\frac{1.3-1}{1.3}} = 484.75\ \text{K}$$

The work input to compressor, Eq. (5.7)

$$W_{in} = \frac{n}{n-1}\, m_a R(T_2 - T_1)$$

$$= \frac{1.3}{1.3-1}\times(2.74\ \text{kg/min})\times(0.287\ \text{kJ/kg}\cdot\text{K})\times(484.75-300)\ (\text{K})$$

= 629.56 kJ/min or **10.49 kW**

Example 5.3 *A single-acting, single-cylinder reciprocating air compressor is compressing 20 kg/min. of air from 110 kPa, 30°C to 600 kPa and delivers it to a receiver. Law of compression is $pV^{1.25}$ = constant. Mechanical efficiency is 80%. Find the power input to compressor, neglecting losses due to clearance, leakages and cooling.*

Solution

Given A single-stage reciprocating air compressor

$\dot{m}_a$ = 20 kg/min p_1 = 110 kPa

T_1 = 30°C = 303 K p_2 = 600 kPa

Law $pV^{1.25}$ = C

η_{mech} = 0.8

To find Power input to compressor.

Assumptions

(i) Negligible clearance volume in the compressor.

(ii) No throttling effects on valve opening and closing.

(iii) Air as an ideal gas with $R = 0.287$ kJ/kg·K.

Analysis The delivery temperature of air

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (303\ \text{K})\times\left(\frac{600}{110}\right)^{\frac{1.25-1}{1.25}}$$

= 425.4 K

The indicated power input to compressor,

$$IP = \frac{n}{n-1}\,\dot{m}_a R(T_2 - T_1)$$

$$= \frac{1.25}{1.25-1}\times\left(\frac{20}{60}\ \text{kg/s}\right)\times(0.287\ \text{kJ/kg}\cdot\text{K})\times(425.4-303)\ (\text{K})$$

= 58.55 kW

The motor (brake) power

$$BP = \frac{IP}{\eta_{mech}} = \frac{58.55\ \text{kW}}{0.8} = \mathbf{73.18\ kW}$$

Example 5.4 *A single-cylinder, double-acting, reciprocating air compressor receives air at 1 bar; 17°C, compresses it to 6 bar according to the law $pV^{1.25}$= constant. The cylinder diameter is 300 mm. The average piston speed is 150 m/min at 100 rpm. Calculate the power required in kW for driving the compressor. Neglect clearance.*

Solution

Given A double-acting, single-cylinder reciprocating air compressor

d = 300 mm = 0.3 m p_1 = 1 bar = 100 kPa

p_2 = 6 bar N = 100 rpm

T_1 = 17°C = 290 K n = 1.25

k = 2 $\mathcal{V}_{piston}$ = 150 m/min

To find Power input to compressor in kW.

Analysis The piston speed is given as

$$\mathcal{V}_{piston} = 2\,LN$$

Stroke of piston;

$$L = \frac{\mathcal{V}_{piston}}{2N} = \frac{150\ \text{m/min}}{2\times(100\ \text{rotation/min})}$$

= 0.75 m

The swept volume of the cylinder per cycle

$$V_s = V_1 = \left(\frac{\pi}{4}\right)d^2 L = \frac{\pi}{4}\times(0.3\ \text{m})^2\times(0.75\ \text{m})$$

= 0.053 m³

The indicated work input to compressor by Eq. (5.9)

$$W_{in} = \frac{n}{n-1}\,p_1 V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.25}{1.25-1}\times 100\times 0.053\times\left[\left(\frac{6}{1}\right)^{\frac{1.25-1}{1.25}} - 1\right]$$

= 11.42 kJ/cycle

For a double-acting reciprocating compressor, the indicated power

$$IP = \frac{W_{in} N k}{60} \text{ (kW)}$$

$$= \frac{11.42 \times 100 \times (2 \text{ for double acting})}{60}$$

$$= \mathbf{38.1\ kW}$$

Example 5.5 *A single-stage, single-acting, reciprocating air compressor takes in 1 m³ air per minute at 1 bar and 17°C and delivers it at 7 bar. The compressor runs at 300 rpm and follows the law $pV^{1.35}$= constant. Calculate the cylinder bore and stroke required, assuming stroke-to-bore ratio of 1.5. Calculate the power of the motor required to drive the compressor, if the mechanical efficiency of the compressor is 85% and that of motor transmissions is 90 %. Neglect clearance volume and take R = 0.287 kJ/kg·K for air.*

Solution

Given A single-stage, single-acting, reciprocating air compressor

$\dot{V} = 1\ \text{m}^3/\text{min}$	$p_1 = 1\ \text{bar} = 100\ \text{kPa}$
$T_1 = 17°\text{C} = 290\ \text{K}$	$p_2 = 7\ \text{bar}$
$N = 300\ \text{rpm}$	$n = 1.35$
$\eta_{transmission} = 0.9$	$\eta_{mech} = 0.85$
$L/d = 1.5$	$R = 0.287\ \text{kJ/kg}\cdot\text{K}$

To find

(i) Cylinder bore, and strokes, and

(ii) Motor power.

Analysis Volume sucked in per cycle

$$V_s = \frac{\dot{V}}{N k} = \frac{1\ \text{m}^3/\text{min}}{(300\ \text{rpm}) \times 1} = \frac{1}{300}\text{m}^3$$

The cylinder (swept) volume also given as,

$$V_s = \frac{\pi}{4} d^2 L = \frac{\pi}{4} d^2 (1.5d) = 1.5 \times \left(\frac{\pi}{4}\right) d^3$$

Equating two equations

$$1.5 \times \left(\frac{\pi}{4}\right) d^3 = \frac{1}{300}$$

We get

cylinder bore,

$$d = 0.1414\ \text{m} = \mathbf{141.4\ mm}$$

and stroke;

$$L = 1.5\, d = \mathbf{212.10\ mm}$$

The mass flow rate of air per minute,

$$\dot{m}_a = \frac{p_1 \dot{V}}{R T_1} = \frac{(100\ \text{kPa}) \times (1\ \text{m}^3)}{(0.287\ \text{kJ/kg}\cdot\text{K}) \times (290\ \text{K})}$$

$$= 1.2\ \text{kg/min}$$

The temperature of air after compression

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (290\,\text{K}) \times \left(\frac{7}{1}\right)^{\frac{1.35-1}{1.35}}$$

$$= 480.28\ \text{K}$$

The rate of work input to compressor, Eq. (5.7)

$$\dot{W} = \frac{n}{n-1}\, \dot{m}_a R (T_2 - T_1)$$

$$= \frac{1.35}{1.35-1} \times 1.2 \times 0.287 \times (480.28 - 290)$$

$$= 252.77\ \text{kJ/min}$$

Indicated power required;

$$IP = \frac{(252.77\ \text{kJ/min})}{(60\ \text{s/min})} = \mathbf{4.21\ kW}$$

The brake power input to compressor;

$$\text{Brake power} = \frac{IP}{\eta_{mech}} = \frac{(4.21\ \text{kW})}{0.85} = 4.956\ \text{kW}$$

The motor power required;

$$\text{Motor power} = \frac{\text{Brake power}}{\eta_{transmission}} = \frac{(4.956\ \text{kW})}{0.9}$$

$$= \mathbf{5.5\ kW}$$

5.6 MINIMIZING COMPRESSION WORK

The work done on the gas for compression can be minimized when the compression process is executed in an internally reversible manner, i.e., by minimizing the irreversibilities. The other way of reducing the compression work is to keep the specific volume of gas as small as possible during compression process. It is achieved by keeping the gas temperature as low as possible during the compression. Since specific volume of gas is proportional to temperature, therefore, the cooling arrangement is provided on the compressor to cool the gas during the compression.

For better understanding of the effect of cooling during compression process, we consider three types of compression processes executed between same pressure levels (p_1 and p_2); an isentropic compression 1–2″ (involves no cooling), a polytropic compression 1–2 (involves partial cooling) and an isothermal compression 1–2′ (involves perfect cooling) as shown in Fig. 5.6.

(a) The indicated compression work per cycle for a polytropic compression process 1–2 is given by Eq. (5.9)

$$W_{Poly} = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

(b) Isentropic compression process 1–2″

An equation for indicated work input can be obtained as Eq. (5.9) by replacing n by γ. That is,

$$W_{isentropic} = \frac{\gamma}{\gamma-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad ...(5.17)$$

(c) Isothermal compression process 1–2′: With perfect cooling ($T_2 = T_1$);

Indicated work input for isothermal compression is given by area c–1–2′–d–c.

Area c–1–2′–d–c = Area a–1–2′–b + Area b–2′–d–0–Area a–1–c–0

$$W_{iso} = -\int_{V_1}^{V_2} p dV + p_2 V_2 - p_1 V_1$$

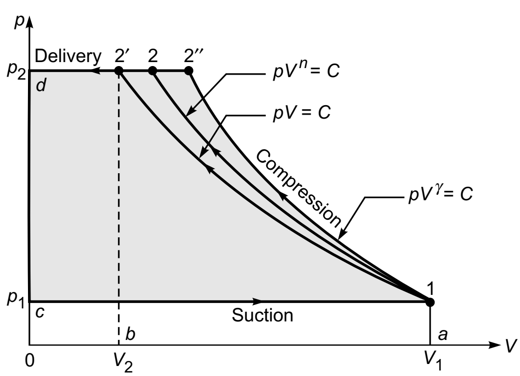


Fig. 5.6 *Three types of compression processes on a p–V diagram*

For an isothermal process; using

$$p = \frac{C}{V} \quad \text{(Since } pV = C\text{)}$$

We get $-\int_{V_1}^{V_2} p dV = p_1 V_1 \ln\left(\frac{V_1}{V_2}\right)$

For isothermal process;

$$p_2 V_2 = p_1 V_1 \text{ and } \frac{V_1}{V_2} = \frac{p_2}{p_1}$$

$$\therefore \quad W_{iso} = p_1 V_1 \ln\left(\frac{p_2}{p_1}\right) \quad ...(5.18)$$

where V_1 is the volume of the air inducted per cycle.

The three processes are plotted on a p–V diagram in Fig. 5.6 for same inlet state and exit pressure. The area of the indicator diagram is the measure of compression work. The only type of compression can influence the magnitude of area of indicator diagram and length of the line 2–d.

It is interesting to observe from this diagram that among the three processes considered, the area with isentropic compression is *maximum.* Thus it requires maximum work input and with isothermal compression, the area of indicator diagram is *minimum.* Thus, the *compressor with isothermal compression will require minimum* work input.

5.6.1 Isentropic Efficiency

This term is seldom used in practice for reciprocating compressors. The isentropic efficiency is also called adiabatic efficiency. The isentropic efficiency of an air compressor is defined as the ratio of isentropic work input to actual work input.

$$\eta_{isentropic} = \frac{\text{Isentropic work input}}{\text{Actual work input}} \quad ...(5.19)$$

5.6.2 Compressor Efficiency

It compares the indicated work input to isothermal work input to the compressor and it is defined as ratio of isothermal work input to indicated work input

$$\eta_{comp} = \frac{\text{Isothermal work input}}{\text{Indicated work input}} \quad ...(5.20)$$

5.6.3 Isothermal Efficiency

It compares the actual work done on the gas with isothermal compression work, and is defined as the ratio of isothermal work input to actual work input during compression, i.e.,

$$\eta_{iso} = \frac{\text{Isothermal work input}}{\text{Actual work input}} \quad ...(5.21)$$

5.6.4 Methods for Improving Isothermal Efficiency

As illustrated with the help of Fig. 5.6, the compression of gas in isothermal manner requires minimum work input. With isothermal compression, the temperature remains constant throughout the compression process.

$$T_2 = T_1$$

Isothermal compression is only possible when all the heat generated during compression is dissipated to cooling medium around the cylinder wall. It is possible, when the compressor runs very slowly.

In actual practice, the compression process should approach isothermal compression even with high-speed compressors. The various methods are adopted to reduce the temperature of gas during the compression and keep it more closely to isothermal compression.

1. Water Spray It was an old method in which water was injected into the cylinder during compression of air to keep the temperature of air constant. But this method has certain disvantages, and thus became obsolete.

2. Water Jacketing It is commonly and successfully used practice for all types of reciprocating compressors. The water is circulated around the cylinder through the water jacket which helps to cool the air during compression.

3. External Fins For a small-capacity compressor, the effective cooling can be achieved by attaching fins of conducting material around the cylinder. The fins increase surface area of the cylinder for heat transfer.

4. Inter-cooler If the very high pressure ratio is required then air is compressed in stages. The intercooler is used between two stages of compression for cooling of compressed air after one stage, before entering the next stage. The water jackets are also used around the cylinder of compressor of each stage.

5. By Suitable Cylinder Proportions If the compressor has large surface to volume ratio, the greater surface area will be available for heat transfer and cooling will be more effective.

It is possible by choosing a cylinder with large bore and short piston stroke. The large cylinder head dissipates heat in a much more effective way, which contains hottest compressed air all the time.

Example 5.6 *A single-acting, single-stage reciprocating air compressor of 250 mm bore and 350 mm stroke runs at 200 rpm The suction and delivery pressures are 1 bar and 6 bar, respectively. Calculate the theoretical power required to run the compressor under each of the following conditions of compression:*

(a) isothermal,
(b) polytropic $n = 1.3$, and
(c) isentropic, $\gamma = 1.4$.

Neglect the effect of clearance and also calculate isothermal efficiency in each of the above cases.

Solution

Given A single-acting, single-stage reciprocating air compressor:

p_1 = 1 bar = 100 kPa $\quad$ p_2 = 6 bar
d = 250 mm = 0.25 m $\quad$ L = 350 mm
N = 200 rpm

and three types of compression with n = 1, 1.3 and 1.4

To find

(i) Theoretical power required to run the compressor for
  (a) Isothermal compression,
  (b) Polytropic compression,
  (c) Isentropic compression.

(ii) Isothermal efficiency in each case.

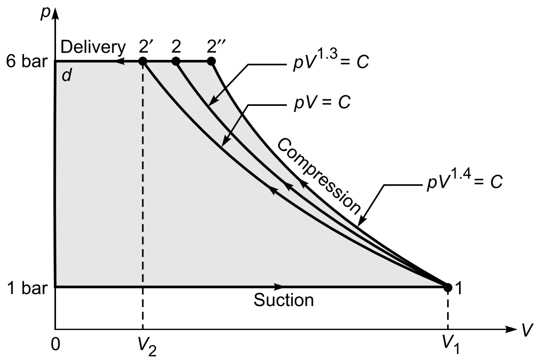


Fig. 5.7

Analysis The cylinder volume (without clearance)

$$V_1 = V_s = \left(\frac{\pi}{4}\right)d^2L = \left(\frac{\pi}{4}\right) \times (0.25 \text{ m})^2 \times (0.35 \text{ m})$$

$$= 0.0171 \text{ m}^3$$

The volume flow rate of air

$$\dot{V}_1 = \text{Cylinder volume} \times \text{No of suctions per second}$$

$$= 0.0171 \times \frac{200}{60} = 0.0572 \text{ m}^3/\text{s}$$

(i) *Theoretical power required*

(a) For isothermal compression,

$$\text{Power } \dot{W}_{iso} = p_1\dot{V}_1 \ln\left(\frac{p_2}{p_1}\right)$$

$$= 100 \times 0.0572 \times \ln\left(\frac{6}{1}\right)$$

$$= 10.261 \text{ kJ/s or } \mathbf{10.261 \text{ kW}}$$

(b) Polytropic compression,

$$\dot{W}_{Poly} = \frac{n}{n-1} p_1\dot{V}_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \left(\frac{1.3}{1.3-1}\right) \times 100 \times 0.0572 \times \left[\left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= \mathbf{12.7 \text{ kW}}$$

(c) Isentropic compression,

$$\dot{W}_{isentropic} = \frac{\gamma}{\gamma-1} p_1\dot{V}_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= \left(\frac{1.4}{1.4-1}\right) \times 100 \times 0.0572 \times \left[\left(\frac{6}{1}\right)^{\frac{1.4-1}{1.4}} - 1\right]$$

$$= \mathbf{13.38 \text{ kW}}$$

(ii) *Isothermal efficiency*

$$\eta_{iso} = \frac{\text{Isothermal work}}{\text{Actual work}} \times 100$$

(a) For isothermal compression,

$$\eta_{iso} = \mathbf{100\%}$$

(b) For polytropic compression,

$$\eta_{iso} = \frac{10.261}{12.7} \times 100 = \mathbf{80.8\%}$$

(c) For isentropic compression,

$$\eta_{iso} = \frac{10.261}{13.38} \times 100 = \mathbf{76.67\%}$$

5.7 CLEARANCE VOLUME IN A COMPRESSOR

The *clearance volume* is the space left in the cylinder when the piston reaches its topmost position, i.e., TDC. It is provided

(i) to avoid the piston striking the cylinder head, and

(ii) to accommodate the valve's actuation inside the cylinder, because suction and delivery valves are located in the clearance volume.

A compressor should have the smallest possible clearance volume, because the compressed air left in the clearance volume, first re-expands in the cylinder during suction stroke, thus reducing suction capacity.

The ratio of clearance volume to swept volume is defined as the *clearance ratio* or *percentage clearance*. The value of clearance ratio may vary from 2 to 10%.

5.7.1 Effects of Clearance Volume

1. The volume of air taken in per stroke is less than the swept volume, thus the volumetric efficiency decreases.
2. More power input is required to drive the compressor for same pressure ratio, due to increase in volume to be handled.
3. The maximum compression pressure is controlled by the clearance volume.

5.7.2 Indicated Compression Work with Clearance

The clearance volume is generally kept very small. The work done on the air in the clearance space during compression stroke is approximately equal to the work done by the air when it re-expands during suction stroke. Therefore, the work of compression is not affected by clearance space in the compressor. But the mass of air inducted is reduced, and thus the volumetric efficiency of the compressor will be less.

Figure 5.8 shows an indicator diagram for a reciprocating air compressor with clearance. After delivery of compressed air, the air remaining in the clearance volume at pressure p_2 expands, when the piston proceeds for the next suction stroke. As soon as the pressure p_1 reaches at the state 4, the induction of fresh charge starts and continues to the end of the stroke at state 1.

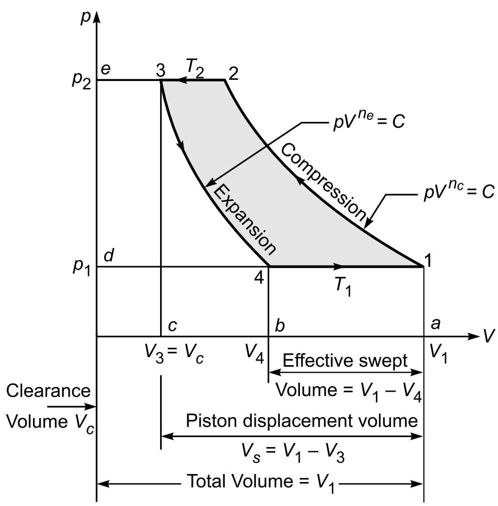


Fig. 5.8 *Indicator diagram for a reciprocating compressor showing effective swept volume, and piston displacement volume*

The indicated work done is given by the area 1–2–3–4–1 on a p–V diagram.

Indicated work

= Area 1–2–3–4–1

= Area 1–2–e–d – Area 3–e–d–4

Similar to derivation of Eq. (5.9), the compression work equivalent to area 1–2–e–d, can be obtained with index of compression n_c.

Area 1–2–e–d;

$$W_{Comp} = \frac{n_c}{n_c - 1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_c - 1}{n_c}} - 1 \right] \quad \text{...(5.22a)}$$

The work done by gas during expansion (index n_e) can be also obtained as above

Area 3–e–d–4;

$$W_{Expan} = \frac{n_e}{n_e - 1} p_4 V_4 \left[\left(\frac{p_3}{p_4} \right)^{\frac{n_e - 1}{n_e}} - 1 \right]$$

$$= \frac{n_e}{n_e - 1} p_1 V_4 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_e - 1}{n_e}} - 1 \right] \quad \text{...(5.22b)}$$

Since $p_4 = p_1$ and $p_3 = p_2$

Net work of compression;

$$W_{in} = \frac{n_c}{n_c - 1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_c - 1}{n_c}} - 1 \right]$$

$$- \frac{n_e}{n_e - 1} p_1 V_4 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_e - 1}{n_e}} - 1 \right] \quad \text{...(5.23)}$$

If indices of compression and expansion are same, i.e., $n_c = n_e = n$, then

$$W_{in} = \frac{n}{n-1} p_1 (V_1 - V_4) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad \text{...(5.24)}$$

Example 5.7 *A single cylinder, single acting air compressor has a cylinder diameter of 15.25 cm and a stroke of 22.8 cm. Air is drawn into the cylinder at a pressure of 1.013 bar and a temperature of 15.6°C. It is compressed adiabatically to 6.1 bar. Calculate the theoretical power required to drive the compressor, if it runs at 100 rpm and the mass of air compressed per minute.*

Solution

Given Single cylinder, single acting air compressor,

d = 15.25 cm L = 22.8 cm
p_1 = 1.013 bar T_1 = 15.6°C = 288.6 K
p_2 = 6.1 bar N = 100 rpm

To Find

(i) Theoretical power required by compressor, and
(ii) Mass of air compressed in kg/min.

Assumptions

(i) Specific gas constant of air as 0.287 kJ/kg · K
(ii) Adiabatic index $\gamma = 1.4$.

Analysis The volume of cylinder

$$\dot{V} = \frac{\pi}{4} d^2 L \times \frac{N}{60}$$

$$= \frac{\pi}{4} \times (0.1525)^2 \times (0.228) \times \frac{100}{60}$$

$$= 6.94 \times 10^{-3} \text{ m}^3/\text{s}$$

(i) Adiabatic power input to compressor

$$IP = \frac{\gamma}{\gamma - 1} p_1 \dot{V}_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$= \frac{1.4}{1.4-1} \times (101.3 \times 6.94 \times 10^{-3}) \times \left[\left(\frac{6.1}{1.013} \right)^{\frac{1.4-1}{1.4}} - 1 \right]$$

$$= \mathbf{1.65\ kW}$$

(ii) Mass of air compressed per minute

$$\dot{m} = \frac{p_1 \dot{V}_1}{RT_1} = \frac{101.3 \times (6.94 \times 10^{-3} \times 60)}{0.287 \times 288.6}$$

$$= \mathbf{0.51\ kg/min}$$

Example 5.8 *An ideal single-stage, single-acting reciprocating air compressor has a displacement volume of 14 litre and a clearance volume of 0.7 litre. It receives the air at a pressure of 1 bar and delivers it at a pressure of 7 bar. The compression is polytropic with an index of 1.3 and re-expansion is isentropic with an index of 1.4. Calculate the net indicated work of a cycle.*

Solution

Given A single-stage, single-acting reciprocating air compressor.

p_1 = 1 bar = 100 kPa p_2 = 7 bar
V_s = 14 litre V_c = 0.7 litre
n_c = 1.3 n_e = 1.4

To find Indicated work input per cycle.

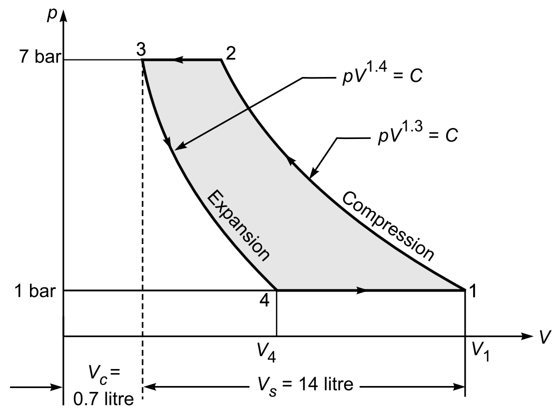


Fig. 5.9

Analysis The total volume of cylinder;

$$V_1 = V_s + V_c$$
$$= 14 + 0.7 = 14.7 \text{ litre} \quad \text{or} \quad 0.0147 \text{ m}^3$$

The volume V_4 after re-expansion of compressed air in clearance space

$$V_4 = V_3 \left(\frac{p_2}{p_1} \right)^{\frac{1}{n_e}} = 0.7 \times \left(\frac{7}{1} \right)^{\frac{1}{1.4}}$$

$$= 2.81 \text{ litre} \quad \text{or} = 0.00281 \text{ m}^3$$

Indicated work input per cycle, Eq. (5.23)

$$W_{in} = \frac{n_c}{n_c - 1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_c-1}{n_c}} - 1 \right] - \frac{n_e}{1 - n_e} p_1 V_4 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_e-1}{n_e}} - 1 \right]$$

$$= \frac{1.3}{1.3-1} \times 100 \times 0.0147 \times \left[\left(\frac{7}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right] - \frac{1.4}{1.4-1} \times 100 \times 0.00281 \times \left[\left(\frac{7}{1} \right)^{\frac{1.4-1}{1.4}} - 1 \right]$$

$$= 3.61 - 0.731 = \mathbf{2.88\ kJ/cycle}$$

Example 5.9 *A single-stage, double-acting reciprocating air compressor takes in 14 m³ of air per minute measured at 1.013 bar and 15°C. The delivery pressure is 7 bar and the compressor speed is 300 rpm. The compressor has a clearance volume of 5% of swept volume with a compression and re-expansion index of n = 1.3. Calculate the swept volume of the cylinder, the delivery temperature and the indicated power.*

Solution

Given A single-stage, double-acting reciprocating air compressor, with

$\dot{V}_1 = 14\text{ m}^3/\text{min}$ $\quad T_1 = 15°\text{C} = 288\text{ K}$

$N = 300\text{ rpm}$ $\quad n = 1.3$

$p_1 = 1.03\text{ bar} = 101.3\text{ kPa}$ $\quad p_2 = 7\text{ bar}$

$V_c = 0.05\,V_s$

To find

(i) The swept volume of cylinder,

(ii) The delivery temperature, and

(iii) Indicated power.

Assumptions

(i) No throttling effect on valve opening and closing.

(ii) Effect of piston rod on underside of cylinder is negligible.

(iii) Air as an ideal gas with specific gas constant R = 0.287 kJ/kg·K.

Analysis The p–V diagram for given data of compressor is shown in Fig. 5.10.

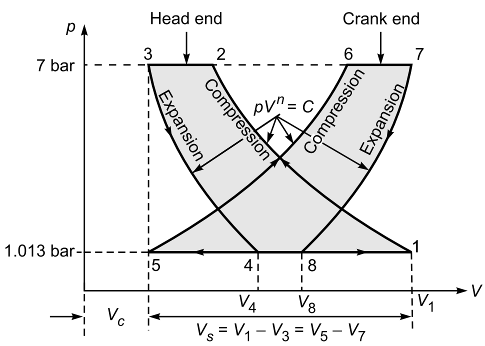


Fig. 5.10 *p–V diagram for double-acting reciprocating air compressor*

(i) *Swept volume of cylinder*

The total volume of cylinder,

$$V_1 = V_s + V_c = V_s + 0.05\,V_s = 1.05\,V_s$$

The volume inducted per cycle

$$V_1 - V_4 = \frac{\text{Volume induction per minute}}{\text{No. of suction per revolution} \times \text{No. of revolution per minute}}$$

$$= \frac{14\text{ m}^3/\text{min}}{2\text{ suction per rev.} \times 300\text{ rev./min}}$$

$$= 0.0233\text{ m}^3/\text{cycle}$$

The volume V_4 after re-expansion of compressed air.

$$V_4 = V_3\left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} = V_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= 0.05\,V_s \times \left(\frac{7}{1.013}\right)^{\frac{1}{1.3}}$$

$$= 0.221\,V_s$$

Then $V_1 - V_4$

$$= 1.05\,V_s - 0.221\,V_s = 0.829\,V_s$$

or $0.829\,V_s = 0.0233\text{ m}^3/\text{cycle}$

∴ swept volume,

$$V_s = \frac{0.0233}{0.829} = \mathbf{0.0281\ m^3/cycle}$$

The swept volume of the cylinder is 0.0281 m³.

(ii) *The delivery temperature of air*

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

$$= 288 \times \left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}}$$

$$= 450\text{ K} = \mathbf{177°C}$$

(iii) *Indicated power*

$$IP = \frac{n}{n-1}\,p_1\dot{V}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times (101.3\text{ kPa})$$

$$\times \left(\frac{14}{60}\text{ kg/s}\right)\left[\left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= \mathbf{57.58\ kW}$$

5.8 ACTUAL INDICATOR DIAGRAM

The actual indicator diagram on a p–V plane for a single-stage reciprocating air compressor is shown in Fig. 5.11. It is similar to theoretical one (Fig. 5.8) except for induction and delivery processes. The variation during these processes is due to valve action effects.

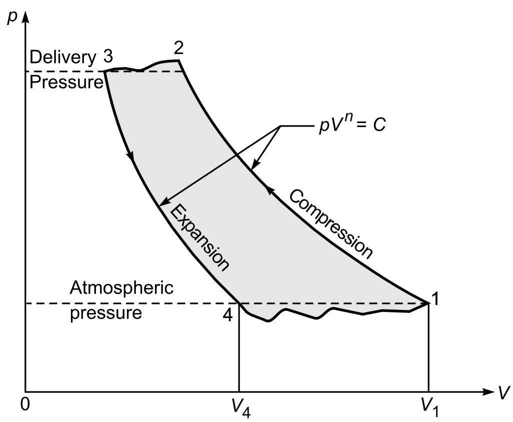


Fig. 5.11 *Actual indicator diagram for reciprocating compressor*

There must be some pressure difference across the valves to operate them. During the suction process 4–1, the pressure drops in the cylinder until the inlet valve is forced by atmospheric air to open. During the suction stroke, the piston creates *vacuum* in the cylinder. Thus pressure reduces and atmospheric air enters the cylinder.

Similarly, during delivery process 2–3, some more pressure is required to open the delivery valve and to displace the compressed air through narrow valve passage. Thus, *gas throttling* takes place during delivery, reducing the pressure gradually to the state 3.

The waviness of lines during these processes is due to valve bounce and wire drawing effect through the valves.

5.9 VOLUMETRIC EFFICIENCY

Actual volume sucked into the cylinder during the suction stroke is always less than the swept volume. It is due to

(i) resistance offered by inlet valve to incoming air,
(ii) temperature of incoming air, and
(iii) back pressure of residual gas left in the clearance volume.

The *volumetric efficiency,* η_{vol} of the air compressor is defined as the ratio of actual volume of air sucked into the compressor, measured at atmospheric pressure and temperature to the piston displacement volume.

In terms of mass ratio, the volumetric efficiency is defined as the ratio of actual mass of air sucked per stroke to the mass of air corresponding to piston displacement volume at atmospheric conditions.

$$\eta_{vol} = \frac{\text{Actual mass sucked}}{\text{Mass corresponding to swept volume at atmospheric pressure and temperature}} \quad ...(5.25)$$

$$= \frac{\text{Effective swept volume}}{\text{Piston displacement volume}} \quad ...(5.26)$$

Figure 5.8 has shown an indicator diagram for a reciprocating air compressor showing effective swept volume, and piston displacement volume. The volumetric efficiency can be expressed in terms of effective volume and piston displacement volume as;

$$\eta_{vol} = \frac{V_1 - V_4}{V_1 - V_c} = \frac{V_s + V_c - V_4}{V_s + V_c - V_c} = \frac{V_s + V_c - V_4}{V_s}$$

$$= 1 + \frac{V_c}{V_s} - \frac{V_4}{V_s} \times \frac{V_c}{V_c} \quad ...(5.27)$$

Introducing $c = \frac{V_c}{V_s}$ as clearance ratio, and using $V_c = V_3$, then

$$\eta_{vol} = 1 + c - c\left(\frac{V_4}{V_3}\right) \quad ...(5.28)$$

For expansion of gas in clearance volume

$$\frac{V_4}{V_3} = \left(\frac{p_3}{p_4}\right)^{\frac{1}{n_e}} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}$$

Then $$\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}} \quad ...(5.29)$$

If index of expansion and index of compression are same, then

$$\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n_c}} = \frac{V_1}{V_2}$$

and $$\eta_{vol} = 1 + c - c\left(\frac{V_1}{V_2}\right) \quad ...(5.30)$$

The volumetric efficiency decreases with pressure ratio $\left(\frac{p_2}{p_1}\right)$ in the compressor, its variation is shown in Fig. 5.12. The factors which lower volumetric efficiency are the following:

1. Too Large Clearance Volume Re-expansion of residual compressed air in the clearance space will reduce effective suction stroke $(V_1 - V_4)$ and therefore, the mass of fresh air entering into the cylinder reduces and volumetric efficiency decreases.

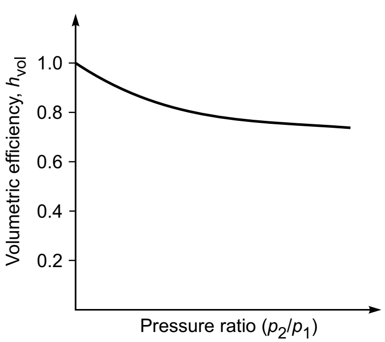


Fig. 5.12 *Effect of pressure ratio on volumetric efficiency of a reciprocating compressor*

2. Obstruction at Intet Valve Obstruction due to narrow valve passage causes throttling of air in the cylinder. Throttling reduces the pressure in the cylinder during intake stroke and discharge pressure during delivery stroke, and thus the pressure ratio in the cylinder decreases. It leads to reduced FAD and volumetric efficiency.

3. High speed of Compressor With high speed of the compressor, the pressure drop across the inlet valve and delivery valve increases. Further, the temperature of compressed air increases due to less available time for cooling. Both these factors reduce volumetric efficiency of the compressor.

4. Heated Cylinder Walls The heated cylinder walls increase the temperature of the intake air. Thus, the specific volume of air increases which will reduce FAD and volumetric efficiency of the compressor.

5. Leakage Past the Piston Leakages across the piston will reduce the vacuum during suction and mass of compressed air above the piston. Both these effects will increase compression work input and decrease in volumetric efficiency.

5.10 FREE AIR DELIVERY (FAD)

The volume of compressed air corresponding to atmospheric conditions is known as *free air delivery* (FAD). FAD is the volume of compressed air measured in m^3/min, reduced to atmospheric pressure and temperature.

The free air delivered volume is less than the compressor displacement volume due to the following reasons:

1. *Obstruction at inlet valve.* It offers the resistance to air flow through the narrow passage of valve.
2. *Re-expansion of high pressure air in clearance volume.* It reduces effective suction stroke.
3. *Presence of hot cylinder walls of compressor.* Air gets heated as it enters the cylinder. Thus, it expands and reducing the mass of air sucked into the cylinder.

In the actual indicator diagram as shown in Fig. 5.11, the air is sucked at a pressure and temperature which are lower than that of free (atmospheric) air. Using the property relation for an ideal gas as

$$\frac{p_f V_f}{T_f} = \frac{p_1 (V_1 - V_4)}{T_1}$$

Then free air delivery (FAD)

$$V_f = \frac{p_1 T_f}{p_f T_1}(V_1 - V_4) \qquad ...(5.31)$$

where the suffix f denotes free (ambient) conditions while the suffix 1 indicates actual suction conditions. Then volumetric efficiency with respect to free air delivery;

$$\eta_{vol,\ overall} = \frac{V_f}{V_1 - V_c} = \frac{p_1 T_f}{p_f T_1}\left(\frac{V_1 - V_4)}{V_1 - V_c)}\right)$$

$$= \frac{p_1 T_f}{p_f T_1}\left[1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}\right] \qquad ...(5.32)$$

Example 5.10 *A single-stage, single-acting reciprocating air compressor receives air at 1.013 bar, 27°C and delivers it at 9.5 bar. The compressor has a bore = 250 mm, and stroke = 300 mm and it runs at 200 rpm. The mass-flow rate of air is 200 kg/h. Calculate the volumetric efficiency of the compressor.*

Solution

Given A single-stage, single-acting reciprocating air compressor

p_1 = 1.03 bar = 101.3 kPa T_1 = 27°C = 300 K
N = 200 rpm p_2 = 9.5 bar
d = 250 mm = 0.25 m L = 300 mm = 0.3 m
$\dot{m}_{act}$ = 200 kg/h

To find The volumetric efficiency of the compressor.

Assumptions

(i) Neglecting clearance volume.
(ii) Specific gas constant of air, R = 0.287 kJ/kg·K.

Analysis The volume swept per cycle

$$V_1 = \left(\frac{\pi}{4}\right)d^2 L$$

$$= \left(\frac{\pi}{4}\right)\times(0.25\text{ m})^2\times(0.3\text{ m})$$

$$= 0.0147\text{ m}^3$$

The mass of air inducted per cycle

$$m_a = \frac{p_1 V_1}{RT_1} = \frac{(101.3\text{ kPa})\times(0.0147\text{ m}^3)}{(0.287\text{ kJ/kg}\cdot\text{K})\times(300\text{ K})}$$

$$= 0.0173\text{ kg/cycle}$$

The mass-flow rate per hour

$$\dot{m}_a = \text{mass per cycle}\times\text{No. of suctions/revolution}\times\text{No. of revolutions/h}$$

$$= (0.0173\text{ kg/cycle})\times(1\text{ suction/rev})\times(200\times 60\text{ rev/h})$$

$$= 207.6\text{ kg/h}$$

The mass of air actually sucked, $\dot{m}_{act}$ = 200 kg/h

Thus
$$\eta_{vol} = \frac{\text{Actual mass sucked}}{\text{Mass corresponds to swept volume at atmospheric pressure and temperature}}$$

$$= \frac{200\text{ kg}}{207.6\text{ kg}} = 0.963 \quad \text{or} \quad \mathbf{96.3\%}$$

Example 5.11 *A single-stage, double-acting reciprocating air compressor has a FAD of 14 m³/min measured at 1.013 bar and 27°C. The pressure and temperature of the cylinder during induction are 0.95 bar and 45°C. The delivery pressure is 7 bar and the index of compression and expansion is 1.3. Calculate the indicated power required and volumetric efficiency. The clearance volume is 5% of the swept volume.*

Solution

Given A single-stage, double-acting reciprocating air compressor

p_f = 1.03 bar = 101.3 kPa
T_f = 27°C = 300 K
p_2 = 7 bar
FAD, V_f = 14 m³/min
p_1 = 0.95 bar = 95 kPa
T_1 = 45°C = 318 K
$n_c = n_e = n$ = 1.3
$V_c = 0.05\ V_s$

To find

(i) Indicated power, and
(ii) Volumetric efficiency.

Assumptions

(i) The compression and expansion are reversible.
(ii) Air as an ideal gas with R = 0.287 kJ/kg·K.

Analysis The p–V diagram for given data is shown in Fig. 5.13.

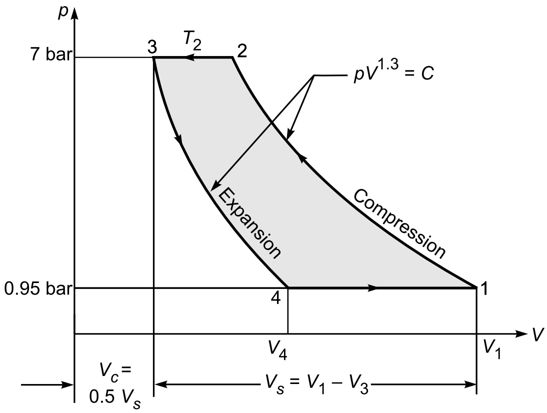


Fig. 5.13

The mass-flow rate corresponding to FAD at atmospheric conditions, p_f and T_f;

$$\dot{m}_a = \frac{p_f V_f}{R T_f}$$

$$= \frac{101.3 \times 14}{0.287 \times 300} = 16.47 \text{ kg/min}$$

The temperature T_2 after compression,

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

$$= (318 \text{ K}) \times \left(\frac{7}{0.95}\right)^{\frac{1.3-1}{1.3}} = 504.18 \text{ K}$$

(i) The indicated power,

$$IP = \frac{n}{n-1} \dot{m}_a R (T_2 - T_1)$$

$$= \frac{1.3}{1.3-1} \times \left(\frac{16.47}{60} \text{kg/s}\right) \times (0.287 \text{ kJ/kg} \cdot \text{K}) \times (504.18 \text{ K} - 318 \text{ K})$$

$= \mathbf{63.56\ kW}$

(ii) The volumetric efficiency can be obtained by using Eq. (5.32)

$$\eta_{vol,\ overall} = \frac{p_1 T_f}{p_f T_1}\left[1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]$$

$$= \frac{0.95 \times 300}{1.013 \times 318} \times \left[1 + 0.05 - 0.05 \times \left(\frac{7}{0.95}\right)^{\frac{1}{1.3}}\right]$$

$= 0.723$ or **72.3%**

Example 5.12 *A single-stage, single-acting reciprocating air compressor has a bore of 20 cm and a stroke of 30 cm. The compressor runs at 600 rpm. The clearance volume is 4% of the swept volume and index of expansion and compression is 1.3. The suction conditions are at 0.97 bar and 27°C and delivery pressure is 5.6 bar. The atmospheric conditions are at 1.01 bar and 17°C. Determine,*

(a) The free air delivered in m^3*/min.*

(b) The volumetric efficiency referred to the free air conditions.

(c) The indicated power.

Solution

Given Single-stage, single-acting reciprocating air compressor

d = 20 cm = 0.2 m	L = 30 cm = 0.3 m
N = 600 rpm	n = 1.3
p_1 = 0.97 bar = 97 kPa	T_1 = 27°C = 300 K
p_f = 1.01 bar = 101 kPa	T_f = 17°C = 290 K
V_c = 0.04 V_s	p_2 = 5.6 bar

To find

(i) Free air delivery, m^3/min,

(ii) Volumetric efficiency at ambient conditions; $\eta_{vol,\ overall}$, and

(iii) Indicated power.

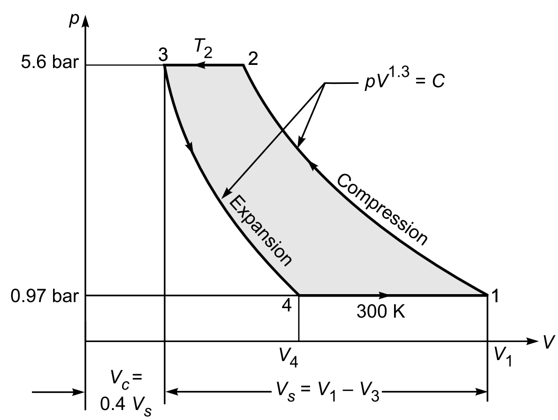


Fig. 5.14

Analysis

(i) *Free air delivery* (FAD)

The swept volume;

$$V_s = V_1 - V_3 = \left(\frac{\pi}{4}\right) d^2 L$$

$$= \left(\frac{\pi}{4}\right) \times (0.2\ \text{m})^2 \times (0.3\ \text{m})$$

$$= 0.009425\ \text{m}^3$$

Clearance volume

$$V_c = 0.04\ V_s = 0.04 \times 0.009425$$

$$= 3.77 \times 10^{-4}\ \text{m}^3$$

Total volume

$$V_1 = V_s + V_c$$

$$= 0.009425 + 3.77 \times 10^{-4}$$

$$= 0.0098\ \text{m}^3$$

The volume V_4, after re-expansion of compressed air in clearance space,

$$\frac{V_4}{V_3} = \left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$V_4 = 3.77 \times 10^{-4} \times \left(\frac{5.6}{0.97}\right)^{\frac{1}{1.3}}$$

$$= 0.00145\ \text{m}^3$$

Effective swept volume,

$$V_1 - V_4 = 0.0098 - 0.00145 = 0.00835\ \text{m}^3$$

The free air delivery per cycle,

$$V_f = \frac{p_1 T_f}{p_f T_1}(V_1 - V_4)$$

$$= \frac{0.97 \times 290}{1.01 \times 300} \times 0.00835$$

$$= 0.00775\ \text{m}^3/\text{cycle}$$

Free air delivered per minute

$$= V_f \times \text{No. of cycle per minute}$$

$$= 0.00775 \times 600 = \mathbf{4.65\ m^3/min}$$

(ii) The *volumetric efficiency referred to free air conditions*

$$\eta_{vol,\ overall} = \frac{V_f}{V_s} = \frac{0.00775}{0.009425}$$

$$= 0.822 \quad \text{or} \quad \mathbf{82.2\%}$$

(iii) *Indicated power*

Indicated work input using Eq. (5.24)

$$W_{in} = \frac{n}{n-1} p_1 (V_1 - V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times 97 \times 0.00835 \times \left[\left(\frac{5.6}{0.97}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= 1.75\ \text{kJ/cycle}$$

Since N cycles take place within a minute, the indicated power

$$IP = W_{in}\left(\frac{N}{60}\right)$$

$$= (1.75\ \text{kJ/cycle}) \times \left(\frac{600}{60}\ \text{cycle/s}\right)$$

$$= \mathbf{17.5\ kW}$$

Example 5.13 *A single-stage, double-acting air compressor delivers air at 7 bar. The pressure and temperature at the end of the suction stroke are 1 bar and 27°C. It delivers 2 m³ of free air per minute when the compressor is running at 300 rpm. The clearance volume is 5% of the stroke volume. The pressure and temperature of the ambient air are 1.03 bar and 20°C. The index of compression is 1.3, and index of expansion is 1.35. Calculate*

(a) Volumetric efficiency of the compressor,

(b) Indicated power of the compressor, and

(c) Brake Power, if mechanical efficiency is 80%.

Solution

Given A single-stage, double-acting reciprocating air compressor

p_1 = 1 bar = 100 kPa	p_2 = 7 bar
N = 300 rpm	T_1 = 27°C = 300 K
$\dot{V}_f$ = 2 m³/min	p_f = 1.03 bar = 103 kPa
V_c = 0.05 V_s	T_f = 20°C = 293 K
η_{mech} = 0.8	n_c = 1.3, and
n_e = 1.35	

To find

(i) Volumetric efficiency,

(ii) Indicated power, and

(iii) Brake Power.

Analysis

(i) *Volumetric efficiency*

Clearance volume; $V_c = V_3 = 0.05\ V_s$

The volume V_4 after re-expansion of compressed air in clearance space

$$V_4 = V_3 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}$$

$$= 0.05\, V_s \times \left(\frac{7}{1}\right)^{\frac{1}{1.35}} = 0.2113\, V_s$$

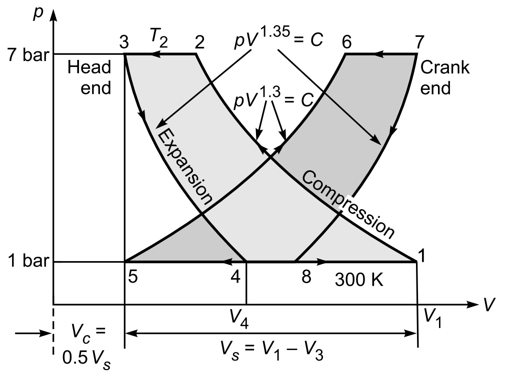


Fig. 5.15 *p-V dagram for double-acting compressor*

The total volume of cylinder;

$$V_1 = V_s + V_c = 1.05\, V_s$$

Effective swept volume

$$V_1 - V_4 = 1.05\, V_s - 0.2113\, V_s = 0.8386\, V_s$$

The volumetric efficiency

$$\eta_{vol} = \frac{V_1 - V_4}{V_s} = \frac{0.8386 V_s}{V_s}$$

$$= 0.8386 \quad \text{or} \quad \mathbf{83.86\%}$$

Free air delivered per cycle is given as

$$V_f = \frac{p_1 T_f}{p_f T_1}\ (V_1 - V_4)$$

Thus volume of air inducted per min at suction condition

$$\dot{V}_1 - \dot{V}_4 = \dot{V}_f \frac{p_f T_1}{p_1 T_f}$$

$$= (2\ \text{m}^3/\text{min}) \times \frac{(1.03\ \text{bar}) \times (300\ \text{K})}{(1\ \text{bar}) \times (293\ \text{K})}$$

$$= 2.109\ \text{m}^3/\text{min}$$

It is the volume sucked per minute, which can also be expressed as

$$\dot{V}_1 - \dot{V}_4 = \eta_{vol} V_s \times N \times \text{Number of suction per revolution}$$

$$2.109\ \text{m}^3/\text{min} = 0.8386\, V_s \times (300\ \text{rpm}) \times 2$$

or $\quad V_s = 0.00419\ \text{m}^3$

Now $\quad V_1 = 1.05 \times 0.00419 = 0.0044\ \text{m}^3$

and $\quad V_4 = 0.2113 \times 0.00419 = 8.856 \times 10^{-4}\ \text{m}^3$

(ii) *Indicated power*

Indicated work input per cycle, Eq. (5.23)

$$W_{in} = \frac{n_c}{n_c - 1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_c - 1}{n_c}} - 1\right] - \frac{n_e}{1 - n_e} p_1 V_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_e - 1}{n_e}} - 1\right]$$

$$= \frac{1.3}{1.3 - 1} \times 100 \times 0.0044 \times \left[\left(\frac{7}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right] - \frac{1.35}{1.35 - 1} \times 100 \times 8.856 \times 10^{-4} \times \left[\left(\frac{7}{1}\right)^{\frac{1.35-1}{1.35}} - 1\right]$$

$$= 1.080 - 0.224 = 0.856\ \text{kJ/cycle}$$

The indicated power input to compressor

$$IP = W_{in}\left(\frac{N}{60}\right) \times 2\ \text{(for double acting)}$$

$$= (0.856\ \text{kJ/cycle}) \times \left(\frac{300}{60}\ \text{cycle/s}\right) \times 2$$

$$= \mathbf{8.558\ kW}$$

(iii) The brake (shaft) power

$$\text{Brake power} = \frac{\text{Indicated power}}{\eta_{mech}} = \frac{8.558}{0.8}$$

$$= \mathbf{10.7\ kW}$$

Example 5.14 *A single-stage, double-acting reciprocating air compressor works between 1 bar and 10 bar. The compression follows the law $pV^{1.35}$ = constant. The piston speed is 200 m/min and the compressor speed is 120 rpm. The compressor consumes an indicated power of 62.5 kW with volumetric efficiency of 90%. Calculate*

(a) diameter and stroke of the cylinder

(b) Clearance volume as percentage of stroke volume

Solution

Given A single-stage, single-acting reciprocating air compressor

$p_1 = 1 \text{ bar} = 100 \text{ kPa}$ $\quad p_2 = 10 \text{ bar}$

$N = 120 \text{ rpm}$ $\quad \mathcal{V}_{piston} = 200 \text{ m/min}$

$\eta_{vol} = 0.9$ $\quad IP = 62.5 \text{ kW}$

and $pV^{1.35} = C$

To find

(i) Cylinder bore and piston stroke, and
(ii) Clearance ratio.

Analysis

(i) *Cylinder bore* and *piston stroke*

The indicated work input per cycle for double acting compressor can be obtained as

$$W_{in} = IP\left(\frac{60}{2N} \text{ s/cycle}\right)$$

$$= (62.5 \text{ kJ/s}) \times \left(\frac{60}{2 \times 120} \text{ s/cycle}\right)$$

$$= 15.625 \text{ kJ/cycle}$$

Further,

$$W_{in} = \frac{n}{n-1} p_1 (V_1 - V_4) \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$15.625 = \frac{1.35}{1.35 - 1} \times (100 \text{ kPa}) \times (V_1 - V_4) \times \left[\left(\frac{10}{1}\right)^{\frac{1.35-1}{1.35}} - 1\right]$$

or $(V_1 - V_4) = 0.0496 \text{ m}^3$

The volumetric efficiency is given as

$$\eta_{vol} = \frac{V_1 - V_4}{V_s}$$

Stroke volume

or $$V_s = \frac{0.0496 \text{ m}^3}{0.9} = 0.0551 \text{ m}^3$$

The piston speed is given by

$$\mathcal{V}_{piston} = 2\,LN$$

Stroke length;

$$L = \frac{200}{2 \times 120} = 0.833 \text{ m or } \mathbf{833 \text{ mm}}$$

Further, the stroke volume can be expressed as

$$V_s = \left(\frac{\pi}{4}\right) d^2 L$$

Bore $$d = \sqrt{\frac{0.0551}{(\pi/4) \times 0.833}}$$

$$= 0.290 \text{ m} \quad \text{or} \quad \mathbf{290 \text{ mm}}$$

(ii) *Clearance ratio*

The volumetric efficiency is given by

$$\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

or $$0.9 = 1 + c - c \times \left(\frac{10}{1}\right)^{\frac{1}{1.35}}$$

Clearance ratio;

$$c = 0.222 \quad \text{or} \quad \mathbf{2.22\%}$$

Example 5.15 *A single-stage, single-acting reciprocating air compressor delivers 0.6 kg/min of air at 6 bar. The temperature and pressure at the suction stroke are 30°C and 1 bar, respectively. The bore and stroke are 100 mm and 150 mm respectively. The clearance volume is 3% of the swept volume and index of expansion and compression is 1.3. Determine,*

(a) the volumetric efficiency of compressor,
(b) the power required, if mechanical efficiency is 85%, and
(c) speed of the compressor.

Solution

Given A single-stage, single-acting reciprocating air compressor

$p_1 = 1 \text{ bar}$ $\quad p_2 = 6 \text{ bar}$

$\dot{m}_a = 0.6 \text{ kg/min}$ $\quad T_1 = 30°\text{C} = 303 \text{ K}$

$L = 150 \text{ cm}$ $\quad d = 100 \text{ mm}$

$c = 0.03$ $\quad n = 1.3$

$\eta_{mech} = 0.85$

To find

(i) Volumetric efficiency,
(ii) Power required by compressor, and
(iii) Speed of the compressor.

Assumptions

(i) Suction takes place at free air conditions.

(ii) The specific gas constant for air R = 0.287 kJ/kg·K.

Analysis

(i) The *volumetric efficiency* is given by

$$\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

or $$= 1 + 0.03 - 0.03 \times \left(\frac{6}{1}\right)^{\frac{1}{1.3}}$$

$$= 0.910 \quad \text{or} \quad \mathbf{91.0\%}$$

(ii) *Power* required by compressor

Indicated power input

$$IP = \frac{n}{n-1}\,\dot{m}_a R\,T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times \left(\frac{0.6\text{ kg}}{60\text{ s}}\right) \times 0.287$$

$$\times 303 \times \left[\left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= 1.929\text{ kW}$$

Brake power required;

$$BP = \frac{IP}{\eta_{mech}} = \frac{1.929}{0.85} = \mathbf{2.27\ kW}$$

(iii) *Speed of the compressor*

The volume-flow rate of air at suction conditions

$$\dot{V}_1 - \dot{V}_4 = \frac{\dot{m}_a R T_1}{p_1}$$

$$= \frac{0.6 \times 0.287 \times 303}{100}$$

$$= 0.5217\text{ m}^3\text{/min}$$

Piston-displacement volume rate

$$\dot{V}_s = \frac{\dot{V}_1 - \dot{V}_4}{\eta_{vol}} = \frac{0.5217\text{ m}^3\text{/min}}{0.91}$$

$$= 0.5733\text{ m}^3\text{/min}$$

Further, $$\dot{V}_s = \left(\frac{\pi}{4}\right) d^2 L N$$

$\therefore$ 0.5733

$$= \left(\frac{\pi}{4}\right) \times (0.1\text{ m})^2 \times (0.15\text{ m}) \times N$$

Speed of compressor;

$$N = \mathbf{487\ rpm}$$

Example 5.16 *A single-stage, double-acting reciprocating air compressor is driven by an electric motor consuming 40 kW, compresses 5.5 m³/min air according to the law $pV^{1.3}$ = constant. It receives atmospheric air at 20°C and 745 mm of Hg barometer and delivers at a gauge pressure of 700 kPa. Calculate isothermal, volumetric, mechanical and overall efficiencies for the following data from an indiactor diagram for head and tail end.*

Length of the indicator diagram = 6.75 cm
Area of the head end = 7.6 cm²
Area of the tail end = 7.8 cm²
Spring scale = 200 kPa/cm

The diameter of the piston and piston rod are 25 and 2.5 cm, respectively. The stroke length is 30 cm. The compress or runs at 300 rpm.

Solution

Given A single-stage, double-acting reciprocating air compressor.

p_1 = 745 mm of Hg	p_{g_2} = 700 kPa
N = 300 rpm	$\dot{V}_f$ = 5.5 m³/min
BP = 40 kW	L = 30 cm
d = 25 cm	d_{rod} = 2.5 cm, n = 1.3
$T_1 = T_f$ = 20°C = 293 K	n = 1.3 cm

For indicator diagram,

a_1 = 7.6 cm²	a_2 = 7.8 cm²
l = 6.75 cm	k = 200 kPa/cm

To find

(i) Overall efficiency,
(ii) Isothermal efficiency,
(iii) Mechanical efficiency, and
(iv) Volumetric efficiency.

Assumptions

(i) Suction takes place at free air conditions.
(ii) Negligible clearance volume on head and tail side of the cylinder.
(iii) Specific gas constant for air R = 0.287 kJ/kg·K.

Analysis The suction pressure corresponds to 745 mm of Hg;

$$p_1 = \frac{745}{760} \times (101.3 \text{ kPa}) = 99.3 \text{ kPa}$$

Absolute delivery pressure,

$$p_2 = p_1 + p_{g_2} = 99.3 + 700 = 799.3 \text{ kPa}$$

Mass of air inducted per second

$$\dot{m}_a = \frac{p_1 \dot{V}_f}{RT_f} = \frac{(99.3 \text{ kPa}) \times (5.5 \text{ m}^3/\text{min})}{(0.287 \text{ kJ/kg}\cdot\text{K}) \times (293 \text{ K})}$$

$$= 6.49 \text{ kg/min or } 0.1082 \text{ kg/s}$$

Temperature after polytropic compression

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

$$= (293 \text{ K}) \times \left(\frac{793.3}{93.3}\right)^{\frac{1.3-1}{1.3}}$$

$$= 474.12 \text{ K}$$

Actual power input to air in the compressor;

$$\dot{W}_{in} = \frac{n}{n-1} \dot{m}_a R(T_2 - T_1)$$

$$= \frac{1.3}{1.3-1} \times 0.1082 \times 0.287 \times (474.12 - 293)$$

$$= 24.37 \text{ kW}$$

(i) *Overall efficiency*

$$\eta_{Overall} = \frac{\dot{W}_{in}}{BP} = \frac{24.37 \text{ kW}}{40 \text{ kW}}$$

$= 0.6093$ or **60.93%**

Isothermal power input to the compressor;

$$\dot{W}_{iso} = p_1 \dot{V}_f \ln\left(\frac{p_2}{p_1}\right)$$

$$= (99.3 \text{ kPa}) \times \left(\frac{5.5 \text{ m}^3}{60 \text{ s}}\right) \times \ln\left(\frac{793.3}{93.3}\right)$$

$$= 19.48 \text{ kW}$$

(ii) *Isothermal efficiency*

$$\eta_{iso} = \frac{\dot{W}_{iso}}{BP} = \frac{19.48}{40} = 0.487 \quad \text{or} \quad \mathbf{48.7\%}$$

For head end of the cylinder

Cross-sectional area,

$$A_1 = \left(\frac{\pi}{4}\right) d^2 = \left(\frac{\pi}{4}\right) \times (0.25 \text{ m})^2$$

$$= 0.049 \text{ m}^2$$

Indicated mean effective pressure;

$$p_{m_1} = \frac{a_1}{l} k = \frac{7.6 \text{ cm}^2}{6.75 \text{ cm}} \times (200 \text{ kPa/cm})$$

$$= 225.18 \text{ kPa}$$

For tail end of the cylinder

Cross-sectional area,

$$A_2 = \left(\frac{\pi}{4}\right)\left[d^2 - d_p^2\right]$$

$$= \left(\frac{\pi}{4}\right) \times \left[(0.25 \text{ m})^2 - (0.025 \text{ m})^2\right]$$

$$= 0.0486 \text{ m}^2$$

Indicated mean effective pressure;

$$p_{m_2} = \frac{a_2}{l} k = \frac{7.8 \text{ cm}^2}{6.75 \text{ cm}} \times (200 \text{ kPa/cm})$$

$$= 231.11 \text{ kPa}$$

Total indicated power input to head and tail sides of the compressor

$$IP = \frac{p_{m_1} L A_1 N}{60} + \frac{p_{m_2} L A_2 N}{60}$$

$$= \frac{225.18 \times 0.3 \times 0.049 \times 300}{60} + \frac{231.11 \times 0.3 \times 0.0486 \times 300}{60}$$

or $IP = 4.965 + 5.054 = 33.4$ kW

(iii) *Mechanical efficiency*

$$\eta_{mech} = \frac{IP}{BP} = \frac{33.4 \text{ kW}}{40 \text{ kW}}$$

$= 0.835$ or **83.5%**

Total displacement volume per minute from head and tail ends of the cylinder;

$$\dot{V}_{total} = (A_1 + A_2) LN$$

$$= (0.049 + 0.0486) \times 0.3 \times 300$$

$$= 8.78 \text{ m}^3/\text{min}$$

(iv) *Volumetric efficiency*

$$\eta_{vol} = \frac{\dot{V}_f}{\dot{V}_{total}} = \frac{5.5 \text{ m}^3/\text{min}}{8.78 \text{ m}^3/\text{min}}$$

$= 0.626$ or **62.6%**

Example 5.17 *During the overhauling of an old compressor, a distance piece of 9 mm thickness is inserted accidentally between the cylinder head and cylinder. Before overhaul, the clearance volume was 3 per cent of the swept volume. The compressor receives* *atmospheric air at 1 bar and it is designed to deliver air at a gauge pressure of 7 bar with a stroke of 75 cm. If the compression and re-expansion follow the law $pV^{1.3}$ = constant, determine the percentage change in*

(a) volume of free air delivered, and

(b) power necessary to drive the compressor.

Solution

Given A reciprocating air compressor before and after overhaul

p_1 = 1 bar	p_{g_2} = 7 bar
t = 9 mm	$V_c = 0.03\,V_s$
L = 75 cm	n = 1.3

To find

(i) Percentage change in volume of free air delivered, and

(ii) Percentage change in power necessary to drive the compressor.

Analysis The absolute pressure of delivered air

$$p_2 = p_1 + p_{g_2} = 1 \text{ bar} + 7 \text{ bar} = 8 \text{ bar}$$

The clearance space

Before overhaul,

$$L_{c_1} = 0.03L = 0.03 \times 75 = 2.25 \text{ cm}$$

After overhaul,

$$L_{c_2} = L_{c_1} + t = 2.25 \text{ cm} + 0.9 \text{ cm} = 3.15 \text{ cm}$$

The clearance volume V_4 after re-expansion of compressed air in clearance space

$$V_4 = V_3\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = V_3 \times \left(\frac{8}{1}\right)^{\frac{1}{1.3}} = 4.95\,V_c$$

Using cylinder cross-section area A, before overhaul;

$$V_4 = 4.95 \times 2.25\,A = 11.14\,A \text{ cm}^3$$

After overhaul,

$$V_{4a} = 4.95 \times 3.15\,A = 15.59\,A \text{ cm}^3$$

Total volume of compressor

Before overhaul;

$$V_1 = V_s + AL_{c_1} = 75\,A + 2.25A = 77.25\,A \text{ cm}^3$$

After overhaul;

$$V_{1a} = V_s + AL_{c_2} = 75\,A + 3.15A = 78.15\,A \text{ cm}^3$$

The effective suction volume

Before overhaul;

$$V_1 - V_4 = 77.25\,A \text{ cm}^3 - 11.14\,A \text{ cm}^3 = 66.11\,A \text{ cm}^3$$

After overhaul;

$$V_{1a} - V_{4a} = 78.15\,A \text{ cm}^3 - 15.59\,A \text{ cm}^3 = 62.56\,A \text{ cm}^3$$

(i) Percentage change in FAD

$$= \frac{66.11A - 62.56\,A}{66.11A} \times 100 = \mathbf{5.37\%}$$

(ii) Indicator work input

$$W_1 = \frac{n}{n-1} p_1(V_1 - V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

In calculation of indicated work input, all terms except $(V_1 - V_4)$ will remain constant before and after overhaul. Therefore, percentage change in indicated work input will be equal to percentage change in FAD, i.e. % change in work input = **5.37%**

Example 5.18 *A 4-cylinder, single-stage, double acting air compressor delivers air at 7 bar. The pressure and temperature at the end of the suction stroke are 1 bar and 27°C. It delivers 30 m^3 of free air per minute when the compressor is running at 300 rpm. The pressure and temperature of the ambient air are 1 bar and 17°C. The clearance volume is 4% of the stroke volume. The stroke-to-bore ratio is 1.2. The index of compression and expansion is 1.32. Calculate*

(a) Volumetric efficiency of the compressor,

(b) Indicated power of the compressor,

(c) Size of motor, if mechanical efficiency is 85%, and

(d) Cylinder dimensions.

Solution

Given A 4-cylinder, single-stage, double-acting reciprocating air compressor

No. of cylinders = 4	k = 2
p_1 = 1 bar = 100 kPa	T_1 = 27°C = 300 K
p_2 = 7 bar	$\dot{V}_f$ = 30 m^3/min
N = 300 rpm	p_f = 1 bar
c = 0.04	T_f = 17°C = 290 K
η_{mech} = 0.85	n = 1.32
L/d = 1.2	

To find

(i) Volumetric efficiency,

(ii) Indicated power,

(iii) Size of motor (brake power), and
(iv) Bore of cylinder and piston stroke.

Analysis

(i) *Volumetric efficiency*

Clearance ratio;

$$c = 0.04$$

The volumetric efficiency;

$$\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{7}{1}\right)^{\frac{1}{1.32}}$$

$$= 0.8653 \quad \text{or} \quad \mathbf{86.53\%}$$

(ii) *Indicated power*

The effective swept volume;

$$\dot{V}_1 - \dot{V}_4 = \frac{p_f \dot{V}_f}{T_f} \times \frac{T_1}{p_1}$$

$$= \frac{(1\text{ bar}) \times (30\text{ m}^3/\text{min})}{(290\text{ K})} \times \left(\frac{300\text{ K}}{1\text{ bar}}\right)$$

$$= 31.03\text{ m}^3/\text{min} \quad \text{or} \quad 0.517\text{ m}^3/\text{s}$$

Indicated power;

$$IP = \frac{n}{n-1} p_1(\dot{V}_1 - \dot{V}_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.32}{1.32 - 1} \times (100\text{ kPa}) \times (0.517\text{ m}^3/\text{s}) \times \left[\left(\frac{7}{1}\right)^{\frac{1.32-1}{1.32}} - 1\right]$$

$$= \mathbf{128.60\ kW}$$

(iii) The motor (brake) power

$$BP = \frac{IP}{\eta_{mech}} = \frac{128.6}{0.85} = 151.3\text{ kW}$$

(iv) *Cylinder dimensions*

The piston displacement volume of one cylinder can be obtained by using volumetric efficiency as

$$V_s = \frac{\dot{V}_1 - \dot{V}_4}{\text{No. of cylinders} \times \eta_{vol}}$$

$$= \frac{31.03}{4 \times 0.8653} = 8.965\text{ m}^3/\text{min}$$

The piston displacement volume rate with $L = 1.2\,d$; for double-acting cylinder can be expressed as

$$V_s = \left(\frac{\pi}{4}\right) d^2 L N k$$

$$= \left(\frac{\pi}{4}\right) d^2 \times (1.2\,d) \times 300 \times 2$$

Bore $\quad d = \sqrt[3]{\dfrac{8.965}{(\pi/4) \times 1.2 \times 300 \times 2}}$

$$= 0.251\text{ m} \quad \text{or} \quad \mathbf{251\ mm}$$

Stroke; $\quad L = 1.2\,d = \mathbf{301.4\ mm}$

Example 5.19 *A twin-cylinder single-stage, single-acting reciprocating air compressor running at 300 rpm has pressure ratio of 8. The clearance is 3 per cent of the swept volume. It compresses 30 m^3/min free air at 101.3 kPa and 20°C according to $pV^{1.3}$ = constant. The temperature rise during suction stroke is 25°C . The loss of pressure through intake and discharge valve is 8 and 150 kPa, respectively. Determine*

(a) Indicated power required by the compressor,
(b) Brake power, assuming compression efficiency of 85% and mechanical efficiency of 95%, and
(c) Cylinder dimesions for same bore and stroke size.

Solution

Given A single-stage, single-acting reciprocating air compressor

No. of cylinders $= 2$	$c = 0.03$
$N = 300$ rpm	$n = 1.3$
$\dot{V}_f = 30\text{ m}^3/\text{min}$	$p_f = 101.3$ kPa
$T_f = 20°\text{C} = 293\text{ K}$	$p_f - p_1 = 8$ kPa
$\dfrac{p_2}{p_1} = 8$	$L = d$
$T_1 = 20°\text{C} + 25°\text{C} = 45°\text{C}$ or 318 K	
$p_2 - p_d = 150$ kPa	
$\eta_{comp} = 0.85$	
$\eta_{mech} = 0.95$	

To find

(i) Indicated power necessary to drive the compressor,
(ii) Brake power, and
(iii) Cylinder dimesions for same bore and stroke size.

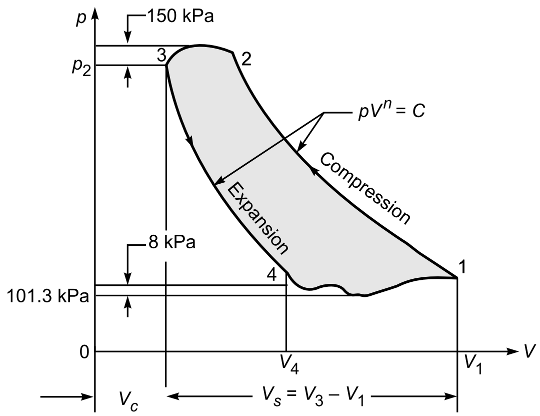


Fig. 5.16

<u>**Analysis**</u> The suction pressure

$$p_1 = p_f - 8 \text{ kPa} = 101.3 - 8 = 93.3 \text{ kPa}$$

Using the property relation for an ideal gas as

$$\frac{p_f \dot{V}_f}{T_f} = \frac{p_1(\dot{V}_1 - \dot{V}_4)}{T_1}$$

It gives

$$\dot{V}_1 - \dot{V}_4 = \frac{p_f \dot{V}_f}{T_f} \times \frac{T_1}{p_1} = \frac{101.3 \times 30}{293} \times \frac{318}{93.3}$$

$$= 35.35 \text{ m}^3/\text{min}$$

(i) *The indicated power input to the compressor*

$$IP = \frac{n}{n-1} p_1(\dot{V}_1 - \dot{V}_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times 93.3 \times \left(\frac{35.5}{60}\right) \times \left[(8)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= \mathbf{146.7\ kW}$$

(ii) *The brake power input to the compressor*

Brake power;

$$BP = \frac{\text{Indicated power}}{\eta_{comp} \times \eta_{mech}} = \frac{146.7}{0.85 \times 0.95}$$

$$= \mathbf{181.67\ kW}$$

(iii) *Cylinder dimesions for same bore and stroke size:*

The volumetric efficiency is given as

$$\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= 1 + 0.03 - 0.03 \times (8)^{1/1.3}$$

$$= 0.8814$$

Further, the volumetric efficiency is also given as

$$\eta_{vol} = \frac{(\dot{V}_1 - \dot{V}_4)}{\dot{V}_s}$$

∴ total displacement volume for two cylinders;

$$\dot{V}_s = \frac{(\dot{V}_1 - \dot{V}_4)}{\eta_{vol}} = \frac{35.35}{0.8814}$$

$$= 40.1 \text{ m}^3/\text{min}$$

For a two-cylinder, single-acting reciprocating air compressor, the displacement volume per minute is also expressed as

$$\dot{V}_s = 2 \times \left(\frac{\pi}{4}\right) d^2 L N$$

For $d = L$;

$$40.1 = 2 \times \left(\frac{\pi}{4}\right) d^3 \times 300$$

It gives bore and stroke sizes as

$$d = L = 0.44 \text{ m} \quad \text{or} \quad \mathbf{440\ mm}$$

5.11 LIMITATIONS OF SINGLE-STAGE COMPRESSION

Usually, the pressure ratio for a single-stage reciprocating air compressor is limited to 7. Increase in pressure ratio in a single-stage reciprocating air compressor causes the following undesirable effects:

1. Greater expansion of clearance air in the cylinder and as a consequence, it decreases effective suction volume (V_1 – V_4) and therefore, there is a decrease in fresh air induction.
2. With high delivery pressure, the delivery temperature increases. It increases specific volume of air in the cylinder, thus more compression work is required.
3. Further, for high pressure ratio, the cylinder size would have to be large, strong and heavy working parts of the compressor will be needed. It will increase balancing problem

and high torque fluctuation will require a heavier flywheel installation.

All the above problems can be reduced to minimum level with multistage compression.

5.12 MULTISTAGE COMPRESSION

As discussed in preceeding sections, the compressor requires minimum work input with isothermal compression. But the delivery temperature T_2 increases with pressure ratio and the volumetric efficiency decreases as pressure ratio increases.

All the above problems can be reduced to minimum level by compressing the air in more than one cylinders with intercooling between stages, for the same pressure ratio. The compression of air in two or more cylinders in series is called *multistage compression.* Air cooling between stages provides the means to achieving an appreciable reduction in the compression work and maintaining the air temperature within safe operating limits.

5.12.1 Advantages of Multistage Compression

1. The gas can be compressed to a sufficiently high pressure.
2. Cooling of air is more efficient with intercoolers and cylinder wall surface.
3. By cooling the air between the stages of compression, the compression can be brought to isothermal and power input to the compressor can be reduced considerably.
4. By multistaging, the pressure ratio of each stage is lowered. Thus, the air leakage past the piston in the cylinder is also reduced.
5. The low pressure ratio in a cylinder improves volumetric efficiency.
6. Due to phasing of operation in stages, in a multistage compressor, the negative and positive forces are balanced to a large extent. Thus, more uniform torque and better mechanical balance can be achieved.
7. Due to low pressure ratio in stages, the compressor speed could be higher for same isothermal efficiency.
8. Low working temperature in each stage helps to sustain better lubrication.
9. The low-pressure cylinder is made lighter and high-pressure cylinders are made smaller for reduced pressure ratio in each stage.

5.12.2 Work Done in Multistage Compressor with Intercooler

Figure 5.17 shows a schematic for two-stage compression. The air at p_1 and T_1 is first drawn into the first stage or low pressure (LP) cylinder. It is partially compressed to some intermediate pressure, p_2 and temperature T_2 and is then discharged to an intercooler which ideally cools the air to its initial temperature T_1. The cooled air then enters the second stage or high pressure (HP) cylinder and is compressed to a delivery pressure p_3 and temperature T_3. The corresponding indicator diagram is shown on a p–V plane in Fig. 5.18.

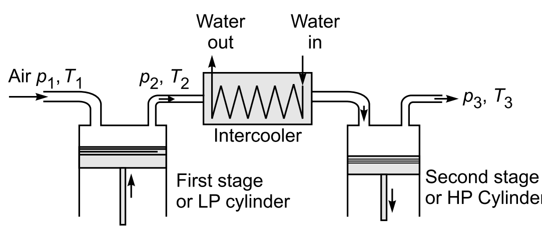


Fig. 5.17 *Two stage compression with intercooler*

The cycle 1–2–3–4–1 represents first-stage compression cycle. The air is discharged from LP cylinder at intermediate pressure p_2 and temperature T_2, related with inlet pressure p_1 and temperature T_1 as

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

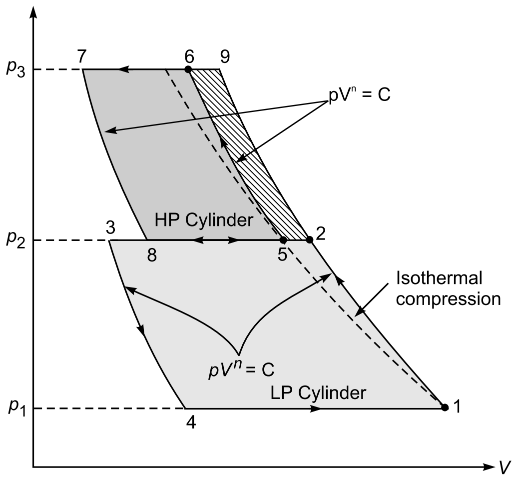


Fig. 5.18 *Indicator diagram for two-stage compressor*

The air is then cooled in an intercooler, if intercooling is complete (perfect), the air will enter the HP cylinder at the same temperature at which it enters the LP cylinder. The second-stage compression cycle in an HP cylinder is shown by cycle 5–6–7–8–5. The line 1–2–9 represents the single-stage compression from initial pressure p_1 to delivery pressure p_3. The shadded area 2–9–6–5–2 represents the saving in compression work obtained by intercooling.

The total indicator work

$$W_{in} = W_{LP} + W_{HP}$$

$$= \frac{n}{n-1} p_1(V_1 - V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} p_2(V_5 - V_8)\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right]$$

In terms of mass of air inducted per cycle;

$$W_{in} = \frac{n}{n-1} m_a R T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} m_a R T_1\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W_{in} = \frac{n}{n-1} \times m_a R T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{(kJ/cycle)} \quad ...(5.33)$$

where, $m_a = \dfrac{p_f V_f}{R T_f} = \dfrac{p_1(V_1 - V_4)}{R T_1} = \dfrac{p_2(V_5 - V_8)}{R T_1}$

Since same mass of air is handled by both cylinders, the suffix f represents free air conditions.

For given mass-flow rate $\dot{m}_a$ (kg/s), the indicated power

$$IP = \frac{n}{n-1} \dot{m}_a R T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{(kW)} \quad ...(5.34)$$

5.12.3 Heat Rejected per Stage of Compression

In a two-stage air compressor, air rejects heat (i) during compression process, and (ii) after compression in intercooler.

$$Q_{stage} = Q_{comp} + Q_{cooling}$$

Heat rejected during polytropic compression process

$$Q_{comp} = m_a C_n (T_2 - T_1) \text{ (kJ)}$$

where $C_n = \dfrac{C_p - n C_v}{n - 1}$, the polytropic specific heat, measured in kJ/kg·K.

For perfect intercooling, the temperature of air after cooling should be reduced to initial temperature T_1. Therefore, the heat rejected in intercooler

$$Q_{cooling} = m_a C_p (T_2 - T_1) \text{ (kJ)} \quad ...(5.35)$$

Using $C_p = \gamma C_v$, we get

$$C_n = \frac{\gamma C_v - n C_v}{n-1} = \frac{(\gamma - n)}{n-1} C_v$$

Then total heat rejected;

$$Q_{stage} = m_a \frac{(\gamma - n)}{n-1} C_v (T_2 - T_1) + m_a C_p (T_2 - T_1)$$

$$= m_a \left[\frac{(\gamma - n)}{n-1} C_v + C_p \right] (T_2 - T_1) \text{ (kJ)} \quad ...(5.36)$$

For heat rejected per kg of air

$$q_{stage} = \left[\frac{(\gamma - n)}{n-1} C_v + C_p \right] (T_2 - T_1) \text{ (kJ/kg)} \quad ...(5.37)$$

5.12.4 Actual Indicator Diagram for a Two-Stage Compressor

The actual indicator diagram on a p–V plane for a two-stage reciprocating air compressor is shown in Fig. 5.19. The variation during suction and delivery processes is due to valve action effects.

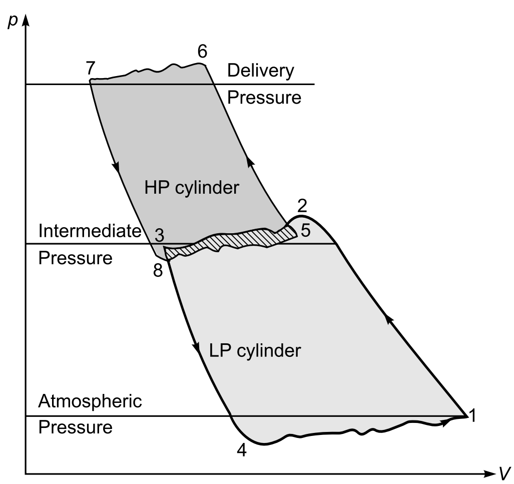


Fig. 5.19 *Actual indicator diagram for two-stage reciprocating air compressor*

The indicator diagrams for low-pressure and high-pressure cylinders overlap due to pressure drop taking place in intercooler and clearance effect.

During the suction process, pressure drops in the cylinder until the inlet valve is forced to open by air. Similarly, during delivery process , some more pressure is required to open the delivery valve and to displace the compressed air through a narrow valve passage. Thus, *gas throttling* takes place during delivery, which reduces the pressure gradually.

5.12.5 Condition for Minimum Compression Work: Optimum Intermediate Pressure

The intermediate pressure p_2 influences the work to be done on the gas and its distribution between stages. The intermediate pressure, which makes work input minimum, is always important.

The total power input to a two-stage reciprocating air compressor with complete intercooling is given by Eq. (5.34);

$$IP = \frac{n}{n-1} \dot{m}_a R T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right]$$

If p_1, T_1 and p_3 are fixed then the optimum value of intermediate pressure p_2 for minimum work input can be obtained by applying condition of minima, i.e.,

$$\frac{d(IP)}{dp_2} = 0$$

$$\frac{n}{n-1} \dot{m}_a R T_1 \frac{d}{dp_2} \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right] = 0$$

$$\left(\frac{1}{p_1} \right)^{\frac{n-1}{n}} \frac{d}{dp_2} (p_2)^{\frac{n-1}{n}} + (p_3)^{\frac{n-1}{n}} \frac{d}{dp_2} (p_2)^{\frac{1-n}{n}} = 0$$

$$\left(\frac{1}{p_1} \right)^{\frac{n-1}{n}} \left(\frac{n-1}{n} \right) (p_2)^{\frac{n-1}{n} - 1} + (p_3)^{\frac{n-1}{n}} \left(\frac{1-n}{n} \right) (p_2)^{\frac{1-n}{n} - 1} = 0$$

or $$\left(\frac{1}{p_1} \right)^{\frac{n-1}{n}} (p_2)^{-\frac{1}{n}} - (p_3)^{\frac{n-1}{n}} (p_2)^{\frac{1-2n}{n}} = 0$$

or $$(p_1)^{\frac{1-n}{n}} (p_2)^{-\frac{1}{n}} = (p_3)^{\frac{n-1}{n}} (p_2)^{\frac{1-2n}{n}}$$

or $$(p_2)^{-\frac{1}{n}+\frac{2n-1}{n}} = (p_1)^{\frac{n-1}{n}} (p_3)^{\frac{n-1}{n}}$$

or $$(p_2)^{\frac{2(n-1)}{n}} = (p_1\, p_3)^{\frac{n-1}{n}}$$

Therefore,

$$p_2^2 = p_1\, p_3 \qquad ...(5.38)$$

or $$\frac{p_2}{p_1} = \frac{p_3}{p_2} \qquad ...(5.39)$$

It is proved that for minimum compression work, the conditions required are the following

1. The pressure ratio of each stage should be the same.
2. The pressure ratio of any stage is the square root of overall pressure ratio, for a two stage compressor.
3. Air after compression in each stage should be cooled to intial temperature of air intake.
4. The work input to each stage is same.

Consider multistage compression with z stages. Then

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \ldots$$

$$= \frac{p_{(z+1)}}{p_z} = X \text{ (say)} \qquad ...(5.40)$$

Then $p_2 = Xp_1;\ p_3 = Xp_2 = X^2 p_1;$

$p_4 = Xp_3 = X^2 p_2 = X^3 p_1$

and $p_{(z+1)} = Xp_z = \ldots = X^z p_1$

or $$X^z = \frac{p_{(z+1)}}{p_1}$$

or $$X = \sqrt[z]{\frac{p_{(z+1)}}{p_1}}$$

$$= \sqrt[z]{(\text{Pressure ratio through compressor})} \qquad ...(5.41)$$

5.12.6 Minimum Compression Work Input for Two-Stage Compression

Inserting Eq. (5.39) in Eq. (5.34), we get total minimum power,

$$IP_{min} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a\, RT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad ...(5.42)$$

$= 2 \times$ power required for one stage

In terms of overall pressure ratio

$$\frac{p_2}{p_1} = \frac{\sqrt{p_1\, p_3}}{p_1} = \sqrt{\frac{p_3}{p_1}} = \left(\frac{p_3}{p_1}\right)^{\frac{1}{2}}$$

Total minimum power;

$$IP_{min} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a\, RT_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] \qquad ...(5.43)$$

This expression can be extended to z stages of compression. Total minimum power;

$$IP_{min} = z \times \left(\frac{n}{n-1}\right) \dot{m}_a\, RT_1 \left[\left(\frac{p_{(z+1)}}{p_1}\right)^{\frac{n-1}{zn}} - 1\right] \qquad ...(5.44)$$

where the pressure ratio in each stage $= \left(\frac{p_{z+1}}{p_1}\right)^{\frac{1}{z}}$

Example 5.20 *Calculate the power required to compress 25 m³/min atmospheric air at 101.3 kPa, 20°C to a pressure ratio of 7 in an LP cylinder. Air is then cooled at constant pressure to 25°C in an intercooler, before entering HP cylinder, where air is again compressed to a pressure ratio of 6. Assume polytropic compression with n = 1.3 and R = 0.287 kJ/kg·K.*

Solution

Given A two-stage reciprocating air compressor with imperfect inercooler;

$\dot{V}_1 = 25$ m³/min $\qquad p_1 = 101.3$ kPa

$T_1 = 20°C = 293$ K $\qquad T_3 = 25°C = 298$ K

$\frac{p_2}{p_1} = 7.0 \qquad \frac{p_3}{p_2} = 6.0$

$n = 1.3 \qquad R = 0.287$ kJ/kg·K

To find Power input to compressor.

Analyis The mass of air compressed per minute, using perfect gas equation

$$\dot{m}_a = \frac{p_1 \dot{V}_1}{RT_1}$$

$$= \frac{(101.3 \text{ kPa}) \times (25 \text{ m}^3/\text{min})}{(0.287 \text{ kJ/kg}\cdot\text{K}) \times (293 \text{ K})}$$

$$= 30.11 \text{ kg/min}$$

Temperature of air after first-stage compression;

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (293\text{K}) \times \left(\frac{7}{1}\right)^{\frac{1.3-1}{1.3}}$$

$$= 459.08 \text{ K}$$

The work input in LP cylinder compressor,

$$IP_{LP} = \frac{n}{n-1} \dot{m}_a R(T_2 - T_1)$$

$$= \frac{1.3}{1.3-1} \times 30.11 \times 0.287 \times (459.08 - 300)$$

$$= 6219.24 \text{ kJ/min} \quad \text{or} \quad \mathbf{103.65 \text{ kW}}$$

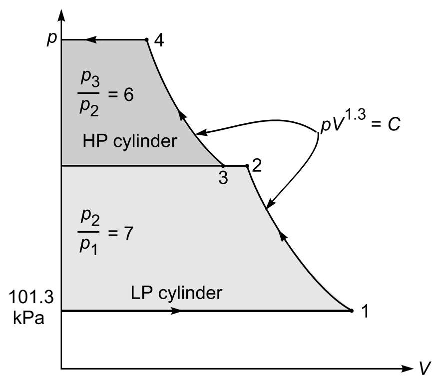


Fig. 5.20 *Two-stage compression*

Temperature of air after second-stage compression;

$$T_4 = T_3\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} = (298\text{K}) \times \left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}}$$

$$= 450.59 \text{ K}$$

The work input in HP cylinder of compressor,

$$IP_{HP} = \frac{n}{n-1} \dot{m}_a R(T_4 - T_3)$$

$$= \frac{1.3}{1.3-1} \times 30.11 \times 0.287 \times (450.59 - 298)$$

$$= 5714 \text{ kJ/min} \quad \text{or} \quad \mathbf{95.23 \text{ kW}}$$

Total power input to the compressor

$$IP = IP_{LP} + IP_{HP}$$

$$= 103.65 \text{ kW} + 95.23 \text{ kW} = \mathbf{198.88 \text{ kW}}$$

Example 5.21 *The LP cylinder of a two-stage double-acting reciprocating air compressor running at 120 rpm has a 50-cm diameter and 75-cm stroke. It draws air at a pressure of 1 bar and 20°C and compresses it adiabatically to a pressure of 3 bar. The air is then delivered to the intercooler, where it is cooled at constant pressure to 35°C and is then further compressed polytropically (index n = 1.3) to 10 bar in HP cylinder. Determine the power required to drive the compressor. The mechanical efficiency of the compressor is 90% and motor effieincy is 86%.*

Solution

Given A two-stage, double-acting reciprocating air compressor with imperfect intercooling;

$N = 120$ rpm	$d_1 = 50$ cm $= 0.5$ m
$L = 75$ cm $= 0.75$ m	$p_1 = 1$ bar
$T_1 = 20°\text{C} = 293$ K	$p_2 = 3$ bar
$p_1 V_1^\gamma = p_2 V_2^\gamma$	$p_2 = p_3 = 3$ bar
$T_3 = 35°\text{C} = 308$ K	$p_4 = 10$ bar
$p_3 V_3^n = p_4 V_4^n$ with $n = 1.3$	$k = 2$
$\eta_{mech} = 0.9$	$\eta_{Motor} = 0.86$

To find Motor power input to drive the compressor.

Assumptions

(i) The effect of the piston rod is negligible on the cylinder volume.

(ii) For air: $R = 0.287$ kJ/kg·K, $C_p = 1.005$ kJ/kg·K and $\gamma = 1.4$.

Analysis The volume-flow rate of air to LP cylinder

$$\dot{V}_1 = \frac{\pi}{4} d_{LP}^2 L \frac{N}{60} k$$

$$= \frac{\pi}{4} \times (0.5 \text{ m})^2 \times (0.75 \text{ m}) \times \left(\frac{120}{60} \text{ rps}\right) \times 2$$

$$= 0.589 \text{ m}^3/\text{s}$$

The density of incoming air

$$\rho_1 = \frac{p_1}{RT_1} = \frac{(100 \text{ kPa})}{(0.287 \text{ kJ/kg}\cdot\text{K}) \times (293 \text{ K})}$$

$$= 1.189 \text{ kg/m}^3$$

The mass-flow rate of air into LP cylinder

$$\dot{m}_a = \rho_1 \dot{V}_1 = 1.189 \times 0.589 = 0.7 \text{ kg/s}$$

Temperature of air after first-stage compression;

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (293\,\text{K}) \times \left(\frac{3}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$= 401.04\,\text{K}$$

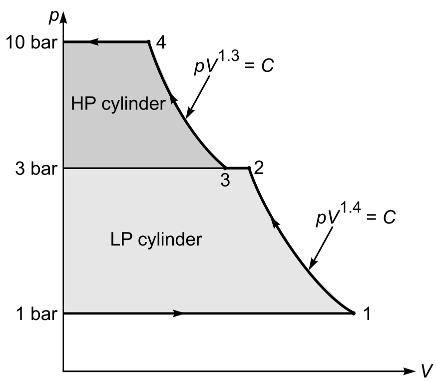


Fig. 5.21 *Two stage compression*

The *IP* input to LP cylinder,

$$IP_{LP} = \frac{\gamma}{\gamma - 1}\, \dot{m}_a R (T_2 - T_1)$$

$$= \frac{1.4}{1.4 - 1} \times 0.7 \times 0.287 \times (401.04 - 293)$$

$$= 75.97\,\text{kW}$$

Temperature of air after second-stage compression;

$$T_4 = T_3 \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} = (308\,\text{K}) \times \left(\frac{10}{3}\right)^{\frac{1.3-1}{1.3}}$$

$$= 406.64\,\text{K}$$

The work input in HP cylinder of compressor,

$$IP_{HP} = \frac{n}{n - 1}\, \dot{m}_a R (T_4 - T_3)$$

$$= \frac{1.3}{1.3 - 1} \times 0.7 \times 0.287 \times (406.64 - 308)$$

$$= 85.87 \text{ kW}$$

Total power input to the compressor

$$IP = IP_{LP} + IP_{HP}$$

$$= 75.97 \text{ kW} + 85.87 \text{ kW} = 161.85 \text{ kW}$$

Motor power input

$$= \frac{IP}{\eta_{mech}\, \eta_{Motor}} = \frac{161.85}{0.9 \times 0.86}$$

$$\mathbf{= 209.1\,kW}$$

Example 5.22 *Find the percentage saving in work input by compressing air in two stages from 1 bar to 7 bar instead of one stage. Assume a compression index of 1.35 in both the cases and optimum pressure and complete intercooling in a two-stage compressor.*

Solution

Given A two-stage air compressor with perfect intercooling

$p_1 = 1$ bar $\quad p_3 = 7$ bar $\quad n = 1.35$

To find Saving in work in comparison with single-stage compression.

Assumptions

(i) Single-acting reciprocating air compressor.

(ii) Compressions and expansions are reversible processes.

(iii) No effect of valve opening and closing on induction and delivery processes.

Analysis *Power required to drive compressor*

The minimum power input for two-stage compression with perfect intercooling;

$$IP_{multi} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a R T_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

$$= 2 \times \left(\frac{1.35}{1.35-1}\right) \times \dot{m}_a RT \times \left[\left(\frac{7}{1}\right)^{\frac{1.35-1}{2\times1.35}} - 1\right]$$

$$= 2.213\, \dot{m} RT$$

Power input with single-stage compression from 1 bar to 7 bar;

$$IP_{single} = \frac{n}{n-1} \dot{m}_a R T_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.35}{1.35-1} \times \dot{m}_a RT \times \left[\left(\frac{7}{1}\right)^{\frac{1.35-1}{1.35}} - 1\right]$$

$$= 2.53\, \dot{m} RT$$

Percentage saving in power due to multistage compression

$$= \frac{IP_{single} - IP_{multi}}{IP_{single}} \times 100 = \frac{2.53 - 2.213}{2.53} \times 100$$

$= \mathbf{12.56\%}$

Example 5.23 *2 kg/s of air enters the LP cylinder of a two-stage, reciprocating air compressor. The overall pressure ratio is 9. The air at inlet to compressor is at 100 kPa and 35°C. The index of compression in each cylinder is 1.3. Find the intercooler pressure for perfect intercooling. Also, find the minimum power required for compression, and percentage saving over single-stage compression.*

Solution

Given A two-stage, single-acting, reciprocating air compressor with perfect intercooling

$\dot{m}_a = 2$ kg/s $\qquad p_1 = 100$ kPa

$T_1 = 35°\text{C} = 308$ K $\qquad p_3 = 9$ bar

$n = 1.3$

To find

(i) Intermediate pressure,

(ii) Power required to drive the compressor, and

(iii) Percentage saving in work in comparison with single-stage compression.

Assumptions

(i) For air; $R = 0.287$ kJ/kg·K and $C_p = 1$ kJ/kg·K.

(ii) No effect of valve opening and closing on induction and delivery processes.

Analysis

(i) *For perfect intercooling*

or $\quad p_2 = \sqrt{p_1 \times p_3} = \sqrt{1 \times 9} = 3$ bar

or $\quad \dfrac{p_2}{p_1} = \mathbf{3}$

(ii) *Power required to drive the compressor*

The minimum power input for two-stage compression with perfect intercooling;

$$IP_{multi} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a R T_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

$$= 2 \times \left(\frac{1.3}{1.3-1}\right) \times 2 \times 0.287 \times 308 \times \left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{2\times 1.3}} - 1\right]$$

$= \mathbf{442.1\ kW}$

(iii) *Percentage saving in work of comparison with single-stage compression*

Power input with single stage compression from 1 bar to 9 bar;

$$IP_{single} = \frac{n}{n-1} \dot{m}_a R T_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times 2 \times 0.287 \times 308 \times \left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$= 505.92$ kW

Saving in power due to multistage compression

$= IP_{single} - IP_{multi} = 505.92 - 442.1$

$= 63.82$ kW

Per cent saving

$$= \frac{IP_{multi} - IP_{single}}{IP_{single}} \times 100 = \frac{63.82}{505.92} \times 100$$

$= \mathbf{12.61\%}$

Example 5.24 *A two-stage, single-acting, reciprocating air compressor takes in air at 1 bar and 300 K. Air is discharged at 10 bar. The intermediate pressure is ideal for minimum work and perfect intercooling. The law of compression is $pV^{1.3}$ = constant. The rate of discharge is 0.1 kg/s. Calculate*

(a) power required to drive the compressor,

(b) saving in work in comparison with single stage compression,

(c) isothermal efficiency, and

(d) heat transferred in intercooler.

Take $R = 0.287$ kJ/kg·K and $C_p = 1$ kJ/kg·K.

Solution

Given A two-stage, single-acting, reciprocating air compressor with perfect intercooling

$\dot{m}_a = 0.1$ kg/s $\qquad p_1 = 1$ bar $= 100$ kPa

$T_1 = 300$ K $\qquad p_3 = 10$ bar

$n = 1.3$ $\qquad R = 0.287$ kJ/kg·K

$C_p = 1$ kJ/kg·K $\qquad k = 1$

To find

(i) Power required to drive the compressor,
(ii) Saving in work in comparison with single-stage compression,
(iii) Isothermal efficiency, and
(iv) Heat transferred in intercooler.

Assumptions

(i) Given conditions leads to perfect intercooling.
(ii) Compressions and expansions are reversible processes.
(iii) No effect of valve opening and closing on induction and delivery processes.

Analysis For perfect intercooling

or $$p_2 = \sqrt{p_1 \times p_3} = \sqrt{1 \times 10} = 3.162 \text{ bar}$$

or $$\frac{p_2}{p_1} = 3.162$$

(i) *Power required to drive compressor*

The minimum power input for two-stage compression with perfect intercooling;

$$IP_{multi} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a R T_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

$$= 2 \times \left(\frac{1.3}{1.3-1}\right) \times 0.1 \times 0.287 \times 300 \times \left[\left(\frac{10}{1}\right)^{\frac{1.3-1}{2\times 1.3}} - 1\right]$$

$$= \mathbf{22.7\,kW}$$

Power input with single-stage compression from 1 bar to 10 bar;

$$IP_{single} = \frac{n}{n-1} \dot{m}_a R T_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times 0.1 \times 0.287 \times 300 \times \left[\left(\frac{10}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= 26.16 \text{ kW}$$

(ii) Saving in power due to multistage compression

$$= IP_{single} - IP_{multi} = 26.16 - 22.7$$

$$= \mathbf{3.46\,kW}$$

Isothermal power input,

$$IP_{iso} = \dot{m}_a R T_1 \ln\left(\frac{p_3}{p_1}\right)$$

$$= 0.1 \times 0.287 \times 300 \times \ln\left(\frac{10}{1}\right)$$

$$= 19.825 \text{ kW}$$

(iii) Isothermal efficiency;

$$\eta_{iso} = \frac{IP_{iso}}{IP_{act}} = \frac{19.825}{22.7}$$

$$= 0.8733 \quad \text{or} \quad \mathbf{87.33\%}$$

Temperature after first-stage compression

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (300\,\text{K}) \times \left(\frac{3.162}{1}\right)^{\frac{1.3-1}{1.3}}$$

$$= 391.3\,\text{K}$$

(iv) Heat rejection rate in the intercooler

$$\dot{Q} = \dot{m}_a C_p (T_2 - T_1)$$

$$= 0.1 \times 1.0 \times (391.3 - 300)$$

$$= \mathbf{9.13\,kW}$$

Example 5.25 *In a three-stage compressor, air is compressed from 98 kPa to 20 bar. Calculate for 1 m^3 of air per second,*

(a) Work under ideal condition for n = 1.3,
(b) Isothermal work,
(c) Saving in work due to multi staging, and
(d) Isothermal efficiency.

Solution

Given An ideal three-stage reciprocating air compressor

$p_1 = 98$ kPa $\qquad p_4 = 20$ bar $= 2000$ kPa
$n = 1.3 \qquad \dot{V}_1 = 1\ \text{m}^3/\text{s}$

To find

(i) Power input to compressor,
(ii) Isothermal work,
(iii) Saving in work due to multi-staging, and
(iv) Isothermal efficiency.

Assumptions

(i) Neglecting clearance volume.
(ii) Perfect inetercooling.

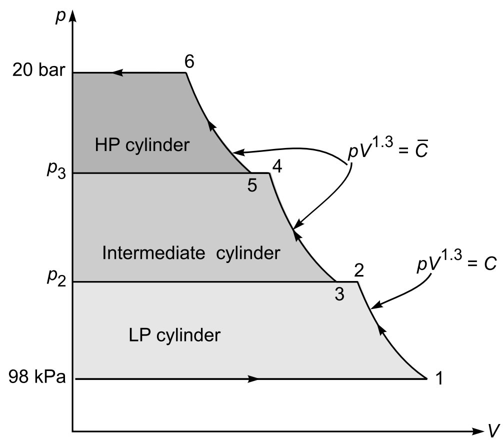


Fig. 5.22

Analysis For perfect intercooling

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \left(\frac{p_4}{p_1}\right)^{\frac{1}{3}} = \left(\frac{2000}{98}\right)^{\frac{1}{3}} = 2.732$$

(i) *Power input to compressor*

$$IP_{3,\,stage} = 3\times\left(\frac{n}{n-1}\right)p_1\dot{V}_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{3n}} - 1\right]$$

$$= 3\times\left(\frac{1.3}{1.3-1}\right)\times(98\,\text{kPa})$$

$$\times(1\,\text{m}^3/\text{s})\times\left[\left(\frac{2000}{98}\right)^{\frac{1.3-1}{3\times1.3}} - 1\right]$$

$$= \mathbf{332.62\,kW}$$

(ii) *Isothermal work*

$$IP_{iso} = p_1\dot{V}_1\ln\left(\frac{p_4}{p_1}\right)$$

$$= (98\,\text{kPa})\times(1\,\text{m}^3/\text{s})\times\ln\left(\frac{2000}{98}\right)$$

$$= \mathbf{295.56\,kW}$$

Power required in single-stage compression,

$$IP_{1,\,stage} = \left(\frac{n}{n-1}\right)p_1\dot{V}_1\left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \left(\frac{1.3}{1.3-1}\right)\times 98\times 1\times\left[\left(\frac{2000}{98}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= 427.08\,\text{kW}$$

(iii) *Saving in work due to multistaging*

$$= IP_{1,stage} - IP_{3,stage}$$

$$= 427.08\text{ kW} - 332.62\text{ kW} = \mathbf{94.42\,kW}$$

Percentage saving;

$$= \frac{\text{Saving}}{IP_{1,stage}}\times 100 = \frac{94.42}{427.08}\times 100$$

$$= \mathbf{22.1\%}$$

(iv) *Isothermal efficiency*

$$\eta_{iso} = \frac{IP_{iso}}{IP_{act}}\times 100 = \frac{295.56}{332.62}\times 100$$

$$= \mathbf{88.85\%}$$

Example 5.26 *A two-stage, single-acting reciprocating air compressor has an LP cylinder bore and stroke of 250 mm each. The clearance volume of a low-pressure cylinder is 5% of the stroke volume of the cylinder. The intake pressure and temperature are 1 bar and 17°C, respectively. Delivery pressure is 9 bar and the compressor runs at 300 rpm. The polytropic index is 1.3 throughout. The intercooling is complete and intermediate pressure is 3 bar. The overall efficiency of the plant including electric driving motor is 70%. Calculate*

(a) the mass-flow rate through the compressor, and

(b) energy input to electric motor.

Solution

Given Two-stage, single-acting reciprocating air compressor

LP cylinder: $d_1 = 250\text{ mm} = 0.25\,\text{m}$

$L_1 = 250\text{ mm} = 0.25\,\text{m}$

$c_1 = 0.05\ V_s$ or $c = 0.05$

$p_1 = 1\text{ bar} = 100\text{ kPa}$

$T_1 = 17°\text{C} = 290\,\text{K}$

$p_2 = 3\,\text{bar}$ $\quad p_3 = 9\text{ bar}$

$n = 1.3$ $\quad k = 1$

$N = 300\text{ rpm}$ $\quad \eta_{Overall} = 0.7$

To find

(i) The mass-flow rate of air through the compressor,

(ii) Power input to electric motor.

Assumptions

(i) Air as an ideal gas, with $R = 0.287$ kJ/kg·K.

(ii) Compression and expansion are reversible polytropic.

Analysis The stroke (swept) volume of LP cylinder

$$V_s = \left(\frac{\pi}{4}\right) d_1^2 L_1 = \left(\frac{\pi}{4}\right) \times (0.25\,\text{m})^2 \times (0.25\,\text{m})$$
$$= 0.01227\,\text{m}^3$$

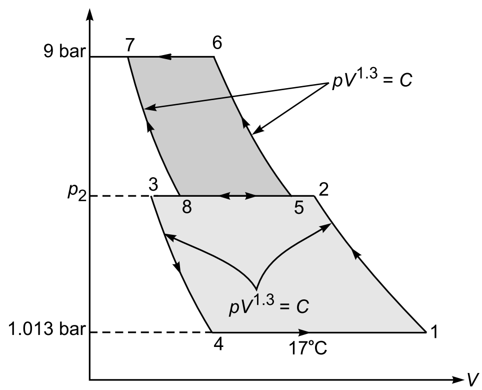


Fig. 5.23 *Two-stage compression with clearance*

The volumetric efficiency of LP cylinder is given by

$$\eta_{vol,\text{LP}} = 1 + c_1 - c_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$
$$= 1 + 0.05 - 0.05 \times \left(\frac{3}{1}\right)^{\frac{1}{1.3}} = 0.933$$

It is also expressed as

$$\eta_{vol,\,LP} = \frac{V_1 - V_4}{V_s}$$

or $$V_1 - V_4 = \eta_{vol,LP}\, V_s = 0.933 \times 0.01227 = 0.01145\,\text{m}^3$$

The mass of air inducted per cycle into LP cylinder

$$m_a = \frac{p_1(V_1 - V_4)}{RT_1}$$

or $$m_a = \frac{(100\text{ kPa}) \times (0.01145\text{ m}^3)}{(0.287\text{ kJ/kg}\cdot\text{K}) \times (290\text{ K})}$$
$$= 0.0137\,\text{kg/cycle}$$

(i) *Mass-flow rate of air*

The mass flow rate of air per minute

$$\dot{m}_a = m_a N k$$
$$= 0.0137 \times 300 \times 1 = 4.13\,\text{kg/min}$$
$$= \mathbf{0.069\,kg/s}$$

(ii) *Motor power input*

The indicated power input to the compressor with

$$\frac{p_2}{p_1} = \left(\frac{p_3}{p_1}\right)^{1/2}$$

$$IP = \frac{2n}{n-1}\, \dot{m}_a\, RT \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$
$$= \frac{2 \times 1.3}{1.3 - 1} \times 0.069 \times 0.287 \times 290 \times \left[\left(\frac{3}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$
$$= 14.32\,\text{kW}$$

Motor power

$$= \frac{IP}{\eta_{Overall}} = \frac{14.32}{0.7} = \mathbf{20.46\,kW}$$

5.13 CYLINDER DIMENSIONS OF A MULTISTAGE COMPRESSOR

In an air compressor, the mass of air inducted through each cylinder is same. Therefore,

$$m_1 = m_2 = m_3 = \ldots$$

or $$\rho_1(V_1 - V_4) = \rho_2(V_5 - V_8) = \rho_3(V_9 - V_{12}) = \ldots \quad \text{...(5.45)}$$

where $(V_1 - V_4)$, $(V_5 - V_8)$ and $(V_9 - V_{12})$ represent effective suction volume per cycle taken in LP, intermediate, and HP cylinders, respectively and ρ_1, ρ_2, ρ_3 are corresponding densities of air. The density of air can be expressed as

$$\rho = \frac{p}{RT}$$

$$\therefore \quad \frac{p_1}{RT_1}(V_1 - V_4) = \frac{p_2}{RT_3}(V_5 - V_8) = \frac{p_3}{RT_5}(V_9 - V_{12}) = \ldots$$

For perfect intercooling, isothermal conditions prevail, i.e.,

$$T_1 = T_3 = T_5 = \ldots$$

$$\therefore \quad p_1(V_1 - V_4) = p_2(V_5 - V_8) = p_3(V_9 - V_{12}) = \ldots \quad \text{...(5.46)}$$

Introducing volumetric efficiency of respective cylinders;

$$p_1\eta_{vol_1}V_{s_1} = p_2\eta_{vol_2}V_{s_2} = p_3\eta_{vol_3}V_{s_3} = \ldots \quad \text{...(5.47)}$$

Piston displacement volume as

$$V_s = \left(\frac{\pi}{4}\right)d^2 L \text{, then}$$

$$p_1\eta_{vol_1}\left(\frac{\pi}{4}\right)d_1^2 L_1 = p_2\eta_{vol_2}\left(\frac{\pi}{4}\right)d_2^2 L_2 = p_3\,\eta_{vol_3}\left(\frac{\pi}{4}\right)d_3^2 L_3 = \ldots \quad \text{...(5.48)}$$

Usually, the stroke length for all cylinders remains same, i.e.,

$$L_1 = L_2 = L_3 = \ldots$$

$$\therefore \quad p_1\eta_{vol_1}d_1^2 = p_2\eta_{vol_2}d_2^2 = p_3\eta_{vol_3}d_3^2 = \ldots \quad \text{...(5.49)}$$

If all cylinders have same clearance ratio, then

$$\eta_{vol_1} = \eta_{vol_2} = \eta_{vol_3} = \ldots$$

and $\quad p_1 d_1^2 = p_2 d_2^2 = p_3 d_3^2 = \ldots$ ...(5.50)

Equations (5.49) and (5.50) are used to calculate the cylinder dimension for multi-stage compressors.

Example 5.27 *A two-stage, single-acting, reciprocating air compressor with complete intercooling receives atmospheric air at 1 bar and 15°C, compresses it polytropically (n = 1.3) to 30 bar. If both cylinders have the same stroke, calculate the diameter of the HP cylinder, if the diameter of the LP cylinder is 300 mm.*

Solution

Given Two-stage, single-acting reciprocating air compressor

LP cylinder: $d_1 = 300$ mm $\quad p_1 = 1$ bar

$T_1 = 15°C = 288\,K \quad T_3 = 15°C = 288\,K$

$p_3 = 30$ bar $\quad n = 1.3 \quad k = 1$

To find Diameter of HP cylinder.

Assumptions

(i) Compression in both cylinders is reversible.

(ii) Negligible clearance in both cylinders.

(iii) No effect of valve opening and closing on induction and delivery processes.

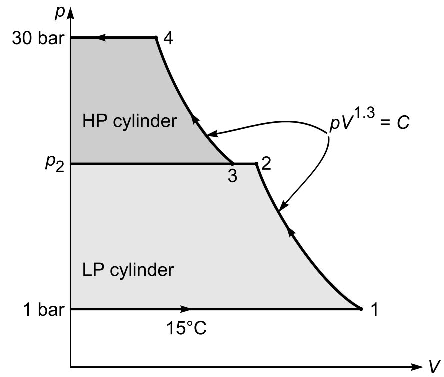


Fig. 5.24

Analysis For perfect intercooling, the pressure ratio per stage

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \sqrt{p_1 \times p_3} = \sqrt{1\times 30} = 5.477$$

$$p_2 = 5.477\,p_1 = 5.477 \times 1 = 5.477\,\text{bar}$$

The total volume of LP cylinder

$$V_1 = V_{s,LP} = \left(\frac{\pi}{4}\right)d_1^2 L$$

The total volume of HP cylinder

$$V_3 = V_{s,HP} = \left(\frac{\pi}{4}\right)d_2^2 L$$

Equation (5.46), for perfect intercooling, without clearance leads to

$$p_1 V_1 = p_2 V_3$$

or $$V_{s,HP} = V_{s,LP}\left(\frac{p_1}{p_2}\right) = V_{s,LP}\times\frac{1}{5.477} = 0.1825\,V_{s,LP}$$

Without clearance, with same pressure ratio and with perfect intercooling, the volumetric efficiency will be same. Expressing the above equation in terms of cylinder diameters

$$\left(\frac{\pi}{4}\right)d_{HP}^2 L = 0.1825\times\left(\frac{\pi}{4}\right)\times(300\text{ mm})^2\times L$$

or $$d_{HP} = \sqrt{0.1825\times(300\text{ mm}^2)} = \mathbf{128.18\,mm}$$

Example 5.28 *A two-stage, single-acting, reciprocating air compressor receives atmospheric air at 15°C, compresses it isentropically in the LP cylinder to inter-*

mediate pressure, where air cools to its initial temperature and then again compresses polytropically (n = 1.3) in the HP cylinder. The clearance volume and pressure ratio in both cylinders are 5% of the swept volume and 2, respectively. Determine the stroke volume of HP cylinder for 60 litre swept volume of LP cylinder.

Solution

Given Two-stage, single-acting reciprocating air compressor

$V_{s,LP} = 60$ litres $\quad V_3 = 0.05\, V_{s,LP} = 3$ litres

$T_1 = 15°C = 288\,K$

$T_3 = 15°C = 288\,K$

$\dfrac{p_2}{p_1} = 2.0 \quad \dfrac{p_3}{p_2} = 2.0$

$V_7 = 0.05\, V_{s,HP} \quad n = 1.3 \quad k = 1$

To find Stroke volume $(V_5 - V_7)$ of HP cylinder.

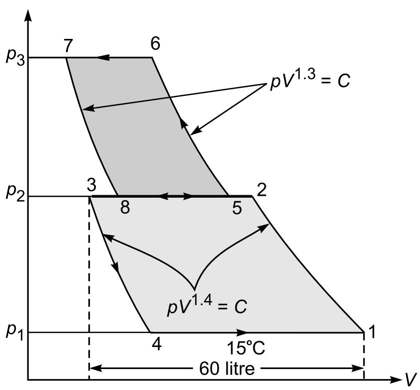


Fig. 5.25

Assumptions

(i) For air $\gamma = 1.4$.

(ii) For LP cylinder, the re-expansion of air is isentropic.

(iii) For HP cylinder, the re-expansion of air is polytropic.

Analysis The total volume of LP cylinder

$$V_1 = V_{s,LP} + V_3 = 60 \text{ lit} + 3 \text{ lit} = 63 \text{ litres}$$

The volume of air after first-stage compression

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

$$= (63 \text{ lit}) \times \left(\frac{1}{2}\right)^{\frac{1}{1.4}} = 38.4 \text{ litres}$$

The temperature of air after first-stage compression

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (288\,K) \times \left(\frac{2}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$= 351\,K$$

The clearance volume of HP cylinder

$$V_{c,HP} = 0.05\, V_{s,HP}$$

or $\quad V_{s,HP} = 20\, V_{c,HP}$

The total volume of HP cylinder

$$V_5 = V_{c,HP} + V_{s,HP} = V_{c,HP} + 20\, V_{c,HP}$$

$$= V_7 + 20\, V_7 = 21\, V_7$$

The volume of air after re-expansion in HP cylinder

$$V_8 = V_7\left(\frac{p_7}{p_8}\right)^{\frac{1}{n}} = V_7 \times \left(\frac{2}{1}\right)^{\frac{1}{1.3}} = 1.704\, V_7$$

In intercooling, air is cooled to initial temperature, and volume of air reduces from $(V_2 - V_3)$ to $(V_5 - V_8)$ at constant pressure. Therefore,

$$\frac{V_2 - V_3}{T_2} = \frac{V_5 - V_8}{T_1}$$

or $\quad V_5 - V_8 = (V_2 - V_3)\dfrac{T_1}{T_2} = (38.4 - 3) \times \dfrac{288}{351}$

$$= 29.046 \text{ litres}$$

Introducing V_5 and V_8 in terms of V_7, then

$$21\, V_7 - 1.704\, V_7 = 29.046 \text{ litres}$$

or $\quad V_7 = \dfrac{29.046}{21 - 1.704} = 1.505$

or $\quad V_5 = 21 \times 1.505 = 31.61$ litres

Stroke volume of HP cylinder

$$V_5 - V_7 = 31.61 - 1.505 = \mathbf{30.10 \text{ litres}}$$

Example 5.29 *In a trial on a two-stage, single-acting, reciprocating air compressor, following data were recorded:*

Free air delivery per min	*= 6 m³*
Free air conditions	*= 1 bar, 27°C*
Delivery pressure	*= 30 bar*
Compressor speed	*= 300 rpm*
Intermediate pressure	*= 6 bar*
Temperature at the inlet of HP cylinder	*= 27°C*

Law of compression $= pV^{1.3}$
Mechanical efficiency = 85%
Stroke to bore ratio for LP cylinder = 1:2
Stroke of HP cylinder = *Stroke of LP cylinder*

Calculate
(a) *Cylinder diameters, and*
(b) *Power input, neglecting clearance volume.*

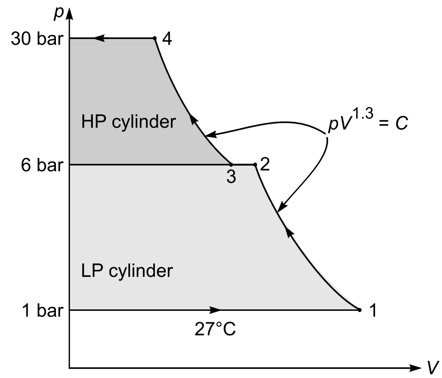


Fig. 5.26

Solution

Given Two-stage, single-acting reciprocating air compressor $k = 1$

$FAD = \dot{V}_1 = 6\,m^3/min$ $\quad p_1 = 1$ bar = 100 kPa
$T_1 = 27°C = 300\,K$ $\quad p_2 = 6$ bar
$p_3 = 30$ bar $\quad T_3 = 27°C = 300\,K$
$n = 1.3$ $\quad N = 300$ rpm
$\eta_{mech} = 0.85$
LP cylinder: $L_1/d_1 = 1.2$
and $L_2 = L_1$
and Negligible clearance

To find
(i) Cylinder dimensions, and
(ii) Power input to compressor.

Assumptions
(i) Air as an ideal gas, with $R = 0.287$ kJ/kg·K.
(ii) Compression and expansion are reversible polytropic.
(iii) Volumetric efficiency of both cylinders as 100%.

Analysis
(i) *Cylinder diameters*
The stroke (swept) volume rate of LP cylinder with $\eta_{vol} = 1.0$

$$\dot{V}_1 = FAD = \left(\frac{\pi}{4}\right) d_1^2 L_1 N k$$

or $$6\ m^3/min = \left(\frac{\pi}{4}\right) \times d_1^2 \times (1.2 d_1) \times 300 \times 1$$

or $d_1 = 0.276$ m or **276 mm**

For same stroke length and with $\eta_{vol} = 1.0$, the diameters of *LP* and *HP* cylinders are related as

$$p_1 d_1^2 = p_2 d_2^2$$
$$1 \times (276)^2 = 6 \times d_2^2$$

or $d_2 =$ **112.67 mm**

(ii) *Power input to compressor*
The mass-flow rate of air into compressor

$$\dot{m}_a = \frac{p_1 \dot{V}_1}{RT_1} = \frac{(100\ \text{kPa}) \times (6\ \text{m}^3/\text{min})}{(0.287\ \text{kJ/kg}\cdot\text{K}) \times (300\ \text{K})}$$
$$= 6.9686\ \text{kg/min} \quad \text{or} \quad 0.116\ \text{kg/s}$$

The indicated power input to two-stage, single-acting air compressor

$$IP = \frac{n}{n-1}\,\dot{m}_a RT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right]$$
$$= \frac{1.3}{1.3-1} \times 0.116 \times 0.287 \times 300 \times \left[\left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}} + \left(\frac{30}{6}\right)^{\frac{1.3-1}{1.3}} - 2\right]$$
$$= 84.9\,\text{kW}$$

Brake power input

$$= \frac{IP}{\eta_{mech}} = \frac{84.9\ \text{kW}}{0.85} = \mathbf{99.88\,kW}$$

Example 5.30 *In a single-acting, two-stage reciprocating air compressor handles 4.5 kg of air per minute, and compresses it from 1.013 bar 17°C through a pressure ratio of 9. The index of compression and expansion in both stages is 1.3. If the intercooling is complete,find the minimum indicated power and cylinder swept volumes required. Assume the clearance volume of both the stages are 5% of their respective stroke volumes and the compressor runs at 300 rpm. Take R = 0.287 kJ/kg·K.*

Solution

Given A single-acting, two-stage reciprocating air compressor with perfect intercooling

$$\dot{m}_a = 4.5 \text{ kg/min} = 0.075 \text{ kg/s}$$
$$k = 1$$
$$p_1 = 1.013 \text{ bar} = 101.3 \text{ kPa}$$
$$T_1 = 17°\text{C} = 290\,\text{K}$$
$$\frac{p_3}{p_1} = 9 \qquad n = 1.3$$
$$c_1 = c_2 = 0.05\,V_s$$
$$N = 300\,\text{rpm} \qquad R = 0.287\,\text{kJ/kg}\cdot\text{K}$$

To find

(i) Minimum indicated power, and
(ii) Swept volumes of LP and HP cylinders.

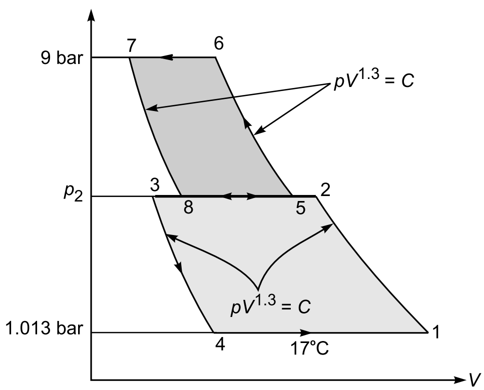


Fig. 5.27

Analysis

(i) *Minimum indicated power*

For perfect intercooling

$$p_2 = \sqrt{p_1 \times p_3} = \sqrt{p_1 \times 9p_1} = 3p_1$$

or $$\frac{p_2}{p_1} = 3$$

The minimum power input, Eq. (5.42)

$$IP_{min} = 2\times\left(\frac{n}{n-1}\right)\dot{m}_a RT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= 2\times\left(\frac{1.3}{1.3-1}\right)\times 0.075\times 0.287$$

$$\times 290\times\left[\left(\frac{3}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right] = \mathbf{15.61\,kW}$$

(ii) *The stroke (swept) volume of LP cylinder*

The mass of air inducted per cycle into LP cylinder

$$m_a = \frac{\dot{m}_a}{N\,k} = \frac{4.5 \text{ kg/min}}{(300 \text{ rotation/min})\times 1}$$

$$= 0.015 \text{ kg/cycle}$$

Efffective swept volume of LP cylinder;

$$(V_1 - V_4) = \frac{m_a RT_1}{p_1} = \frac{0.015\times 0.287\times 290}{101.3}$$

$$= 0.0123\,\text{m}^3\text{/cycle}$$

The volumetric efficiency of LP and HP cylinders

$$\eta_{vol} = 1 + c_1 - c_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= 1 + 0.05 - 0.05\times\left(\frac{3}{1}\right)^{\frac{1}{1.3}} = 0.933$$

It is also expressed as

$$\eta_{vol} = \frac{V_1 - V_4}{V_{s,\,LP}}$$

or $$V_{s,LP} = \frac{V_1 - V_4}{\eta_{vol}} = \frac{0.0123}{0.933}$$

$$= \mathbf{0.0131\,m^3/cycle}$$

The swept volume of LP cylinder is $0.0131\,\text{m}^3$/cycle.

Using Eq. (5.46) for ratio of LP and HP cylinders,

$$p_1(V_1 - V_4) = p_2(V_5 - V_8)$$

$$V_5 - V_8 = \frac{p_1}{p_2}(V_1 - V_4) = \frac{1}{3}\times(0.0123)$$

$$= 0.0041\,\text{m}^3\text{/cycle}$$

Then swept volume of HP cylinder

$$V_{s,HP} = \frac{V_5 - V_8}{\eta_{vol}} = \frac{0.0041}{0.933} = \mathbf{0.00440\,m^3}$$

Example 5.31 *A single-acting, two-stage reciprocating air compressor with complete intercooling delivers 10.5 kg/min of air at 16 bar. The compressor takes in air at 1 bar and 27°C. The compression and expansion follow the law $pV^{1.3}$ = Const. Calculate*

(a) Power required to drive the compressor
(b) Isothermal efficiency
(c) Free air delivery
(d) Heat transferred in intercooler

If the compressor runs at 440 rpm, the clearance ratios for LP and HP cylinders are 0.04 and 0.06, respectively, calculate the swept and clearance volumes for each cylinder.

Solution

Given A single-acting, two-stage reciprocating air compressor with perfect intercooling

$\dot{m}_a$ = 10.5 kg/min = 0.175 kg/s

p_1 = 1 bar = 100 kPa T_1 = 27°C = 300 K

p_3 = 16 bar n = 1.3

c_1 = 0.04 c_2 = 0.06

N = 440 rpm k = 1

To find

(i) Indicated power,
(ii) Isothermal efficiency,
(iii) FAD,
(iv) Heat transfer in intercooler, and
(v) Swept volumes of LP and HP cylinders.

Assumptions

(i) Air an ideal gas, R = 0.287 kJ/kg·K,
(ii) Compressions and expansions are reversible processes, and
(iii) Suction takes place at free air conditions.

Analysis For perfect intercooling, the pressure ratio per stage

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \sqrt{p_1 \times p_3} = \sqrt{1 \times 16} = 4$$

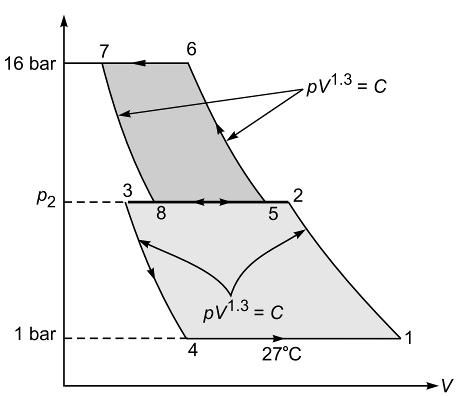


Fig. 5.28

(i) *Indicated power input for two-stage compressor*

$$IP_{min} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= 2 \times \left(\frac{1.3}{1.3-1}\right) \times 0.175 \times 0.287$$

$$\times 300 \times \left[\left(\frac{4}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= \mathbf{49.23\ kW}$$

(ii) *Isothermal efficiency*

Power input for isothermal compression from 1 bar to 16 bar pressure

$$IP_{iso} = \dot{m}_a R T_1 \ln\left(\frac{p_3}{p_1}\right)$$

$$= 0.175 \times 0.287 \times 300 \times \ln\left(\frac{16}{1}\right)$$

$$= 41.77\ \text{kW}$$

$$\eta_{iso} = \frac{\text{Isothermal power}}{\text{Actual power}} = \frac{41.77\ \text{kW}}{49.23\ \text{kW}}$$

$$= 0.848 \quad \text{or} \quad \mathbf{84.8\%}$$

(iii) *Free air delivery* (*FAD*)

$$\dot{V}_f = \dot{V}_1 - \dot{V}_4 = \frac{\dot{m}_a R T_1}{p_1}$$

$$= \frac{(10.5\ \text{kg/min}) \times (0.287\ \text{kJ/kg}\cdot\text{K}) \times (300\ \text{K})}{(100\ \text{kPa})}$$

$$= \mathbf{9.04\ m^3/min}$$

(iv) *Heat transferred in intercooler*

Temperature after compression in each stage

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 300 \times \left(\frac{4}{1}\right)^{\frac{1.3-1}{1.3}}$$

$$= 413.1\ \text{K}$$

Heat transfer rate in intercooler

$$\dot{Q} = \dot{m}_a C_p (T_2 - T_1)$$

$$= 10.5 \times 1.005 \times (413.1 - 300)$$

$$= 1193.5\ \text{kJ/min or} = \mathbf{19.89\ kW}$$

(v) *Swept volumes*

The volumetric efficiency of *LP* cylinder

$$\eta_{vol,LP} = 1 + c_1 - c_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{4}{1}\right)^{\frac{1}{1.3}}$$

$$= 0.923 \quad \text{or} \quad 92.3\%$$

The volume handled per cycle by *LP* cylinder

$(V_1 - V_4)$

$$= \frac{FAD}{\text{No. of cycle per minute}} = \frac{\dot{V}_f}{N\,k} = \frac{9.04}{440 \times 1}$$

$$= 0.0205\,\text{m}^3/\text{cycle}$$

Further, it can also be expressed as

$$\eta_{vol,LP} = \frac{V_1 - V_4}{V_{s,\,LP}}$$

or $$V_{s,LP} = \frac{V_1 - V_4}{\eta_{vol,\,LP}} = \frac{0.0205}{0.923}$$

$$= \mathbf{0.0222\ m^3/cycle}$$

For *HP* cylinder

$$\eta_{vol,HP} = 1 + c_2 - c_2\left(\frac{p_3}{p_2}\right)^{\frac{1}{n}}$$

$$= 1 + 0.06 - 0.06 \times \left(\frac{4}{1}\right)^{\frac{1}{1.3}}$$

$$= 0.885 \quad \text{or} \quad 85.5\%$$

The effective swept volume handled by *HP* cylinder with perfect intercooling

$$p_2(V_5 - V_8) = p_1(V_1 - V_4)$$

or $$(V_5 - V_8) = \frac{p_1}{p_2} \times (V_1 - V_4)$$

$$= \frac{1}{4} \times 0.0205 = 0.005125\ \text{m}^3/\text{cycle}$$

The piston displacement volume of *HP* cylinder;

$$V_{s,HP} = \frac{V_5 - V_8}{\eta_{vol,HP}} = \frac{0.005125}{0.885}$$

$$= \mathbf{0.0058\ m^3/cycle}$$

Example 5.32 *A two-stage, double-acting, reciprocating air compressor operating at 300 rpm, receives air at 1 bar and 27°C. The bore of LP cylinder is 360 mm and its stroke is 400 mm. Both cylinders have equal stroke and equal clearance of 4%. The LP cylinder discharges air at a pressure of 5 bar. The air then passes through an intercooler to cool air to its initial temperature. Pressure drops in intercooler to 4.75 bar. Finally air is discharged from HP cylinder at 15 bar. The index of compression and expansion in both cylinders is 1.3.* $C_p = 1.005$ *kJ/kg·K, and* $R = 0.287$ *kJ/kg·K.*

Calculate

(a) Heat rejected in the intercooler,
(b) Diameter of HP cylinder, and
(c) Power required to drive HP cylinder.

Solution

Given Two-stage, double-acting reciprocating air compressor

$k = 2$	$N = 300\,\text{rpm}$
$p_1 = 1\,\text{bar} = 100\,\text{kPa}$	$T_1 = 27°\text{C} = 300\,\text{K}$
$p_2 = 5\,\text{bar}$	$p_5 = 4.75\,\text{bar}$
$T_3 = 27°\text{C} = 300\,\text{K}$	$p_6 = 15\,\text{bar}$
$n = 1.3$	$d_1 = 360\,\text{mm}$
$L_1 = 400\,\text{mm}$	$c_1 = c_2 = 0.04$
$C_p = 1.005\,\text{kJ/kg}\cdot\text{K}$	$R = 0.287\,\text{kJ/kg}\cdot\text{K}$
$L_2 = L_1$	

To find

(i) Heat rejected in the intercooler,
(ii) Diameter of HP cylinder, and
(iii) Power required to drive HP cylinder.

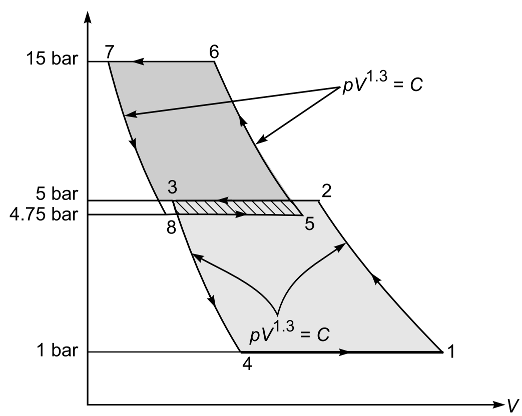


Fig. 5.29

Analysis The stroke (swept) volume per minute of *LP* cylinder

$$\dot{V}_{s,LP} = \dot{V}_1 - \dot{V}_3 = \left(\frac{\pi}{4}\right) d_1^2 L_1 N k$$

$$= \left(\frac{\pi}{4}\right) \times (0.36\ \text{m})^2 \times (0.4\ \text{m}) \times 300 \times 2$$

$$= 24.42\ \text{m}^3/\text{min}$$

The pressure ratio for LP cylinder;

$$\frac{p_2}{p_1} = 5$$

The volumetric efficiency of LP cylinder;

$$\eta_{vol,LP} = 1 + c_1 - c_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{5}{1}\right)^{\frac{1}{1.3}} = 0.902$$

It is also expressed using effective stroke volume rate $(\dot{V}_1 - \dot{V}_4)$ of LP cylinder as

$$\eta_{vol,\,LP} = \frac{\dot{V}_1 - \dot{V}_4}{\dot{V}_{s,\,LP}}$$

or $\dot{V}_1 - \dot{V}_4 = \dot{V}_{s,LP} \times \eta_{vol,LP}$

$$= (24.42\,\text{kg/min}) \times 0.902 = 22.02\ \text{m}^3/\text{min}$$

The mass flow rate of air into the LP cylinder;

$$\dot{m}_a = \frac{p_1(\dot{V}_1 - \dot{V}_4)}{RT_1} = \frac{(100\ \text{kPa}) \times (22.02\ \text{m}^3/\text{min})}{(0.287\ \text{kJ/kg}\cdot\text{K}) \times (300\ \text{K})}$$

$$= 25.584\ \text{kg/min} \quad \text{or} \quad 0.426\ \text{kg/s}$$

Temperature after compression in each stage

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 300 \times \left(\frac{5}{1}\right)^{\frac{1.3-1}{1.3}}$$

$$= 434.93\,\text{K}$$

(i) *Heat transfer rate in the intercooler*

$$\dot{Q} = \dot{m}_a C_p (T_2 - T_1)$$

$$= 25.584 \times 1.005 \times (434.93 - 300)$$

$$= 3469.33\,\text{kJ/min} \quad \text{or} \quad \mathbf{57.82\,kW}$$

(ii) *Diameter of HP cylinder*

The effective swept volume rate of HP cylinder

$$\dot{V}_5 - \dot{V}_8 = \frac{\dot{m}_a R T_1}{p_5}$$

$$= \frac{25.584 \times 0.287 \times 300}{475}$$

$$= 4.637\,\text{m}^3/\text{min}$$

The volumetric efficiency of HP cylinder;

$$\eta_{vol,HP} = 1 + c_2 - c_2 \left(\frac{p_3}{p_5}\right)^{\frac{1}{n}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{15}{4.75}\right)^{\frac{1}{1.3}} = 0.943$$

Further, the volumetric efficiency can be expressed in terms of piston displacement volume rate of HP cylinder as

$$\eta_{vol,\,HP} = \frac{\dot{V}_5 - \dot{V}_8}{\dot{V}_{s,\,HP}}$$

or $\dot{V}_{s,HP} = \dfrac{\dot{V}_5 - \dot{V}_8}{\eta_{vol,HP}} = \dfrac{4.637}{0.943} = 4.917\,\text{m}^3/\text{min}$

which can be further, expressed as

$$\dot{V}_{s,HP} = \left(\frac{\pi}{4}\right) d_2^{\,2} L_1 N k$$

or $4.917 = \left(\dfrac{\pi}{4}\right) \times d_2^2 \times 0.4 \times 300 \times 2$

or $d_2 = 0.1615\,\text{m}$ or **161.5 mm**

(iii) *Power required to drive HP cylinder*

$$IP = \frac{n}{n-1} p_5(\dot{V}_5 - \dot{V}_8)\left[\left(\frac{p_3}{p_5}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times 4.75 \times 10^2 \times \left(\frac{4.637}{60}\right) \times \left[\left(\frac{15}{4.75}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= \mathbf{48.34\,kW}$$

Example 5.33 *A three-stage, double-acting, reciprocating air compressor operating at 300 rpm, receives air at 1 bar and 27°C. The bore of LP cylinder is 360 mm and its stroke is 400 mm. Intermediate cylinder and HP cylinder have same stroke as LP cylinder. The clearance volume in each cylinder is 4 % of the stroke volume . The LP cylinder dischrges air at a pressure of 5 bar, the intermediate cylinder discharges at 20 bar and air is finally discharged by the HP cylinder at 75 bar. The air is cooled in intercoolers to initial temperature after each stage of compression. A Pressure drop of 0.2 bar takes place in intercooler after each stage. The index of compression and expansion for an LP cylinder is 1.3, for intermediate cylinder is 1.32 and for HP cylinder is 1.35. Neglect the effect of piston rod and assume C_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.*

Calculate

(a) Heat rejected in each stages in intercooler and during compression,

(b) Heat rejected in after-cooler, if delivered air is cooled to initial temperature,

(c) Diameter of intermediate and HP cylinders, and

(d) Power required to drive compressor, if its mechanical efficiency is 85%.

Solution

Given Three-stage, double-acting reciprocating air compressor

$k = 2$	$N = 300$ rpm
$p_1 = 1$ bar $= 100$ kPa	$T_1 = 27°C = 300$ K
$p_2 = 5$ bar	$p_5 = 4.8$ bar
$T_3 = 27°C = 300$ K	$p_6 = 20$ bar
$p_9 = 19.8$ bar	$p_{10} = 75$ bar
$n_1 = 1.3$	$n_2 = 1.32$
$n_3 = 1.35$	$d_1 = 360$ mm
$L_1 = 400$ mm	$c_1 = c_2 = c_3 = 0.04$
$C_p = 1.005$ kJ/kg·K	$R = 0.287$ kJ/kg·K
$L_3 = L_2 = L_1$	$\eta_{mech} = 0.85$

To find

(i) Heat rejected in each stage in intercooler and during compression,

(ii) Heat rejected in after-cooler,

(iii) Diameter of intermediate and HP cylinders, and

(iv) Power required to drive compressor.

Analysis The stroke (swept) volume per minute of *LP* cylinder

$$\dot{V}_{s,LP} = \dot{V}_1 - \dot{V}_3 = \left(\frac{\pi}{4}\right) d_1^2 L_1 N k$$

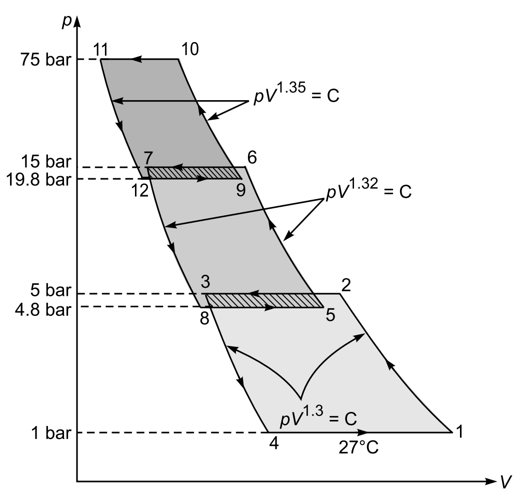


Fig. 5.30

$$= \left(\frac{\pi}{4}\right) \times (0.36 \text{ m})^2 \times (0.4 \text{ m}) \times 300 \times 2$$

$$= 24.42 \text{ m}^3/\text{min}$$

The volumetric efficiencies of *LP*, intermediate and *HP* cylinders;

$$\eta_{vol,LP} = 1 + c_1 - c_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n_1}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{5}{1}\right)^{\frac{1}{1.3}} = 0.902$$

For *IP* cylinder

$$\eta_{vol,IP} = 1 + c_2 - c_2 \left(\frac{p_6}{p_5}\right)^{\frac{1}{n_2}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{20}{4.8}\right)^{\frac{1}{1.32}} = 0.922$$

For *HP* cylinder

$$\eta_{vol,IP} = 1 + c_3 - c_3 \left(\frac{p_{10}}{p_9}\right)^{\frac{1}{n_3}}$$

$$= 1 + 0.04 - 0.04 \times \left(\frac{75}{19.8}\right)^{\frac{1}{1.35}} = 0.9327$$

The effective stroke volume $(\dot{V}_1 - \dot{V}_4)$ of air per minute in *LP* cylinder;

$$\dot{V}_1 - \dot{V}_4 = \dot{V}_{s,LP} \times \eta_{vol,LP}$$

$$= (24.42 \text{ m}^3/\text{min}) \times 0.902 = 22.02 \text{ m}^3/\text{min}$$

The mass flow rate of air in the compressor;

$$\dot{m}_a = \frac{p_1(\dot{V}_1 - \dot{V}_4)}{RT_1} = \frac{(100 \text{ kPa}) \times (22.02 \text{ m}^3/\text{min})}{(0.287 \text{ kJ/kg} \cdot \text{K}) \times (300 \text{ K})}$$

$$= 25.584 \text{ kg/min} \quad \text{or} \quad 0.426 \text{ kg/s}$$

Temperature after compression in each stage

After first stage,

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n_1 - 1}{n_1}}$$

$$= 300 \times \left(\frac{5}{1}\right)^{\frac{1.3-1}{1.3}} = 434.93 \text{ K}$$

After second stage,

$$T_6 = T_5\left(\frac{p_6}{p_5}\right)^{\frac{n_2-1}{n_2}}$$

$$= 300\times\left(\frac{20}{4.8}\right)^{\frac{1.32-1}{1.32}} = 424.00\,\text{K}$$

After third stage

$$T_{10} = T_9\left(\frac{p_{10}}{p_9}\right)^{\frac{n_3-1}{n_3}}$$

$$= 300\times\left(\frac{75}{19.8}\right)^{\frac{1.35-1}{1.35}} = 423.71\,\text{K}$$

Specific heat at constant volume;

$$C_v = \frac{R}{\gamma-1} = \frac{0.287}{1.4-1} = 0.717\text{ kJ/kg}\cdot\text{K}$$

(i) *Heat rejection rate in each stage*

$$\dot{Q}_{comp} = \dot{m}_a\frac{(\gamma-n)}{n-1}C_v\,(T_2-T_1)$$

$$\dot{Q}_{intercooler} = \dot{m}_a C_p\,(T_2-T_1)$$

Heat rejcetion rate in **first** stage (during compression and in intercooler);

$$\dot{Q}_1 = \dot{m}_a\left[\frac{(\gamma-n_1)}{n_1-1}C_v+C_p\right](T_2-T_1)$$

$$= 25.584\times\left[\frac{(1.4-1.3)}{1.3-1}\times 0.717+1.005\right]\times(434.93-300)$$

$$\mathbf{= 4295\ kJ/min}$$

Heat rejcetion rate in **second** stage (during compression and in intercooler);

$$\dot{Q}_2 = \dot{m}_a\left[\frac{(\gamma-n_2)}{n_2-1}C_v+C_p\right](T_6-T_5)$$

$$= 25.584\times\left[\frac{(1.4-1.32)}{1.32-1}\times 0.717+1.005\right]\times(424-300)$$

$$\mathbf{= 3757\,kJ/min}$$

Heat rejction rate in **third** stage compression;

$$\dot{Q}_3 = \dot{m}_a\left(\frac{\gamma-n_3}{n_3-1}\right)C_v\,(T_{10}-T_9)$$

$$= 25.584\times\left(\frac{1.4-1.35}{1.35-1}\right)\times 0.717\times(423.71-300)$$

$$\mathbf{= 324.18\,kJ/min}$$

Heat rejection rate in after-cooler

$$\dot{Q}_3 = \dot{m}_a C_p\,(T_{10}-T_1)$$

$$= 25.584\times 1.005\times(423.71-300)$$

$$\mathbf{= 3180.82\,kJ/min}$$

(ii) *Diameter of IP and HP cylinders*

The effective swept volume rate of *IP* cylinder

$$\dot{V}_5-\dot{V}_8 = \frac{\dot{m}_a RT_1}{p_5}$$

$$= \frac{25.584\times 0.287\times 300\text{ K}}{480}$$

$$= 4.588\,\text{m}^3/\text{min}$$

Further, the volumetric efficiency can be expressed using piston displacement volume rate of *IP* cylinder as

$$\eta_{vol,IP} = \frac{\dot{V}_5-\dot{V}_8}{\dot{V}_{s,\,IP}}$$

$$\therefore\ \dot{V}_{s,IP} = \frac{\dot{V}_5-\dot{V}_8}{\eta_{vol,IP}} = \frac{4.588}{0.922} = 4.976\text{ m}^3/\text{min}$$

which can be further, expressed as

$$\dot{V}_{s,IP} = \left(\frac{\pi}{4}\right)d_2{}^2 L_1 N k$$

$$\text{or } 4.976 = \left(\frac{\pi}{4}\right)\times d_2^2\times 0.4\times 300\times 2$$

or $d_2 = 0.1624$ m or **162.4 mm**

Similarly, for *HP* cylinder;

The effective swept volume rate of *HP* cylinder

$$\dot{V}_9-\dot{V}_{12} = \frac{\dot{m}_a RT_9}{p_9}$$

$$= \frac{25.584\times 0.287\times 300\text{ K}}{1980}$$

$$= 1.1125\,\text{m}^3/\text{min}$$

$$\dot{V}_{s,HP} = \frac{\dot{V}_9-\dot{V}_{12}}{\eta_{vol,HP}} = \frac{1.1125}{0.9327} = 1.192\text{ m}^3/\text{min}$$

which can be further, expressed as

$$1.192 = \left(\frac{\pi}{4}\right)\times d_3^2\times 0.4\times 300\times 2$$

or $d_3 = 0.07954$ m or **79.54 mm**

The power input to compressor

Since $T_1 = T_5 = T_9$,

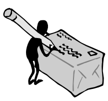
$$IP = \dot{m}_a RT_1\left[\frac{n_1}{n_1-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n_1-1}{n_1}}-1\right\}+\frac{n_2}{n_2-1}\left\{\left(\frac{p_6}{p_5}\right)^{\frac{n_2-1}{n_2}}-1\right\}+\frac{n_3}{n_3-1}\left\{\left(\frac{p_{10}}{p_9}\right)^{\frac{n_3-1}{n_3}}-1\right\}\right]$$

$$= 25.584 \times 0.287 \times 300$$

$$\times\left[\frac{1.3}{1.3-1}\left\{\left(\frac{5}{1}\right)^{\frac{1.3-1}{1.3}}-1\right\}+\frac{1.32}{1.32-1}\left\{\left(\frac{20}{4.8}\right)^{\frac{1.32-1}{1.32}}-1\right\}+\frac{1.35}{1.35-1}\left\{\left(\frac{75}{19.8}\right)^{\frac{1.35-1}{1.35}}-1\right\}\right]$$

$$= 2202.78 \times (1.949 + 1.705 + 1.59)$$

$= 11{,}551.4$ kJ/min or **192.34 kW**



Summary

- An air compressor is a machine that decreases the volume and increases the pressure of a quantity of air by mechanical means. The reciprocating compressor handles a small quantity of gas and produces very high pressure, while rotary compressors are used to handle a large volume of gas and to produce low and medium pressures.
- The volume flow rate to a single-acting, single-cylinder reciprocating compressor is

$$\dot{V} = \frac{\pi}{4}d^2L\frac{N}{60}$$

- For the *double-acting reciprocating compressor*, the induction takes place on both sides of the piston for each revolution. Thus

$$\dot{V} = \frac{\pi}{4}d^2L\left(\frac{2N}{60}\right)$$

where N is the speed of compressor in rotations per minute.

- *The capacity of a compressor* is the actual quantity of air delivered per unit time at atmospheric conditions. *Free Air delivery* (*FAD*) is the discharge volume of the compressor corresponding to ambient conditions.
- Piston speed is the linear speed of the piston measured in m/min. It is expressed as

$$v_{piston} = 2LN$$

- The indicated work input to a single-stage, single-acting reciprocating compressor without clearance is

$$W_{in} = \frac{n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right] \text{ (kJ/cycle)}$$

- The clearance volume is provided in the cylinder to accomodate valves. The clearance ratio c is the ratio of clearance volume to the swept volume. The clearance ratio for a reciprocating air compressor is usually 2 to 10%. The work input to the compressor with clearance ratio c is

$$W_{in} = \frac{n_c}{n_c-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n_c-1}{n_c}}-1\right] - \frac{n_e}{n_e-1}p_1V_4\left[\left(\frac{p_2}{p_1}\right)^{\frac{n_e-1}{n_e}}-1\right]$$

where n_c = index of compression and n_e = index of expansion.

If both indices are same, i.e., $n_c = n_e$ then

$$W_{in} = \frac{n}{n-1}p_1(V_1 - V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]$$

- Isothermal efficiency of a compressor is defined as the ratio of isothermal work input to actual work input.
- The indicated power required by an air compressor is given as

$$IP = W_{in} \text{ per cycle} \times \text{No of compression per unit time} = W_{in}\left(\frac{Nk}{60}\right)$$

- From an indicated diagram, the indicated power is obtained in terms of indicated mean effective pressure, p_m as

$$IP = \frac{p_m \, L \, A \, N \, k}{60} \text{ (kW)}$$

where $k = 1$ for single acting and $k = 2$ for double acting reciprocating compressor.

- The mechanical efficiency of the compressor is given by

$$\eta_{mech} = \frac{\text{Indicated power}}{\text{Brake power}}$$

- The brake power is derived from a driving motor or engine. The input of the driving motor can be expressed as

Motor power

$$= \frac{\text{Shaft power (or Brake power)}}{\text{Mechanical efficiency of motor and drive}}$$

- The volumetric efficiency of a reciprocating air compressor is

$$\eta_{vol} = \frac{\text{Actual Mass sucked}}{\text{Mass corresponds to swept volume at atmospheric pressure and temperature}}$$

$$= 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}$$

- Multistage compression with intercooling reduces the compression work.
- For a two-stage reciprocating air compressor with intercooler, the work input per cycle is

$$W_{in} = \frac{n}{n-1} m_a R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right]$$

where p_2 and T_2 are intermediate pressure and temperature, respectively.

- The heat rejected with intercooler is

$$Q_{cooling} = m_a C_p (T_2 - T_1) \text{ (kJ)}$$

and heat rejected during polytropic compression is

$$Q_{comp} = m_a \frac{(\gamma - n)}{n - 1} C_v (T_2 - T_1)$$

- The compression work in a reciprocating compressor would be minimum when a stage pressure ratio is

$$\frac{p_2}{p_1} = \frac{p_3}{p_2}$$

and the minimum compression power for a two-stage compressor

$$IP_{min} = 2 \times \left(\frac{n}{n-1}\right) \dot{m}_a R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Glossary

Reciprocating compressor A reciprocating machine, used to compress the air during each stroke of piston

Rotary compressor A machine which compresses the air by dynamic action

Single-acting compressor A compressor in which all actions take place only one side of the piston during a cycle

Double-acting compressor A compressor in which suction, compression and delivery of gas take place on both sides of the piston

Single-stage compressor A compressor in which the compression of gas to final delivery pressure is carried out in one cylinder only

Multistage compressor A compressor which compresses the gas to the final pressure in more than one cylinder in series

Pressure ratio The ratio of absolute discharge pressure to absolute suction pressure

Free air Air that exists under atmospheric condition

Free Air delivery (FAD) Discharge volume of compressor corresponding to ambient conditions

Compressor Capacity Quantity of air delivered per unit time at atmospheric conditions

Inter-stage coolers Used to cool the air in between stages of compression

After coolers Used to remove the moisture in the air by cooling it

Air-dryers Removes traces of moisture after after-cooler is used

Moisture drain traps used for removal of moisture in the compressed air

Air receiver Storage tank used to store the compressed air

Review Questions

1. What is an air compressor? Why is it an important machine?
2. Write the uses of compressed air.
3. Classify the air compressors.
4. How do the suction and delivery valve activate in reciprocating air compressor?
5. State the main parts of reciprocating air compressor.
6. Differentiate between
   (i) Single-acting and double-acting compressors
   (ii) Single-stage and multistage compressors.
7. Why is a cooling arrangement provided with all compressors?
8. Define swept volume, and deduce it for single-cylinder, single-acting and double-acting compressor having bore d, stroke L, and speed N rpm.
9. Write the construction of a single-acting, single-stage reciprocating air compressor.
10. Explain the working of a single-acting reciprocating air compressor.
11. Explain the working of double-acting reciprocating air compressor.
12. Derive an expression for indicated work of a reciprocating air compressor by neglecting clearance.
13. Why is the clearance volume provided in each reciprocating compressor? Is it desirable to have a high clearance volume in a compressor?
14. What is clearance ratio? Write the effect of clearance volume on performance of a reciprocating compressor.
15. Derive an expression for indicated work of a reciprocating air compressor by considering its clearance.
16. Define volumetric efficiency and prove that
$$\eta_{vol} = 1 + c - c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n_e}}$$
where each term has its usual meaning.
17. Define volumetric efficiency. How is it affected by (i) pressure ratio, (ii) speed of compressor, and (iii) throttling across the valves? Explain in brief.
18. What are the advantages of multistage compression over single-stage compression?
19. Why is the intercooler provided between stages?
20. Prove that in a reciprocating air compressor, with perfect intercooling, the work done for compressing the air is equal to heat rejected by the air.
21. What is an after-cooler? Why is it provided with an air compressor.
22. Prove that for complete intercooling between two stages, the compression work would be minimum when intermediate pressure
$$p_2 = \sqrt{p_1 \times p_3}$$
23. Define overall volumetric efficiency. Discuss the parameters in brief, which affect it.
24. Show the effect of increase in compression ratio in single-stage reciprocating compressor on a p–V diagram and give its physical explanation.
25. Draw the indicator diagram for single-stage, double-acting reciprocating air compressor on a p–V diagram.
26. What are the advantages of using an after-cooler with an air compressor, when air under pressure has to be stored over long periods?
27. What is the effect of intake temperature and pressure on output of an air compressor?

Problems

1. A single acting reciprocating air compressor delivers air at 70 bar from an induction pressure of 1 bar. The rate of air compression is 2.4 m^3/min. measured at free air condition of 1.013 bar and 15°C. The temperature at the end of induction stroke is 32°C. Calculate the indicated power input, if compression is carried out in two stages with an ideal intermediate pressure and perfect intercooling and the index of compression and expansion for both stages is 1.25.

[1282.2 kW]

2. Calculate the bore of the cylinder for a double-acting, single-stage reciprocating air compressor runs at 100 rpm with average piston speed of 150 m/min. The indicated power input is 50 kW. It receives air at 1 bar and 15°C and compresses it according to $pV^{1.2}$ = constant to 6 bar.

[349 mm]

3. A single-acting, single-cylinder reciprocating air compressor has a cylinder diameter of 300 mm and a stroke of 400 mm. It runs at 100 rpm. Air enters the cylinder at 1 bar; 20°C. It is then compressed to 5 bar. Calculate the mean effective pressure and indicated power input to compressor, when compression is
   (a) isothermal,
   (b) according to the law $pV^{1.2}$ = constant,
   (c) adiabatic.

   Calculate isothermal efficiency for each case. Neglect clearance.

[(a) 1.61 bar, 7.58 kW, 100% (b) 1.85 bar, 8.7 kW, 87.2% (c) 2.043 bar, 9.63 kW, 78.8%]

4. An air compressor takes in air at 100 kPa, 300 K. The air delivers at 400 kPa, 200°C at the rate of 2 kg/s. Determine minimum compressor work input. *[312.7 kW]*

5. A single-acting, single-cylinder reciprocating air compressor receives 30 m^3 of atmospheric air per hour at 1 bar and 15°C . It runs at 450 rpm and discharges air at 6.5 bar. It has a mechanical efficiency of 80% and a clearance ratio of 8.9%. Calculate
   (a) the volumetric efficiency,
   (b) mean effcetive pressure,
   (c) brake power.

[(a) 75% (b) 1.06 bar (c) 1.48 kW]

6. Calculate the power required to drive a single-stage, single-acting reciprocating air compressor to compress 8 m^3/min of air, receiving at 1 bar, 20°C to 7 bar. The index of compression is 1.3.

   Also, calculate the percentage saving in indicated power by compressing the same mass of air
   (a) in two stages with optimum intercooler pressure and perfect intercooling,
   (b) in two stages with imperfect intercooling to 27°C, intercooler pressure remaining the same as in case in (a),
   (c) in three stages with optimum intercooler and perfect cooling.

[32.8 kW (a) 11.3% (b) 10.2% (c) 14.63%]

7. A power cylinder of 0.5 m^3 capacity is charged with compressed air without after-cooling it at 170 bar from a four-stage compressor with perfect intercooling between stages and working in best conditions. What are the most economical intermediate pressure?

[3.611bar 13.04 bar and 47.08 bar]

8. The free air delivered by a single-stage, double-acting reciprocating air compressor, measured at 1 bar and 15°C is 16 m^3/min. The suction takes place at 96 kPa and 30°C and delivery pressure is 6 bar. The clearance volume is 4% of swept volume and mean piston speed is limited to 300 m/min. Determine
   (a) power input to compressor, if mechanical efficiency is 90% and compression efficiency is 85%
   (b) Bore and stroke if compressor runs at 500 rpm

   Assume index of compression and expansion = 1.3.

[(a) 83.6 kW (b) 290 mm and 300 mm]

9. A double-acting, single-stage reciprocating air compressor has a bore of 330 mm, stroke of 350 mm, clearance of 5%, and runs at 300 rpm. It receives air at 95 kPa and 25°C . The delivery pressure is 4.5 bar and the index of compression is 1.25. The free air conditions are 1.013 bar and 20°C. Determine
   (a) FAD,
   (b) heat rejected during the compression, and
   (c) power input to compressor, if its mechanical efficiency is 80%.

[(a) 14.51 m^3/min (b) 817.4 kJ/min. (c) 56.82 kW]

10. A Four cylinder, double acting reciprocating air compressor is used to compress 30 m^3/min of air at 1 bar and 27°C to a pressure of 16 bar. Calculate the size of motor required and cylinder dimensions for the following data:

speed of compressor = 320 rpm, clearance ratio 4%, stroke to bore ratio 1.2, η_{mech} = 82%, index of compression and expansion, n = 1.3.

Assume air gets healed by 12°C during suction.

[241 kW, 263 mm, 315.6 mm]

11. A single-acting, single-cylinder air compressor running at 300 rev/min is driven by an electric motor. Using the data given below, and assuming that the bore is equal to the stroke, calculate

(a) free air delivery,
(b) volumetric efficiency,
(c) bore and stroke.

Data: Air inlet conditions = 1.013 bar and 15°C; delivery pressure = 8 bar; clearance volume = 7% of swept volume; index of compression and re-expansion = 1.3; mechanical efficiency of the drive between motor and compressor = 87%; motor power output = 23 kW

(4.47 m^3/min;72.7% 297 mm)

12. The LP cylinder of a two-stage, double-acting reciprocating air compressor running at 120 rpm has a 50-cm diameter and 75-mm stroke. It receives air at 1 bar and 20°C and compresses it adiabatically to 3 bar. Air is then delivered to an intercooler, where it is cooled at constant pressure to 35°C and then furter compressed to 10 bar in HP cylinder. Determine the power required of an electric motor to drive a compressor. Assume the mechanical efficiency of the compressor as 90% and of the motor as 86%. *[212.9 kW]*

13. A reciprocating air compressor takes in air at 40°C and 1.013 bar in the daytime.

(a) Calculate the percentage increase of mass output in the night, if the night temperature is 10°C.
(b) If the compressor is shifted to a hill station, where the barometric pressure is 0.92 bar, calculate percentage decrease in output, assuming suction temperature to be same at two places.
(c) Calculate the pressure ratio of the compressor at two places, if the law of compression is $pV^{1.25}$ = constant, if delivery gauge pressure is 7 bar at both places.

[(a) 10.61% (b) 9.18% (c) at first place 4.81% and second place 5.24]

14. A three-stage single-acting reciprocating air compressor has perfect intercooling. The pressure and temperature at the end of the suction stroke in an LP cylinder are 1.013 bar and 15°C, respetively. If 8.4 m^3 of free air is delivered by the compressor at 70 bar per minute and work done is minimum, calculate

(a) LP and IP cylinder delivery pressures,
(b) ratio of cylinder volumes,
(c) total indicated power.

Neglect clearance and assume n = 1.2.

[(a)4.16 bar, 17.05 bar (b) 2.02 (c) 676.8 kW]

Objective Questions

1. For isothermal compression in a compressor, the compressor should run at
   (a) very high speed (b) very slow speed
   (c) constant speed (d) none of above
2. A reciprocating compressor handles
   (a) large volume for high pressure ratio
   (b) large volume for low pressure ratio
   (c) small volume for high pressure ratio
   (d) small volume for low pressure ratio
3. Usually, the index of actual compression is
   (a) near to 1 (b) 1.3 to 1.4
   (c) 1.1 to 1.3 (d) 1.4 to 1.6
4. Which of the following process takes place in an air compressor?
   (a) Specific volume of air decreases
   (b) Pressure of air increases
   (c) Mechanical energy is supplied
   (d) All of above
5. For which one of the following applications, the compressed air is not used?
   (a) Driving air motors
   (b) Oil and gas transmission
   (c) Starting of I.C. engines
   (d) Transmission of electrical energy

6. Reciprocating compressor is
   (a) a positive displacement machine
   (b) a negative displacement machine
   (c) a dynamic action machine
   (d) none of above
7. Air dryers are used in an air compressor
   (a) before air entry into cylinder
   (b) before entering air receiver
   (c) between two stages
   (d) after leaving air receiver
8. Air receiver used in an air compressor is used to
   (a) cool the air after compression
   (b) eliminate the pulsation
   (c) supply the air to utility
   (d) to separate the moisture
9. In a reciprocating air compressor, inlet and delivery valves actuate
   (a) by separate cam mechanism
   (b) by pressure difference
   (c) by use of compressed air
   (d) none of the above
10. In a reciprocating air compressor, the work input is minimum when compression is
   (a) isentropic (b) polytropic
   (c) isothermal (d) isobaric
11. What is the sequence of processes in a reciprocating air compressor?
   (a) Compression, expansion, and constant volume discharge
   (b) Induction, compression and constant pressure discharge
   (c) Induction, expansion and constant pressure discharge
   (d) Induction, compression and constant volume discharge
12. Work input in a reciprocating air compressor is given by
   (a) $\dfrac{n-1}{n} p_1 v_1 \left[1+\left(\dfrac{p_2}{p_1}\right)^{\frac{n}{n-1}}\right]$
   (b) $\dfrac{n-1}{n} p_1 v_1 \left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n}{n-1}} -1\right]$
   (c) $\dfrac{n-1}{n} p_1 v_1 \left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} -1\right]$
   (d) $\dfrac{n-1}{n} p_1 v_1 \left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} +1\right]$
13. The isothermal efficiency of a reciprocating air compressor is given by
   (a) $\dfrac{\text{Indicated power}}{\text{Isothermal power}}$
   (b) $\dfrac{\text{Isothermal power}}{\text{Indicated power}}$
   (c) $\dfrac{\text{Isothermal power}}{\text{Brake power}}$
   (d) $\dfrac{\text{Brake power}}{\text{Isothermal power}}$
14. The compressor efficiency of a reciprocating air compressor is given by
   (a) $\dfrac{\text{Indicated power}}{\text{Isothermal power}}$
   (b) $\dfrac{\text{Isothermal power}}{\text{Indicated power}}$
   (c) $\dfrac{\text{Isothermal power}}{\text{Brake power}}$
   (d) $\dfrac{\text{Brake power}}{\text{Isothermal power}}$
15. The isothermal efficiency of a reciprocating air compressor can be improved by use of
   (a) water jacketing (b) external fins
   (c) intercooler (d) all of the above
16. The clearance volume in a reciprocating air compressor
   (a) reduces work input
   (b) reduces suction capacity
   (c) reduces discharge pressure
   (d) all of the above
17. The volumetric efficiency of a reciprocating air compressor is defined as

(a) $1+c-c\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

(b) $1-c+c\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

(c) $1+c-c\left(\dfrac{p_2}{p_1}\right)^{\frac{1}{n}}$

(d) $1-c+c\left(\dfrac{p_2}{p_1}\right)^{\frac{1}{n}}$

18. The maximum pressure ratio in a single-stage reciprocating air compressor is limited to
   (a) 2 (b) 4
   (c) 7 (d) 10
19. Multistage compression in a reciprocating air compressor improves
   (a) isothermal efficiency
   (b) volumetric efficiency
   (c) mechanical balance
   (d) all of above
20. Ideal intermediate pressure p_2 for two-stage reciprocating air compressor is given by
   (a) $p_1 \times p_3$ (b) $\sqrt{p_1 \times p_3}$
   (c) $\sqrt{\dfrac{p_3}{p_1}}$ (d) $\sqrt{\dfrac{p_3}{p_1}}$
21. Heat rejection rate per stage of air with perfect intercooler is given by
   (a) $\dot{m}_a\left[\dfrac{(\gamma-n)}{n-1}C_v+C_p\right](T_2-T_1)$
   (b) $\dot{m}_a\left[C_p-\dfrac{(\gamma-n)}{n-1}C_v\right](T_2-T_1)$
   (c) $\dot{m}_a\left[C_p-\dfrac{(n-1)}{\gamma-1}C_v\right](T_2-T_1)$
   (d) $\dot{m}_a\left[C_p+\dfrac{(n-1)}{\gamma-1}C_v\right](T_2-T_1)$

Answers

1. (b)	2. (c)	3. (c)	4. (d)	5. (d)	6. (a)	7. (b)	8. (b)
9. (b)	10. (c)	11. (b)	12. (c)	13. (c)	14. (b)	15. (d)	16. (b)
17. (c)	18. (c)	19. (d)	20. (b)	21. (a)			

6 CHAPTER

Rotary Compressor

Introduction

Rotary compressors are used to supply continuous pulsation free compressed air. They have rotor(s) and casing in place of piston cylinder arrangement. They are compact, well balanced, and high speed compressors. They have low starting torque thus they are directly coupled with prime-mover. They handle large mass of gas and are suitable for low and medium pressure ratios.

The special features of rotary compressors are:

- √ Designed to provide pulsation-free air,
- √ 100% continuous duty,
- √ Quiet operation,
- √ Energy efficient at full load,
- √ Extended service intervals,
- √ Reliable and long life,
- √ Improved air quality,
- √ Low starting torque.

Fans and blowers are also rotary machines, which are used for supplying air or gas. These machines are differentiated by the method used to move the air, and by the system pressure, at which they are operating. Normally, the ratio of the discharge pressure to the suction pressure is used to distinguish the fans, blowers and compressors as shown in Table 6.1.

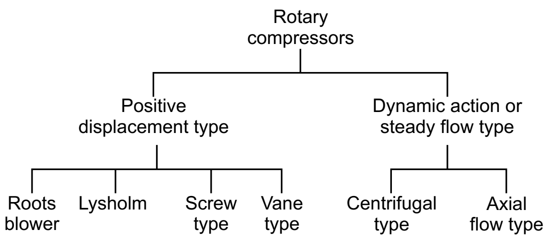
Table 6.1 *Difference between fans, blowers and compressors*

Equipment	*Pressure ratio*	*Pressure rise in mm of water*
Fan	up to 1.11	1136
Blower	1.11 to 1.20	1136 - 2066
Compressor	more than 1.20	more than 2066

Some compressors are suitable only for low pressure ratio work such as for scavenging and supercharging of engines and for various applications of exhausting and vacuum pumping. For a pressure ratio above 9 bar, vane-type rotary machines can be used to boost the pressure.

6.1 CLASSIFICATION OF ROTARY COMPRESSORS

The air compressors are either reciprocating type or rotary type. The rotary air compressors can broadly be classified as



In a rotary *positive displacement type* of air compressor, the air is compressed by being trapped in the reduced space formed by two sets of engaging surfaces. In a *non-positive displacement* or steady flow type of compressor, the air flows continuously and pressure is increased due to dynamic action.

The rotary compressors have adiabatic compression. They have high speed and no cooling arrangement is provided during compression. Comparison of reciprocating type and rotary type compressor is presented in Table 6.1.

Table 6.1 *Comparison between Reciprocating and Rotary Compressors*

S. No.	*Aspect*	*Reciprocating compressor*	*Rotary compressor*
1.	Pressure ratio	Discharge pressure of air is high. The pressure ratio per stage may be in order of 4 to 7.	Discharge pressure of air is low. The pressure ratio per stage is limited to 5.
2.	Handled volume	Quantity of air handled is low and is limited to 50 m^3/s.	Larger quantity of air can be handled and it is about 500 m^3/s or more.
3.	Speed of compressor	Low speed of compressor.	High speed of compressor.
4.	Ideal compression process	Isothermal process	Isentropic process
5.	Cooling arrangement	Effective cooling is required	No cooling is required
6.	Vibrational Problem	Due to reciprocating action, greater vibrational problem, the parts of machine are poorly balanced.	Rotary parts of machine, thus it has less vibrational problems. The machine parts are fairly balanced.
7.	Size	Size of compressor is bulky for given discharge volume.	Compressor size is small for given discharge volume.
8.	Air supply	Air supply is intermittent.	Air supply is steady and continuous.
9.	Purity of compressed air	Air delivered from the compressor is dirty, since it comes in contact with lubricating oil and cylinder surfaces.	Air delivered from the compressor is clean and free from dirt.
10.	Compression efficiency	Higher with pressure ratio more than 2.	Higher with compression ratio less than 2.
11.	Maintanance	Higher due to reciprocating parts.	Lower due to balanced rotary parts.
12.	Mechanical efficiency	Lower due to several sliding parts.	Higher due to less sliding parts.
13.	Lubrication	Complicated lubrication system.	Simple lubrication system.
14.	Initial cost	Higher.	Lower.
15.	Flexibility	Greater flexibility in capacity and pressure range.	No flexibility in capacity and pressure range.
16.	Suitability	For medium and high pressure ratio and low and medium gas volume.	For low and medium pressures and large volumes.

6.2 ROOTS BLOWER COMPRESSOR

Roots blower is a positive dispacement compressor. It is also called *lobe compressor.* The roots blower is essentially a low-pressure blower and is limited to a discharge pressure of 1 bar in single-stage design and up to 2.2 bar in a two-stage design. Its discharge capacity is limited to 1500 m^3/min and it can run up to 7000 rpm.

Construction This type of rotary compressor consists of two or more lobed rotors in the casing with inlet and outlet passage of air. The lobed rotors rotate in an air tight casing with the help of gears in external housing. The compressor inlet is open to atmospheric air at one side and it is open to delivery side at the other side. The two lobed roots blower is shown in Fig. 6.1(a).

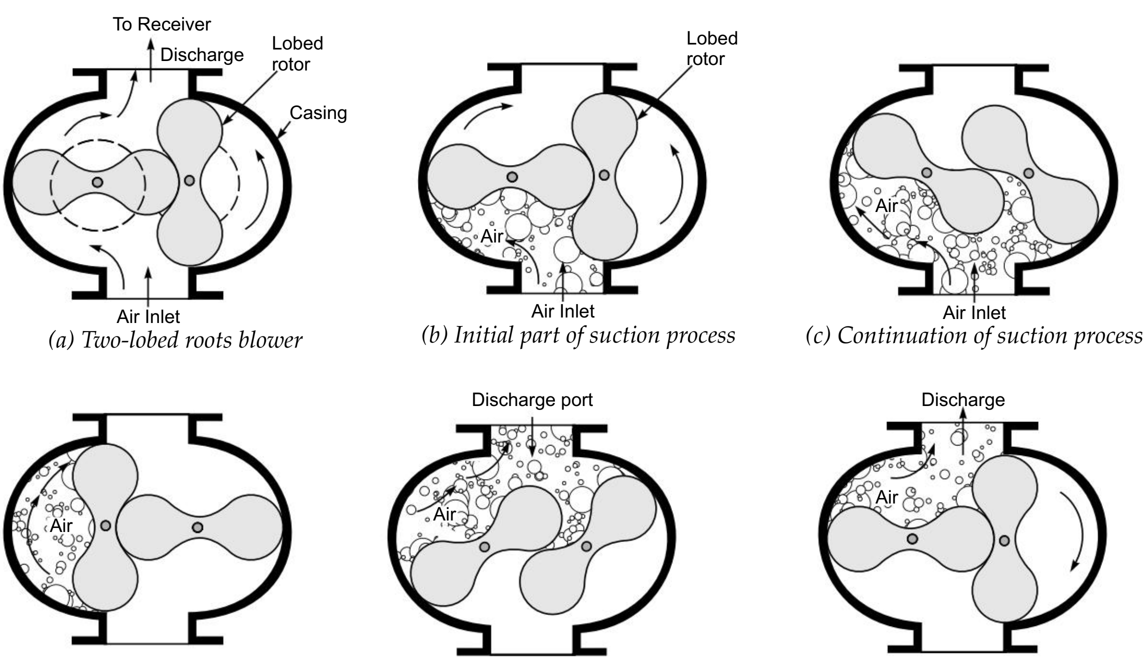
One lobed rotor is connected to drive. The second lobed rotor is gear driven from the first. Thus, both rotors rotate with the same speed. The profile of the lobes is made cycloidal or involute in order to seal the inlet side from the delivery side.

Working The rotation of rotors creates space in the casing at the entry port as shown in Fig. 6.1(b). The air is drawn into the casing to fill the space. The flow of gas in the casing space continues till both rotors change their position as shown in Fig. 26(c).

With further movement of the lobed rotor, the air is trapped between one rotor, when its tip tuches the casing as shown in Fig. 26(d). The air flows into the space created by rotation of other rotor. This rotor is also carrying out the same cycle as first rotor after 90°.

The trapped volume of air is not internally compressed, it is only displaced at high speed from suction side to delivery side. Continued rotation of lobes opens the discharge port as shown in Fig. 26(e).

Since the compressed air at higher pressure is present at the delivery side, when the rotor lobe uncovers the exit port, some pressurised air enters into the space between the rotor and casing of the compressor. This flow of air is called *back flow* of



(a) Two-lobed roots blower *(b) Initial part of suction process* *(c) Continuation of suction process*

(d) Compression process *(e) Start of discharge process* *(f) Completion of discharge process*

Fig. 6.1

air. This back flow of air continues until the pressure in the blower gets equalised. After back flow, the air is compressed irreversibly at constant volume. Finally, at higher pressure, the air is delivered from the blower to receiver as shown in Fig. 6.1(f). The process of compression can be represented by constant volume line on p–V plane as shown in Fig. 6.2.

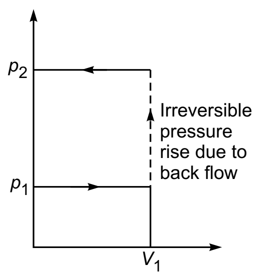


Fig. 6.2 *p-V diagram for roots blower*

Consider a volume of V_1 is trapped between the lobe and casing at atmospheric pressure p_1. The air is compressed to delivery pressure p_2. The actual work done on air;

$$W_{act} = -\int V dp = V_1(p_2 - p_1) \qquad ...(6.1)$$

The ideal work input for compression is isentropic work input. The theoretical work input to compress air from atmospheric pressure p_1 to delivery pressure p_2.

$$W_{isen} = \frac{\gamma}{\gamma - 1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} \qquad ...(6.2)$$

The efficiency of the roots blower can be defined as the ratio between adiabatic work to the actual work input. Mathematically;

$$\eta_{roots} = \frac{\text{Adiabatic work input}}{\text{Actual work input}}$$

$$= \frac{\frac{\gamma}{\gamma - 1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}}{V_1(p_2 - p_1)}$$

Let $\frac{p_2}{p_1} = r_p$, pressure ratio. Dividing numerator and denominator by $p_1 V_1$, we get

$$\eta_{roots} = \frac{\gamma}{\gamma - 1} \times \frac{\left\{ (r_p)^{\frac{\gamma-1}{\gamma}} - 1 \right\}}{(r_p - 1)} \qquad ...(6.3)$$

The efficiency of roots blower decreases with increase in pressure ratio. However, the compressor is suitable to give a pressure ratio between range of 1 to 2.2.

Applications

1. The root blowers are used for scavenging and supercharging of two-stroke internal combustion engines.
2. It is also used for low-pressure supply of air in steel furnaces, sewage disposal plants, low-pressure gas boosters, and for blower service in general.

Limitations

1. A significant limitation of root blower is its low pressure ratio due to the nature of the compression process.
2. The back flow of air in working chamber during its discharge.

Example 6.1 *A root blower compresses 1 m*^{3} *of air per second from a pressure of 1.01325 bar to 1.8 bar. Find the power required to run the compressor and its efficiency.*

Solution

Given A root blower with

$\dot{V}$ = 1 m^3/s
p_1 = 1.01325 bar = 101.325 kPa
p_2 = 1.8 bar = 180 kPa

To find

(i) Power required to run the compressor, and
(ii) Compressor efficiency.

Assumptions

(i) Compression of air at constant volume.
(ii) Compression is reversible.
(iii) Index of isentropic compression is 1.4.

Analysis

(i) The actual power required to run the compressor

$$IP = \dot{V}(p_2 - p_1)$$
$$= (1\ \text{m}^3/\text{s}) \times (180 - 101.325)\ (\text{kPa})$$
$$= \mathbf{78.675\ kW}$$

Ideal power required to compress the air

$$IP_{ise} = \frac{\gamma}{\gamma - 1} p_1 \dot{V}_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}$$

$$= \frac{1.4}{1.4 - 1} \times 101.325 \times 1 \left\{ \left(\frac{1.8}{1.01325} \right)^{\frac{1.4-1}{1.4}} - 1 \right\}$$

$$= 63.28\ \text{kW}$$

(ii) Compressor efficiency;

$$\eta_{roots} = \frac{\text{Adiabatic work input}}{\text{Actual work input}}$$

$$= \frac{63.28\ \text{kW}}{78.675\ \text{kW}} = 0.8043 \quad \text{or} \quad \mathbf{80.43\%}$$

Example 6.2 *1 kg of air per second is taken into a root blower compressor at 1 bar and 27°C. The delivery pressure of air is 1.5 bar. Calculate the motor power required to run the compressor, if mechanical efficiency is 80%.*

Solution

Given A root blower with

$\dot{m}$ = 1 kg/s $\quad$ p_1 = 1 bar = 100 kPa

T_1 = 27°C = 300 K $\quad$ p_2 = 1.5 bar = 150 kPa

η_{mech} = 0.8

To find Power required to run the compressor.

Assumptions

(i) Compression of air at constant volume.
(ii) Compression is reversible.
(iii) Specific gas constnat for air as 0.287 kJ/kg·K.

Analysis The volume flow rate of air taken into the compressor

$$\dot{V} = \frac{\dot{m} R T_1}{p_1}$$

$$= \frac{(1\ \text{kg/s}) \times (0.287\ \text{kJ/kg}\cdot\text{K}) \times (300\ \text{K})}{(100\ \text{kPa})}$$

$$= 0.861\ \text{m}^3/\text{s}$$

The power required to run the compressor

$$IP = \dot{V}(p_2 - p_1)$$
$$= (0.861\ \text{m}^3/\text{s}) \times (150 - 100)\ (\text{kPa})$$
$$= 43.05\ \text{kW}$$

The mechanical efficiency is given by

$$\eta_{mech} = \frac{IP}{BP}$$

or Brake power,

$$BP = \frac{IP}{\eta_{mech}} = \frac{43.05}{0.8} = \mathbf{53.81\ kW}$$

6.3 VANE-TYPE COMPRESSOR

Construction An arrangement of a typical vane-type compressor is shown in Fig. 6.3. It consists of an air-tight circular casing, in which a drum rotates about an ecentric centre of casing. The drum consists of a set of spring-loaded vanes. The slots are cut in the drum to accomodate the vanes. The drum rotates in anticlockwise direction. During the rotation of the drum, the vanes remain in contact with the casing. Size of inlet passage is larger than the size of outlet in the compressor.

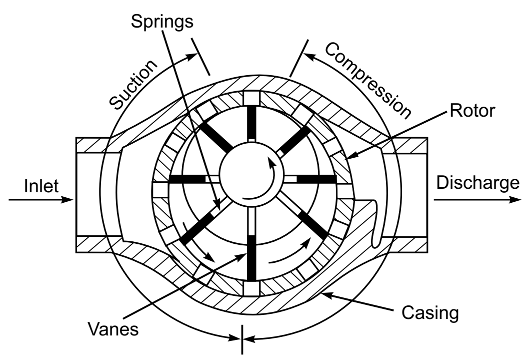


Fig. 6.3 *Vane-type compressor*

Working As the drum rotates, the volume of air V_1 at atmospheric pressure p_1 is trapped between the vanes, drum and casing. Air gets compressed due to two operations performed on air. First the compression begins due to decreasing volume between the drum and casing. The volume is reduced to V_2 and pressure increases to p_2. Secondly, the air is compressed due to back flow of compressed air in the receiver. Then the air is compressed at constant volume to a pressure p_3. The first part of compression follows adiabatic compression process and the second part follows constant-volume process. The process of compression is shown on the $p–V$ diagram in Fig. 6.4

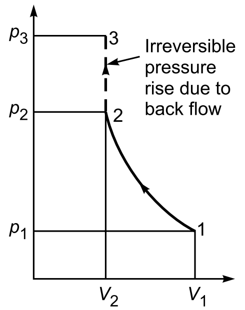


Fig. 6.4 *p–V diagram for Vane type compressor*

Work input for adiabatic compression;

$$W_{1-2} = \frac{\gamma}{\gamma-1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}$$

Work input for constant volume compression;

$$W_{2-3} = V_2(p_3 - p_2)$$

Total work input for compression within a vane;

$$W_{vane} = \frac{\gamma}{\gamma-1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} + V_2(p_3 - p_2)$$

If there are N vanes within the drum, then total work input;

$$W_{N\,vane} = N\left[\frac{\gamma}{\gamma-1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} + V_2(p_3 - p_2) \right] \quad ...(6.4)$$

The efficiency of a vane compressor can be expressed as

$$\eta_{vane,\,comp} = \frac{\text{Work input for constant-volume compression}}{\text{Total work input for compression}}$$

$$= \frac{V_2(p_3 - p_2)}{\frac{\gamma}{\gamma-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + V_2(p_3 - p_2)} \quad ...(6.5)$$

1. The vane-type compressor requires less work input compared to roots blower for same capacity and pressure ratio.
2. Vane-type compressors are commonly used to deliver air up to 150 m^3/min at a pressure ratio up to 8.5.
3. Vane-type compressors can run up to 3000 rpm.
4. Vane-type compressors are used for supercharging of IC engines and supply of air to cupola.
5. These are portable compressors used for construction purpose.

Example 6.3 *Calculate the power required to run the vane compressor and its efficiency, when it handles 6 m^3 of air per minute from 1 bar to 2.2. bar. The pressure rise due to compression in the compressor is limited to 1.6 bar. Take the mechanical efficiency of compressor as 80%.*

Solution

Given A vane compressor with

$\dot{V}$ = 6 m^3/min = 0.1 m^3/s p_1 = 1 bar = 100 kPa

p_2 = 1.6 bar = 160 kPa p_3 = 2.2 bar = 220 kPa

η_{mech} = 0.8

To find

(i) Brake power required to run the compressor,
(ii) Efficiency of the compressor.

Assumptions

(i) Compression is reversible.
(ii) Specific heat ratio for air as 1.4.

Analysis The isentropic compression power from state1 to state 2;

$$IP_1 = \frac{\gamma}{\gamma-1} p_1 \dot{V}_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}$$

$$= \frac{1.4}{1.4-1} \times (100 \text{ kPa}) \times (0.1 \text{ m}^3/\text{s}) \times \left\{ \left(\frac{1.6}{1} \right)^{\frac{1.4-1}{1.4}} - 1 \right\}$$

$= 5.03$ kW

The volume of air after compression to 1.6 bar;

$$\dot{V}_2 = \dot{V}_1 \left(\frac{p_1}{p_2} \right)^{\frac{1}{\gamma}} = (0.1 \text{ m}^3/\text{s}) \times \left(\frac{1}{1.6} \right)^{\frac{1}{1.4}}$$

$$= 0.07148 \text{ m}^3/\text{s}$$

Work done due to back flow from the state 2 to state 3

$$IP_2 = \dot{V}_2 (p_3 - p_2)$$

$$= (0.07148 \text{ m}^3/\text{s}) \times (220 - 160) \text{ (kPa)}$$

$$= 4.29 \text{ kW}$$

Total power input for compression of air

$$IP = IP_1 + IP_2 = 5.03 + 4.29 = 9.32 \text{ kW}$$

(i) The Brake power required to run the motor or shaft power

$$BP = \frac{IP}{\eta_{mech}} = \frac{9.32}{0.8} = \mathbf{11.65 \text{ kW}}$$

(ii) Efficiency of the vane compressor, Eq. (6.5);

$$\eta_{vane\ comp} = \frac{\text{Work input for constant volume compression}}{\text{Total work input for compression}}$$

$$= \frac{4.29 \text{ kW}}{9.32 \text{ kW}} = 0.46 \quad \text{or} \quad \mathbf{46\%}$$

Example 6.4 *Compare the work required for compression in a root blower and vane blower compressors for the following particulars:*

Intake volume = 0.05 m^3 per revolution

Inlet pressure = 1.013 bar

Pressure ratio = 1.5

For the vane-type compressor, internal compression takes place through half the pressure range.

Solution

Given Compression with

$V_1 = 0.05$ m^3/rev

$p_1 = 1.013$ bar $= 101.3$ kPa

Pressure ratio $= 1.5$

To find

(i) Work input to root blower and vane-type compressor.

Assumptions

(i) Compression is reversible.

(ii) Specific heat ratio for air as 1.4.

Analysis

(i) For a root blower compressor,

Pressure ratio, $\frac{p_2}{p_1} = 1.5$

or $p_2 = 1.5\, p_1 = 1.5 \times 1.013$

$= 1.5195$ bar or 151.95 kPa

Work input per revolution

$$W_{root} = V_1 (p_2 - p_1) = 0.05 \times (151.95 - 101.3)$$

$$= \mathbf{2.53 \text{ kJ}}$$

(ii) For vane-type compressor:

Pressure ratio, $\frac{p_3}{p_1} = 1.5$

or $p_3 = 1.5\, p_1$

$= 151.95$ kPa

Intermediate pressure;

$$p_2 = p_1 + 0.5\,(p_3 - p_1)$$

$$= 101.3 + 0.5 \times (151.95 - 101.3)$$

$$= 126.625 \text{ kPa}$$

The volume of air after isentropic compression

$$V_2 = V_1 \left(\frac{p_1}{p_2} \right)^{\frac{1}{\gamma}} = (0.05 \text{ m}^3) \times \left(\frac{101.3}{126.625} \right)^{\frac{1}{1.4}}$$

$$= 0.0426 \text{ m}^3/\text{rev}$$

Total work input per revolution for vane-type compressor

$$W_{Vane} = \frac{\gamma}{\gamma - 1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} + V_2 (p_3 - p_2) = 0$$

$$= \frac{1.4}{1.4-1} \times 101.3 \times 0.05 \times \left\{ \left(\frac{126.625}{101.3} \right)^{\frac{1.4-1}{1.4}} - 1 \right\} + 0.0426 \times (151.95 - 126.625)$$

$$= 1.167 + 1.079 = \mathbf{2.246 \text{ kJ/rev}}$$

Comment The vane-type compressor requires less work input for compression of given volume of air for same pressure ratio.

6.4 LYSHOLM COMPRESSOR—A SCREW COMPRESSOR

The *screw compressors* are most commonly used rotary air compressor. They are single stage helical or spiral lobe, oil flooded screw air compressor. They are simple in design and have few wearing parts, and thus easy to install, maintain and operate.

They are available in wide range of pressure ratio and capacity. Lubricated types are available in sizes ranging from 200 to 2000 m^3 per hour with discharge pressure up to 10 bar.

The oil free rotary screw air compressor uses specially designed machine. It compresses air without oil in the compression chamber. These compressors are air or water cooled machines. They deliver oil-free air up to 30,000 m^3 per hour and pressure upto 15 bar.

Alf Lysholm had developed the modern screw compressor, which is known as *Lysholm compressor*. Its construction and working are discussed below.

6.4.1 Construction

Screw compressors are equipped with two mating helical grooved rotors housed within a cylindrical casing equipped with inlet and discharge ports as shown in Fig. 6.5.

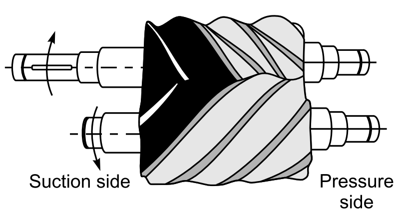


Fig. 6.5 *Screw compressor*

The main (male) rotor is normally driven by either an electric motor or an engine and transforms about 85–90% of the energy received at the coupling into pressure and heat energy. The number of lobes (or valleys) on the rotors will vary from one compressor manufacturer to another. Usually, the male rotor is made with four lobes along the length of the rotor that meshes with similarly formed correspondingly six helical flute on the auxiliary (female) rotor.

The auxiliary rotors seal the working space between the suction and pressure side. In the course of rotation, main and auxiliary rotors generate a *V*-shaped space for the air drawn in, which becomes smaller and smaller right up to the end, between the rotor lobes and the cylinder walls.

Because of the number of male lobes, there are four compression cycles per revolution which means that the resulting compressed air has small pulsations compared to a reciprocating compressor.

6.4.2 Working

During operation, the air is drawn through the inlet port to fill the space between the male lobe and female flute. As rotors continue to rotate, the air is moved past the suction port and sealed in the interlobe space. The trapped air is moved axially and radially and is compressed by direct volume reduction as enmeshing of lobes of compressor progressively reduces the space occupied by the gas with increase in pressure. Simultaneously, with this process, the oil is injected into the system. The oil seals the internal clearances and it absorbs the heat energy generated during compression. The compression of air continues until the interlobe space communicates with discharge port in the casing. The compressed air leaves the casing through the discharge port.

The working parts of compressor never get severe operating temperatures, since the cooling takes place during compression process. The oil is separated out in the oil separator and cooled down in an oil cooler and is returned back to compressor through an oil filter.

The *internal volume ratio* of a screw compressor is defined as the ratio of the volume of flute at the start of compression process to volume of the same flute as it begins to open to discharge port.

6.5 CENTRIFUGAL COMPRESSOR

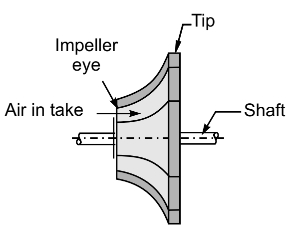
The centrifugal compressors are dynamic action compressors. These compressors have appreciably different characteristics as compared to reciprocating machines. A small change in compression ratio produces a marked change in compressor output and efficiency. Centrifugal machines are better suited for applications requiring very high capacities, typically above 3000 m^3/min and a moderate pressure ratio of 4 to 5. They are preferred due to their simplicity, light weight and ruggedness.

The centrifugal air compressor is an oil-free compressor by design. The oil-lubricated running gear is separated from the air by shaft seals and atmospheric vents. It is a continuous duty compressor, with few moving parts, and is particularly suited to high volume applications, especially where oil-free air is required.

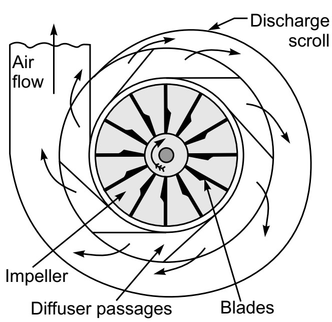
The centrifugal compressors find their applications in oil refineries, gas turbines, refrigeration and air conditioning, HVAC, turbochargers and superchargers of automobiles etc.

6.5.1 Construction

The basic components of a typical centrifugal compressor includes an impeller, diffuser and casing as shown in Fig. 6.6. The impeller is a radial disc with a series of radial blades (vanes). The



(a) Impeller eye



(b) Sectional view of centrifugal compressor

Fig. 6.6 *Centrifugal compressor*

impeller rotates inside the casing. The impeller is usually forged or die casting of aluminium alloy.

The centre of the impeller is called the *eye*. The eye of the impeller is connected with the drive shaft. The casing of the compressor has a volute shape. A diffuser ring is housed in the radial portion of the casing.

6.5.2 Working

As the impeller rotates, the air enters radially into the impeller eye with low velocity $\mathcal{V}_1$ at atmospheric pressure p_1. Due to centrifugal action of the impeller, the air comes radially out and during its movement, it is guided by the blades within the impeller.

The high velocity of the impeller increases the momentum of air, causing rise in static pressure, temperature and kinetic energy of air. The pressure, temperature and velocity of air leaving the impeller are p_2, T_2 and $\mathcal{V}_2$, respectively.

The air leaving the outside edge of the impeller enters into the diffuser ring where its kinetic energy is converted into pressure energy. Thus, the static pressure of air is further increased. The air is then collected in the volute casing and discharged from the compressor. About half of the pressure rise per stage is normally achieved in the rotating impeller and remaining half in the diffuser and volute casing. The change in pressure and velocity of air passing the impeller and diffuser passage are shown in Fig. 6.7.

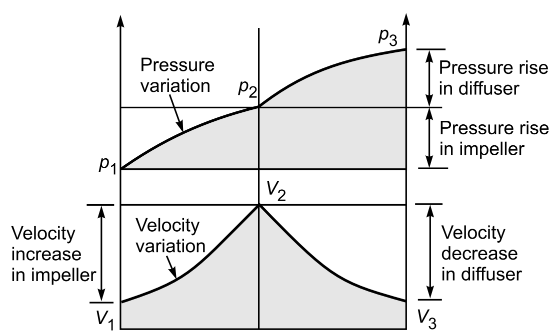


Fig. 6.7 *Velocity and pressure variation of air passing through impeller and diffuser*

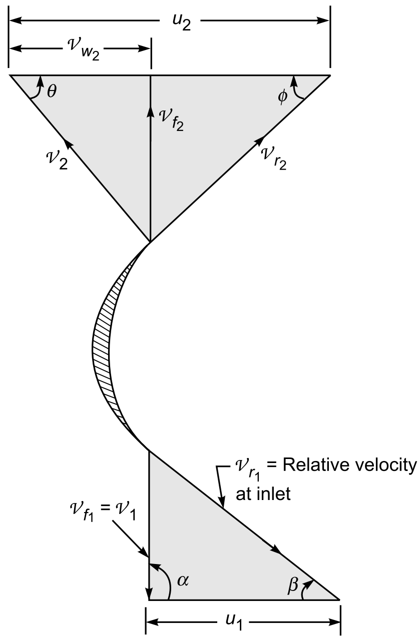


Fig. 6.8 *Velocity diagram at inlet and outlet of impeller blade*

6.5.3 Velocity Diagram

In a centrifugal air compressor, the air enters the eye axially and impeller radially. Hence the blades are designed in such a way that the air enters and leaves the blades without shock.

Using usual notation, let

u_1 = Blade velocity at inlet

$\mathcal{V}_{r_1}$ = Relative velocity of blades at inlet

$\mathcal{V}_1$ = Absolute velocity of inlet air

$\mathcal{V}_{f_1}$ = Inlet flow velocity

α = Air inlet angle

β = Blade angle at inlet

u_2, $\mathcal{V}_{r_2}$, $\mathcal{V}_2$, $\mathcal{V}_{f_2}$, θ, and ϕ are the corresponding values at outlet.

Work input per kg of air

$$w = u_1\mathcal{V}_{w_1} + u_2\mathcal{V}_{w_2}$$

Since the working fluid enters impeller radially, i.e., $\alpha = 90°$, thus $\mathcal{V}_{w_1} = 0$

Hence work input by blade per kg of air

$$w = u_2\mathcal{V}_{w_2} \text{ (J/kg)} \qquad \text{...(6.6)}$$

The above equation is known as *Euler's equation* or *Euler's work*.

If $\dot{m}$ is the mass flow rate of air in kg/s, then power input to compressor

$$P = \dot{m}\,u_2\mathcal{V}_{w_2} \text{ (Joules)} \qquad \text{...(6.7)}$$

6.5.4 Width of Blades of Impeller and Diffuser

For constant mass flow rate of air, the width of the blades of impeller can be calculated as follows:

$$\dot{m} = \frac{A\mathcal{V}_1}{v_1} \text{ (Continuity equation)}$$

where v_1 = specific volume of air at inlet

$A = \pi D_1 B_1$, with D_1, diameter of impeller and B_1 as width of impeller blades at inlet.

As air trapped radially, $\mathcal{V}_1 = \mathcal{V}_{f_1}$

$$\therefore \quad \dot{m} = \frac{\pi D_1 B_1}{v_1}\mathcal{V}_{f_1} \qquad \text{...(6.8)}$$

or

$$B_1 = \frac{\dot{m}\,v_1}{\pi\, D_1\,\mathcal{V}_{f_1}} \qquad \text{...(6.9)}$$

Similarly, the width of impeller blade at the outlet can be obtained by using the suffix 2 in Eq. (6.9)

$$B_2 = \frac{\dot{m}\,v_2}{\pi\, D_2\,\mathcal{V}_{f_2}} \qquad \text{...(6.10)}$$

If the number and thickness of blades are considered then the effective area of the blade will be

$$A = (\pi D - nt)B$$

where n = number of blades and t = thickness of blade.

Then

$$B_1 = \frac{\dot{m}\,v_1}{(\pi\, D_1 - n\,t)\mathcal{V}_{f_1}} \qquad \text{...(6.11)}$$

and

$$B_2 = \frac{\dot{m}\,v_2}{(\pi\, D_2 - n\,t)\mathcal{V}_{f_2}} \qquad \text{...(6.12)}$$

6.5.5 Degree of Reaction

The *degree of reaction* is defined as the ratio of static pressure rise in the impeller to the total static pressure rise in the compressor. The pressure rise in the impeller is equal to change in kinetic energy of air in the impeller, i.e.,

$$\Delta ke = \frac{\mathcal{V}_{r_1}^2 - \mathcal{V}_{r_2}^2}{2} + \frac{u_2^2 - u_1^2}{2} \quad ...(6.13)$$

The *first* term above Eq. (6.13) indicates the pressure rise in the compressor due to diffusion action, and the *second* term represents the pressure rise in the impeller due to centrifugal action of the impeller.

Total pressure rise in the compressor is equal to work input to the compressor.

$\therefore$ degree of reaction

$$R_d = \frac{\text{Pressure rise in the impeller}}{\text{Pressure rise in the compressor}}$$

$$= \frac{\frac{\mathcal{V}_{r_1}^2 - \mathcal{V}_{r_2}^2}{2} + \frac{u_2^2 - u_1^2}{2}}{u_2 \mathcal{V}_{w_2}} = \frac{(\mathcal{V}_{r_1}^2 - \mathcal{V}_{r_2}^2) + (u_2^2 - u_1^2)}{2u_2 \mathcal{V}_{w_2}}$$

or $$R_d = \frac{(u_2^2 - \mathcal{V}_{r_2}^2) + (\mathcal{V}_{r_1}^2 - u_1^2)}{2u_2 \mathcal{V}_{w_2}} \quad ...(6.14)$$

From the inlet velocity triangle of Fig. 6.8;

$$\mathcal{V}_{r_1}^2 - u_1^2 = \mathcal{V}_{f_1}^2 = \mathcal{V}_{f_2}^2 \ (\text{since } \mathcal{V}_{f_1} = \mathcal{V}_{f_2}) \quad ...(6.15)$$

From outlet velocity trinagle of Fig. 6.8;

$$\mathcal{V}_{f_2}^2 + (u_2 - \mathcal{V}_{w_2})^2 = \mathcal{V}_{r_2}^2$$

or $$\mathcal{V}_{r_2}^2 = \mathcal{V}_{f_2}^2 + u_2^2 + \mathcal{V}_{w_2}^2 - 2u_2 \mathcal{V}_{w_2}$$

or $$u_2^2 - \mathcal{V}_{r_2}^2 = 2u_2 \mathcal{V}_{w_2} - \mathcal{V}_{f_2}^2 - \mathcal{V}_{w_2}^2 \quad ...(6.16)$$

Using Eqs. (6.15) and (6.16) in Eq. (6.14), we get

$$R_d = \frac{2u_2 \mathcal{V}_{w_2} - \mathcal{V}_{f_2}^2 - \mathcal{V}_{w_2}^2 + \mathcal{V}_{f_2}^2}{2u_2 \mathcal{V}_{w_2}}$$

or $$R_d = 1 - \frac{\mathcal{V}_{w_2}}{2u_2} \quad ...(6.17)$$

6.5.6 Combined Velocity Diagram

Since the flow velocity remains constant at inlet and outlet, therefore, the inlet and exit velocity triangles can be drawn on the common base. Since air enters at right angles to the impeller blade, thus the air inlet angle at the impeller is equal to 90°, i.e., α = 90°. Combined velocity triangle for centrifugal compressor is shown in Fig. 6.9.

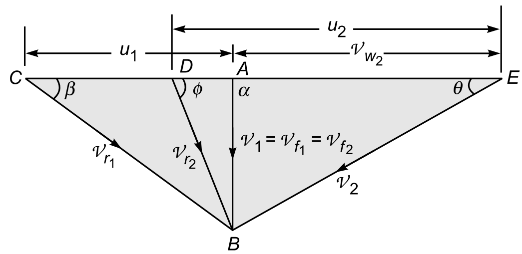


Fig. 6.9 *Combined velocity triangle for centrifugal compressor*

Procedure

1. Draw a vertical line AB to represent the flow velocity and it remains constant at inlet and exit.
2. The horizontal line CA represents the blade velocity u_1 at inlet.
3. The line CB inclined at the blade angle at inlet β represents relative velocity $\mathcal{V}_{r_1}$ of blade at the inlet.
4. The line DB inclined at blade angle at inlet ϕ represents the relative velocity $\mathcal{V}_{r_2}$ of the blade at the outlet.
5. The line DE represents the blade velocity u_2 of impeller at outlet.
6. Join the line EB. It represents absolute velocity $\mathcal{V}_2$ of air at the outlet inclined at angle θ with respect to the horizontal.

If the air delivery through the impeller blade is radial (ideal case), the velocity diagram at the outlet takes the shape as shown in Fig. 6.10.

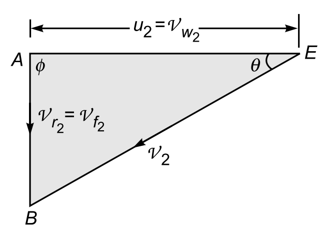


Fig. 6.10 *Velocity triangle at outlet for ideal radial air flow*

In this case, the blade velocity at the outlet becomes equal to whirl velocity at the outlet, i.e., $u_2 = \mathcal{V}_{w_2}$. The work input per kg of air is

$$w = u_2^2 \text{ (J/kg)} \quad ...(6.18)$$

The exit whirl velocity $\mathcal{V}_{w_2}$ of air cannot be greater than the blade tip velocity. Thus, it is the limiting case and it is the *maximum work supplied* to air per kg.

6.5.7 Efficiency of Centrifugal Compressor

The work transfer per kg of air to centrifugal compressor can also be obtained by using steady-flow energy equation

$$q - w = (h_2 - h_1) + \left(\frac{\mathcal{V}_2^2}{2} - \frac{\mathcal{V}_1^2}{2}\right) + \Delta pe$$

For centrifugal air compressor, $q = 0$,

Change in potential energy, $\Delta pe = 0$;

Omitting the negative sign for work input, the compresson work per kg of air

$$w = \left(h_2 + \frac{\mathcal{V}_2^2}{2}\right) - \left(h_1 + \frac{\mathcal{V}_1^2}{2}\right)$$

The enthalpy h $(= C_pT)$ and kinetic energy $\dfrac{\mathcal{V}^2}{2}$ always appear in flow analysis and for convenience their summation $\left(h + \dfrac{\mathcal{V}^2}{2}\right)$ is considered as a single quantity, called stagnation enthalpy or total enthalpy of fluid and designated as h_0.

Then the isentropic work input in terms of stagnation enthalpies

$$w = (h_{02s} - h_{01}) = C_p(T_{02s} - T_{01})$$

$$= C_pT_{01}\left(\frac{T_{02s}}{T_{01}} - 1\right) \quad \ldots(6.19)$$

where h_{02s} and T_{02s} are stagnation enthalpy and stagnation temperature, respectively after isentropic compression. The static states and stagnation states are shown in Fig. 6.11. The suffix 1 indicates the inlet state, 2s indicates the state after isentropic expansion, 2; the state after actual expansion and the properties with suffix 0 with a number indicate corresponding stagnation properties.

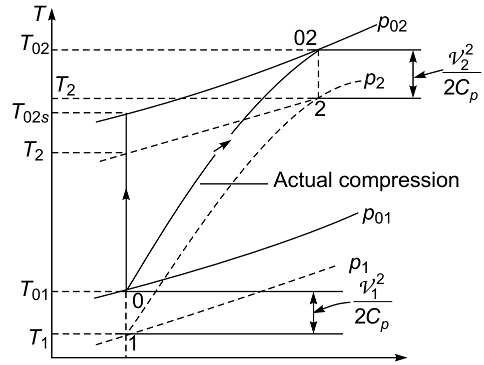


Fig. 6.11 *Static and stagnation properties during adiabatic compression*

The stagnation temperature and stagnation pressure p_0 for an isentropic flow can be related as

$$\frac{T_{01}}{T_1} = \left(\frac{p_{01}}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

And $$\frac{T_{02s}}{T_{01}} = \left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} \quad \ldots(6.20)$$

Inserting Eqn. (6.19) in Eqn. (6.20), then the isentropic work input to compressor per kg of air

$$w = C_pT_{01}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \quad \ldots(6.21)$$

Figure 6.11 also shows isentropic and actual compression processes. The actual work input for the same pressure ratio is more due to irreversibilities.

Since the cooling arrangement is not provided in rotary compressors, the ideal compression process is *isentropic compression*. But the actual work input is always more than isentropic work input for compression through same pressure ratio.

The *isentropic efficiency* (η_C) of a dynamic compressor may be defined as ratio of isentropic work input to actual work input and is given by

$$\eta_C = \frac{\text{Isentropic work input}}{\text{Actual work input}} = \frac{h_{02s} - h_{01}}{h_{02} - h_{01}} \quad \ldots(6.22)$$

If specific heat C_p of air remains constant then

$$\eta_C = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} \qquad \ldots(6.23)$$

If inlet velocity of air is equal to air exit velocity, i.e., $\mathcal{V}_1 = \mathcal{V}_2$ then

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{\text{Isentropic temperature rise}}{\text{Actual temperature rise}} \qquad \ldots(6.24)$$

Thus the isentropic efficiency of a rotary compressor may also be defined as ratio of isentropic temperature rise to actual temperature rise.

6.5.8 Losses During Compression

The following losses occur in a centrifugal compressor, when air flows through the impeller:

1. Friction between moving air layers and impeller blades and friction between air layers moving with relative velocities,
2. Shock at entry, and exit,
3. Eddies formation during flow, and
4. Turbulence caused in air.

These losses cause an increase in enthalpy of air without increasing the pressure of air. Therefore, the actual temperature of air coming out of the compressor is more than the temperature of air at the inlet.

6.5.9 Slip and Slip Factor

The difference between blade tip velocity u_2 and whirl velocity $\mathcal{V}_{w2}$ is known as *slip*, i.e., slip = $u_2 - \mathcal{V}_{w2}$.

The *slip factor* is defined as the ratio of whirl velocity to blade tip velocity. It is designated as ϕ_s and expressed as

$$\text{Slip factor,} \quad \phi_s = \frac{\mathcal{V}_{w2}}{u_2} \qquad \ldots(6.25a)$$

The slip factor depends on number of vanes in the impeller; more number of vanes, smaller is the slip. An empirical relation to obtain the slip factor is

$$\phi_s = 1 - \frac{0.63}{n} \qquad \ldots(6.25b)$$

where n is number of vanes.

1. In actual practice, the whirl velocity $\mathcal{V}_{w2}$ is always less than the blade tip velocity u_2. With increase in slip factor, $\mathcal{V}_{w2}$ increases and when it approaches u_2, the centrifugal compressor has maximum work input.
2. The impeller with large number of vanes has high value of slip factor and effective area of flow decreases.

6.5.10 Work Factor and Pressure coefficient

The actual work input per kg of air by compressor is always greater than the impeller work $u_2\mathcal{V}_{w_2}$ due to fluid friction and windage losses. *Work factor* or *power input factor* is defined as ratio of actual work input to Euler work input. It is designated as ϕ_w and is given as

$$\phi_w = \frac{\text{Actual work input}}{\text{Euler work input}}$$

$$= \frac{C_p(T_{02} - T_{01})}{u_2\mathcal{V}_{w_2}} \qquad \ldots(6.26)$$

Pressure coefficient is defined as the ratio of isentropic work to Euler work. It is designated as ϕ_p.

$$\phi_p = \frac{\text{Isentropic work input}}{\text{Euler work input}}$$

$$= \frac{C_p(T_{02s} - T_{01})}{u_2\mathcal{V}_{w_2}} \qquad \ldots(6.27)$$

Using Eq. (6.23), and assuming radial vanes of impeller ($u_2 = \mathcal{V}_{w_2}$), then

$$\phi_p = \frac{\eta_C C_p(T_{02} - T_{01})}{u_2^2} \qquad \ldots(6.28)$$

From Eq. (6.26), we get

$$C_p(T_{02} - T_{01}) = \phi_w u_2 \mathcal{V}_{w_2}$$

Substituting the V_{w_2} as $\phi_s u_2$; from Eq. (6.25), we get

$$C_p(T_{02} - T_{01}) = \phi_w \phi_s u_2^2$$

Using in Eq. (6.28), we get;

$$\phi_p = \frac{\eta_C \phi_w \phi_s u_2^2}{u_2^2} = \phi_w \phi_s \eta_C \qquad ...(6.29)$$

6.5.11 Effect of Impeller Blade Shape on Compressor Performance

There are usually three types of impeller blade shapes used in a centrifugal compressor. These are

1. Backward curved blades ($\theta < 90°$)
2. Radial blades ($\theta = 90°$)
3. Forward curved blades ($\theta > 90°$)

Figure 6.12 shows the geometry of backward, radial and forward curved vanes and performance of these vanes. The centrifugal action on the curved vanes creates bending moment and induces bending stresses.

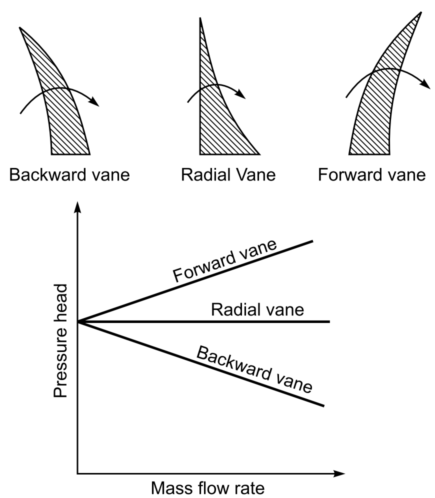


Fig. 6.12 *Charecteristic of backward, radial and forward curved vanes*

The pressure head of delivered air decreases with increase in mass-flow rate for backward curved vanes, while the pressure head increases for forward curved vanes. But for radial curved vanes, the pressure head of delivered air remains constant with mass-flow rate.

The backward vanes are normally used with θ = 20° to 25°, except for delivery of air at high head. The radial vane is the compromise between backward and forward curved vanes. Therefore, the radial vane impeller is commonly used in a centrifugal compressor due to the following reasons:

1. Radial-vane geometry is simple, thus vanes can be manufactures easily.
2. Radial vanes have lowest bending stress for given diameter and speed as compared to forward and backward curved vanes.
3. Radial vanes have a constant pressure head in the impeller as well as in the diffuser.
4. A radial-vane impeller has good efficiency and high pressure head.

6.5.12 Diffuser System

In a centrifugal compressor, the diffuser converts kinetic energy of air into static pressure head. For a radial vane impeller, the diffuser contributes about one half of the overall static pressure rise.

A diffuser consists of curved vanes, which are used to minimize the whirl of high speed air with smallest possible flow path and diameter. The flow of air through the diffuser vanes may be approximated as *logarithmic spiral path.* For fast diffusion, the axis of vanes is straight and tangential to spiral. Usually, the number of diffuser passages are less than the impeller passages for more uniform and smooth flow.

Figure 6.13 shows a typical curved vanes diffuser along with an impeller. The clearance provided between the impeller and diffuser rings acts as a vaneless diffuser and it functions to

1. smooth out velocity variation betweem the impeller tip and diffuser vanes
2. reduce circumferential pressure gradient at the impeller tip
3. reduce the velocity at the entry of vanes

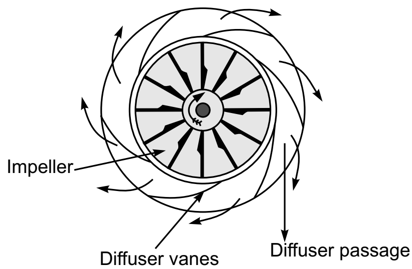


Fig. 6.13 *Typical vaned diffuser*

6.5.13 Comparison between Reciprocating and Centrifugal Compressors

Sr. No.	*Reciprocating Compressor*	*Centrifugal Compressor*
1.	Geater noise and vibrations	Compratively salient operation
2.	Poor mechanical efficiency due to large sliding parts	Better mechanical efficiency due to absence of sliding parts
3.	Installation cost is higher	Installation cost is lower
4.	Pressure ratio up to 5 to 8	Pressure ratio up to 4
5.	Higher pressure ratio up to 500 atm. is possible with multistaging of compressor	It is not suitable for multistaging
6.	It runs intermittantly and delivers pulsating air	It runs continuously and delivers steady and pulsating free air
7.	Less volume is handled	Large volume is handled
8.	More maintenance is required	Less maintenance is required
9.	Weight of compressor is more	Comparatively less weight
10.	It operates at low speed	It operates at high speed
11.	Isothermal efficiency should be better	Isentropic efficiency should be better
12.	Higher compression efficiency at pressure ratio more than 2	Higher compression efficiency, if pressure ratio less than 2
13.	Suitable for low discharge and high pressure ratio	Suitable for high discharge and low pressure ratio

Contd.

Example 6.5 *In a centrifugal compressor, the air enters at 27°C and leaves at 105°C. The air is compressed through a pressure ratio of 2. Calculate the isentropic efficiency and power required by the compressor, if 30 kg of air is compressed per minute. Take $C_p = 1.00$ kJ/kg·K and $C_v = 0.716$ kJ/kg·K.*

Solution

Given A centrifugal compressor with

$$\dot{m} = 30 \text{ kg/min} = 0.5 \text{ kg/s}$$
$$\frac{p_2}{p_1} = 2.0$$
$$T_1 = 27°C = 300 \text{ K}$$
$$T_2 = 105°C = 378 \text{ K}$$
$$C_p = 1.00 \text{ kJ/kg}\cdot\text{K}$$
$$C_v = 0.716 \text{ kJ/kg}\cdot\text{K}$$

To find

(i) Isentropic efficiency of the compressor, and

(ii) Power required to run the compressor.

Analysis The ratio of two specific heats

$$\gamma = \frac{C_p}{C_v} = \frac{1.00}{0.716} = 1.4$$

The temperature of air after isentropic compression

$$T_{2s} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (300 \text{ K}) \times \left(\frac{2}{1}\right)^{\frac{1.4-1}{1.4}}$$
$$= 365.7 \text{ K}$$

(i) Isentropic efficiency is given by;

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{\text{Isentropic temperature rise}}{\text{Actual temperature rise}}$$
$$= \frac{365.7 - 300}{378 - 300} = 0.8423 \quad \text{or} \quad \mathbf{84.23\%}$$

(ii) The power input for compression

$$P = \dot{m}C_p(T_2 - T_1) = 0.5 \times 1.0 \times (378 - 300)$$
$$= \mathbf{39\ kW}$$

Example 6.6 *A centrifugal compressor compresses air from 1 bar at 15°C to 2.15 bar, 95°C. The mass of air delivered is 2.2 kg/s and no heat is added to the air from external sources during compression. Find the efficiency of the compressor relative to ideal adiabatic compression and estimate the power absorbed. Also, find the change in entropy of air during compression.*

Solution

Given A centrifugal compressor with

$\dot{m} = 2.2$ kg/s	$p_1 = 1$ bar
$T_1 = 15°C = 288$ K	$p_2 = 2.15$ bar
$T_2 = 95°C = 368$ K	$Q = 0$

To find

(i) Isentropic efficiency of the compressor,
(ii) Power required to run the compressor,
(iii) Entropy change during compression process.

Assumptions

1. Negligible effect of kinetic energy on fluid temperatures.
2. For air $C_p = 1.00$ kJ/kg·K
   $\gamma = 1.4$ and $R = 0.287$ kJ/kg·K

Analysis The temperature of air after isentropic compression

$$T_{2s} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (288\text{ K}) \times \left(\frac{2.15}{1}\right)^{\frac{1.4-1}{1.4}}$$
$$= 358.4\text{ K}$$

(i) *Isentropic efficiency* is given by

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{358.4 - 288}{368 - 280}$$
$$= 0.880 \quad \text{or} \quad \mathbf{80\%}$$

(ii) *The power input for compression*

In absence of change in kinetic and potential energies and heat transfer, the steady-flow energy equation for compressor leads to

$$w_{in} = h_2 - h_1 = C_p(T_2 - T_1)$$
$$= 1.005 \times (358.4 - 288)$$
$$= 70.75\text{ kJ/kg}$$

The power input

$$P = \dot{m}w_{in}$$
$$= 2.2 \times 70.75 = \mathbf{155.65\ kW}$$

(iii) *Entropy change during compression process:*

The entropy change during a process can be obtained as

$$\Delta S = \dot{m}\left[C_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{p_2}{p_1}\right)\right]$$
$$= 2.2 \times \left[1.005 \times \ln\left(\frac{368}{288}\right) - 0.287 \times \ln\left(\frac{2.15}{1}\right)\right]$$
$$= \mathbf{0.586\ kJ/K}$$

Example 6.7 *A centrifugal compressor running at 2000 rpm has internal and external diameters of the impeller as 300 mm and 500 mm, respectively. The blade angles at inlet and outlet are 22° and 40°, respectively. The air enters the impeller radially. Determine the work done by the compressor per kg of air and degree of reaction.*

Solution

Given A centrifugal compressor with

$N = 2000$ rpm	$D_1 = 300$ mm $= 0.3$ m
$\beta = 22°$	$D_2 = 500$ mm $= 0.5$ m
$\phi = 40°$	

To find

(i) Work input to the compressor,
(ii) Degree of reaction.

Assumptions

1. Negligible effect of kinetic energy on fluid temperatures.
2. Negligible change in potential energy.
3. Constant fluid properties.

Analysis The linear velocity of impeller at inlet

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.3 \times 2000}{60}$$
$$= 31.416\text{ m/s}$$

The linear velocity of impeller at the outlet

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.5 \times 2000}{60} = 52.36\text{ m/s}$$

With the use of blade velocities and blade angles at the inlet and outlet, the velocity triangle can be constructed as shown in Fig. 6.14 and procedure is given below.

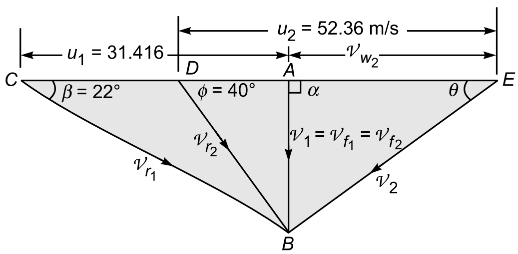


Fig. 6.14 *Combined velocity triangle for centrifugal compressor*

(i) The horizontal line CA represents the blade velocity $u_1 = 31.416$ m/s at inlet.

(ii) Draw line CB inclined at 22°.

(iii) From the point D, draw line DB inclined at angle $\phi = 40°$; which cuts line CB at point B.

(iv) Draw a vertical line AB to represent the flow velocity.

(v) Draw the line DE to represent blade velocity $u_2 =$ 52.36 m/s.

(vi) Join the line EB to represent exit velocity of air $\mathcal{V}_2$.

From the velocity triangles, the measurements give

$$\mathcal{V}_{w_2} = \text{Length of } AE = 37.3 \text{ m/s},$$
$$\mathcal{V}_{r_1} = \text{Length of } CB = 34.1 \text{ m/s},$$
$$\mathcal{V}_{r_2} = \text{Length of } DB = 19.9 \text{ m/s}.$$

(i) Work input per kg of air

$$w = \mathcal{V}_{w_2} u_2 = 37.3 \times 52.36 = 1953.02 \text{ J/kg}$$
$$= \mathbf{1.953\ kJ/kg}$$

(ii) Degree of reaction;

$$R_d = \frac{\text{Pressure rise in the impeller}}{\text{Pressure rise in the compressor}}$$
$$= \frac{(u_2^2 - \mathcal{V}_{r_2}^2) + (\mathcal{V}_{r_1}^2 - u_1^2)}{2u_2\mathcal{V}_{w_2}}$$
$$= \frac{(52.36^2 - 19.9^2) + (34.1^2 - 31.416^2)}{2 \times 52.36 \times 37.3}$$
$$= 0.6455 \quad \text{or} \quad \mathbf{64.55\%}$$

Example 6.8 *A centrifugal compressor running at 1440 rpm, handles air at 101 kPa and 20°C and compresses it to a pressure of 6 bar isentropically. The inner and outer diameters of the impeller are 14 cm and 28 cm, respectively. The width of the blade at the inlet is 2.5 cm. The blade angles are 16° and 40° at entry and exit. Calculate mass-flow rate of air, degree of reaction, power input and width of blades at outlet.*

Solution

Given A centrifugal compressor with

$N = 1440$ rpm	$D_1 = 14$ cm $= 0.14$ m
$\beta = 16°$	$D_2 = 28$ cm $= 0.28$ m
$\phi = 40°$	$p_1 = 101$ kPa
$p_2 = 6$ bar $= 600$ kPa	$T_1 = 20°$C $= 293$ K
$B_1 = 2.5$ cm $= 0.025$ m	

To find

(i) Mass flow rate of air,

(ii) Power input to the compressor,

(iii) Degree of reaction, and

(iv) Width of blade at outlet.

Assumptions

1. The specific gas constant as 0.287 kJ/kg·K.
2. Negligible effect of kinetic energy on fluid temperatures.

Analysis The linear velocity of impeller at the inlet

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.14 \times 1440}{60} = 10.56 \text{ m/s}$$

The linear velocity of impeller at the outlet

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.28 \times 1440}{60} = 21.12 \text{ m/s}$$

With the use of blade velocities and blade angles at inlet and outlet, the velocity triangle can be constructed as shown in Fig. 6.15.

From the velocity triangles, the measurements give

$$\mathcal{V}_{w_2} = \text{Length of } AE = 17.5 \text{ m/s}$$
$$\mathcal{V}_{r_1} = \text{Length of } CB = 11 \text{ m/s}$$
$$\mathcal{V}_{r_2} = \text{Length of } DB = 4.7 \text{ m/s}$$
$$\mathcal{V}_{f_1} = 3 \text{ m/s}$$

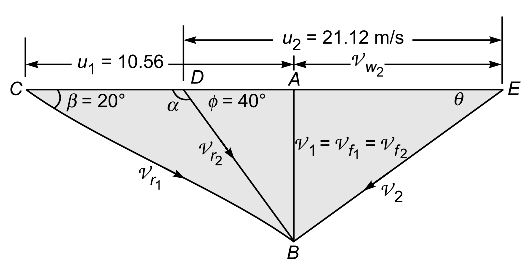


Fig. 6.15 *Combined velocity triangle for centrifugal compressor*

Work input per kg of air

$$w = \mathcal{V}_{w_2} u_2 = 17.5 \times 21.12 = 369.6 \text{ J/kg}$$

(i) *Mass-flow rate of air*

$$\dot{m} = \rho_1 A_1 \mathcal{V}_{f_1} = \rho_1 \pi D_1 B_1 \mathcal{V}_{f_1}$$

Density of air

$$\rho = \frac{p_1}{RT_1} = \frac{101}{0.287 \times 293}$$

$$= 1.201 \text{ kg/m}^3$$

$$\therefore\ \dot{m} = 1.201 \times \pi \times 0.14 \times 0.025 \times 3$$

$$= \mathbf{0.0396\ kg/s}$$

(ii) *Power input to run the compressor*

$$P = \dot{m} w = 0.0396 \times 369.6 = \mathbf{14.61\ W}$$

(iii) *Degree of reaction*

$$R_d = \frac{(u_2^2 - \mathcal{V}_{r_2}^2) + (\mathcal{V}_{r_1}^2 - u_1^2)}{2 u_2 \mathcal{V}_{w_2}}$$

$$= \frac{(21.12^2 - 4.7^2) + (11^2 - 10.56^2)}{2 \times 17.5 \times 21.12}$$

$$= 0.5864 \quad \text{or} \quad \mathbf{58.64\%}$$

(iv) *Width of blade at outlet*

Flow velocity at outlet

$$\mathcal{V}_{f_2} = \mathcal{V}_{f_1} = 3.0 \text{ m/s}$$

Temperature of air at the outlet

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 293 \times \left(\frac{600}{101}\right)^{\frac{1.4-1}{1.4}}$$

$$= 487.48 \text{ K}$$

Density of air at the outlet

$$\rho_2 = \frac{p_2}{RT_2} = \frac{600}{0.287 \times 487.48}$$

$$= 4.288 \text{ kg/m}^3$$

From continuity equation;

$$\dot{m} = \rho_2 \pi D_2 B_2 \mathcal{V}_{f_2}$$

$$\text{or } B_2 = \frac{\dot{m}}{\rho_2 \pi D_2 \mathcal{V}_{f_2}} = \frac{0.0396}{4.288 \times \pi \times 0.28 \times 3}$$

$$= 0.0035 \text{ m} = \mathbf{3.5\ mm}$$

Example 6.9 *A centrifugal compressor runing at 12000 rpm delivers 600 m³/min of free air. The air is compressed from 1 bar and 27°C to a pressure ratio of 4 with an isentropic efficiency of 85%. The blades are radial at the impeller outlet and flow velocity of 60 m/s may be assumed throughout constant. The outer radius of the impeller is twice the inner one and slip factor is 0.9. Calculate*

(a) *Final temperature of air*
(b) *Power input to compressor*
(c) *Impeller diameter at inlet and outlet*
(d) *Width of impeller at inlet*

Solution

Given A centrifugal compressor with

$N = 12000$ rpm $\qquad r_2 = 2 r_1$

$p_1 = 1$ bar $= 100$ kPa $\qquad T_1 = 27°\text{C} = 300$ K

$\frac{p_2}{p_1} = 4 \qquad \mathcal{V}_{f_1} = 60$ m/s

$\eta_C = 0.85$

$\dot{V} = 600 \text{ m}^3/\text{min} = 10 \text{ m}^3/\text{s}$

$\phi_s = 0.9$

To find

(i) Final temperature of air,
(ii) Power input to the compressor,
(iii) Impeller diameter at inlet and outlet, and
(iv) Width of impeller at inlet.

Assumption The specific heat of air as 1005 J/kg·K and $\gamma = 1.4$

Analysis

(i) *Final temperature of air*

$$T_{2s} = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= 300 \times (4)^{\frac{1.4-1}{1.4}} = 445.79 \text{ K}$$

The isentropic efficiency is given by

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1}$$

Final temperature of air is given by

$$T_2 = \frac{T_{2s} - T_1}{\eta_C} + T_1$$

$$= \frac{445.79 - 300}{0.85} + 300$$

$$= 471.52 \text{ K} \quad \text{or} \quad \mathbf{198.52°C}$$

The mass flow rate of air

$$\dot{m} = \frac{p\dot{V}}{RT} = \frac{100 \times 10}{0.287 \times 300} = 11.61 \text{ kg/s}$$

(ii) *Power input to compressor*

Work input per kg of air

$$w = C_p(T_2 - T_1) = 1.005 \times (471.52 - 300)$$
$$= 172.37 \text{ kJ/kg}$$

Power input

$$P = \dot{m}w = 11.61 \times 172.37 = \mathbf{2001.3\ kW}$$

(iii) *Impeller diameters at inlet and outlet*

For radial blades, the work input to compressor with slip is given by

$$w = \phi_s u_2^2 \text{ (J/kg)}$$

Using numerical values and equating with work obtained above

$$0.9 \times u_2^2 = 172.37 \times 10^3 \text{ (J/kg)}$$

It gives $u_2 = 437.63$ m/s

The linear blade velocity at impeller tip is given by

$$u_2 = \frac{\pi D_2 N}{60}$$

It gives impeller diameter at outlet

$$D_2 = \frac{437.63 \times 60}{\pi \times 12000}$$
$$= 0.6965 \text{ m} \quad \text{or} \quad \mathbf{69.65\ cm}$$

Impeller diameter at inlet

$$D_1 = \frac{D_2}{2} = \frac{69.65}{2}$$
$$= \mathbf{34.825\ cm}$$

(iv) Width of impeller at inlet, Eq. (6.12)

$$B_1 = \frac{\dot{m} v_1}{\pi D_1 \mathcal{V}_{f_1}} = \frac{\dot{V}}{\pi D_1 \mathcal{V}_{f_1}}$$
$$= \frac{10 \text{ m}^3/\text{s}}{\pi \times 0.34825 \times 60}$$
$$= 0.1523 \text{ m} \quad \text{or} \quad \mathbf{15.23\ cm}$$

Example 6.10 *A centrifugal compressor handles 600 kg/min of air.The ambient air conditions are 1 bar and 27°C. The compressor runs at 18000 rpm with an isentropic efficiency of 80%. The air is compressed in the compressor from 1 bar static pressure to 4 bar total pressure. The air enters the impeller eye with a velocity of 150 m/s with no prewhirl. Take the ratio of whirl speed to tip speed as 0.9. Calculate*

(a) rise in total temperature during compression if change in kinetic energy is negligible

(b) tip diameter of impeller

(c) power required

(d) Eye diameter, if hub diameter is 10 cm

Solution

Given A centrifugal compressor with

$N = 18000$ rpm, $D_h = 10$ cm $= 0.1$ m

$p_1 = 1$ bar $= 100$ kPa, $T_1 = 27°C = 300$ K

$p_{02} = 4$ bar $= 400$ kPa, $\mathcal{V}_1 = 150$ m/s

$\dfrac{\mathcal{V}_{w2}}{u_2} = 0.9$, $\eta_C = 0.8$

$\dot{m} = 600$ kg/min $= 10$ kg/s, $\mathcal{V}_{w_1} = 0$

To find

(i) Rise in total temperature of air,

(ii) Tip diameter of the impeller,

(iii) Power input, and

(iv) Eye diameter.

Assumption The specific heat of air as 1005 J/kg·K and $\gamma = 1.4$

Analysis The schematic for Example 6.10 is shown in Fig. 6.16.

(i) *Rise in total temperature of air*

Stagnation temperature at the inlet of compressor

$$T_{01} = T_1 + \frac{\mathcal{V}_1^2}{2C_p} = 300 + \frac{150^2}{2 \times 1005}$$
$$= 311.19 \text{ K}$$

The stagnation pressure at compressor inlet

$$p_{01} = p_1 \left(\frac{T_{01}}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$$
$$= (1 \text{ bar}) \times \left(\frac{311.19}{300}\right)^{\frac{1.4}{1.4-1}}$$
$$= 1.137 \text{ bar}$$

Stagnation temperature after isentropic compression

$$T_{02s} = T_{01} \left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}}$$
$$= 311.19 \times \left(\frac{4}{1.137}\right)^{\frac{1.4-1}{1.4}}$$
$$= 445.79 \text{ K}$$

Isentropic rise in stagnation temperature

$$= T_{02s} - T_{01}$$
$$= 445.79 - 311.19 = 134.6°\text{C}$$

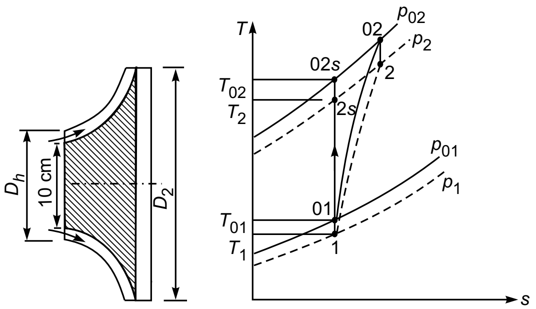


Fig. 6.16 *Schematic*

The isentropic efficiency in terms of stagnation temperatures may be given as

$$\eta_C = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}}$$

Actual rise in stagnation temperature is given by

$$T_{02} - T_{01} = \frac{T_{02s} - T_{01}}{\eta_C} = \frac{134.6}{0.8}$$

$$= \mathbf{168.25°C}$$

(ii) *Tip diameter of impeller, D_2*

Work input to compressor per kg of air

$$w = C_p(T_{02} - T_{01}) = 1.005 \times 168.25$$
$$= 169.09 \text{ kJ/kg}$$

Work input per kg of air to compressor is also given by

$$w = u_2 \mathcal{V}_{w_2} \text{ (J/kg)}$$

or $169.09 \times 10^3 = 0.9\, u_2^2 \qquad \left(\because \frac{\mathcal{V}_{w_2}}{u_2} = 0.9\right)$

or $u_2 = 433.45$ m/s

The blade velocity is given by

$$u_2 = \frac{\pi D_2 N}{60}$$

or $$D_2 = \frac{u_2 \times 60}{\pi N} = \frac{433.45 \times 60}{\pi \times 18000}$$

$$= 0.4599 \text{ m} \approx \mathbf{46\ cm}$$

(iii) *Power input to compressor*

$$P = \dot{m}w = 10 \times 169.09 = \mathbf{1690.9\ kW}$$

(iv) *Eye diameter, D_1*

The density of air at compressor inlet is given by

$$\rho_1 = \frac{p_1}{RT_1} = \frac{100}{0.287 \times 300}$$

$$= 1.161 \text{ kg/m}^3$$

The mass-flow rate through impeller eye can be given by

$$\dot{m} = \frac{\pi}{4}\left(D_1^2 - D_h^2\right) \times \mathcal{V}_1 \times \rho_1$$

or $$10 = \frac{\pi}{4}\left(D_1^2 - 0.1^2\right) \times 150 \times 1.161$$

or $$D_1 = \sqrt{\frac{10 \times 4}{\pi \times 150 \times 1.161} + 0.1^2}$$

$$= 0.288 \text{ m} = \mathbf{28.8\ cm}$$

Example 6.11 *An aircraft engine is fitted with a single-sided centrifugal compressor. The aircraft flies with a speed of 900 km/h at an altitude, where the pressure is 0.23 bar and temperature is 217 K. The inlet duct of the impeller eye contains fixed vanes, which gives the air prewhirl of 25° at all radii. The inner and outer diameter of the eye are 180 and 330 mm, respectively. The diameter of the impeller tip is 540 mm and rotational speed is 15000 rpm. Estimate the stagnation pressure at compressor outlet when the mass flow rate is 210 kg/minute.*

Neglect losses in inlet duct and fixed vanes, and assume that the isentropic efficiency of the compressor is 80%. Take slip factor as 0.9 and power input factor as 1.04.

Solution

Given A single-sided centrifugal compressor of an airfraft with

$N = 15000$ rpm $\qquad D_h = 180$ mm $= 0.15$ m

$D_1 = 330$ mm $= 0.33$ m $\qquad D_2 = 540$ mm $= 0.54$ m

$p_1 = 0.23$ bar $= 23$ kPa $\qquad T_1 = 217$ K

$\mathcal{V}_1 = 900$ km/h $\qquad \eta_C = 0.8$

$\dot{m} = 210$ kg/min $= 3.5$ kg/s

$\phi_s = \frac{\mathcal{V}_{w_2}}{u_2} = 0.9 \qquad \phi_w = 1.04$

To find Stagnation pressure at compressor outlet.

Assumption The specific heat of air as 1005 J/kg·K and $\gamma = 1.4$

Analysis The velocity of air with reference to aircraft

$$\mathcal{V}_1 = \frac{900 \times 1000}{3600} = 250 \text{ m/s}$$

Stagnation temperature at the inlet of compressor

$$T_{01} = T_1 + \frac{V_1^2}{2C_p} = 217 + \frac{250^2}{2 \times 1005} = 248.1 \text{ K}$$

The stagnation pressure at compressor inlet

$$p_{01} = p_1 \left(\frac{T_{01}}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$$

$$= (0.23 \text{ bar}) \times \left(\frac{248.1}{217}\right)^{\frac{1.4}{1.4-1}}$$

$$= 0.3675 \text{ bar}$$

The blade velocity at exit is given by

$$u_2 = \frac{\pi D_2 N}{60}$$

or $$u_2 = \frac{\pi \times 0.54 \times 15000}{60} = 424.11 \text{ m/s}$$

The whirl velocity at exit

$$V_{w_2} = \phi_s u_2 = 0.9 \times 424.11 = 381.7 \text{ m/s}$$

The power input factor is given as

$$\phi_w = \frac{C_p(T_{02} - T_{01})}{u_2 V_{w_2}}$$

The stagnation temperature at exit

$$T_{02} = \frac{\phi_w u_2 V_{w_2}}{C_p} + T_{01}$$

$$= \frac{1.04 \times 424.11 \times 381.7}{1005} + 248.1$$

$$= 415.62 \text{ K}$$

The isentropic efficiency in terms of stagnation temperatures may be given as

$$\eta_C = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}}$$

Isentropic stagnation temperature at exit is given by

$$T_{02s} = \eta_C (T_{02} - T_{01}) + T_{01}$$

$$= 0.8 \times (415.62 - 248.1) + 248.1$$

$$= 382.11 \text{ K}$$

Stagnation pressure after compression

$$p_{02} = p_{01} \left(\frac{T_{02s}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}}$$

$$= 0.3675 \times \left(\frac{382.11}{248.1}\right)^{\frac{1.4}{1.4-1}}$$

$$= \mathbf{1.667 \text{ bar}}$$

6.6 AXIAL COMPRESSOR

Axial compressors are aerofoil (blade) based rotary compressors. The gas flows parallel to the axis of rotation in axial flow compressors and gas is continuously compressed. The several rows of aerofoil blades are used to achieve large pressure rise in the compressor.

The axial compressors are generally multi-stage machines; each stage can give a pressure ratio of 1.2 to 1.3. The axial flow compressors are suitable for higher pressure ratios and are generally more efficient than radial compressors.

Axial compressors are widely used in gas turbine plants and small power stations. They are also used in industrial applications such as blast-furnace, large-volume air-separation plants, and propane dehydrogenation. Axial compressors are also used for supercharging. They are also used to boost the power of automotive reciprocating engines by compressing the intake air.

6.6.1 Construction

An axial air compressor consists of a large number of rotating blade rows, fixed on a rotating drum, and stator (fixed) blade rows fixed on the casing of the compressor as shown in Fig. 6.17. A pair of rotating and stationary blades is called a *stage*.

The moving blades act as a series of fans and the fixed blades act as guide vanes and diffuser. The moving blades are imparting energy into the fluid, and the fixed blades convert a part of kinetic energy of the fluid into pressure energy through diffusion-process and then guide and redirect the fluid onto the next stage of moving blades without shock.

The blades are made in aerofoil section to reduce the losses caused by shocks, turbulence and boundary separation. The annular area for air flow is gradually reduced from the inlet to the outlet of the compressor. The rotor of an axial compressor is made aerodynamic.

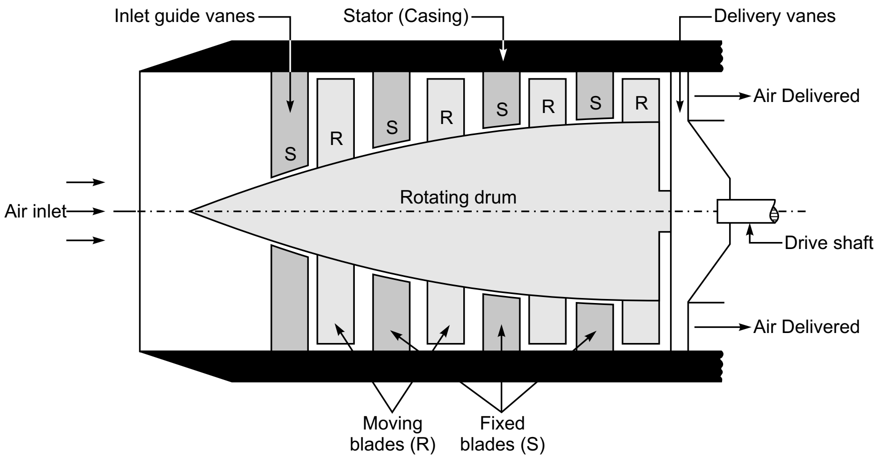


Fig. 6.17 *Axial-flow compressor*

6.6.2 Working

The work input to a rotor shaft is transferred by moving blades to air, thus accelerating the air flow. The spaces between moving blades and casing form the diffuser passages, and thus the velocity of air decreases as air passes through them and results into increase in pressure and enthalpy. The air is then further diffused in stator blades which are also arranged to form diffuser passages. The fixed blades also guide the air to flow at an angle for smooth entry of next row of moving blades.

The temperature rise of air is almost same in moving as well as in fixed blades and axial velocity of air remains constant throughout the compressor.

6.6.3 Velocity Triangles for Axial Flow Compressor

The velocity triangles at inlet and outlet of moving blades of an axial-flow air compressor is shown in Fig. 6.18. The following points should be considered for the construction of a velocity triangle for an axial-flow compressor.

1. The blade velocity remains same at inlet and outlet, i.e., $u_1 = u_2 = u$.
2. Flow velocity also remains constant, i.e., $\mathcal{V}_{f_1} = \mathcal{V}_{f_2}$.
3. Relative velocity at outlet is less than that at

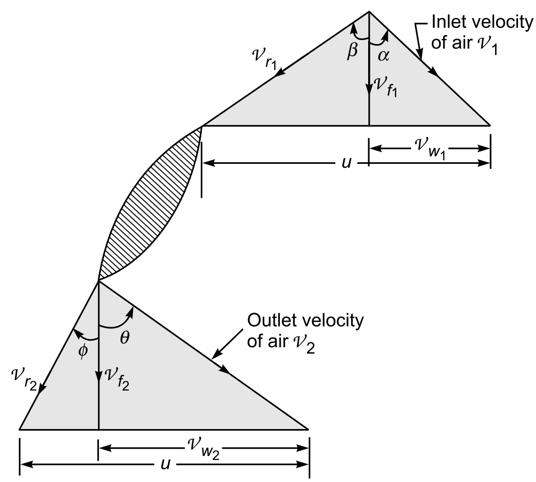


Fig. 6.18 *Inlet and out velocity triangles for axial compressor*

   inlet, i.e., $\mathcal{V}_{r_2} < \mathcal{V}_{r_1}$.
4. Both the whirl velocity components lie in the same plane.

The work input per stage per kg of air

$$w = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1}) \quad \text{...(6.30)}$$

Power required to drive the compressor

$$P = \dot{m}\,w = \dot{m}\,u\,(\mathcal{V}_{w_2} - \mathcal{V}_{w_1}) \quad \text{...(6.31)}$$

From inlet and outlet velocity triangles, we get

$$\mathcal{V}_{w_1} = \mathcal{V}_{f_1} \tan \alpha$$

and $$\mathcal{V}_{w_2} = \mathcal{V}_{f_2} \tan \theta \qquad ...(6.32)$$

given that $\mathcal{V}_{f_1} = \mathcal{V}_{f_2} = \mathcal{V}_f$ (say)

Therefore, work input to axial flow compressor can also be given by

$$w = u\mathcal{V}_f(\tan \theta - \tan \alpha) \qquad ...(6.33)$$

From Eqs. (6.30) and (6.33), we get

$$u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1}) = u\mathcal{V}_f(\tan \theta - \tan \alpha) \qquad ...(6.34)$$

6.6.4 Combined Velocity Triangles for Axial Flow Compressor

Since the blade speed u and flow velocity $\mathcal{V}_f$ remain constant at the inlet and outlet, thus the combined velocity triangle can be constructed shown in Fig. 6.19 below.

1. Draw the line AB to represent the blade velocity u.
2. Through the point A, draw a line at an inclination of $90° - \alpha$.
3. Through the point B, draw a line at an inclination of $90° - \beta$.
4. The above two lines intersect at the point C, line AC represents the air inlet velocity $\mathcal{V}_1$ and line BC represents relative velocity of air $\mathcal{V}_{r_1}$ at inlet.
5. The vertical line CD through the point C represents the velocity $\mathcal{V}_{f_1}$ at inlet.
6. Through the point B draw a line at an inclination of $90° - \phi$.
7. Along the inclined line, locate the point E with line EF, representing velocity $\mathcal{V}_{f_2}(=\mathcal{V}_{f_1})$ at the outlet.
8. Join the line AE to represent absolute velocity $\mathcal{V}_2$ of air at the outlet.

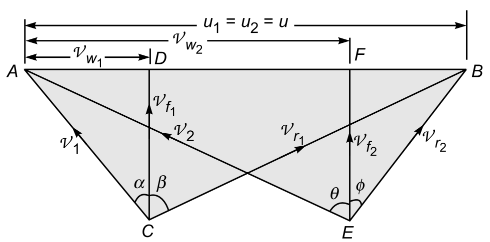


Fig. 6.19 *Combined velocity triangle for axial flow compressor*

6.6.5 Degree of Reaction

The degree of reaction of an axial compressor is defined as the ratio of static pressure rise in rotor to the total static pressure rise in the compressor. It is denoted by R_d. Mathematically,

$$R_d = \frac{\text{Pressure rise in rotor blade}}{\text{Pressure rise in the stage}}$$

The total pressure rise in an axial compressor stage is equal to work input per stage and is given by Eqn. (6.34)

$$(\Delta p)_{stage} = w = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})$$
$$= u\mathcal{V}_f(\tan \theta - \tan \alpha)$$

The pressure rise in rotor blade $(\Delta p)_{rotor}$ is outcome of change in relative kinetic energy of fluid.

$$(\Delta p)_{rotor} = \frac{\mathcal{V}_{r_1} - \mathcal{V}_{r_2}}{2} \qquad \ldots(6.35)$$

It can be obtained from the inlet velocity triangle *CDB and* outlet velocity triangle *BEF in Fig. 6.19*

$$\mathcal{V}_{r_1}^2 = \mathcal{V}_{f_1}^2 + (\mathcal{V}_{f_1} \tan \beta)^2$$
$$= \mathcal{V}_{f_1}^2 + \mathcal{V}_{f_1}^2 \tan^2 \beta \qquad \ldots(i)$$

and

$$\mathcal{V}_{r_2}^2 = \mathcal{V}_{f_2}^2 + \mathcal{V}_{f_2}^2 \tan^2 \phi \qquad \ldots(ii)$$

Since $\mathcal{V}_{f_1} = \mathcal{V}_{f_2} = \mathcal{V}_f$ (say)

Then $$(\Delta p)_{rotor} = \frac{\mathcal{V}_{r_1} - \mathcal{V}_{r_2}}{2} = \frac{\mathcal{V}_f^2}{2}(\tan^2 \beta - \tan^2 \alpha) \qquad \ldots(6.36)$$

Substituting Eqn. (6.34) and (6.36), the degree of reaction is expressed as

$$R_d = \frac{\mathcal{V}_f^2(\tan^2 \beta - \tan^2 \phi)}{2u\mathcal{V}_f(\tan \theta - \tan \alpha)} \qquad ...(6.37)$$

From symmetry of velocity triangles, we get

$$\beta = \theta \quad \text{and} \quad \alpha = \phi$$

Usually, the degree of reaction in axial flow compressor is taken 50%, i.e., $R_d = \frac{1}{2}$;

$$\therefore \qquad \frac{1}{2} = \frac{\mathcal{V}_f(\tan \beta + \tan \phi)}{2u}$$

or $$\frac{u}{\mathcal{V}_f} = \tan\beta + \tan\phi \qquad ...(6.38)$$

With 50% reaction blading, the axial compressor has symmetrical blades and losses in the compressor are drastically reduced.

6.6.6 Polytropic Work Input

The total work input per kg of air is given by

$$w = C_p(T_{02} - T_{01})$$

which can be written as

$$w = C_p\left(\frac{T_{02} - T_{01}}{T_{02s} - T_{01}}\right) \times (T_{02s} - T_{01})$$

$$= \frac{C_p(T_{02s} - T_{01})}{\eta_C}$$

$$= \frac{C_p \times T_{s01}\left(\frac{T_{02s}}{T_{01}} - 1\right)}{\eta_C}$$

$$= \frac{C_p \times T_{s01}\left(\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)}{\eta_C} \qquad ...(6.39)$$

Equation (6.39) demonstrate that for a given pressure ratio and isentropic efficiency, the work input to the compressor is directly proportional to the initial temperature of air in the stage. Thus, a compressor consisting of more than one stage of equal isentropic efficiency, will require more work input, because it receives fluid at increased temperature from preceeding stage. The two axial compressors of different pressure ratios will have different work input and overall efficiency.

Number of stages in axial flow air compressor can be obtained by dividing total work input by stage work input, providing all stages have equal isentropic efficiency of compression.

$$N_{stages} = \frac{\text{Total work input to compressor per kg of air}}{\text{Work input to one stage of compression}}$$

$$= \frac{C_p(T_{02} - T_{01})}{u(\mathcal{V}_{w2} - \mathcal{V}_{w2})\phi_h} \qquad ....(6.40)$$

6.6.7 Polytropic Efficiency

The *polytropic efficeincy* is the isentropic efficiency of one stage of a multistage, axial flow air compressor. The stage efficiency remains constant for all stages of the compressor.

Figure 6.20 shows a stage of multistage axial flow compressor. ΔT_{0s} is the isentropic stagnation temperture drop and ΔT_0 is the actual temperature drop during the stage. Thus, the polytropic effciency is defined as

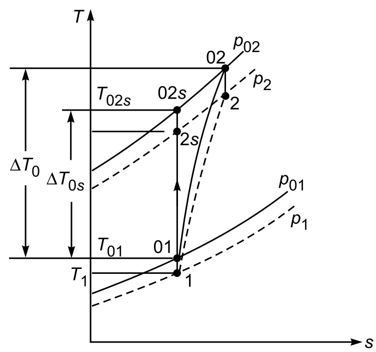


Fig. 6.20 *Polytropic compression process*

$$\eta_{poly} = \frac{\text{Isentropic tempature drop}}{\text{Actual temperature drop}} = \frac{dT_{0s}}{dT_0} \qquad ...(6.41)$$

For an irreversible polytropic process, the stagnation pressure and specific volume are related as

$$p_0 v_0^n = Z\,(\text{compressibility factor; a constant})$$

or $$p_0 = Z\rho_0^n$$

Differentiating both sides, we get

$$dp_0 = nZ\rho_0^{n-1}d\rho_0 = n\frac{p_0}{\rho_0}d\rho_0 \qquad ...(6.42)$$

From characteristic gas equation, ρ_0 can be expressed as

$$\rho_0 = \frac{p_0}{RT_0}$$

On differentiation; we get

$$d\rho_0 = \frac{1}{R}\left[\frac{T_0 dp_0 - p_0 dT_0}{T_0^2}\right] \qquad ...(6.43)$$

Using $d\rho_0$ in Eq. (6.41);

$$dp_0 = n\frac{p_0}{\rho_0} \times \frac{1}{R}\left[\frac{T_0 dp_0 - p_0 dT_0}{T_0^2}\right]$$

$$= nRT_0 \times \frac{1}{R}\left[\frac{T_0 dp_0 - p_0 dT_0}{T_0^2}\right]$$

$$= n\left[\frac{T_0 dp_0 - p_0 dT_0}{T_0}\right]$$

or $$dp_0 = n\,dp_0 - np_0\frac{dT_0}{T_0}$$

$$np_0\frac{dT_0}{T_0} = ndp_0 - dp_0 = dp_0(n-1)$$

Actual stage temperature;

$$dT_0 = dp_0\left(\frac{n-1}{n}\right)\frac{T_0}{p_0} \qquad \text{...(6.44a)}$$

Similarly, for isentropic compression path, stagnation pressure and temperature are related as

$$dT_{0s} = dp_0\left(\frac{\gamma-1}{\gamma}\right)\frac{T_0}{p_0} \qquad \text{...(6.44b)}$$

Subsituting Eqs. (6.44a) and (6.44b) in Eq. (6.40) to express the polytropic efficiency;

$$\eta_{poly} = \frac{dp_0\left(\frac{\gamma-1}{\gamma}\right)\frac{T_0}{p_0}}{dp_0\left(\frac{n-1}{n}\right)\frac{T_0}{p_0}}$$

$$= \left(\frac{\gamma-1}{\gamma}\right)\left(\frac{n}{n-1}\right) \qquad \text{...(6.45)}$$

Equation (6.45) expresses the polytropic efficiency in terms of polytropic and isentropic exponents.

Using $\left(\frac{n}{n-1}\right) = \frac{T_0}{p_0}\frac{dp_0}{dT_0}$ from Eq. (6.43), and subsituting in Eq. (6.45),

$$\eta_{poly} = \left(\frac{\gamma-1}{\gamma}\right)\frac{dp_0}{p_0} \times \frac{T_0}{dT_0}$$

or $$\eta_{poly}\left(\frac{dT_0}{T_0}\right) = \left(\frac{\gamma-1}{\gamma}\right)\frac{dp_0}{p_0}$$

Integrating between two states, we get

$$\eta_{poly}\ln\left(\frac{T_{02}}{T_{01}}\right) = \left(\frac{\gamma-1}{\gamma}\right)\ln\left(\frac{p_{02}}{p_{01}}\right)$$

or $$\eta_{poly} = \left(\frac{\gamma-1}{\gamma}\right)\frac{\ln\left(\frac{p_{02}}{p_{01}}\right)}{\ln\left(\frac{T_{02}}{T_{01}}\right)} \qquad \text{...(6.46)}$$

Equation (6.46) expresses the polytropic efficiency in terms of stagnaton pressure ratio and stagnation temperature ratio.

6.6.8 Flow Coefficient, Work Coefficient, and Pressure Coefficient

1. Flow Coefficient It is defined as the ratio of axial velocity ($\mathcal{V}_f$) to blade velocity (u) . It is designated as ϕ_f.

$$\phi_f = \frac{\mathcal{V}_f}{u}$$

From Eq. (6.38), $u = \mathcal{V}_f(\tan\alpha + \tan\beta)$

$$\therefore \qquad \phi_f = \frac{\mathcal{V}_f}{\mathcal{V}_f(\tan\alpha + \tan\beta)}$$

$$= \frac{1}{\tan\alpha + \tan\beta} \qquad \text{...(6.47)}$$

From symmetry of inlet and outlet velocity triangles for 50% reaction,

$$\beta = \theta \quad \text{and} \quad \alpha = \phi$$

The flow coefficient can also be expressed as

$$\phi_f = \frac{1}{\tan\phi + \tan\theta} \qquad \text{...(6.48)}$$

2. Work Coefficient It is defined as the ratio of actual work input to kinetic energy corresponding to mean peripheral velocity. It is also called *head coefficient* and designated as ϕ_h.

$$\phi_h = \frac{C_p\Delta T_{act}}{u^2/2} = \frac{u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})}{u^2/2} = \frac{2(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})}{u}$$

$$= 2\left(\frac{\tan\beta - \tan\phi}{\tan\theta + \tan\phi}\right) \qquad \text{...(6.49)}$$

3. Pressure Coefficient It is defined as ratio of isentropic work input to kinetic energy corresponding to mean peripheral velocity. It is denoted by ϕ_p.

$$\phi_p = \frac{C_p \Delta T_{isen}}{u^2/2} = \eta_C \, \phi_h \qquad ...(6.50)$$

6.6.9 Losses in Axial-Flow Compressor

The losses in an axial-flow compressor can be divided in three groups:

1. Profile Losses The blade geometry of an axial compressor is two dimensional. The air flow along the profile of blade experiences skin friction. Further, the different streams of air are mixed after passing on blades. These losses lead to pressure loss of compressed air.

2. Annulus Losses When air flows through the annulus passage of the compressor, it experiences growth of boundary layer and skin friction. Therefore, there is loss of pressure of compressed air.

3. Secondary Losses In an axial-flow air compressor, the certain secondary flows are generated by combined effect of curvature of blade and growth of boundary layer in the annulus. The air is deflected by curvature of blades and bends in pipe, etc. It causes loss in pressure of compressed air.

6.7 CHOKING, SURGING AND STALLING

6.7.1 Choking

At constant rotor speed of compressor, the decrease in pressure ratio leads to an increase in mass-flow rate, consequently, increase in fluid velocity. On further, reduction in pressure ratio, a situation attains, when fluid velocity reaches a sonic velocity and mass flow rate through the compressor reaches a maximum value. The slope of the characteristic curve decreases and finally the point A is reached as shown in Fig. 6.21. The mass-flow rate of fluid cannot be increased beyond the point A. This point is called *choking state.*

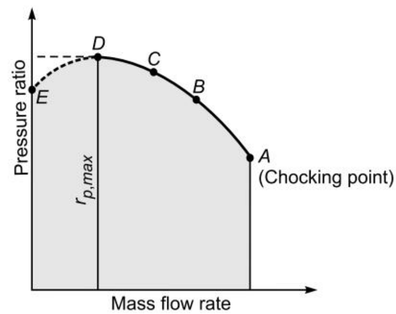


Fig. 6.21 *Typical characteristic of rotary compressor at constant speed*

At choking condition, the pressure ratio in the compressor becomes unity, i.e., there is no compression. Choking means constant mass flow irrespective of pressure ratio.

6.7.2 Surging

Surging is defined as a *self-oscillation* of the discharge pressure and flow rate, including a flow reversal. Every centrifugal or axial compressor has a characteristic combination of maximum head and minimum flow. Beyond this point, surging will occur.

Consider a compressor is running at constant speed and the valve is open on delivery side at a pressure ratio located by the point B on the characteristic curve shown in Fig. 6.21. If any resistance in delivery path of a compressor, or by partial closing of the valve, the mass-flow rate decreases with an increase in pressure ratio, and the operating point will shift towards left on characteristic curve, say at the point C from the point B and on further increase in resistance, point D is reached, the mass-flow rate of air through the compressor decreases and the compressor is operating for its maximum pressure ratio.

If resistance is further increased, the mass-flow rate will decrease and reach the zone *D–E* on the curve, with decrease in pressure ratio also. In this situation, the pressure in downstream line will be more than the pressure of air at the actual delivery of the compressor. This situation leads to a stop of fluid flow and sometimes, flow of fluid in reverse direction. This situation of instability is known as *surging* or *pumping*.

Surging is caused when mass-flow rate of fluid is reduced to a value which is less than the predetermined value. Surging brings the compressor into the unstable area of the operating curve. During *surging,* the compressor shows cyclic flow and back-flow of the compressed air resulting into high vibrations, pressure shock, overheating and sometimes heavy damages.

Effect of Surging Consequences of surging can include

1. Rapid flow and pressure oscillations causing process instabilities
2. Rising temperatures inside the compressor
3. Tripping of the compressor
4. Mechanical damage

Mechanical damage can include

✓ Radial bearing load during the initial phase of surging
✓ Thrust bearing load due to loading and unloading
✓ Seal rubbing
✓ Stationary and rotating part contact, if thrust bearing is overloaded

6.7.3 Stalling

Stalling occurs on blades of an axial flow compressor. It is a situation of abnormal airflow resulting from a reduction in the lift coefficient generated by an airfoil within the compressor. The stalling is the separation of flow from blade surface as shown in Fig. 6.22.

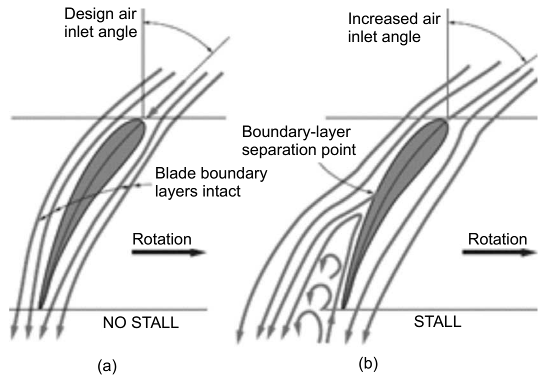


Fig. 6.22 *Flow of air over a blade at designed condition and with an increase in air inlet angle*

The flow around the blade is controlled by its *boundary layer***.** When moving air stream meets the airfoil blade at a certain angle of attack, a lift and drag are produced. Angle of attack is the inclination of chord of blade and air stream. The aerodynamic force acts through centre of pressure and normal to the centre line as shown in Fig. 6.23. This force has two components

1. Drag force, which is parallel to air stream

$$= \frac{1}{2} C_d \rho \mathcal{V}^2 A$$

2. Lift, that is a force perpendicular to air stream

$$= \frac{1}{2} C_l \rho \mathcal{V}^2 A$$

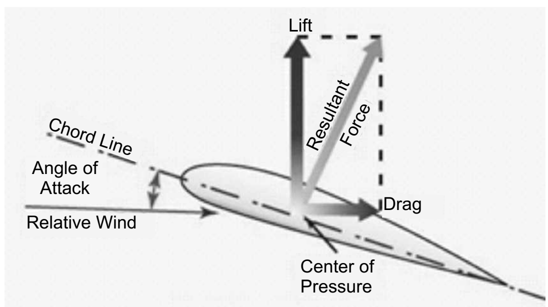


Fig. 6.23 *Forces on airfoil blade*

Where C_d and C_l are coefficient of drag and lift, respectively. The variation of lift coefficient with change in angle of incidence for a typical cascade (row of airfoils) test is plotted in Fig. 6.24.

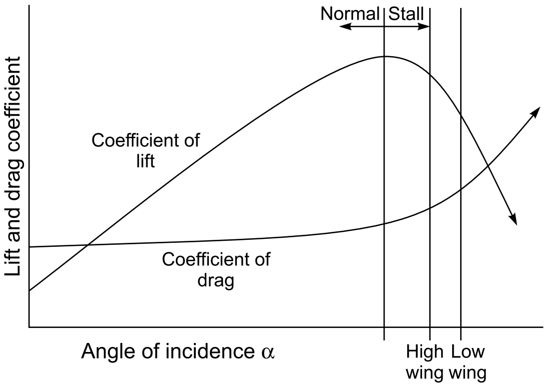


Fig. 6.24 *Variation of drag and lift coefficients with angle of incidence*

It is observed that lift coefficient C_l increases with increase in angle of incidence upto certain maximum value. With further increase in angle of incidence, the lift coefficient declines, but drag coefficient increases rapidly. After angle of incidence corresponds to maximum lift coefficient, the flow separation of fluid occurs on the surface of blade as shown in Fig. 6.22(b). This separation causes violent eddies and distortion in fluid flow. With an increase in angle of incidence, area of separation flow goes on increasing with an increase in drag coefficient.

At the large value of incidence, the flow separation occurs at suction side of the blade, which is referred as *positive stalling*. *Negative stalling* is due to separation of flow occurring on the delivery side of the blade due to large value of negative incidence.

6.8 DIFFERENCE BETWEEN CENTRIFUGAL AND AXIAL FLOW COMPRESSORS

Sr. No.	*Centrifugal Compressor*	*Axial-flow Compressor*
1.	Air flows radially in the compressor.	Air flows parallel to axis of shaft.
2.	Low maintenance and running cost.	High maintenance and running cost.
3.	Low starting torque required.	Requires high starting torque.
4.	Not suitable for multistaging.	Suitable for only multistaging.
5.	Suitable for low pressure ratios up to 4.	Suitable up to a pressure ratio of 10.
6.	For given mass-flow rate, it requires, larger frontal area.	It requires less frontal area.
7.	Isentropic efficiency 80 to 82%.	Isentropic efficiency 86 to 88%.
8.	Better performance at part load.	Poor performance at part load.

Example 6.12 *An axial-flow compressor having 10 stages works with 50% degree of reaction. It compresses air with a pressure ratio of 5. The inlet conditions of air are 27°C and 100 kPa. The air enters the compressor with a velocity of 110 m/s. The mean speed of the rotor blade is 220 m/s. The isentropic efficiency of the compressor is 85%. Calculate the work input per kg of air and blade angles.*

Solution

Given An axial-flow air compressor with

$R_d = 0.5$ $\qquad N_{stages} = 10$

$p_1 = 100$ kPa $\qquad T_1 = 27°C = 300$ K

$p_2 = 500$ kPa $\qquad \mathcal{V}_1 = 110$ m/s

$\eta_C = 0.85$

$u = 220$ m/s

To find

(i) Power input, and
(ii) Blade angles.

Assumptions

1. The specific heat of air as 1005 J/kg·K and $\gamma = 1.4$
2. Effect of kinetic energy change on fluid properties is negligible.

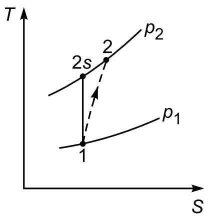


Fig. 6.25

Analysis The temperature after isentropic compression

$$T_{2s} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300 \times (5)^{\frac{1.4-1}{1.4}}$$

$$= 475.14 \text{ K}$$

The isentropic efficiency is given by

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1}$$

Final temperature of air is given by

$$T_2 = \frac{T_{2s} - T_1}{\eta_C} + T_1 = \frac{475.14 - 300}{0.85} + 300$$

$$= 506 \text{ K}$$

(i) Work input to compressor per kg of air

$$w = C_p(T_2 - T_1) = 1.005 \times (506 - 300)$$
$$= \mathbf{207 \text{ kJ/kg}}$$

Work input per kg is also given by Eq. (6.33),

$$w = u\mathcal{V}_f(\tan\theta - \tan\alpha) \times \text{No. of stages}$$

Using symmetry of velocity triangles, $\theta = \beta$ and numerical values;

$$207 \times 10^3 = 220 \times 110 \times (\tan\beta - \tan\alpha) \times 10$$

$$\tan\beta - \tan\alpha = 0.855 \quad \text{(i)}$$

Further form Eq. (6.38) with $\theta = \beta$;

$$\frac{u}{\mathcal{V}_f} = \tan\beta + \tan\alpha$$

or $$\frac{220}{110} = \tan\beta + \tan\alpha$$

or $$\tan\beta + \tan\alpha = 2 \quad \text{...(ii)}$$

Solving Eqs. (i) and (ii), we get

$$2\tan\beta = 2.855$$

or $$\beta = \mathbf{55°}$$

and $$\alpha = \mathbf{29.79°}$$

Example 6.13 *A 50% reaction axial flow compressor runs at a mean blade velocity of 240 m/s. For pressure ratio of 1.5, determine blade and air angles, if mean flow velocity is 160 m/s. Inlet conditions are 1 bar and 300 K.*

Solution

Given An axial flow compressor with

$R_d = 0.5$ $\quad u = 240$ m/s
$r_p = 1.5$ $\quad \mathcal{V}_f = 160$ m/s
$p_1 = 1$ bar $\quad T_1 = 300$ K

To find

1. Blade angle
2. Air angle

Analysis For inlet conditions of axial flow compressor, the temperature after compression

$$\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$T_{2s} = 300 \times (1.5)^{\frac{1.4-1}{1.4}} = 336.85 \text{ K}$$

Stage temperature difference

$$\Delta T_{st} = T_{2s} - T_1 = 336.85 \text{ K} - 300 = 36.85 \text{ K}$$

For 50% degree of reaction in axial compressor

$$\frac{u}{\mathcal{V}_f} = \tan\beta + \tan\alpha$$

$$\frac{240}{160} = \tan\beta + \tan\alpha$$

or $$\tan\beta + \tan\alpha = 1.5 \quad \text{…(i)}$$

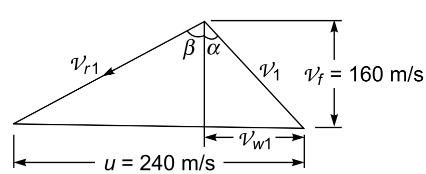


Fig. 6.26

Further work input per stage for blade symmetry

$$C_p(T_{2s} - T_1) = u\mathcal{V}_f(\tan\beta - \tan\alpha)$$

$$\tan\beta - \tan\alpha = \frac{1005 \times 36.85}{240 \times 160} = 0.964 \quad \ldots\text{(ii)}$$

Adding Eqns (i) and (ii), we get

$$2\tan\beta = 3.22$$

$$\tan\beta = 1.61$$

or $$\boldsymbol{\beta = 72.74°}$$

On subtracting Eqn (i) from (ii), we get

$$2\tan\alpha = 0.535$$

$$\tan\alpha = 0.267$$

or $$\boldsymbol{\alpha = 15°}$$

Example 6.14 *An axial flow compressor compresses air from an inlet pressure of 1 bar and 290 K to a delivery pressure of 5 bar with an overall isentropic efficiency of 87%. The degree of reaction is 50% and blade angles at inlet and outlet are 44 and 13 degree, respectively. The mean velocity is 180 m/s and axial velocity is constant throughout the compressor and work done factor is 0.87. Calculate the number of stages.*

Solution

Given Multistage axial compressor

$p_1 = 1$ bar $\quad T_1 = 290$ K $\quad p_2 = 5$ bar
$\eta_C = 0.87 \quad R_d = 0.5 \quad u = 180$ m/s
$\phi_h = 0.87 \quad \beta = 44° \quad \alpha = 13°$

To find Number of stages

Assumptions

(i) For air $\gamma = 1.4$
(ii) Specific heat of air at constant pressure

$$C_p = 1.005 \text{ kJ/kg K}$$

Analysis Temperature after isentropic compression

$$T_{2s} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 290 \times \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = 459.3 \text{ K}$$

$$(\Delta_T)_{overall} = \frac{T_{2s} - T_1}{\eta_C} = \frac{459.3 - 290}{0.87} = 194.6 \text{ K}$$

The blade velocity and flow velocity are related as

$$\frac{u}{\mathcal{V}_f} = \tan\beta + \tan\alpha = \tan 44° + \tan 13° = 1.196$$

or $$\mathcal{V}_f = \frac{180}{1.196} = 150.43 \text{ m/s}$$

whirl velocity at inlet

$$\mathcal{V}_{w_1} = \mathcal{V}_f \tan\phi = \mathcal{V}_f \tan\alpha = 150.43 \times \tan 13° = 34.73 \text{ m/s}$$

whirl velocity at exit

$$\mathcal{V}_{w_1} = \mathcal{V}_f \tan\beta = 150.43 \times \tan 44° = 145.27 \text{ m/s}$$

Work input per stage of compression

$$w_1 = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})\phi_h$$
$$= 180 \times (145.27 - 34.73) \times 0.87$$
$$= 17310.56 \text{ J/kg} = 17.31 \text{ kJ/kg}$$

Total work input to compressor

$$W_{th} = C_p(\Delta T)_{overall}$$
$$= 1.005 \times (194.6) = 195.57 \text{ kJ/kg}$$

Number of stages of compression

$$N_{stages} = \frac{w_{th}}{w_1} = \frac{195.57}{17.31} = \mathbf{11.29 \approx 12\ stages}$$

Example 6.15 *An axial-flow compressor draws air at 20°C and delivers it at 50°C. Assuming 50% reaction, calculate the velocity of flow, if blade velocity is 100 m/s, work factor is 0.85. Take C_p = 1 kJ/kg·K, Assume α = 10°, and β = 40°. Find the number of stages.*

Solution

Given An axial-flow air compressor with

$R_d = 0.5 \quad T_1 = 20°\text{C} = 293$ K
$T_2 = 50°\text{C} = 323$ K $\quad \alpha = 10°$
$\beta = 40° \quad u = 100$ m/s
$\phi_h = 0.85 \quad C_p = 1$ kJ/kg·K

To find

(i) Flow velocity, and
(ii) Number of stages.

Analysis The blade velocity and flow velocity are related as

$$\frac{u}{\mathcal{V}_f} = \tan\beta + \tan\alpha$$

or $$\frac{100}{\mathcal{V}_f} = \tan 40° + \tan 10°$$

or $\quad 100 = 1.015\mathcal{V}_f$

or $\quad \mathcal{V}_f = \mathbf{98.48\ m/s}$

Further,

$$\mathcal{V}_{w_1} = \mathcal{V}_f \tan\alpha = 98.48 \times \tan 10° = 17.36 \text{ m/s}$$

and $$\mathcal{V}_{w_2} = \mathcal{V}_f \tan\theta = \mathcal{V}_f \tan\beta = 98.48 \times \tan 40° = 82.63 \text{ m/s}$$

Work done per stage

$$w_1 = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})\phi_h = 100 \times (82.63 - 17.36) \times 0.85 = 5548.33 \text{ J/kg}$$

Theoretical work required for a compressor

$$w = C_p(T_2 - T_1) = 1005 \times (323 - 293) = 30150 \text{ J/kg}$$

Number of stages

$$= \frac{\text{Theoretical work}}{\text{Work input per stage}} = \frac{30150}{5548.33} = 5.43 \approx \mathbf{6\ stages}$$

Example 6.16 *In an axial flow compressor, overall stagnation pressure ratio achieved is 4 and overall stagnation isentropic efficiency is 85%. The inlet stagnation pressure and temperature are 1 bar and 300 K, respectively. The mean blade velocity is 180 m/s. The degree of reaction is 50% at mean radius with relative air angles of 12° and 32° at rotor inlet and outlet, respectively. The work done factor is 0.9. Calculate*

(a) Stagnation polytropic efficiency,
(b) Inlet temperature and pressure,
(c) Number of stages,
(d) Blade height in first stage, if ratio of hub-to-tip diameter is 0.42, mass-flow rate is 19.5 kg/s.

Solution

Given An axial-flow air compressor with

$R_d = 0.5$ $\qquad p_{01} = 100$ kPa

$T_{01} = 300$ K $\qquad \dot{m} = 19.5$ kg/s

$\dfrac{p_{02}}{p_{01}} = 4$ $\qquad u = 180$ m/s

$\eta_C = 0.85$ $\qquad \phi_h = 0.9$

$\alpha = \phi = 12°$ $\qquad \beta = \theta = 32°$

$r_h = 0.42 r_1$

To find

(i) Stagnation Polytropic efficiency,
(ii) Inlet temperature and pressure,
(iii) Number of stages, and
(iv) Blade height in first stage.

Assumption The specific heat of air as 1005 J/kg·K and $\gamma = 1.4$.

Analysis Inlet and outlet velocity triangles are shown in Fig. 6.27.

(i) *Stagnation polytropic efficiency:*

Stagnnation temperature after isentropic compression

$$T_{02} = T_{01}\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} = 300 \times (4)^{\frac{1.4-1}{1.4}} = 445.8 \text{ K}$$

The isentropic efficiency is given by

$$\eta_C = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}}$$

Final stagnation temperature of air

$$T_{02} = \frac{T_{02s} - T_{01}}{\eta_C} + T_{01} = \frac{445.8 - 300}{0.85} + 300 = 471.52 \text{ K}$$

The stagnation polytropic efficiency

$$\eta_{poly} = \left(\frac{\gamma-1}{\gamma}\right)\frac{\ln\left(\frac{p_{02}}{p_{01}}\right)}{\ln\left(\frac{T_{02}}{T_{01}}\right)} = \left(\frac{1.4-1}{1.4}\right)\frac{\ln\left(\frac{4}{1}\right)}{\ln\left(\frac{471.52}{300}\right)}$$

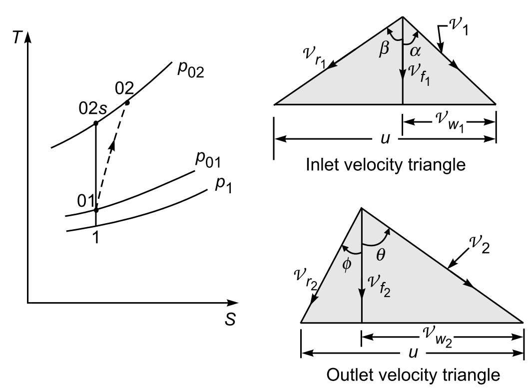


Fig. 6.27

$= 0.8759$ or **87.59%**

(ii) *Inlet temperature and pressure:*

The blade velocity-to-flow velocity ratio is given by Eq. (6.38);

$$\frac{u}{\mathcal{V}_f} = \tan\alpha + \tan\beta = \tan 12° + \tan 32°$$
$$= 0.8374$$

or $$\mathcal{V}_f = \frac{180}{0.8374} = 214.95 \text{ m/s}$$

Inlet air velocity;

$$\mathcal{V}_1 = \frac{\mathcal{V}_f}{\cos\alpha} = \frac{214.95}{\cos 12°} = 219.75 \text{ m/s}$$

Whirl velocity at inlet;

$$\mathcal{V}_{w_1} = \mathcal{V}_f \tan\alpha = 214.95 \times \tan 12°$$
$$= 45.69 \text{ m/s}$$

Whirl velocity at outlet;

$$\mathcal{V}_{w_2} = \mathcal{V}_f \tan\theta = 214.95 \times \tan 32°$$
$$= 134.31 \text{ m/s}$$

Work input per kg of air per stage

$$w_{stage} = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})\,\phi_h$$
$$= 180 \times (134.31 - 45.69) \times 0.9$$
$$= 14357.46 \text{ J/kg}$$

The stganation temperature is given by

$$T_{01} = T_1 + \frac{\mathcal{V}_1^2}{2C_p}$$

The static temperature at inlet

$$T_1 = T_{01} - \frac{\mathcal{V}_1^2}{2C_p} = 300 - \frac{219.75^2}{2 \times 1005}$$
$$= \mathbf{276\ K}$$

Inlet pressure can be expressed as

$$p_1 = p_{01}\left(\frac{T_1}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}} = 1 \times \left(\frac{276}{300}\right)^{\frac{1.4}{1.4-1}}$$
$$= \mathbf{0.746\ bar}$$

(iii) *Number of stages*

Total work input to compressor per kg of air in all stages;

$$w_T = C_p(T_{02} - T_{01})$$
$$= 1005 \times (471.52 - 300)$$
$$= 172385 \text{ J/kg}$$

No. of stages

$$= \frac{w_T}{w_{stage}} = \frac{172385}{14356.44} \approx \mathbf{12}$$

(iv) *Blade height in first stage*

Density of air entering the first stage

$$\rho = \frac{p_1}{RT_1} = \frac{0.746 \times 100}{0.287 \times 276} = 0.9425 \text{ kg/m}^3$$

The mass-flow rate is given by

$$\dot{m} = \rho A_c \mathcal{V}_f = \rho\pi(r_1^2 - r_h^2)\mathcal{V}_f$$

or $$19.5 = 0.9425 \times \pi \times r_1^2(1 - 0.42^2) \times 214.95$$
$$(\because\ r_h = 0.42 r_1)$$

or $$r_1^2 = 0.03720$$

It gives $r_1 = 0.1928$ m

and $r_h = 0.42\, r_1 = 0.081$ m

Height of blade in first stage

$$= r_1 - r_h = 0.1928 - 0.081$$
$$= 0.1118 \text{ m or } \mathbf{11.18\ cm}$$

Example 6.17 *An axial-flow compressor has a constant axial velocity of 150 m/s and 50% reaction . The mean daimeter of the blade ring is 35 cm and speed is 15,000 rpm.. The exit angle of the blade is 27°. Calculate blade angle at inlet and work done per kg of air.*

Solution

Given An axial flow air compressor with

$R_d = 0.5$ $\qquad \mathcal{V}_{f_1} = 150$ m/s

$D = 35$ cm $= 0.35$ m $\qquad \alpha = \phi = 27°$

$N = 15{,}000$ rpm

To find

(i) Blade angle β at inlet, and

(ii) Work input per kg of air.

Analysis The mean velocity of the blade ring

$$u = \frac{\pi DN}{60} = \frac{\pi \times 0.35 \times 15000}{60}$$
$$= 274.89 \text{ m/s}$$

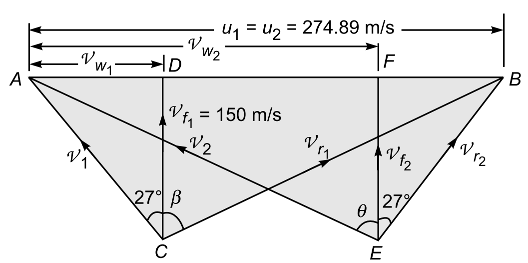


Fig. 6.28

Construction of velocity triangles, Fig. 6.28

(i) Draw a horizontal line *AB* to represent the blade velocity $u = 274.89$ m/s.

(ii) Through the point *A*, draw an inclined line at angle $90° - \alpha = 63°$.

(iii) Mark the intersection point *C* and draw a vertical line *CD* representing flow velocity of 150 m/s at inlet.

(iv) Through the point *B*, draw an inclined line *BE* at angle $90° - 27° = 63°$.

(v) Mark the intersection point *E* and draw a vertical line *EF* representing flow velocity of 150 m/s at outlet.

(vi) Join the point *E* with *A* and the point *C* with *B*.

(vii) Line *CB* represents relative velocity $\mathcal{V}_{r_1}$ at the inlet and the line *EA* represents relative velocity $\mathcal{V}_{r_2}$ at the outlet.

(viii) Measure the $\angle AEF$ or $\angle DCB = 53°$.

(ix) Measure $AD = \mathcal{V}_{w_1} = 76.5$ m/s, and $AF = \mathcal{V}_{w_2} = 198.5$ m/s.

Work input per kg of air

$$w = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})$$
$$= 274.89 \times (198.5 - 76.5)$$
$$= 33536.5 \text{ J/kg} \quad \text{or} \quad \mathbf{33.53\ kJ/kg}$$

Example 6.18 *An axial-flow compressor of 50% reaction has a blade outlet angle of 30°. The flow velocity is 0.5 times the mean blade velocity. The speed of the rotor is 7500 rpm. The stagnation condition of air at the entry is 1.013 bar and 5°C and the static pressure at this section is 0.91 bar. Draw the velocity triangle and find the power required to run the compressor, mass-flow rate and mean diameter of rotor. The mean flow area is 0.35 m^2.*

Solution

Given An axial-flow air compressor with

$R_d = 0.5$ $\qquad \mathcal{V}_{f_1} = 0.5\,u$

$p_{01} = 1.013$ bar $\qquad T_{01} = 5°\text{C} = 278$ K

$p_1 = 0.91$ bar $\qquad \alpha = \phi = 30°$

$N = 7500$ rpm $\qquad A = 0.35$ m^2

To find

(i) Mean diameter of rotor,

(ii) Power input per kg of air, and

(iii) Mass flow rate of air.

Assumption The specific heat of air as 1005 J/kg·K and $\gamma = 1.4$

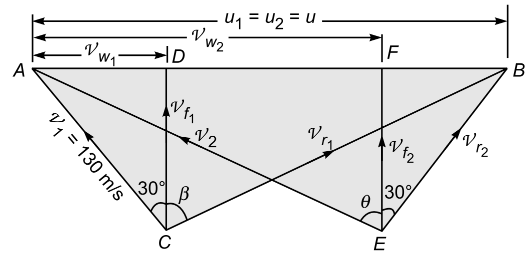


Fig. 6.29

Analysis The static temperature

$$T_1 = T_{01}\left(\frac{p_1}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} = 278 \times \left(\frac{0.91}{1.013}\right)^{\frac{1.4-1}{1.4}}$$
$$= 269.6 \text{ K}$$

The stagnation temperature is given by

$$T_{01} = T_1 + \frac{\mathcal{V}_1^2}{2C_p}$$

It gives

$$\mathcal{V}_1 = \sqrt{2C_p(T_{01} - T_1)} = \sqrt{2 \times 1005 \times (278 - 269.6)}$$
$$= 130 \text{ m/s}$$

Flow velocity

$$\mathcal{V}_{f_1} = \mathcal{V}_{f_2} = \mathcal{V}_1 \cos\alpha$$
$$= 130 \times \cos 30° = 112.5 \text{ m/s}$$

$$\text{Blade velocity, } u = \frac{\mathcal{V}_f}{0.5} = \frac{112.5}{0.5} = 225 \text{ m/s}$$

For 50% reaction in axial flow compressor

$$\alpha = \phi, \quad \text{and} \quad \beta = \theta$$

Construct the combined velocity triangles as shown in Fig. 6.29 with following steps:

(i) Draw an inclined line *AC* to represent air velocity, $\mathcal{V}_1 = 130$ m/s at an angle $(90°–30°) = 60°$.

(ii) Draw a vertical line *CD*, which represents axial velocity of air. Its length is equivalent to $\mathcal{V}_{f_1}$ 112.5 m/s.

(iii) The line segment *AD*, represents whirl velocity at the inlet. Its length is equivalent to $\mathcal{V}_{w_1} = 65$ m/s.

(iv) Extend the line *AD* to the point *B* to represent $u = 225$ m/s.

(v) From the point *B,* draw a line *BE* inlined at angle $(90° - 30°) = 60°$.

(vi) Locate the point *E* on the line *BE* by drawing a vertical line $EF = \mathcal{V}_{f_2} = \mathcal{V}_{f_1} = 112.5$ m/s.

(vii) Length of line *BE* represents relative velocity at outlet $\mathcal{V}_{r_2} = \mathcal{V}_1$.

(viii) From the point *B,* meet the point C by a line *BC* to represent $\mathcal{V}_{r_1} = \mathcal{V}_2 = 195.5$ m/s.

(ix) Length of line *AF* represents $\mathcal{V}_{w_2} = 160$ m/s.

Mean diameter of rotor:

The mean velocity of blade ring,

$$u = \frac{\pi DN}{60}$$

or $$D = \frac{60u}{\pi N} = \frac{60 \times 225}{\pi \times 7500} = \mathbf{0.573\ m}$$

Power input per kg of air

$$w = u(\mathcal{V}_{w_2} - \mathcal{V}_{w_1})$$
$$= 225 \times (160 - 65)$$
$$= 21375 \text{ J/kg} \quad \text{or} \quad \mathbf{21.37\ kJ/kg}$$

Mass-flow rate of air

Density of air $$\rho = \frac{p_1}{RT_1} = \frac{0.91}{0.287 \times 278}$$
$$= 1.176 \text{ kg/m}^3$$

Mass-flow rate, $$\dot{m} = \rho A \mathcal{V}_{f_1} = 1.176 \times 0.35 \times 112.5$$
$$= \mathbf{46.3\ kg/s}$$

Summary

- Rotary compressors have rotors in place of piston in reciprocating ones and give a continuous, pulsation-free compressed air. The rotary compressors are mainly classified as rotary positive-displacement type compressor and steady-flow type compressor.
- In a positive-displacement type compressor, the air is compressed by being trapped in the reduced space formed by two sets of engaging surfaces. In a non-positive dispacement or steady-flow type compressor, the air flows continuously through them and pressure is increased due to dynamic action.
- The *roots blower* is essentially a low-pressure blower and is limited to a discharge pressure of 1 bar in single-stage design and up to 2.2 bar in a two-stage design.
- *The Lysholm compressor* is a patented screw compressor, a single-stage helical lobe, oil flooded air compressor.
- The *centrifugal compressors* are dynamic action compressors. The centrifugal air compressor is an oil free compressor by design. Centrifugal machines are better suited due to their simplicity, light weight and ruggedness.
- *Axial compressors* are dynamic action, rotating, aerofoil blade compressors. Axial flow compressors produce a continuous flow of compressed gas, and have the benefits of high efficiencies and large mass flow capacity, particularly in relation to their cross-section.

Glossary

Degree of Reaction Ratio of static pressure rise in impeller to the total static pressure rise in the compressor

Slip factor Ratio of whirl velocity to blade tip velocity

Euler's work Product of blade velocity and whirl velocity ($w = u_2\mathcal{V}_{w_2}$)

Work factor Ratio of actual work input to Euler work input

Pressure coefficient Ratio of isentropic work to Euler work

Choking State of maximum mass-flow rate

Surging State of alternatively forward and backward flow of fluid

Stalling Separation of flow from blade surface

Polytropic Efficiency The isentropic efficiency of one stage of a multistage axial flow air compressor

Review Questions

1. Define rotary compressor. Classify them.
2. Differentiate between positive displacement and negative displacement compressors.
3. Compare reciprocating compressor with a rotary compressor.
4. Explain construction and working of a roots blower.
5. Explain construction and working of a vane-type compressor.
6. Explain working and construction of a screw compressor.
7. Describe the principle of operation, construction and working of centrifugal compressor.
8. Discuss the effect of impeller blade shape on performance of centrifugal compressor.
9. Define slip, slip factor and pressure coefficient.
10. Explain the construction and working of a diffuser in a centrifugal compressor.
11. Explain the phenomenon of surging and its effects in the compressor.
12. Prove that the work input per kg of air in an axial flow compressor is $w = u\mathcal{V}_w(\tan\beta - \tan\alpha)$
13. Compare axial flow compressor with centrifugal one.
14. What is stalling in an axial flow compressor?

Problems

1. Compare the work inputs required for a roots blower and a vane-type compressor having same volume inducted of 0.3 m^3/rev. The inlet pressure is 1.013 bar and pressure ratio is 1.5 in a compressor. For vane-type compressor, assume half the compression takes place through half the pressure range. *[1.52 kJ, 1.352 kJ]*

2. A roots blower compresses 0.08 m^3 of air from 100 kPa to 150 kPa per revolution. Calculate compressor efficiency. *[85.95%]*

3. A centrifugal compressor running at 2000 rpm has internal and external diameters of impeller as 300 mm and 500 mm, respectively. The vane angle at inlet and outlet are 22° and 40°, respectively. The air enters the impeller radially. Determine the work done by a compressor per kg of air. Also calculate the degree of reaction.

[1.95 kJ/kg, 64.5%]

4. A rotary compressor handles 3 kg of air per second, runing at 2400 rpm. The internal and external diameters of the impeller are 120 mm and 240 mm, respectively. The impeller angle at the exit is 35°. The air enters the impeller radially with 7 m/s. Calculate the vane angle at inlet. Also calculate the power required to drive the compressor. *[25°, 1.83 kW]*

5. A centrifugal compressor runs at 8000 rpm, handles 4.8 m^3/s from 1 bar and 20°C to 1.5 bar. The index of compression is 1.5. The flow velocity is 65 m/s, same at the inlet and outlet of compressor. The inlet and outlet impeller diameters are 320 mm and 620 mm, respectively. Calculate (a) blade angle at inlet and outlet, (b) absolute angle at tip of impeller, and (c) width of blade at inlet and outlet.

[(a) β = 25.88°, ϕ = 34.2°, (b) θ = 21.6°, (c) 7.34 cb and 2.89 cm]

6. A centrifugal compressor runs at 1440 rpm, compresses air from 101 kPa , 20° to a pressure of 6 bar isentropically. The inner and outer diameters of the impeller are 140 mm and 280 mm, respectively. The width of blades at inlet is 25 mm. The blade angles are 16° and 40° at entry and exit. Determine the mass-flow rate of air, degree of reaction, power developed and width of blade at outlet.

[0.4 kg/s, 14.78 W, 58.6%, 3.5 mm]

7. A centrifugal compressor handles 16.5 kg/s of air with total pressure ratio of 4. The speed of the compressor is 15000 rpm. Inlet stagnation temperature is 20°, slip factor is 0.9, power input factor is 1.04 and isentropic efficiency as 80%.

Calculate diameter of impeller and power input to the compressor. *[55.67 cm, 2951.7 kW]*

8. An axial-flow compressor stage has a mean diameter of 600 mm and runs at 15000 rpm. The mass-flow rate through the compressor is 50 kg/s. Calculate the power required to drive the compressor and degree of reaction, if the inlet angle is 12°. The blade angle at the inlet and exit are 35° and 27°, respectively. *[2318 kW, 66.55]*

9. An axial flow compressor, with a compression ratio of 4, draws air at 20°C and delivers it at 97°C. The blade velocity and flow velocity are constant throughout the compressor. The blade velocity is 300 m/s. Air enters the blade at an angle of 12°. Calculate the flow velocity, work done per kg of air and degree of reaction. Take the inlet stagnation temperature of 305 K.
[152 m/s, 77.38 kg/s, 46.2%]

10. An axial flow compressor with 50% degree of reaction has blades with inlet and outlet angles of 45° and 10°, respectively. The pressure ratio is 6 and isentropic efficiency is 85%, when the air inlet temperature is 40°C. The blade velocity is 200 m/s. The blade velocity and axial velocity are constant throughout the compressor. Calculate the number of stages required, when work factor is (a) unity, and (b) 0.89 for all stages.
[(a) 9 (b)10]

11. Air from a quiescent atmosphere, at pressure of 1 bar and 300 K enters a centrifugal compressor, fitted with radial vanes and the air leaves the diffuser with negligible velocity. The tip diameter of the impeller is 450 mm and the compressor rotates at 18000 rpm. Neglect all losses and calculate the temperature and pressure of air as it leaves the compressor. Take $\gamma = 1.4$, C_p = 1.005 kJ/kg·K, and $\phi_s = 0.9$. *[188°C, 4.5 bar]*

Objective Questions

1. Which one of the following is a non-positive type rotary compressor?
   (a) Vane blower
   (b) roots blower
   (c) Centrifugal compressor
   (d) Lysholm compressor
2. A machine is called a compressor when it has a pressure ratio
   (a) up to 1.11 (b) up to 1.2
   (c) more than 1.2 (d) none of the above
3. In a roots blower, the compression process can be represented by
   (a) isothermal line
   (b) isentropic line
   (c) constant-volume line
   (d) constant-pressure line
4. In a roots blower, the pressure is increased due to
   (a) rotation of lobes
   (b) increase in mass
   (c) back flow of air
   (d) reduction in volume of air
5. In a centrifugal compressor, the increase in pressure is due to
   (a) back flow of air
   (b) dynamic action
   (c) intermittent flow
   (d) reduction in volume
6. The efficiency of a roots blower is given as
   (a) $\dfrac{\text{Isentropic work input}}{\text{Actual work input}}$
   (b) $\dfrac{\text{Actual work input}}{\text{Isentropic work input}}$
   (c) $\dfrac{\text{Actual work input}}{\text{Isothermal work input}}$
   (d) $\dfrac{\text{Isothermal work input}}{\text{Actual work input}}$
7. The stages are arranged in a multisatge compressor in
   (a) parallel (b) series
   (c) cross connected (d) none of the above
8. In a centrifugal compressor, the pressure rise takes place in

(a) impeller only
(b) diffuser only
(c) casing only
(d) impeller and diffuser both

9. The degree of reaction in a centrifugal compressor is defined as

(a) $\dfrac{\text{Pressure rise in diffuser}}{\text{Pressure rise in impeller}}$

(b) $\dfrac{\text{Pressure rise in impeller}}{\text{Pressure rise in compressor}}$

(c) $\dfrac{\text{Pressure rise in diffuser}}{\text{Pressure rise in compressor}}$

(d) none of the above

10. Pressure coefficient is defined as

(a) $\dfrac{\text{Actual work input}}{\text{Euler work input}}$

(b) $\dfrac{\text{Blade velocity}}{\text{Whirl velocity}}$

(c) $\dfrac{\text{Isentropic work input}}{\text{Euler work input}}$

(d) none of the above

11. Forward curved impeller vane has a blade exit angle of

(a) $< 90°$ (b) $> 90°$
(c) $= 90°$ (d) none of the above

12. At the state of choking in compressor,

(a) mass-flow rate reaches a minimum value
(b) mass-flow rate reaches a maximum value
(c) pressure ratio reaches a maximum value
(d) pressure ratio reaches a minimum value

Answers

1. (c)	2. (c)	3. (c)	4. (c)	5. (b)	6. (a)	7. (b)	8. (d)
9. (b)	10. (c)	11. (b)	12. (b)				

7 CHAPTER

Refrigeration

Introduction

A refrigerator and a heat pump are both heat engines operating in reverse direction. In operation, the reversed heat engine transfers heat energy from a low-temperature source to a high-temperature sink. The work input is required to transfer the heat from cooler to hotter region. A refrigerator or air conditioner removes heat energy from a low-temperature region while a heat pump delivers heat to a high-temperature region. The gas refrigeration cycle, vapour compression refrigeration cycle, vapour absorption cycle, heat pump cycle and refrigerant properties are explained in this chapter.

7.1 REFRIGERATION

Refrigeration is a branch of science which deals with the transfer of heat energy from a low temperature region to a high-temperature region, in order to maintain a desired region at a temperature below than that of surroundings.

In the refrigeration process, heat is continuously removed from a low temperature region to a high-temperature medium by using a low boiling point refrigerant. External power is required to carry out this process. Therefore, the refrigeration systems are power-absorbing devices.

7.1.1 Unit of Refrigeration

The capacity of refrigerating machines is often measured in terms of tonnes of refrigeration (TR). *One tonne of refrigeration* is the amount of *refrigerating effect* (heat removed) produced by uniform melting of 1 tonne (1000 kg) of ice from and at 0°C in 24 hours. Using specific enthalpy of fusion of ice as 333.43 kJ/kg, then

$$1\,TR = \frac{(1000 \text{ kg}) \times (333.43 \text{ kJ/kg})}{(24 \text{ h}) \times (60 \text{ min/h})}$$
$$= 231.5 \text{ kJ/min}$$

In practical calculations, 1 *TR* is taken as 211 kJ/min or 3.517 kW.

7.1.2 Applications of Refrigeration

1. Refrigeration is used for preservation of food items, fruits, vegetables, dairy products, fish and meats, etc.
2. Refrigeration is used for preservation of life-saving drugs, vaccines, etc., in hospitals and medical stores.
3. It is used in operation theaters and intensive care units (ICU) of hospitals.
4. It is used for making ice and preservation of ice-creams.

5. It is used for providing comfort air-conditioning in offices, houses, restaurants, theatres, hotels, etc.
6. It is used in industries for improving working environment for their employees.
7. It is used for providing suitable working environment for some precision machines and precision measurement.
8. It is used in cold storages for preservation of seasonal vegetables, fruits, etc.
9. It is used for preservation of photographic goods, archeological and important documents.
10. It is used for chilling beverages (soft drinks).
11. It is used for cooling of drinking water.
12. It is used for liquefaction of gases.
13. It is used for processing of textiles, printing work, precision articles, photographic materials, etc.
14. It is used in control rooms and air-crafts.

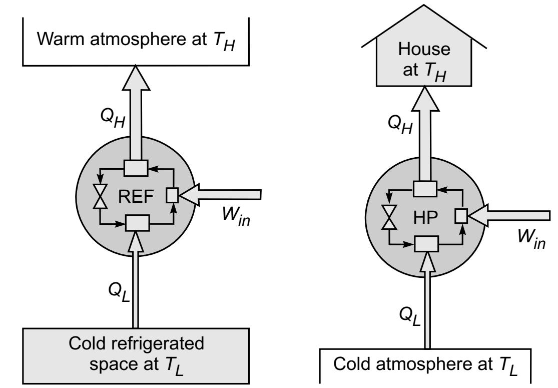
7.2 REFRIGERATORS AND HEAT PUMPS

The device that produces refrigeration effect is called refrigerator and cycle on which it operates is called refrigeration cycle. The working fluid used in refrigeration cycle is called refrigerant.

Another device that also transfers heat from a low-temperature region to a high temperature region is called *heat pump*. The refrigerators and heat pumps operate on the same cycle, but they differ in their objectives. A heat pump maintains a region at a higher temperature than that of its surroundings. It absorbs heat from a low-temperature region and supplies it to a warmer region. Schematics of a refrigerator and heat pump are shown in Fig. 7.1.

The refrigerator removes heat Q_L from the low-temperature region. While a heat pump supplies heat Q_H to a high-temperature region.

The performance of a refrigerator and a heat pump is expressed in terms of coefficient of performance (*COP*), defined as



(a) Schematic of refrigerator *(b) Schematic of heat pump*

Fig. 7.1

$$(COP) = \frac{\text{Desired output}}{\text{Work input}}$$

For a refrigerator, it is the ratio of refrigerating (cooling) effect to the work input required to achieve that effect. Thus

$$(COP)_R = \frac{\text{Refrigerating effect}}{\text{Work input}} = \frac{\text{Cooling effect}}{\text{Work input}}$$

Referring Fig. 7.1(a)

$$(COP)_R = \frac{Q_L}{W_{in}} \quad \text{...(7.1)}$$

Similarly, for a heat pump, supplying heat Q_H to a high-temperature region at the cost of work input W_{in}, Fig. 7.1(b), the coefficient of performance

$$(COP)_{HP} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{in}} = \frac{W_{in} + Q_L}{W_{in}} = 1 + \frac{Q_L}{W_{in}} \quad \text{...(7.2)}$$

where quantities

Q_L = quantity of heat removed from a low-temperature region,

Q_H = quantity of heat supplied to a high-temperature region,

W_{in} = work input to cycle.

From Eqs. (7.1) and (7.2), it is revealed that

$$(COP)_{HP} = (COP)_R + 1$$

Example 7.1 *A household refrigerator with a COP of 1.8 removes heat from a refrigerated space at a rate of 90 kJ/min. Determine*

(a) Electrical power consumed; and

(b) Heat rejected to surroundings

Solution

Given A household refrigerator

$(COP)_R = 1.8$ $\quad\quad$ $RE = 90$ kJ/min

To find

(i) Electrical power consumption, and

(ii) Heat rejected to surroundings.

Analysis

(i) The coefficient of performance of a refrigerator is given as

$$(COP)_R = \frac{\text{Refrigerating Effect}}{\text{Work input}} = \frac{RE}{W_{in}}$$

Using the given values

$$1.8 = \frac{90 \text{ kJ/min}}{W_{in}}$$

or $\quad W_{in} = 50 \text{ kJ/min} = \mathbf{0.833\ kW}$

The electrical power consumption is 0.833 kW.

(ii) Heat rejected to surroundings

$$Q_H = RE + W_{in}$$
$$= 90 \text{ kJ/min} + 50 \text{ kJ/min} = \mathbf{140\ kJ/min.}$$

Example 7.2 *An ice plant produces 10×10^3 kg of ice per day at 0°C using water at a temperature of 23°C. Estimate the power required by the compressor motor, if the COP of the plants is 3.5 and the transmission efficiency is 85%. Also find the amount of heat transferred from the system per minute.*

Take C_p (water) $= 4.1868$ kJ/kg·K, and

$h_{fg(ice)} = 334.5$ kJ/kg

Solution

Given Production of ice at 0°C

$$\dot{m}_{ice} = 10 \times 10^3 \text{ kg per day}$$
$$T_L = 0°\text{C} = 273 \text{ K}$$
$$T_i = 23°\text{C} = 296 \text{ K}$$
$$(COP)_R = 3.5$$
$$\eta_{transmission} = 0.85$$
$$C_{pw} = 4.1868 \text{ kJ/kg}\cdot\text{K}$$
$$h_{fg} = 334.5 \text{ kJ/kg}$$

To find

(i) Power required by compressor motor,

(ii) Amount of heat transferred by the system per minute.

Analysis The amount of heat removed by the system per day

Q_L = Heat removed from water + Heat absorbed during phase change of water to ice

$$= \dot{m} C_{pw} (T_i - 0°\text{C}) + \dot{m} h_{fg}$$
$$= 10 \times 10^3 \times 4.1868 \times (23 - 0) + 10 \times 10^3 \times 334.5$$
$$= 430.8 \times 10^4 \text{ kJ per day}$$

Rate of heat removal per minute

$$RE = \frac{430.8 \times 10^4 (\text{kJ/day})}{(24\text{h} \times 60 \text{ min/h})} = \mathbf{2991.67\ kJ/min}$$

The *COP* of a refrigerator is given as

$$(COP)_R = \frac{RE}{\dot{W}_{in}}$$

or

$$\dot{W}_{in} = \frac{RE}{(COP)_R} = \frac{2991.67}{3.5} = 854.76 \text{ kJ/min}$$
$$= 14.25 \text{ kW}$$

Power required by compressor motor

$$P = \frac{\dot{W}_{in}}{\eta_{transmission}} = \frac{14.25}{0.85} = \mathbf{16.76\ kW}$$

7.3 REFRIGERATION TERMINOLOGY

1. Refrigeration Load It is the amount of heat which must be removed per unit time from the cold region. It is also known as the *refrigeration capacity*. It is measured in Tonnes of Refrigeration (*TR*) and is designated as RE. Sometimes, it is also referred as refrigeraton effect.

2. Mass Flow Rate of Refrigerant The refrigeration capacity of a system decides the mass flow rate of a given refrigerant, when working under specified conditions, i.e., Mass flow rate of refrigerant

$$\dot{m} = \frac{\text{Refrigeration capacity}}{\text{Refrigerating effect per kg of refrigerant}} \quad ...(7.3)$$

3. Evaporation Capacity It is the refrigeration effect produced at the evaporator of a refrigeration system, thus equivalent to refrigeration capacity.

The *evaporator* is the device that consists of the long thin tubing where the refrigerant evaporates by absorbing its heat of evaporation from the surrounding medium.

4. Freezer It is the term used in a household refrigerator. It is a place very near to evaporator coils, so the liquid can easily be freezed to solid state. It is also called *chiller*.

5. Frosting It is the deposition of ice on the evaporating coil due to its very low temperature. When air passes over the extremely cold coils, the moisture in the air separates and deposits on coils and gets solidified to form frosting. This frosting (ice) is a very bad conductor of heat, and reduces the heat transfer from cold region to refrigerant. Thus the refrigeration load increases. Therefore, it is recommended that this frost (ice) must be removed from the evaporator coils regularly. The *removal of ice from the evaporator coil* is called the *defrostation*.

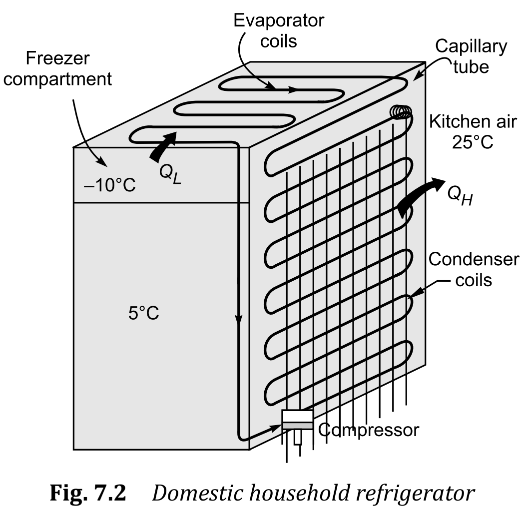


Fig. 7.2 *Domestic household refrigerator*

6. Capillary Tube The capillary tube is a long and narrow tube connecting the condenser directly to the evaporator. The refrigerant passes through this tube by capillary action (drop by drop), thus the fluid friction and flashing of liquid refrigerant into vapour due to heat transfer, cause the pressure drop in the tube. For a given state of the refrigerant, the pressure drop is directly proportional to length of the tube and inversely proportional to diameter of the tube.

7.4 TYPES OF REFRIGERATION SYSTEMS

The commonly used refrigeration systems are listed below:

(i) Gas refrigeration system
(ii) Vapour compression refrigeration system
(iii) Vapour absorption refrigeration system
(iv) Steam refrigeration system

7.5 GAS REFRIGERATION SYSTEMS

In a gas refrigeration cycle, the gas such as air is used as a refrigerant. It transfers only its sensible heat and does not undergo a change of phase. The gas refrigeration systems have a number of applications. They are used to achieve very low temperature for liquefaction of gases and for aircraft cooling. The gas power cycles can be used as gas refrigeration by simply reversing the direction of process involved in these cycles. The reversed Carnot cycle, reversed Brayton cycle and reversed Stirling cycle are some gas refrigeration cycles.

The main advantages of air refrigeration is its free availability, light weight and eco-friendly.

7.5.1 Reversed Carnot Cycle

As studied earlier in Section 6.22, if the direction of Carnot engine is reversed, then the cycle works as a *refrigeration cycle.* The reversed Carnot cycle will theoretically have a maximum possible coefficient of performance, but it is not possible to construct a refrigerating machine which will work on the reversed Carnot cycle.

A reversed Carnot cycle using air as working medium is shown on *p–V* and *T–S* diagrams in Fig. 7.3(a) and (b), respectively.

The cycle consists of four reversible processes in sequence.

Process 1–2 Isentropic expansion of air from higher temperature T_H to lower temperature T_L

Process 2–3 Heat removal from cold space in isothermal manner at temperature T_L

Process 3–4 Isentropic compression of air from low temperature T_L to high temperature T_H

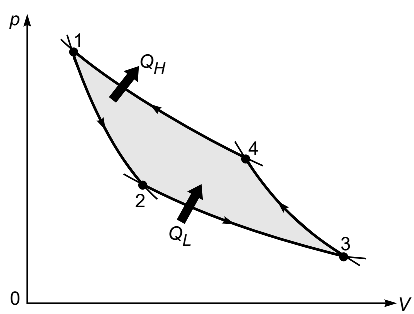
Process 4–1 Heat rejection isothermally to a medium at temperature T_H

The refrigeration effect = Heat absorbed by air during isothermal process 2–3 at temperature T_L

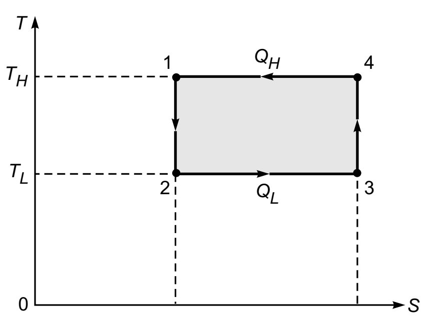
$$RE = T_L(S_3 - S_2) \quad \text{...(7.4)}$$

Heat rejected at temperature T_H

$$Q_H = T_H(S_3 - S_2) \quad \text{...(7.5)}$$



(a) p–V diagram



(b) T–S diagram

Fig. 7.3 *Reversed Carnot Cycle*

Net work input to cycle,

$$\begin{aligned} W_{in} &= Q_H - Q_L \\ &= T_H(S_3 - S_2) - T_L(S_3 - S_2) \\ &= (T_H - T_L)(S_3 - S_2) \end{aligned} \quad \text{...(7.6)}$$

The coefficient of performance of the reversed Carnot cycle operation as refrigerator

$$(COP)_{R,\,rev} = \frac{RE}{W_{in}} = \frac{T_L(S_3 - S_2)}{(T_H - T_L)(S_3 - S_2)}$$

$$= \frac{T_L}{T_H - T_L} \quad \text{...(7.7)}$$

The coefficient performance of reversed Carnot cycle operating as heat pump

$$(COP)_{HP,\,rev} = \frac{\text{Heat rejected at } T_H}{\text{Work input}} = \frac{Q_H}{W_{in}}$$

$$= \frac{T_H(S_3 - S_2)}{(T_H - T_L)(S_3 - S_2)} = \frac{T_H}{T_H - T_L} \quad \text{...(7.8)}$$

7.5.2 Practical Limitations of Carnot Refrigeration Cycle

Although the reversed Carnot cycle gives maximum *COP* between two fixed temperatures and is useful as a criterion of perfection, it has inherent drawbacks, when a gas is used as a refrigerant.

1. It is almost impractical to transfer heat at constant temperature during processes 2–3 and 4–1. Such heat transfer is only possible with a very slow process.
2. Secondly, with the irreversibilities of the compressor and expander, the isentropic compression and expansion cannot be achieved
3. Further, isothermal processes are extremely slow processes while isentropic processes are extremely fast processes. Therefore, combination of these two processes within a cycle is impossible.

Example 7.3 *A refrigerator has a working temperature in the evaporation and condenser coils of – 30°C and 30°C, respectively. What is the maximum possible COP of the refrigerator?*

Solution

Given A refrigerator operating between

$T_L = -30°C = 243$ K

$T_H = 30°C = 303$ K

To find Maximum possible *COP* of a refrigerator.

Analysis The reversed Carnot cycle can give maximum $(COP)_R$, thus

$$(COP)_{R,\,rev} = \frac{T_L}{T_H - T_L} = \frac{243}{303 - 243} = \mathbf{4.05}$$

Example 7.4 *A Carnot cycle machine operates between the temperature limits of 47°C and –30°C. Determine the COP when it operates as (i) refrigerating machine (ii) A heat pump (iii) A heat engine.*

Solution

Given A Carnot cycle machine operating as refrigerator, heat pump and heat engine.

$T_H = 47°C = 320$ K

$T_L = -30°C = 243$ K

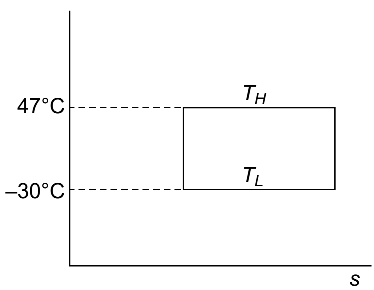


Fig. 7.4

To find

(i) *COP*, if refrigerator

(ii) *COP*, if operates as heat pump

(iii) Efficiency of Carnot engine

Analysis

(i) When Carnot cycle operates as reversible refrigerator

$$(COP)_R = \frac{T_L}{T_H - T_L} = \frac{243}{320 - 243} = \mathbf{3.155}$$

(ii) When Carnot cycle operates as reversible heat pump

$$(COP)_{HP} = \frac{T_H}{T_H - T_L} = \frac{320}{320 - 243} = \mathbf{4.155}$$

(iii) When Carnot cycle operates as reversible heat engine

$$\eta_{rev} = 1 - \frac{T_L}{T_H} = 1 - \frac{243}{320} = 0.240 \text{ or } \mathbf{24\%}$$

Example 7.5 *A reversed Carnot cycle is used for making ice at –5°C from water at 25°C. The temperature of the brine is –10°C. Calculate the quantity of ice formed per kWh of work input. Assume the specific heat of ice as 2 kJ/kg·K, latent heat of ice as 335 kJ/kg and specific heat of water as 4.18 kJ/kg·K.*

Solution

Given Formation of ice at –5°C from water at 25°C

$T_L = -10°C = 263$ K

$T_H = 25°C = 298$ K

$T_{ice} = -5°C$

$W_{in} = 1$ kWh $= 3600$ kJ

$C_{p,\,ice} = 2$ kJ/kg·K

$C_{pw} = 4.18$ kJ/kg·K

$h_{fg} = 335$ kJ/kg

To find The mass of ice formed.

Assumption The ambient temperature to be 25°C

Analysis The schematic is shown in Fig. 7.5.

Amount of heat removed from water–ice system per kg.

Q_L = Heat removed in cooling of water from 25°C to 0°C + Heat removed during formation of ice at

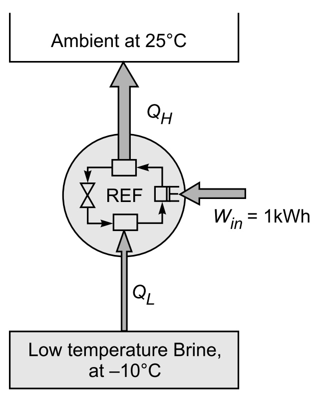


Fig. 7.5

0°C + Heat removed from ice during its cooling from 0°C to –5°C.

$= C_{pw}(T_H - 0°C) + h_{fg} + C_{p,ice}(0 - T_{ice})$

$= 4.18 \times (25 - 0) + 355 + 2.0 \times [0 - (-5)]$

$= 449.5$ kJ/kg

The *COP* of a Carnot refrigerator

$$(COP)_{R,\,rev} = \frac{T_L}{T_H - T_L} = \frac{263}{298 - 263} = 7.514$$

Further, the *COP* can also be expressed as

$$COP = \frac{RE}{W_{in}}$$

or $$7.514 = \frac{RE}{3600 \text{ kJ}}$$

or $$RE = 27051.43 \text{ kJ}$$

Thus, the mass of ice formed with this refrigerating effect

$$\dot{m}_{ice} = \frac{RE}{Q_L} = \frac{27051.43}{449.5} = \mathbf{60.18\ kg/kWh}$$

7.6 REVERSED BRAYTON REFRIGERATION CYCLE: BELL COLEMAN CYCLE

The Bell Coleman refrigeration cycle is the reverse of the closed Brayton power cycle. The schematic and *T–s* diagrams of the reversed Brayton cycle are shown in Fig. 7.6.

The refrigerant gas (may be air) enters the compressor at the state 1 and is compressed to the state 2. The gas is then cooled at constant pressure in a heat exchanger to the state 3. During cooling process, the gas rejects heat to the surroundings and approaches the temperature of the warm environment. The gas is then expanded in an expander to the state 4, where it attains a temperature, that is well below the temperature of the cold region. The refrigeration effect is achieved through the heat transfer from the cold region to gas as it passes from state 4 to 1 and the cycle completes.

The *T–s* diagram for an ideal Brayton cycle is represented by the cycle 1–2*s*–3–4*s*–1, in which all processes are internally reversible, and compression and expansion are isentropic. The cycle 1–2–3–4–1 includes the effect of irreversibilities during compression and expansion. The frictional pressure drops have been ignored.

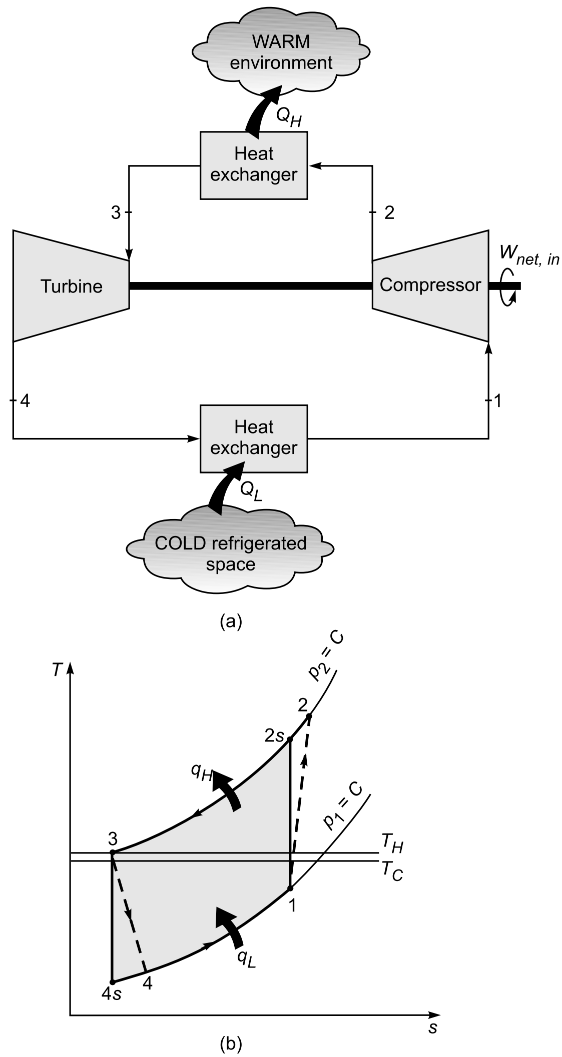


Fig. 7.6 *Brayton refrigeration cycle*

Cycle Analysis At steady state, the compression and expansion work per kg of gas flow in the system are,

$$w_{in} = h_2 - h_1 \quad \text{and} \quad w_{out} = h_3 - h_4$$

The net work input to the cycle is

$$w_{net} = w_{in} - w_{out}$$
$$= (h_2 - h_1) - (h_3 - h_4) \qquad ...(7.9)$$

The refrigeration effect produced in the cycle

RE = Heat transfer from the cold region to the refrigerant gas

$$= h_1 - h_4 \qquad ...(7.10)$$

The coefficient of performance of the cycle is the ratio of the refrigeration effect to the net work input

$$(COP)_R = \frac{RE}{w_{net}} = \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)} \qquad ...(7.11)$$

The gas refrigeration cycle deviates from the reversed Carnot cycle because heat transfer processes are not isothermal. In fact, the gas temperature varies considerably during the heat-transfer processes. Figure 7.7 shows a T–s diagram which compares the reversed Carnot cycle 1–2′–3–4′–1 and reversed Brayton cycle 1–2–3–4–1. It reveals the following facts:

1. The net work (area 1–2′–3–4′–1) required by the reverse Carnot cycle is a fraction of that required by the reverse Brayton cycle.
2. The reverse Carnot cycle produces greater refrigeration effect (area under line 4′–1) as compare to reverse Brayton cycle (area under curve 4–1).
3. The mean temperature of heat rejection is much greater and that of heat absorption is much lower in reverse Brayton cycle.

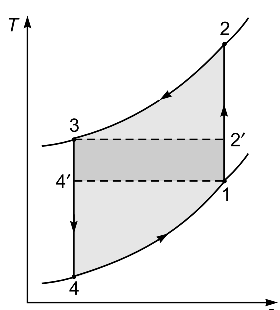


Fig. 7.7 *Comparison of reverse Brayton cycle with reversed Carnot cycle*

Therefore, the *COP* of the reverse Brayton cycle is much lower than that of the vapour compression cycle or the reversed Carnot cycle.

Advantages

1. Air is a freely and easily available fluid.
2. There is no danger of fire and toxic effects, if it leaks.
3. The weight of air refrigeration system per tonne of refrigeration is less compared with other refrigeration systems.
4. An air cycle can work as an open or closed cycle system.
5. It is eco-friendly.

Disadvantages

1. The main drawback of an air refrigeration system is its very low value of *COP*.
2. Air has only sensible heat and cannot transfer heat at constant temperature across heat exchangers. Thus, as the temperature difference decreases, its heat transfer capacity decreases.
3. The quantity of air required per tonne of refrigeration capacity is much larger than the liquid refrigerants.
4. A turbine is required for the expansion of air instead of a throtting device. Since air contains some water vapour, thus danger of frosting during expansion is more possible.
5. It has more running cost than other systems.

Example 7.6 *Air enters the compressor of an ideal Brayton refrigeration cycle at 1 atm and 270 K with a volumetric flow rate of 1.5 m^3/s. If the compressor pressure ratio is 3 and the turbine inlet temperature is 300 K, determine*

(a) the net power input,
(b) the refrigeration capacity, and
(c) coefficient of performance.

Solution

Given An ideal Brayton refrigeration cycle operates with air.

Compressor inlet, $T_1 = 270$ K, $p_1 = 1$ atm, $\dot{V} = 1.5$ m^3/s

Pressure ratio, $p_2/p_1 = 3$

Turbine inlet, $T_3 = 300$ K

To find

(i) Net power input in the cycle,

(ii) Refrigeration capacity, and

(iii) The *COP* of the cycle.

Schematic with given data

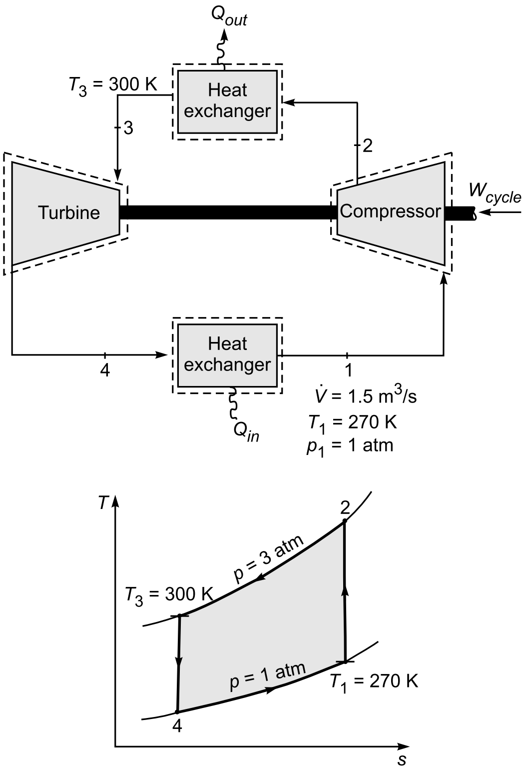


Fig. 7.8

Assumptions

(i) Each component of the cycle is analyzed as a control volume at steady state.

(ii) The turbine and compressor processes are isentropic.

(iii) There are no pressure drop through the heat exchangers.

(iv) Kinetic and potential energy effects are negligible.

(v) The working fluid is modelled as an ideal gas.

(vi) Cold air standard assumptions with C_p = 1.005 kJ/kg·K and γ = 1.4.

Analysis The specific enthalpy at each state of the cycle.

State 1: $T_1 = 270$ K, $p_1 = 1$ atm

$$h_1 = C_p T_1 = 1.005 \times 270 = 271.35 \text{ kJ/kg}$$

State 2: $p_2 = 3p_1 = 3$ atm

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 270 \times (3)^{\frac{1.4-1}{1.4}} = 370 \text{ K}$$

$$h_2 = C_p T_2 = 1.005 \times 370 = 371.85 \text{ kJ/kg}$$

State 3: $T_3 = 300$ K, $p_3 = 3$ atm,

$$h_3 = C_p T_3 = 1.005 \times 300 = 301.5 \text{ kJ/kg}$$

State 4: $p_4 = p_1 = 1$ atm

$$T_4 = \frac{T_3}{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{300}{\left(\frac{3}{1}\right)^{\frac{1.4-1}{1.4}}} = 219.35 \text{ K},$$

$$h_4 = C_p T_4 = 1.005 \times 219.35 = 220.44 \text{ kJ/kg}$$

The specific volume of air

$$v_1 = \frac{RT_1}{p_1} = \frac{(0.287 \text{ kJ/kg·K}) \times (270 \text{ K})}{(101.325 \text{ kPa})} = 0.765 \text{ m}^3/\text{kg}$$

The mass flow rate of air

$$\dot{m} = \frac{\dot{V}}{v} = \frac{1.5}{0.765} = 1.96 \text{ kg/s}$$

(i) Net power input in the cycle

$$\dot{W}_{net,in} = \dot{m}[(h_2 - h_1) - (h_3 - h_4)]$$
$$= 1.96 \times [(371.85 - 271.35) - (301.5 - 220.44)]$$
$$= \mathbf{38.13 \text{ kW}}$$

(ii) The refrigeration capacity

$$RE = \dot{m}(h_1 - h_4)$$
$$= 1.96 \times [271.35 - 220.44] = \mathbf{99.78 \text{ kW}}$$

(iii) The coefficient of performance

$$(COP)_{R,\ Brayton} = \frac{RE}{\dot{W}_{net,in}} = \frac{99.78}{38.13} = \mathbf{2.62}$$

Example 7.7 *Air enters the compressor of an aircraft cooling system at 100 kPa, and 283 K. Air is now compressed to 2.5 bar with an isentropic efficiency of 72%. After being cooled to 320 K at constant pressure in a heat exchanger, the air then expands in a turbine to 1 bar with an isentropic efficiency of 75%. The cooling load of* *the system is 3 tonnes of refrigeration. After absorbing heat at constant pressure, the air re-enters the compressor, which is driven by the turbine. Find the COP of the refrigerator, driving power required and air mass flow rate.*

Solution

Given An aircraft cooling system

Compressor inlet, $p_1 = 100$ kPa, $T_1 = 283$ K

$\eta_C = 0.72$

Pressure ratio, $r_p = 2.5$

$\eta_T = 0.75$

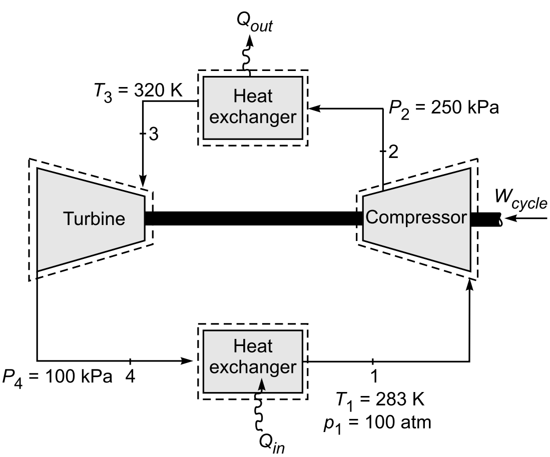
Turbine inlet, $T_3 = 320$ K

$RE = 3$ TR $= 10.55$ kW

To find

(i) *COP* of refrigerator,
(ii) Mass flow rate of air, and
(iii) Power input.

Schematic with given data



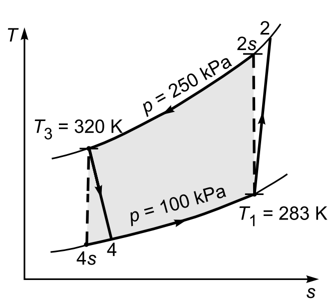


Fig. 7.9

Assumptions

(i) Each component of the cycle is analyzed as a control volume at steady state.
(ii) There are no pressure drops through the heat exchangers.
(iii) Kinetic and potential energy effects are negligible.
(iv) The working fluid is modelled as an ideal gas.
(v) Cold air standard assumptions with C_p = 1.005 kJ/kg·K and γ= 1.4.

Analysis Analysing each component separately;

Compressor The temperature after isentropic compression

$$T_{2s} = T_1 (r_p)^{\frac{\gamma-1}{\gamma}} = 283 \times (2.5)^{\frac{1.4-1}{1.4}} = 367.7 \text{ K}$$

The isentropic efficiency of the compressor is given by

$$\eta_C = \frac{\text{Isentropic work}}{\text{Actual work}} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

or

$$T_2 = 283 + \frac{367.7 - 283}{0.72} = 400.62 \text{ K}$$

The actual work input to compressor w_{in}

$$w_{in} = h_2 - h_1 = C_p(T_2 - T_1)$$
$$= 1.005 \times (400.62 - 283) = 118.21 \text{ kJ/kg}$$

Turbine The temperature T_{4s} after isentropic expansion;

$$T_{4s} = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}} = \frac{320}{(2.5)^{\frac{1.4-1}{1.4}}} = 246.3 \text{ K}$$

The isentropic efficiency of the turbine is given as

$$\eta_T = \frac{\text{Actual work output}}{\text{Isentropic work output}} = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

Actual temperature after expansion in turbine,

$$T_4 = 320 - 0.75 \times (320 - 246.3) = 264.47 \text{ K}$$

Turbine work output per kg of air

$$w_{out} = h_3 - h_4 = C_p(T_3 - T_4)$$
$$= 1.005 \times (320 - 264.72) = 55.556 \text{ kJ/kg}$$

Net work input per kg to the plant

$$w_{net} = w_{in} - w_{out}$$
$$= 118.21 - 55.556 = 62.65 \text{ kJ/kg of air}$$

Heat absorbed per kg of air

$$q_{in} = h_1 - h_4 = C_p(T_1 - T_4)$$
$$= 1.005 \times (283 - 264.72) = 18.37 \text{ kJ/kg}$$

1. *COP of refrigerator*

$$(COP)_R = \frac{\text{Heat absorbed per kg of air}}{\text{Net work input per kg of air}}$$

$$= \frac{q_{in}}{w_{net}} = \frac{18.37}{62.65} = \mathbf{0.293}$$

2. *Mass flow rate of air in the system*

$$\dot{m}_a = \frac{\text{Refrigerating effect}}{\text{Heat removed per kg of air}}$$

$$= \frac{RE}{q_{in}} = \frac{10.55 \text{ kJ/s}}{18.37 \text{ kJ/kg}} = \mathbf{0.573 \text{ kg/s}}$$

3. *Power input to refrigerator*

$$\dot{W}_{input} = \dot{m}_a \, w_{net}$$

$$= (0.573 \text{ kg/s}) \times (62.65 \text{ kJ/kg})$$

$$= \mathbf{35.91 \text{ kW}}$$

7.7 IDEAL VAPOUR COMPRESSION REFRIGERATION CYCLE

A schematic of an ideal vapour compression refrigeration cycle and its *T–s* diagram are shown in Fig. 7.10.

The vapour compression refrigeration cycle consists of four processes discussed below:

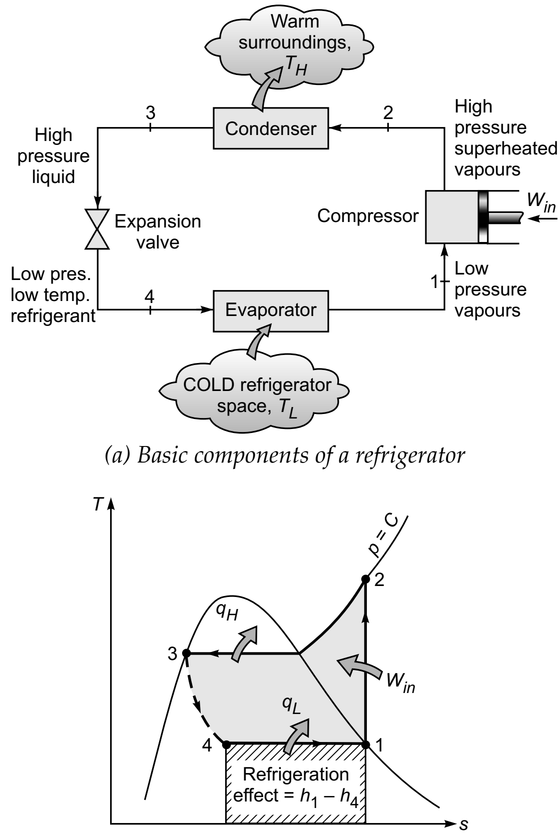
Process 1–2 Isentropic compression of saturated vapour in the compressor,

Process 2–3 Constant pressure heat rejection in the condenser,

Process 3–4 Throttling of refrigerant in an expansion device, and

Process 4–1 Constant pressure heat absorption in evaporator.

In an ideal vapour compression cycle, the refrigerant enters the compressor at the state 1, as dry and saturated vapour, where it is compressed to a relatively high pressure and temperature, the state 2. The refrigerant in the superheated state 2, enters the condenser and leaves as saturated liquid at the state 3, as a result of heat rejection to the surroundings. The saturated liquid refrigerant at the state 3 is throttled to the evaporator pressure by passing through an expansion valve or a capillary tube. The temperature of refrigerant at the state 4 drops below the temperature of the refrigerated space.



(a) Basic components of a refrigerator

(b) T–s diagram for ideal vapour compression cycle.

Fig. 7.10

At the state 4, the refrigerant as mixture of liquid and vapour, passes through an evaporator at constant pressure. The liquid refrigerant evaporates by absorbing its latent heat from cold temperature (refrigerated) space. The latent heat absorbed during evaporation of refrigerant has higher magnitude at lower pressures.

The only throttling process 3–4 is an irreversible process in the cycle and hence it is shown by dotted lines. The remaining three processes are considered reversible. Therefore, the vapour compression cycle is not a reversible cycle.

7.7.1 Vapour Compression Cycle on Pressure–Enthalpy Diagram

The *pressure–enthalpy diagram* is more convenient to represent refrigeration cycles, since the enthalpy

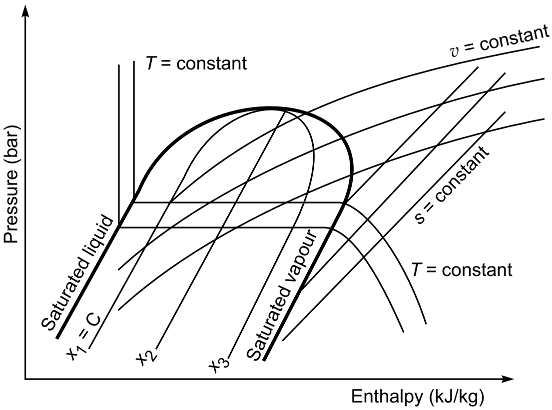


Fig. 7.11 *Schematic of pressure enthalpy chart for a refrigerant*

required for the calculation can directly be read off. The essential features of the diagram are given in Fig. 7.11 and a typical refrigeration cycle is shown in Fig. 7.12. The points 1, 2, 3 and 4 represent the same state of refrigerant as on *T–s* diagram in Fig. 7.10(b).

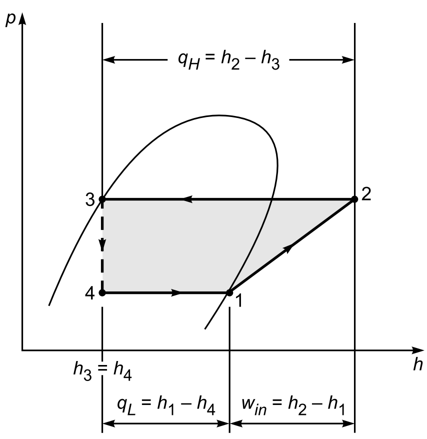


Fig. 7.12 *Vapour compression cycle on a p-h diagram*

7.7.2 Analysis of Vapour Compression Cycle

All four components in a vapour compression cycle are steady flow devices. Thus processes they perform are steady flow processes. In absence of any kinetic and potential energy changes, the steady flow energy equation on unit mass basis reduces to

$$q - w = \Delta h$$

1. Evaporator ($w = 0$)

$$q_L = h_1 - h_4 \qquad \text{...(7.12)}$$

2. Compressor ($q = 0$)

$$w_{in} = h_2 - h_1 \qquad \text{...(7.13)}$$

3. Condenser ($w = 0$)

$$q_H = h_2 - h_3 \qquad \text{...(7.14)}$$

4. Expansion valve ($q = 0$, $w = 0$)

$$h_4 = h_3 \qquad \text{...(7.15)}$$

The coefficient of performance of refrigerator and heat pump can be expressed as

$$(COP)_R = \frac{q_L}{w_{in}} = \frac{h_1 - h_4}{h_2 - h_1} \qquad \text{...(7.16)}$$

$$(COP)_{HP} = \frac{q_H}{w_{in}} = \frac{h_2 - h_3}{h_2 - h_1} \qquad \text{...(7.17)}$$

where $h_1 = h_g @ p_1$

$h_3 = h_f @ p_2$

Example 7.8 *A vapour compression refrigerator cycle works between temperature limits of 25°C and – 10°C. The vapour at the end of isentropic compression is just dry. Assuming there is no subcooling, find the COP of the system. Also find the capacity of refrigerator,if mass flow rate of refrigerant is 5 kg/min.The properties of refrigerant are tabulated below.*

Temperature	*Enthalpy, kJ/kg*		*Entropy of liquid,*
T, K	h_f	h_g	$kJ/kg \cdot K$
298 K	298.9	1465.84	1.1242
263	135.37	1433.05	0.5443

Solution

Given A vapour compression cycle as shown on *p–h* diagram in Fig. 7.13.

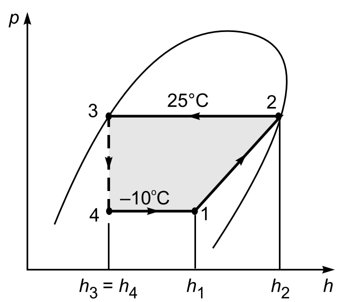


Fig. 7.13

$$\dot{m}_R = 5 \text{ kg/min.}$$
$$T_1 = -10°\text{C} = 263 \text{ K}$$
$$T_3 = 25°\text{C}$$

and tabulated properties of refrigerant.

To find

(i) $(COP)_R$, and

(ii) Capacity of refrigerator.

Analysis Properties of refrigerant at end states:

$$h_2 = 1465.84 \text{ kJ/kg}$$
$$h_3 = 298.9 \text{ kJ/kg}$$
$$h_4 = 298.9 \text{ kJ/kg}$$

Using the Clapeyron equation for calculating entropy change s_{fg_2} during evaporation

$$s_{fg_2} = \frac{h_{fg_2}}{T_{sat_2}} = \frac{h_{g_2} - h_{f_2}}{T_{sat_2}}$$
$$= \frac{1465.84 - 298.9}{298} = 3.9159$$

Entropy at the state 2

$$s_2 = s_{f_2} + s_{fg_2} = 1.1242 + 3.9159$$
$$= 5.0401 \text{ kJ/kg}\cdot\text{K}$$

Similarly, $$s_{fg_1} = \frac{h_{fg_1}}{T_{sat_1}} = \frac{h_{g_1} - h_{f_1}}{T_{sat_1}}$$
$$= \frac{1433.05 - 135.37}{263} = 4.9341$$

Entropy at the state 1

$$s_1 = s_{f_1} + x_1 s_{fg_1} = 0.5443 + 4.9341 x_1$$

For isentropic compression,

$$s_1 = s_2 = 5.0401 \text{ kJ/kg}\cdot\text{K}$$

or $$5.0401 = 0.5443 + 4.9341 x_1$$

or $$x_1 = 0.911$$

Enthalpy at the state 1

$$h_1 = h_{f_1} + x_1 h_{fg_1}$$
$$= 135.37 + 0.911 \times (1433.05 - 135.37)$$
$$= 1317.55 \text{ kJ/kg}$$

(i) *COP of refrigerator*

The compression work

$$w_{in} = h_2 - h_1 = 1465.84 - 1317.55$$
$$= 148.28 \text{ kJ/kg}$$

Refrigeration effect

$$q_{in} = h_1 - h_4$$
$$= 1317.55 - 298.9 = 1018.65 \text{ kJ/kg}$$

Therefore, $$(COP)_R = \frac{q_{in}}{w_{in}} = \frac{1018.65}{148.28} = \mathbf{6.87}$$

(ii) *Capacity of refrigerator*

$$RE = \dot{m}_R \times q_{in}$$
$$= 5 \times 1018.65$$
$$= 5093.25 \text{ kJ/min or } \mathbf{84.88 \text{ kW}}$$

Example 7.9 *A refrigerator used R-12 as a working fluid and it operates on an ideal vapour compression cycle. The temperature of refrigerant in the evaporator is –20°C and in the condenser is 40°C. The refrigerant is circulated at the rate of 0.03 kg/s. Determine the coefficient of performance and capacity of refrigeration plant in the TR.*

Solution

Given A vapour compression cycle as shown on T–s diagram

$$T_1 = -20°\text{C} = 253 \text{ K}$$
$$T_3 = 40°\text{C}$$
$$\dot{m}_R = 0.03 \text{ kg/s}$$

To find

(i) $(COP)_R$, and

(ii) Capacity in TR.

Analysis From R-12 table, we have saturation states

At –20°C $h_1 = 178.61$ kJ/kg,

$s_1 = 0.7082$ kJ/kg

At 40°C $h_3 = h_{f_3} = 74.53$ kJ/kg

For process 1 – 2;

$$s_1 = s_2 = 0.7082 \text{ kJ/kg}\cdot\text{K}$$

It gives $h_2 = 211.38$ kJ/kg

The compression work

$$w_{in} = h_2 - h_1 = 211.38 - 178.61$$
$$= 32.77 \text{ kJ/kg}$$

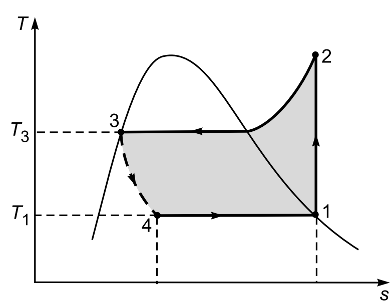


Fig. 7.14

For the process 3 – 4

$$h_4 = h_3 = 74.53 \text{ kJ/kg}$$

Heat absorbed per kg of refrigerant

$$q_{in} = h_1 - h_4$$
$$= 178.61 - 74.53 = 104.08 \text{ kJ/kg}$$

Therefore, $(COP)_R = \dfrac{q_{in}}{w_{in}} = \dfrac{104.08}{32.77} = \mathbf{3.176}$

Refrigerating Capacity,

$$RE = \dot{m}_R \times q_{in} = (0.03 \text{ kg/s}) \times (104.08 \text{ kJ/kg})$$
$$= 3.12 \text{ kJ/s} = 3.12 \text{ kW}$$
$$= \frac{3.12}{3.5} = \mathbf{0.89\ TR}$$

Example 7.10 *An ideal vapour compression system uses R-12 as the refrigerant. The system uses an evaporation temperature of 0°C and a condenser temperature of 40°C. The capacity of the system is 7 TR. Determine*

(a) The mass flow rate of refrigerant,
(b) Power required to run the compressor,
(c) Heat rejected in the condenser,
(d) COP of the system.

Use the properties of R-12 from the table given below:

Temp. °C	*Pressure* bar	h_f kJ/kg	h_g kJ/kg	s_f kJ/kg	s_g kJ/kg·K
0	3.087	36.05	187.53	0.142	0.696
40	9.609	74.59	203.2	0.727	0.682

Take C_p for superheated vapour as 0.6 kJ/kg·K.

Solution

Given An ideal vapour compression system as shown in Fig. 7.14.

$$h_1 = 187.53 \text{ kJ/kg.}$$
$$s_1 = 0.696 \text{ kJ/kg·K}$$
$$h_3 = h_4 = 74.59 \text{ kJ/kg·K}$$
$$s_{g_2} = 0.682 \text{ kJ/kg·K}$$

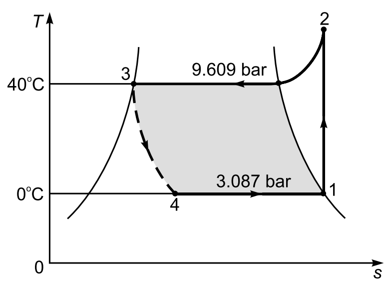


Fig. 7.15

To find

(i) Mass flow rate of refrigerant,
(ii) Power input to compressor,
(iii) Heat rejected in the condenser,
(iv) *COP* of the system.

Assumptions

(i) Refrigerant leaving the evaporator as saturated vapour.
(ii) Compression is isentropic.
(iii) No pressure drop in condenser and evaporator.

Analysis The state 2 can be obtained by equating entropy at two states during compression.

$$s_1 = s_2 = s_g + C_{ps} \ln\left(\frac{T_{sup}}{T_{sat}}\right)$$

$$0.696 = 0.682 + 0.6 \times \ln\left(\frac{T_{sup}}{40 + 273}\right)$$

or $\ln\left(\dfrac{T_{sup}}{313}\right) = 0.0233$

or $T_{sup} = 320.4$ K

The enthalpy after isentropic compression

$$h_2 = h_g + C_{ps}(T_{sup} - T_{sat})$$
$$= 203.2 + 0.6 \times (320.4 - 313)$$
$$= 207.63 \text{ kJ/kg}$$

(i) *Mass flow rate of refrigerant*

$$RE = 7\ TR$$
$$= 7 \times 211 \text{ kJ/min} = 1477 \text{ kJ/min}$$

The refrigerating effect per kg of refrigerant.

$$q_{in} = h_1 - h_4$$
$$= 187.53 - 74.59 = 112.94 \text{ kJ/kg}$$

The mass flow rate of refrigerant

$$\dot{m}_R = \frac{RE}{q_{in}} = \frac{1477}{112.94} = \mathbf{13.07\ kg/min}$$

(ii) *Power input to compressor*

$$P = \dot{m}_R(h_2 - h_1)$$
$$= \left(\frac{13.07}{60}\text{kg/s}\right) \times (207.63 - 187.53) \text{ (kJ/kg)}$$
$$= \mathbf{4.35\ kW}$$

(iii) *Heat rejected in the condenser*

$$= \dot{m}_R(h_2 - h_3)$$
$$= 13.07 \times (207.63 - 74.59)$$
$$= 1730.85 \text{ kJ/min}$$

(iv) *COP* of the system

$$(COP)_R = \frac{\text{Refrigerating effect}}{\text{Work input}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{187.53 - 74.59}{207.63 - 187.53} = \mathbf{5.62}$$

Example 7.11 *A refrigerator operates between temperature limits of 30°C and –5°C. The refrigerant is 0.97 dry after leaving the evaporator coil. Find the condition of refrigerant entering the evaporator and COP of system. If the temperature rise of water circulating through the condenser is limited to 20°C, calculate mass flow rate of the coolant.*

Use the properties of refrigerant from table given below:

Temp. °C	*Enthalpy, kJ/kg* h_f	h_g	*Entropym kJ/kg·K* s_f	s_g	*Specific heat, kJ/kg·K* $C_{p,L}$	$C_{p,g}$
30	323.22	1465.38	1.2037	4.9839	5.024	3.35
–5	158.26	1431.89	0.63	5.4072	–	–

Take C_p for superheated vapour as 3.35 kJ/kg·K.

Solution An ideal vapour compression system as shown in Fig. 7.15.

$$T_1 = -5°C = 268 \text{ K}$$
$$T_2 = 30°C = 303 \text{ K}$$
$$x_1 = 0.97$$
$$(\Delta T)_w = 20°C,$$
$$C_{ps} = 3.5 \text{ kJ/kg·K}$$

To find

(i) Dryness fraction of refrigerant, x_4
(ii) *COP* of the system, and
(iii) Coolant rate in the condenser.

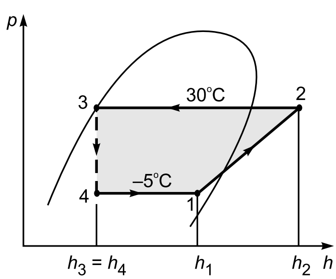


Fig. 7.16

Assumptions

(i) Refrigerant leaving the evaporator as saturated vapour.
(ii) Compression is isentropic.
(iii) No pressure drop in condenser and evaporator.
(iv) Specific heat of water as $C_{pw} = 4.187$ kJ/kg·K.

Analysis The enthalpy at the state 1

$$h_1 = h_{f_1} + x_1(h_{fg_1} - h_{f_1})$$
$$= 158.26 + 0.97 \times (1431.89 - 158.26)$$
$$= 1393.68 \text{ kJ/kg}$$

The entropy at the state 1

$$s_1 = s_{f_1} + x_1(s_{fg_1} - s_{f_1})$$
$$= 0.63 + 0.97 \times (5.4072 - 0.63)$$
$$= 5.2639 \text{ kJ/kg·K}$$

The state 2 can be obtained by equating entropy at the two states during compression.

$$s_1 = s_2 = s_g + C_{ps} \ln\left(\frac{T_{sup}}{T_{sat}}\right)$$

$$5.2639 = 4.9839 + 3.35 \times \ln\left(\frac{T_2}{30 + 273}\right)$$

or $$\ln\left(\frac{T_2}{303}\right) = 0.0836$$

or $$T_2 = 329.41 \text{ K}$$

The enthalpy after isentropic compression

$$h_2 = h_g + C_{ps}(T_{sup} - T_{sat})$$
$$= 1465.38 + 3.35 \times (329.41 - 303)$$
$$= 1553.86 \text{ kJ/kg}$$

Enthalpy at the end of condensation at the state 3;

$$h_3 = h_{f@\,30°C} = 323.22 \text{ kJ/kg·K}$$

Enthalpy after throttling, at the state 4;

$$h_4 = h_3 = 323.22 \text{ kJ/kg·K}$$

(i) *State of refrigerant after throttling*

Enthalpy at state 4 can be expressed as;

$$h_4 = h_{f_4} + x_4(h_{fg_4} - h_{f_4})$$
$$323.22 = 158.26 + x_4 \times (1431.89 - 158.26)$$

or $x_4 = \mathbf{0.13}$

(ii) *COP of refrigerator*

The refrigerating effect per kg of refrigerant.

$$q_{in} = h_1 - h_4$$
$$= 1393.68 - 323.22 = 1070.46 \text{ kJ/kg}$$

Work input per kg of refrigerant

$$w_{in} = h_2 - h_1$$
$$= 1553.86 - 1393.68 = 160.18 \text{ kJ}$$

$$(COP)_R = \frac{q_{in}}{w_{in}} = \frac{1070.46}{160.18} = \mathbf{6.685}$$

(iii) *Mass flow rate of water in condenser*

Heat rejected per kg of refrigerant

$$q_{out} = h_2 - h_3 = 1553.86 - 323.22$$
$$= 1230.64 \text{ kJ/kg}$$

Water flow rate per kg of refrigerant

$$q_{out} = \dot{m}_w C_{pw} (\Delta T)_w$$
$$1230.64 = \dot{m}_w \times 4.187 \times (20°\text{C})$$
$$\text{or} \quad \dot{m}_w = \mathbf{14.7 \text{ kg/kg of refrigerant}}$$

Example 7.12 *A simple R-12 plant is to develop 4 tonnes of refrigeration. The condenser and evaporator temperatures are 35°C and –15°C, respectively. Determine*

(a) The mass flow rate of refrigerant in kg/s,
(b) Volume flow rate handled by compressor in m³/s,
(c) The compressor discharge temperature,
(d) The pressure ratio,
(e) Heat rejected to condenser in kW,
(f) Flash gas percentage after throttling,
(g) The COP, and
(e) Power required to drive the compressor.

Compare this COP with COP of Carnot refrigerator operating between temperatures of 35°C and –15°C.

Solution

Given A vapour compression cycle as shown on the *p–h* diagram in Fig. 7.16.

$$T_1 = -15°\text{C} = 258 \text{ K}$$
$$T_3 = 35°\text{C} = 308 \text{ K}$$
$$RE = 4 \text{ TR} = 4 \times 3.517 = 14.06 \text{ kW}$$

To find

(i) The mass flow rate of refrigerant in kg/s,
(ii) Volume flow rate handled by compressor in m³/s,
(iii) The compressor discharge temperaturem T_2,
(iv) The pressure ratio p_2/p_1,
(v) Heat rejected to condenser in kW,
(vi) Flash gas percentage after throttling,
(vii) The *COP*,
(viii) Power required to drive the compressor,
(ix) *COP* of Carnot refrigerator, and its comparison with *COP* of VCC.

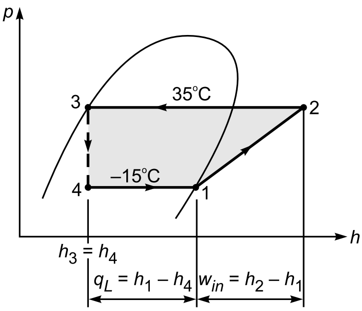


Fig. 7.17

Analysis From R12 refrigeration table and *p–h* diagram, the properties at various states.

At –15°C $h_1 = h_{g_1} = 345$ kJ/kg,

$$s_1 = 1.563 \text{ kJ/kg}$$
$$v_{g_1} = 0.091 \text{ m}^3\text{/kg}$$
$$p_1 = 1.8 \text{ bar}$$
$$h_{f_1} = 186.28 \text{ kJ/kg}$$

We have $s_1 = s_2 = 1.563$ kJ/kg, from *p–h* chart, we get

$$h_2 = 373 \text{ kJ/kg}$$
$$T_2 = 46°\text{C}$$
$$p_2 = 9 \text{ bar}$$

At 35°C $h_3 = h_{f_3} = 233.5$ kJ/kg

For process 3 – 4

$$h_4 = h_3 = 233.5 \text{ kJ/kg}$$

(i) *The mass flow rate of refrigerant in kg/s*

Heat removed per kg of refrigerant

$$q_{in} = h_1 - h_4 = 345 - 233.5 = 111.5 \text{ kJ/kg}$$

The mass flow rate of refrigerant

$$\dot{m}_R = \frac{\text{Refrigerating effect}}{\text{Heat removed per kg of air}}$$
$$= \frac{RE}{q_{in}} = \frac{14.06 \text{ kJ/s}}{111.5 \text{ kJ/kg}} = \mathbf{0.126 \text{ kg/s}}$$

(ii) The volume flow rate of refrigerant

$$\dot{V}_R = \dot{m}_R v_{g_1} = 0.126 \times 0.091$$
$$= \mathbf{0.01145 \text{ m}^3\text{/s}}$$

(iii) *The compressor discharge temperature* T_2 = **46°C**

(iv) *The pressure ratio* p_2/p_1

$$\frac{p_2}{p_1} = \frac{9 \text{ bar}}{1.8 \text{ bar}} = \mathbf{5}$$

(v) *Heat rejected to condenser*

Heat rejected per kg of refrigerant

$$q_{out} = h_2 - h_3 = 373 - 233.5 = 139.5 \text{ kJ/kg}$$

Heat removal rate

$$\dot{Q}_{out} = \dot{m}_R q_{out}$$
$$= 0.126 \times 139.5$$
$$= \mathbf{17.5\ kW}$$

(vi) *Flash gas percentage after throttling*

$$h_4 = h_{f_1} + x(h_{g_1} - h_{f_1})$$
$$233.5 = 186.28 + x(345 - 186.28)$$

or $$x = \frac{233.5 - 186.28}{345 - 186.28} = \mathbf{0.297}$$

(vii) *COP*

The compression work

$$w_{in} = h_2 - h_1 = 373 - 345 = 28 \text{ kJ/kg}$$
$$(COP)_R = \frac{q_{in}}{w_{in}} = \frac{111.5 \text{ kJ/kg}}{28 \text{ kJ/kg}} = \mathbf{3.982}$$

(viii) *Power input to refrigerator*

$$\dot{W}_{input} = \dot{m}_R w_{in}$$
$$= (0.126 \text{ kg/s}) \times (28 \text{ kJ/kg}) = \mathbf{3.528\ kW}$$

(ix) *COP of Carnot refrigerator*

$$(COP)_{R,Carnot} = \frac{T_1}{T_2 - T_1} = \frac{258 \text{ K}}{(308 - 258)\text{ (K)}}$$
$$= \mathbf{5.16}$$

$$\text{Relative } COP = \frac{(COP)_R}{(COP)_{R,Carnot}} = \frac{3.982}{5.16} = \mathbf{0.771}$$

7.7.3 Effect of Operating Conditions on Vapour Compression Cycle

The following analysis will show the effect of change in operating conditions on the performance of the vapour compression cycle.

(*a*) *Effect of Evaporator Pressure*

The evaporator pressure is reduced to $p_{1'}$ from the existing pressure p_1. The effect on the cycle is shown on T–s and p–h diagrams of Fig. 7.18. By reducing the evaporator pressure, the evaporator temperature decreases (favourable) but the refrigeration effect ($h_1' - h_4'$) (area under the curve on T–s diagram) decreases. The work input to the compressor ($h_2' - h_1'$) increases. The specific volume of vapour at low pressure is large, thus the volumetric efficiency will also decrease. Therefore, it is not desirable to decrease the evaporator pressure.

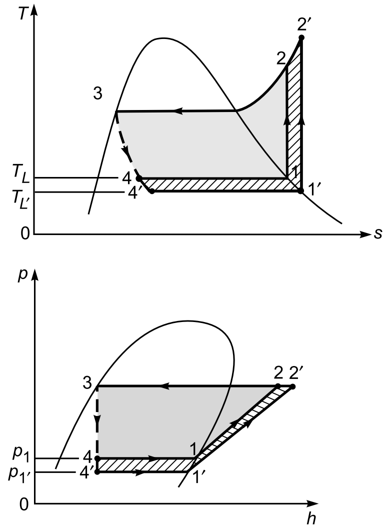


Fig. 7.18 *Effect of evaporator pressure*

(*b*) *Effect of Condenser Pressure*

It is evident from T–s and p–h diagrams of Fig. 7.19, that by increasing the condenser pressure

(i) The work input to the compressor increases from 1–2 to 1–2′.

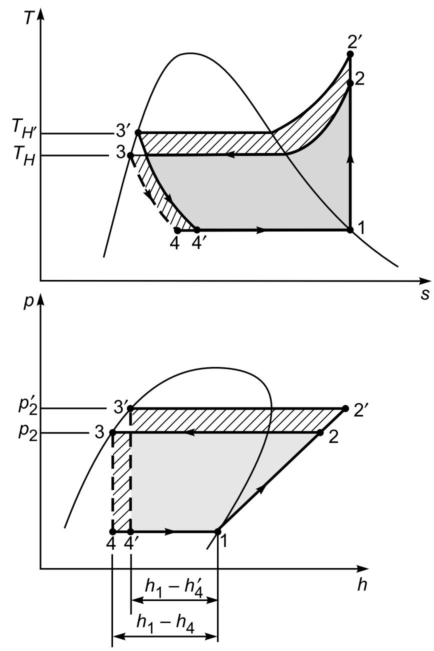


Fig. 7.19 *Effect of condenser pressure*

(ii) Refrigerant effect decreases from 4–1 to 4′–1.
(iii) The condenser temperature increases, thereby increasing heat rejection.
(iv) The *COP* of the system decreases.

(*c*) *Effect of Suction Vapour Superheat*

The effect of superheated vapour at the compression suction is shown with the help of *T–s* and *p–h* diagrams of Fig. 7.20.

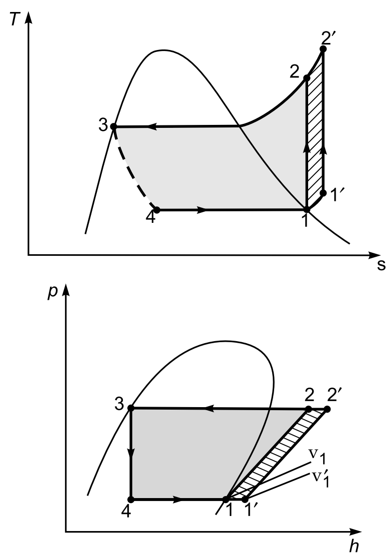


Fig. 7.20 *Effect of suction vapour superheat*

- It increases the specific volume of refrigerant vapour at suction from v_1 to v_1', thus the compression work input increases as shown by hatched area.
- It increases refrigeration effect from $(h_1 - h_4)$ to $(h_1' - h_4)$.
- The *COP* of the new system is given by

$$COP' = \frac{h_1' - h_4}{h_2' - h_1'}$$

$$= \frac{(h_1 - h_4) + (h_1' - h_1)}{(h_2 - h_1) + [(h_2' - h_1') - (h_2 - h_1)]} \quad ...(7.18)$$

Both the numerator and denominator increase, thus the *COP* of the new system may increase, decrease or remain the same.

It is the practical necessity to allow the refrigerant vapour to become slightly superheated at the suction of the compressor in order to avoid any carry over of liquid refrigerant into the compressor. The amount of superheat should be kept minimum in order to keep the compression work minimum.

(*d*) *Effect of Liquid Subcooling*

The condensed liquid can be cooled to a temperature below the saturation temperature corresponding to the condenser pressure. This effect is shown with the help of *T–s* and *p–h* diagrams of Fig. 7.21 in which the constant pressure line is shown left to the saturated liquid line. The effect of *subcooling* (also called *undercooling*) is to move the line 3-4 (throttling process) to the left of the diagrams. It increases the refrigeration effect without increase in compression work. Thus the *COP* of the system increases. Therefore, the subcooling of condensed liquid is desirable.

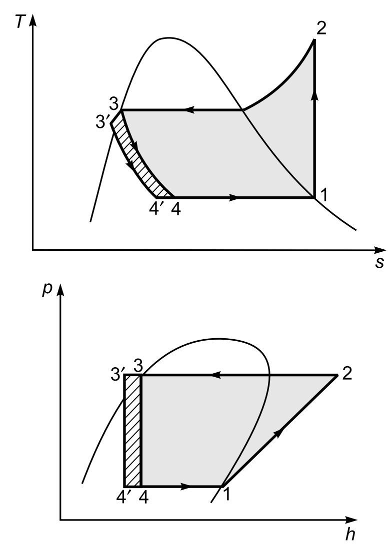


Fig. 7.21 *Effect of subcooling of the condensed liquid*

Example 7.13 *The pressure in the evaporator of an ammonia refrigerator is 1.902 bar and the pressure in the condenser is 12.37 bar. Calculate the refrigeration effect per unit mass of the refrigerant and $(COP)_R$ for the following cycles:*

(a) The dry saturated vapour delivered to the condenser after isentropic compression and no un-

dercooling of the condensed liquid and then throttling of refrigerant to evaporator pressure.

(b) The dry saturated vapour delivered to the compressor, where it is compressed isentropically to the condenser pressure.

(c) The dry saturated vapour delivered to the compressor and liquid after condensation is undercooled by 10°C.

Solution

Given A refrigerator operating between a condenser pressure of 12.37 bar and evaporator pressure of 1.902 bar.

To find

(i) Refrigerating effect/ kg of refrigerant, and

(ii) *COP* of the system.

Assumptions

(i) Each component in the all cycles is analysed as a control volume at steady state.

(ii) Compression, condensation and evaporation processes are internally reversible.

(iii) The kinetic and potential energy changes are negligible.

Analysis

(i) *The dry saturated vapour delivered to the condenser and after condensation throttled without sub-cooling*

At 12.37 bar from properties of ammonia; Table

$$h_3 = h_f = 332.8 \text{ kJ/kg}\cdot\text{K}$$
$$h_2 = h_g = 1469.9 \text{ kJ/kg}\cdot\text{K}$$
$$s_2 = 4.962 \text{ kJ/kg}\cdot\text{K}$$

At 1.902 bar

$$h_{f_1} = 89.8 \text{ kJ/kg},$$
$$h_g = 1420 \text{ kJ/kg}$$
$$s_{f_1} = 0.368 \text{ kJ/kg}\cdot\text{K},$$
$$s_g = 5.623 \text{ kJ/kg}\cdot\text{K}$$

(a) The quality of refrigerant at the state 1;

$$s_2 = s_1 = s_{f_1} + x_1\, s_{fg_1} = s_{f_1} + x_1\,(s_{g_1} - s_{f_1})$$
$$4.962 = 0.368 + x_1\,(5.623 - 0.368)$$
$$\text{or} \quad x_1 = 0.874$$

The specific enthalpy of refrigerant at the state 1;

$$h_1 = h_{f_1} + x_1 (h_{g_1} - h_{f_1})$$
$$= 89.8 + 0.874 \times (1420 - 89.8)$$
$$= \mathbf{1251.8 \text{ kJ/kg}}$$

The refrigerating effect

$$q_{in} = h_1 - h_4 = h_1 - h_3 \quad (\because h_4 = h_3)$$
$$= 1251.8 - 332.8 = \mathbf{919 \text{ kJ/kg}}$$

(b) The *COP* of the system;

$$(COP)_R = \frac{\text{Refrigeration effect}}{\text{Work input}} = \frac{q_{in}}{h_2 - h_1}$$
$$= \frac{919}{1469.9 - 1251.8} = \mathbf{4.21}$$

(ii) *The dry saturated liquid delivered to the compressor*

The properties of the refrigerant, Table B-10, 11

$$h_1 = h_g \text{ @ } 1.902 \text{ bar} = 1420 \text{ kJ/kg}$$
$$h_3 = h_f \text{ @ } 12.37 \text{ bar} = 332.8 \text{ kJ/kg}$$
$$h_4 = h_3 = 332.8 \text{ kJ/kg}$$
$$s_1 = s_g = 5.623 \text{ kJ/kg}\cdot\text{K @ } 1.902 \text{ bar}$$
$$s_{g_2} = 4.962 \text{ kJ/kg}\cdot\text{K}$$
$$s_{g_2} \text{ @ } 12.37 < s_2 = 5.623$$

Thus state 2 is superheated.

Using *p-h* chart

$$h_2 = 1698.5 \text{ kJ/kg}$$

(a) The refrigeration effect;

$$q_{in} = h_1 - h_4 = 1420 - 332.8$$
$$= \mathbf{1087.2 \text{ kJ/kg}}$$

(b) The coefficient of performance;

$$COP = \frac{\text{Refrigeation effect}}{\text{Work input}} = \frac{q_{in}}{h_2 - h_1}$$
$$= \frac{1087.2}{1698.5 - 1420} = \mathbf{3.9}$$

(iii) *Dry saturated refrigerant vapour delivered to compressor and liquid after condensation is undercooled by 10°C*

$$h_1 = 1420 \text{ kJ/kg}$$
$$h_2 = 1698.5 \text{ kJ/kg}$$
$$h_3 = h_4$$
$$h_3 = h_f \text{ @ } T_3$$
$$T_3 = T_3' - 10 = 32 - 10 = 22°\text{C}$$
$$h_3 \text{ @ } 22°\text{C} = 284.6 \text{ kJ/kg}$$

(a) Refrigeration effect

$$q_{in} = h_1 - h_4$$
$$= 1420 - 284.6 = \mathbf{1135.4 \text{ kJ/kg}}$$

(b) *COP* of the system

$$(COP)_R = \frac{q_{in}}{h_2 - h_1} = \frac{1135.4}{1698.5 - 1420} = \mathbf{4.08}$$

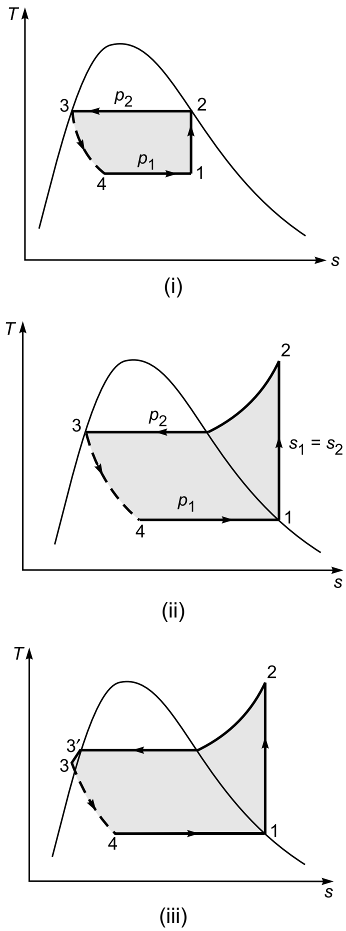


Fig. 7.22

Example 7.14 *Consider a vapour compression refrigeration system using R-12 as a refrigerant. The maximum and minimum pressure of the cycle are 8 bar and 1.2 bar respectively. At the compressor inlet , the vapour temperature is –12°C and the temperature of liquid at the condenser outlet is 30°C. The required refrigerating load is 2.2 kW. The compressor runs to 600 rpm and has a volumetric efficiency of 75%. Find COP and swept volume of compressor.*

Solution

Given A vapor compression refrigeration system:

$p_1 = 1.2$ kPa, $p_2 = 8$ bar

$T_1 = -12°C = 285$ K, $T_3 = 30°C$

$RE = 2.2$ kW, $\eta_{vol} = 0.75$, $N = 600$ rpm

To find

(i) $(COP)_R$, and

(ii) Swept volume of compressor.

Assumptions

(i) Each component in the all cycle is analysed as a control volume at steady state.

(ii) Compression, condensation and evaporation processes are internally reversible.

(iii) The kinetic and potential energy changes are negligible.

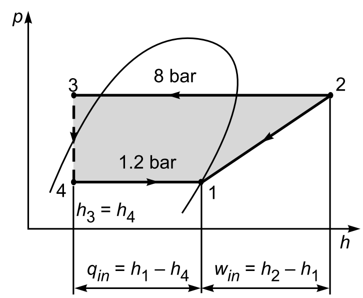


Fig. 7.23

Analysis Properties of refrigerant R-12 at end state:

$h_1 = 354$ kJ/kg, $v_{g_1} = 0.135$ m^3/kg

$h_2 = 388$ kJ/kg, $h_3 = h_4 = 226$ kJ/kg,

(i) *COP of the system*

Heat removed per kg of refrigerant

$$q_{in} = h_1 - h_4 = 354 - 226 = 128 \text{ kJ/kg}$$

The compression work

$$w_{in} = h_2 - h_1 = 388 - 354 = 34 \text{ kJ/kg}$$

$$(COP)_R = \frac{q_{in}}{w_{in}} = \frac{128 \text{ kJ/kg}}{28 \text{ kJ/kg}} = \mathbf{3.764}$$

(ii) *Swept volume of compressor*

The mass flow rate of refrigerant

$$\dot{m}_R = \frac{\text{Refrigerating effect}}{\text{Heat removed per kg of air}}$$

$$= \frac{RE}{q_{in}} = \frac{2.2 \text{ kJ/s}}{128 \text{ kJ/kg}} = 0.01718 \text{ kg/s}$$

Volume flow rate through compressor

$$\dot{V}_R = \dot{m}_R v_{g_1} = 0.01718 \times 0.135$$

$$= 0.00232 \text{ m}^3\text{/s}$$

The volumetric efficiency is given by

$$\eta_{vol} = \frac{\text{Actual volume rate}}{\text{Swept volume rate}} = \frac{\dot{V}_R}{\dot{V}_s}$$

or $$\dot{V}_s = \frac{\dot{V}_R}{\eta_{vol}} = \frac{0.00232}{0.75} = 0.00309 \text{ m}^3/\text{s}$$

Swept volume rate can be expressed as

$$\dot{V}_s = V_s \frac{N}{60}$$

or $$V_s = 0.00309 \times \frac{60}{600}$$

$$= 0.000309 \text{ m}^3 \text{ or } \mathbf{309\ mm^3}$$

Example 7.15 In a 15 TR ammonia refrigeration plant, the condensing temperature is 25°C and evaporating temperature is –10°C. The refrigerant ammonia is sub cooled by 5°C before passing through the throttle valve. The vapor leaving the evaporator is 0.97 dry. Find *COP* and power required to drive the plant? Take C_{pl} = 4.6 kJ/kg·K. C_{ps} = 2.8 kJ/kg·K respectively.

Solution

Given An ammonia refrigeration plant

RE = 1.5 TR = 15 × 3.517 = 52.755 kW

T_c = 25°C T_E = –10°C T_3 = 20°C

x_1 = 0.97 C_{pl} = 4.6 kJ/kg·K C_{ps} = 2.8 kJ/kg·K

To find

(i) *COP* of plant, and

(ii) Power input to refrigerator

Analysis Properties of ammonia.

At –10°C

h_f = 154.03 kJ/kg h_g = 1450.42 kJ/kg

s_f = 0.8294 kJ/kg·K s_g = 5.756 kJ/kg·K

At 25°C h_f = 317.92

h_g = 1483.1 kJ/kg s_g = 5.312 kJ/kg·K

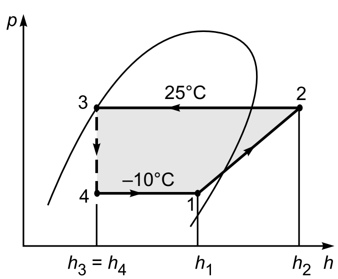


Fig. 7.24 *p-h diagram for given data*

Enthalpy at state 1

$$h_1 = [h_f + x_1(h_g - h_f)]_{@-10°C}$$

$$= 154.03 + 0.97 \times (1450.42 - 154.03)]$$

$$= 1411.53 \text{ kJ/kg}$$

Entropy at state 1

$$s_1 = [s_f + x(s_g - s_f)]_{@-10°C}$$

$$= 0.8294 + 0.97 \times [5.756 - 0.8294]$$

$$= 5.608 \text{ kJ/kg·K}$$

Equating entropy at state 1 and state 2

$$s_1 = s_g + C_{ps} \ln\left(\frac{T_2}{T_{sat}}\right)$$

$$5.608 = 5.312 + 2.8 \ln\left(\frac{T_2}{298}\right)$$

it gives T_2 = 331.22 K or 58.22°C

Enthalpy of refrigerant at state 2

$$h_2 = h_g + C_{ps}(T_2 - T_{sat})$$

$$= 1483.1 + 2.8 \times (58.22 - 25) = 1576.14 \text{ kJ/kg}$$

Enthalpy at state 3,

$$h_3 = h_{f@25°C} - C_{pl}(T_{sat} - 5)$$

$$= 317.92 - 4.6 \times (25 - 5) = 225.92 \text{ kJ/kg}$$

Enthalpy at state 4,

$$h_4 = h_3 = 225.92 \text{ kJ/kg}$$

Mass flow rate of refrigerant

$$\dot{m}_R = \frac{RE}{h_1 - h_4} = \frac{52.755}{1411.53 - 225.92} = 0.044 \text{ kg/s}$$

(i) *COP* of refrigerating plant

$$(COP)_R = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1411.53 - 225.92}{1576.14 - 1411.53} = \mathbf{7.2}$$

(ii) Power input to compressor

$$P = m_R(h_2 - h_1)$$

$$= 0.044 \times (1576.14 - 1411.53)$$

$$= \mathbf{7.24\ kW}$$

Example 7.16 *The bore and stroke of a single cylinder, single acting reciprocating compressor using R-134a refrigerant are 100 mm and 80 mm respectively. The compressor runs at 1500 RPM. If the condensing temperature is 40°C and evaporator temperature (a) 10°C (b) –10°C.*

Find the following:

(i) Mass of refrigerant circulated per minute,

(ii) Refrigerating capacity,

(iii) Power per ton of refrigeration, and
(iv) Total power required to drive the compressor.

Solution

Given A refrigerating system with compressor using R_{134a}

$d = 100$ mm $L = 80$ mm
$N = 1500$ rpm $T_c = 40°C$
(a) $T_{E_1} = 10°C$ (b) $T_{E_2} = -10°C$

To find

(i) Mass of refrigerant per minute,
(ii) Refrigerating capacity,
(iii) Power per ton of refrigeration, and
(iv) Total power input.

Assumption Ideal vapour compression cycle

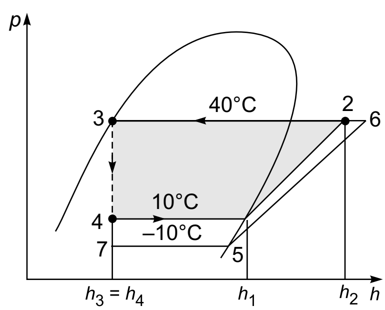


Fig. 7.25

Analysis Properties of R_{134a} at salient points

$h_1 = 405$ kJ/kg $h_2 = 425$ kJ/kg
$\rho_1 = 21$ kg/m^3 $\Rightarrow$ $v_1 = 0.0476$ m^3/kg
$h_3 = h_4 = h_7 = 258$ kJ/kg
$h_5 = 392$ kJ/kg, $h_6 = 430$ kJ/kg, $\rho_5 = 10.5$ kg/m^3

Volume flow rate of

$$\dot{V}_R = \frac{\pi}{4} d^2 L N = \frac{\pi}{4} \times (0.1)^2 \times (0.08) \times 1500$$
$$= 0.942 \text{ m}^3/\text{min}$$

(a) When evaporator temperature is lowered to 10°C

(i) Mass flow rate of refrigerant

$$\dot{m}_R = \dot{V}_R \rho_1 = 0.942 \times 21 = \mathbf{19.8\ kg/min}$$

(ii) Refrigeration effect

$$RE = m_R(h_1 - h_4) = 19.8 \times (405 - 258)$$
$$= 2910.6 \text{ kJ/min}$$
$$= \frac{2910.6}{211} = \mathbf{13.8\ TR}$$

(iii) Power per ton of refrigeration

Mass of refrigerant per ton

$$\dot{m}_1 = \frac{3.517 \text{ kW/TR}}{405 - 258} = 0.0239 \text{ kg/TR}$$

Power, $P_1 = \dot{m}_1 (h_2 - h_1)$

$$= 0.0239 \times (425 - 405) = 0.478 \text{ kW/TR}$$

(iv) Total power

$$P = \dot{m}_R (h_2 - h_1)$$
$$= \left(\frac{19.8}{60} \text{ kg/s}\right) \times (425 - 405)$$
$$= \mathbf{6.6\ kW}$$

(b) When evaporator temperature is lowered to –10°C

(i) Mass of refrigerant

$$\dot{m}_R = \dot{V}_R \rho_5 = 0.942 \times 10.5 = 9.89 \text{ kg/min}$$

(ii) Refrigeration effect in TR

$$RE = \frac{\dot{m}_R(h_5 - h_7)}{211} = \frac{9.89 \times (392 - 258)}{211}$$
$$= \mathbf{6.28\ TR}$$

(iii) Power per ton, $P_1 = \dfrac{3.517}{(392 - 258)} \times (430 - 392)$

$$= \mathbf{0.997\ kW/TR}$$

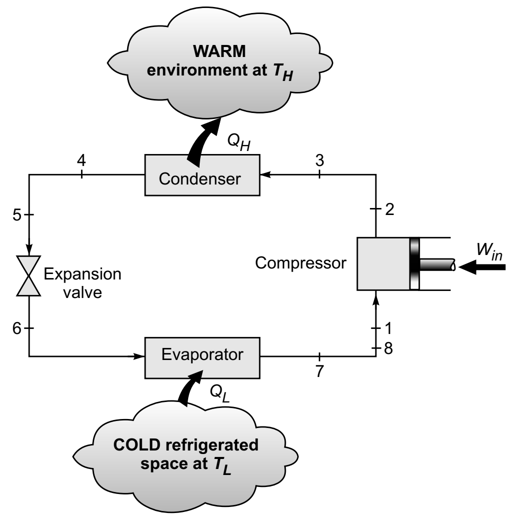
(iv) Total power input

$$P = \dot{m}_R \times (h_6 - h_5) = \frac{9.89}{60} \times (430 - 392)$$
$$= \mathbf{6.26\ kW}$$

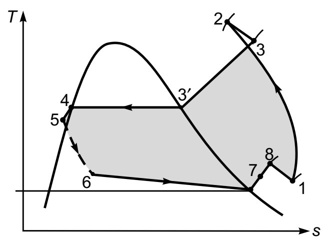
7.7.4 Actual Vapour Compression Cycle: Deviation From Ideal Cycle

The actual refrigeration cycle deviates from the ideal vapour compression cycle, because of pressure drop in tubes associated with fluid friction and heat transfer to or from the surroundings. Further, the compression is also polytropic involving friction and heat transfer, thus migrating from an isentropic one. The actual vapour compression cycle may have some or all the processes deviate from an ideal ones. The actual cycle might be approached one shown in Fig. 7.26 with the help of (b) *T–s* diagram and (c) *p–h* diagram.

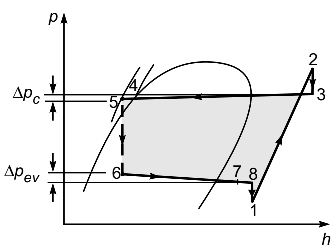
The refrigerant leaving the evaporator at the state 7 is slightly superheated in order to ensure



(a) Schematic arrangement of actual vapour-compression cycle



(b) Actual VCC on T–s diagram



(c) Actual VCC on p-h diagram

Fig. 7.26 *Actual vapour compression cycle*

completely dry refrigerant vapour entry to the compressor. In tube line connecting the evaporator with the compressor, the fluid friction causes a pressure drop of Δp_{ev} and temperature difference between refrigerant and surroundings causes a heat transfer to the refrigerant from the state 7 to 8. As a result, the refrigerant vapour at the state 8 gets superheated further, causing an increase in its specific volume, and thus increase in compression work.

Due to suction effect of the compressor, the sudden pressure drop from the state 8 to the state 1 takes place. The compression begins from the state 1 and continues to the state 2. Actual compression is polytropic with friction and heat transfer, and thus departs from isentropic compression. At the compressor discharge, a sudden pressure drop is observed from the state 2 to the state 3.

During the process 3–4, the refrigerant is first desuperheated to the state 3′ and then condensed during the process 3′-4. A pressure drop further takes place in the tube line connecting the condenser and throttle valve. The liquid refrigerant is slightly subcooled in the condenser from the state 4 to 5 in order to have at least saturated liquid before throttling at the state 5.

The throttling process is represented by a dotted line 5–6 and the refrigeration effect starts from the state 6 to 7. A slight pressure drop due to friction during evaporation makes the process 6–7 irreversible.

Further, in order to have the effective heat transfer between the refrigerant and hot or cold region, the refrigerant temperature in the evaporator is less than the cold region and that in the condenser is higher than the warm atmospheric temperature. It makes the actual cycle further irreversible.

Example 7.17 *Refrigerant 134a is the working fluid in an ideal vapour compression refrigeration cycle, that operated between a cold region at 0°C and a warm region at 26°C. The saturate vapour enters the compressor at –10°C and the saturate liquid leaves the condenser at a pressure of 9 bar. Determine for $\dot{m}$ = 0.08 kg/s (a) compression power in kW, (b) refrigeration capacity in tonnes and (c) coefficient of performance.*

Solution

Given An ideal vapour compression refrigeration cycle operates with refrigerant 134*a* as the working fluid

Mass flow rate, $\dot{m}$ = 0.08 kg/s

Cold region temperature = 0°C

Warm region temperature = 26°C

Temperature of refrigerant vapour at compressor inlet, $T_1 = -10°C$

Pressure of saturate liquid leaving the condenser = 9 bar

To find

(i) Power input in compressor in kW,

(ii) Refrigerating capacity in tonnes, and

(iii) *COP* of the system.

Schematic with given data

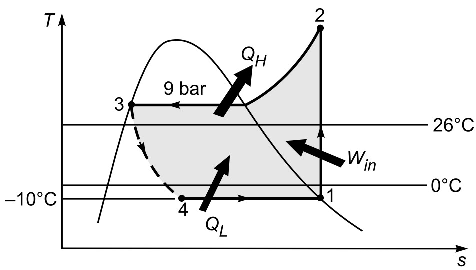


Fig. 7.27

Assumptions

(i) All the components of the cycle are analysed as a control volume at steady state.

(ii) All processes except throttling process are internally reversible.

(iii) Kinetic and potential energy effects are negligible.

(iv) Saturate vapour leaves the evaporator and saturate liquid leaves the condenser.

Analysis The properties of refrigerant 134*a*

State 1: Saturate refrigerant vapour $T_1 = -10°C$

$h_1 = 241.35$ kJ/kg $\qquad s_1 = 0.9253$ kJ/kg·K

State 2: Superheated refrigerant vapour $p_2 = 9$ bar

$s_2 = s_1 = 0.9253$

$h_2 = 272.4$ kJ/kg (by interpolation)

State 3: Saturate liquid at 9 bar

$h_3 = 99.56$ kJ/kg

State 4: Liquid vapour mixture after throttling

$h_4 = h_3 = 99.56$ kJ/kg

(i) Compressor work input;

$$\dot{W}_{comp} = \dot{m}(h_2 - h_1)$$
$$= 0.08 \times (272.4 - 241.35)$$
$$= \mathbf{2.484\ kW}$$

(ii) Refrigerating capacity;

$$RE = \dot{m}(h_1 - h_4)$$
$$= 0.08\ (241.35 - 99.56)$$
$$= 11.343 \text{ kW}$$
$$= \frac{(11.34 \text{ kW})}{(3.517 \text{ kW/TR})}$$
$$= \mathbf{3.24} \text{ ton of Refrigeration}$$

(iii) Coefficient of performance;

$$(COP)_R = \frac{\text{Refrigerating effect}}{\text{Work input}}$$
$$= \frac{11.343}{2.484} = \mathbf{4.567}$$

Example 7.18 *The refrigerant R-12 enters the compressor of a refrigerator as a superheated vapour at 0.14 MPa and –20°C at a rate of 0.05 kg/s and leaves at 0.8 MPa and 50°C. The refrigerant is cooled in the condenser to 26°C and 0.72 MPa and is throttled to 0.15 MPa. Neglecting any heat transfer and pressure drop in the connecting line between the components, determine*

(a) rate of heat removal from the refrigerated space,

(b) power input to the compressor,

(c) the isentropic efficiency of the compressor, and

(d) coefficient of performance of the refrigerator.

Solution

Given A refrigerator operates with *R*-12 refrigerant with the operating conditions

$\dot{m}_R = 0.05$ kg/s

To find

(i) Refrigeration effect,

(ii) Work input to the compressor,

(iii) Isentropic efficiency of the compressor, and

(iv) *COP* of the system.

Assumptions

(i) All the components of the cycle are analysed as a control volume in steady state.

(ii) Neglect kinetic and potential energy effects.

Schematic with given data

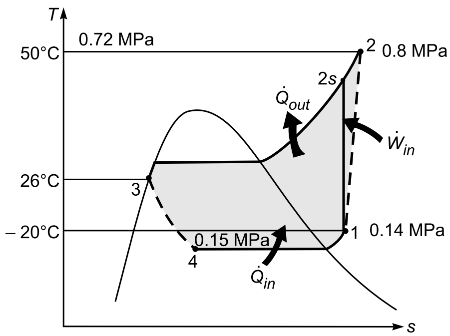


Fig. 7.28

Analysis The properties of R-12 from Table B-5

State 1: Superheated vapour,

p_1 = 0.14 MPa,

T_1 = –20°C

h_1 = 179.01 kJ/kg, s_1 = 0.7147 kJ/kg.

State 2: Superheated compressed vapour at

p_2 = 0.8 MPa, T_2 = 50°C

h_2 = 213.45 kJ/kg

h_{2s} = 210.08 kJ/kg $s_2 = s_1$ = 0.7147 kJ/kg·K

State 3: Undercooled liquid refrigerant from Table

p_3 = 0.72 MPa, T_3 = 26°C

$h_3 = h_f$ @ 26°C = 60.68 kJ/kg

State 4: Liquid-vapour mixture of refrigerant after throttling

$h_4 = h_3$ = 60.68 kJ/kg

(i) Refrigerating effect

$$RE = \dot{Q}_{in} = \dot{m}_R\,(h_1 - h_4)$$
$$= 0.05 \times (179.01 - 60.68)$$
$$= \mathbf{5.92\ kW} \text{ or } \mathbf{1.69\ TR}$$

(ii) Work input in compressor

$$\dot{W}_{in} = \dot{m}_r(h_2 - h_1)$$
$$= 0.05 \times (213.45 - 179.01)$$
$$= \mathbf{1.722\ kW}$$

(iii) Isentropic efficiency η_C of the compressor

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{210.08 - 179.01}{213.45 - 179.01}$$
$$= 0.902 = \mathbf{90.2\%}$$

(iv) Coefficient of performance

$$(COP)_R = \frac{RE}{\dot{W}_{in}} = \frac{5.92\ \text{kW}}{1.722\ \text{kW}} = \mathbf{3.43}$$

7.7.5 Advantages and Disadvantages of Vapour Compression Refrigeration System over Air-Refrigeration System

The followings are the advantages and disadvantages of vapour compression refrigeration system over air refrigeration system:

Advantages

1. The vapour compression refrigeration system has a high value of coefficient of performance.
2. The vapour compression refrigeration system approaches reversed Carnot cycle except for expansion of refrigerant in expansion device.
3. The size of refrigeration system per tonne of refrigeration is smaller.
4. It has less running cost.
5. Since refrigerant has latent heat while air has sensible heat only.
6. The refrigerant has large value of latent heat at lower pressure, and heat is absorbed by evaporation of low pressure liquid refrigerant.
7. The temperature of refrigerant in the evaporator can easily be controlled by regulating the expansion valve.

Disadvantages

1. It has high initial cost.
2. The leakage of refrigerant may cause harmful effect.
3. The production of some refrigerant may be hazardous to the environment.

7.8 VAPOUR ABSORPTION REFRIGERATION CYCLE

In a vapour compression refrigeration cycle, the temperature of saturated vapour leaving the evaporator is increased by a compression process. Since the specific volume of vapour is relatively large, therefore, the input work to the compressor is also large. The input work to the compressor can be reduced significantly, if the refrigerant is compressed in liquid state. The absorption refrigeration is based on this approach.

A vapour absorption system operates with a condenser, a throttle valve and an evaporator in the same way as in vapour compression system, but the compressor is replaced by an absorber, pump and generator units as shown in Fig. 7.29.

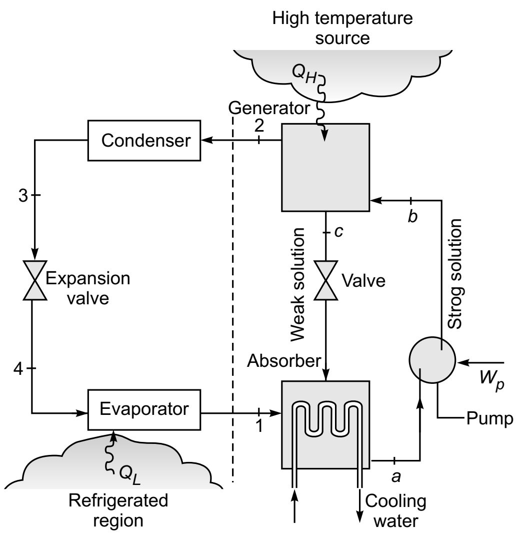


Fig. 7.29 *Schematic a vapour absorption refrigeration cycle*

The low-pressure refrigerant vapour leaving the evaporator is absorbed by a secondary substance, called an *absorbent* to form a strong liquid solution. This liquid solution is then pumped at a higher pressure to the generator. The specific volume of the liquid solution is much less than that of a refrigerant vapour, and thus significant less work is required in the pump. The heat is supplied in the generator, where the refrigerant vapourises from the solution and leaves weak solution in the generator. The refrigerant vapour enters the condenser and the weak solution is again sent back to the absorber through a pressure relief valve.

The coefficient of vapour absorption refrigeration system can be expressed as

$$(COP)_R = \frac{\text{Heat absorbed in evaporator}}{\text{Heat input in generator + pump work}}$$

$$= \frac{Q_L}{Q_H + W_p}$$

7.8.1 Ammonia–water Absorption Refrigeration System

Figure 7.30 shows a schematic arrangement of ammonia–water vapour absorption system using the solar energy for generator heating. The ammonia is used as refrigerant and water is used as absorbent. In the absorber, the ammonia vapour coming out the evaporator at the state 1 is absorbed by liquid water. The formation of this liquid solution is exothermic, thus heat is released. The solvency of ammonia in the water decreases as temperature increases. Thus the cooling arrangement is required in the absorber to absorb the energy released due to absorption of ammonia in the water. The strong ammonia–water solution is pumped to the generator through a heat exchanger, where it is preheated with the help of hot weak solution returning to absorber, thereby, reducing the heat supply in the generator. In the generator, the heat transfer from the source (solar energy) drives the ammonia vapour out of the solution (endothermic process), leaving a weak ammonia–water solution in the generator.

The ammonia vapour liberated passes to the condenser at the state 2 through a *rectifier*. The rectifier removes the traces of water from the refrigerant if any, before it enters the condenser. It avoids the formation of ice in the system. The remaining weak solution returns back to the absorbent through heat exchanger and valve. The condensed ammonia is expanded through an expansion valve and then enters the evaporator, where it absorbs heat from the low-temperature region.

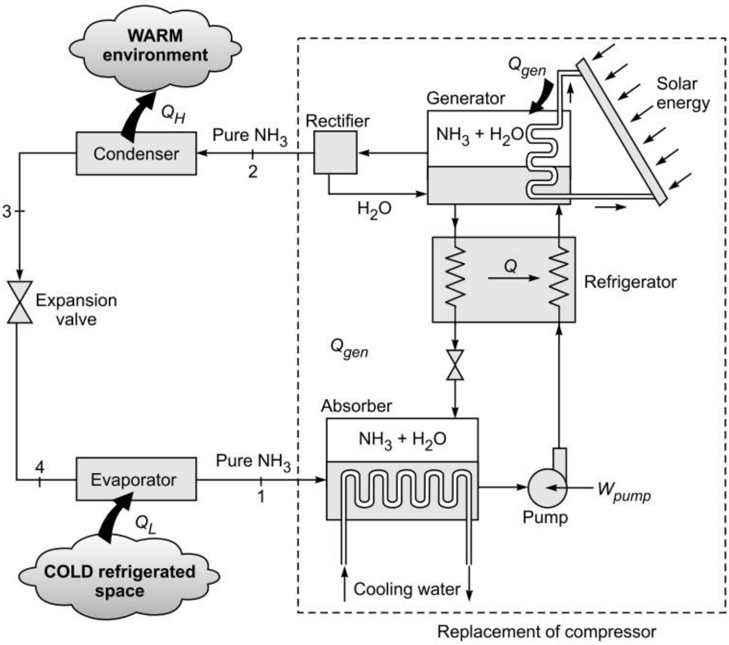


Fig. 7.30 *Ammonia-absorption refrigeration cycle*

7.8.2 Electrolux Refrigeration System

The electrolux absorption principle works on a three-fluid system. It uses natural circulation in absence of any pump. Figure 7.31 shows a schematic of an electrolux refrigeration system. The refrigerant used in the system is ammonia, the absorbent is water and the third fluid hydrogen remains mainly in the evaporator for reducing the partial pressure of the refrigerant to enable it to evaporate at low pressure and low temperature. The total pressure is constant throughout the system.

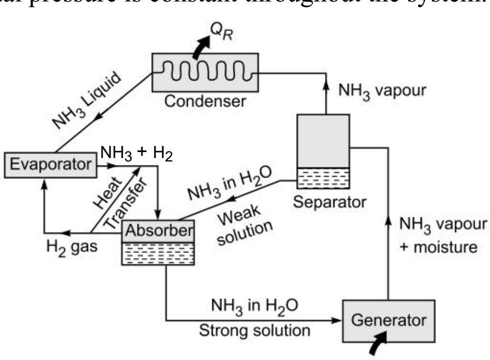


Fig. 7.31 *Electrolux refrigeration system*

The ammonia liquid leaving the condenser enters the evaporator, where it evaporates in the presence of H_2 gas at low temperature corresponding to its partial pressure. The ammonia and H_2 gas mixture passes to the absorber, where it mixes with weak ammonia–water solution coming from the separator. The water absorbs the ammonia vapour and the solution becomes strong. The H_2 gas separates from the solution and returns to the evaporator. The strong ammonia solution passes the generator, where it is heated externally and ammonia vapour drives out and rises to the separator. The moisture from the vapour is separated out and a weak ammonia solution returns back to the absorber. The ammonia vapour from the separator enters the condenser, where it is condensed and then returns to the evaporator.

With the certain modifications, the system can be used in places, where electricity is not available. It requires energy only in the form of heat (waste heat or solar energy) and no pump is necessary.

7.9 COMPARISON OF VAPOUR ABSORPTION SYSTEM WITH VAPOUR COMPRESSION SYSTEM

S.No.	*Aspect*	*Vapour Absorption System*	*Vapour Compression System*
1.	Energy input	The vapour absorption system takes in low-grade energy such as waste heat from furnace, exhaust steam or solar heat for its operation.	Vapour compression system takes in high grade energy such as electrical or mechanical energy for operation of compressor used in the cycle.
2.	Moving part	It uses a small pump as moving part, which is run by a small motor.	It uses a compressor driven by an electric motor or engine.
3.	Evaporator pressure	It can operate with reduced evaporator pressure, with little decrease in refrigeration capacity.	The refrigeration capacity decreases with lowered evaporator pressure.
4.	Load variation	The performance of vapour absorption system does not change with load variation.	The performance of vapour compression system is very poor at partial load.
5.	Evaporator exit	In vapour absorption system, the liquid refrigerant leaving the evaporator does not put any bad effect on the system except to reduce the refrigeration effect.	In a vapour compression system, it is desirable to superheat vapour before leaving the evaporator, so no liquid can enter the compressor.
6.	*COP*	The *COP* of the system is poor.	The *COP* of the system is excellent.
7.	Capacity	It can be built in capacities well above 1000 *TR*.	For a single compression system, it is not possible to have a system with more than 1000 *TR* capacity.
8.	Refrigerant	Water or ammonia is used as refrigerant.	Chlorofluorocarbon, hydrochlofluoro carbon and hydrofluorocarbon are used in most of the system.
9.	Lowest temperature	Since water is used as refrigerant, thus the lowest temperature attained is above 0°C.	With cascading, the temperature can be lowered upto –150°C or even less temperature.

7.10 STEAM JET REFRIGERATION

It uses water as refrigerant, which is quite safe like air. If the pressure exerted on the surface of water is reduced then saturation temperature also lowers and water starts evaporating at lower temperature due to reduced pressure. It is the basic principle for steam jet refrigeration system.

The layout of steam jet refrigeration system is shown in Fig. 7.32. It consists of an evaporator (flash chamber), a steam nozzle, an ejector and a condenser. The steam expands through a nozzle to form a high speed, jet, which draws (sucks) the water from the flash chamber into the ejector. Therefore, the pressure in the flash chamber gets reduced, thereby results into further formation of vapour (evaporation) in the (flash chamber). This evaporation extracts heat (latent heat which is higher at lower pressures) for phase change, thus reducing

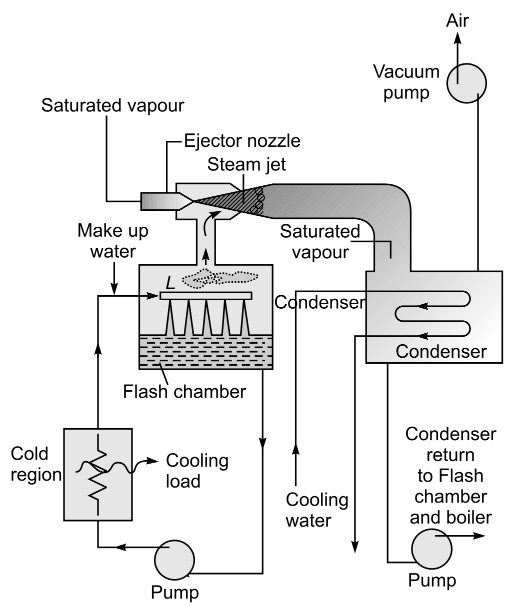


Fig. 7.32 *Steam jet refrigeration system*

the temperature of water in the flash chamber. This cold water is used for refrigeration. The mixture of steam and water vapour is diffused in the diverging part of the ventury tube to the exhaust pressure and fed to the condenser. After condenser some of water is returned to flash chamber as a make up water, and rest is used as feed water to the boiler.

The steam jet refrigeration system uses waste steam returning to condenser. Thus it is quite cheap, but the steam jet refrigeration systems are not used, when temperature below 5°C is required. It is widely used in food precessing plants for precooling of vegetables and concentration of fruit juices, gas plants, paper mills, breweries etc.

7.11 HEAT PUMP

A refrigerator maintains a region at low temperature by removing heat Q_L and it rejects heat Q_H to a high-temperature environment. However, the same basic cycle could maintain a region at higher temperature by supplying heat Q_H, while absorbing heat Q_L from low temperature medium such as atmosphere, lake, etc. Then the device, which maintains a region such as commercial building at higher temperature than that of its surroundings is called a *heat pump*. Figure 7.33 shows a schematic of a heat pump.

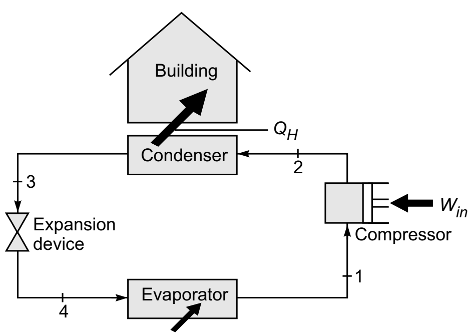


Fig. 7.33 *Schematic of a heat pump*

The modern air-conditioning unit combines both heating and cooling arrangement as shown in Fig. 7.34. When cooling is required, it is operated in the refrigeration mode and removes heat Q_L from living space and rejects heat Q_H outside the building environment. When heating of living space is required in the winter, it absorbs heat Q_L from the environment and supplies heat Q_H to the living space.

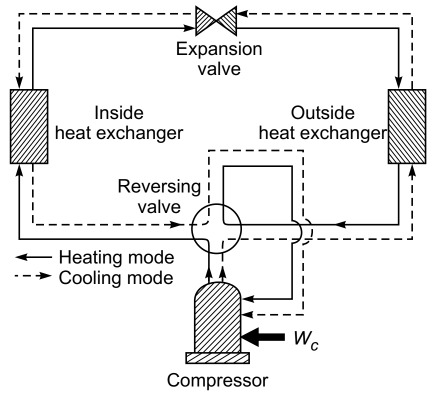


Fig. 7.34 *Example of an air-to-air reversing heat pump*

In the most common type of vapour-compression heat pump for space heating, the evaporator communicates thermally with the outside air. Such *air-source heat pumps* can also be used to provide cooling in the summer with the use of a reversing valve, as illustrated in Fig. 7.34. The solid lines show the flow path of the refrigerant in the heating mode as discussed above. To use the same components for cooling effect, the reverse valve is actuated and the refrigerant follows the path shown by the dashed line. In the cooling mode, the outside heat exchanger becomes the condenser and the inside heat exchanger becomes the evaporator.

Industrial Applications of Heat Pump The heat pump is used for warming of homes and offices in extreme cold climate. The heat pump also offers distinct opportunity for industrial applications. A heat pump can provide heating and cooling of two dissimilar fluids simultaneously. The different industrial applications of heat pump are for

1. Purification of salty water.
2. Concentration of juices, milk and syrups, dyes, chemicals, etc.

3. Preparation of powder milk and table salt.
4. For recovery of valuable solvents from different manufacturing processes.
5. For year round air-conditioning.

7.12 REFRIGERANTS

The *refrigerant* is a heat-carrying medium, which undergoes the theromdynamics cycle of refrigeration (i.e., compression, condensation, expansion and evaporation). In a refrigeration system, it absorbs the heat from a low-temperature medium and discards the absorbed heat to a high-temperature environment.

7.12.1 Desirable Properties of a Refrigerant

A good refrigerant should have the following properties:

(i) The *saturation pressure* of the refrigerant at a desired low temperature should be above or equal to the atmospheric pressure in order to avoid leakage in the evaporator. The pressure at the condenser must not be excessively high for the same reasons.

(ii) The *latent heat of evaporation* at low temperature should be as high as possible to give a reasonably low mass flow rate of refrigerant for a given refrigeration capacity.

(iii) The size of the compressor depends on the *specific volume* of the refrigerant at evaporator pressure. Thus, the specific volume of the refrigerant at the compressor suction should not be high in order to avoid the large compressor for the required mass flow rate.

(iv) The refrigerant should be *chemically stable* and should not react with lubricant used in reciprocating compressor and should be miscible with oil.

(v) It should not be *non-flammable, non*-explosive.

(vi) It should be *non-toxic*. If toxic, then to a limit, below the acceptable level.

(vii) It should have low specific heat of liquid for better heat transfer in condenser.

(viii) The refrigerant should give high value of *COP* with low power input per tonne of refrigeration.

(ix) The refrigerant should have good thermal conductivity for better heat transfer in the condenser and evaporator.

(x) The refrigerant must have freezing point temperature well below the lowest temperature in the cycle.

(xi) Other considerations are chemical stablility, non-corrosiveness, low cost and overall *eco-friendliness*.

7.12.2 Classification of Refrigerants

Refrigerants may be broadly classified into following groups:

1. Primary refrigerants, and
2. Secondary refrigerants.

The refrigerant which directly undergo the refrigeration cycle are called *primary refrigerants*, whereas the *secondary refrigerants* act only as heat carrier. They are first cooled by primary refrigerants, and then are circulated through other media to absorb heat.

The primary refrigerants are further classified into the following groups:

1. Halocarbons All the halocarbon refrigerants are divided into three subgroups according to their constituents.

(a) *Chlorofluorocarbons (CFCs)* composed of chlorine, fluorine and carbon atoms Examples are *R*-11, *R*-12, and *R*-114.

(b) *Hydro chlorofluorocarbons (HCFCs)* composed of hydrogen, chlorine, fluorine and carbon atoms. Examples are *R*-22 and *R*-123.

(c) *Hydro fluorocarbons (HFCs)* composed of hydrogen, fluorine and carbon. Examples are *R*-134*a* (HFC-134*a*), and *R*-125 (HFC-125).

2. Inorganic refrigerant such as air, water, ammonia and carbon dioxide.

3. Hydrocarbon refrigerants, such as ethane, propane, butane, etc.

There are also mixtures of above substances that are used as refrigerants. These are *azeotropes* and *zeotropes*.

Azeotropes are mixtures that behave as a single substance. All components of the mixture evaporate and condense at the same conditions. The frequently used azeotropes are *R*-500, a CFC/HFC mixtures, and *R*-502—a HCFC/CFC mixture.

Zeotropes, or *blends,* are mixtures that do not always behave as a single substance. For instance, they may not evaporate or condense at a constant temperature (called the *temperature glide.*)

7.12.3 Designation of Refrigerants

Refrigerants are designated by letter **R** followed by a unique number. From the number one can predict some useful informations about the type of refrigerant, its chemical composition, molecular weight etc.

(i) Designation of Halocarbon Refrigerants These refrigerants are derived from alkanes (C_nH_{2n+2}) such as methane (CH_4), ethane (C_2H_6). These refrigerants are designated by R(C-1)(H+1)F, where:

C indicates the number of Carbon atoms
H indicates number of Hydrogen atoms, and
F indicates number of Fluorine atoms

Any unsaturated place is filled by Chlorine atoms.

For example **R 22**

Its correct designation is R022

$C - 1 = 0 \Rightarrow$ No. of Carbon atoms, $C = 0 + 1 = 1$ $\Rightarrow$ derived from methane (CH_4)

$H + 1 = 2 \Rightarrow$ No. of Hydrogen atoms, $H = 2 - 1 = 1$

$F = 2 \Rightarrow$ No. of Fluorine atoms = 2

The balance = 4 – no. of (H + F) atoms = 4 – (1 + 2) = 1 $\Rightarrow$ No. of Chlorine atoms = 1

$\therefore$ The chemical formula of R 22 = $CHClF_2$; that is mono-chloro-di-fluoro-methane.

Similarly, the chemical formula of R12 and R134a can be written as:

R12 = CCl_2F_2 that is di-chloro-di-fluoro-methane.

R134a = $C_2H_2F_4$ that is tetra-fluoro-ethane.

(ii) Designation of Inorganic Refrigerants These refrigerants are designated by 700 series. The number 7 is followed by the molecular weight of the refrigerant.

For example

Ammonia: Molecular weight is 17, $\therefore$ the designation is R 717

Carbon dioxide: Molecular weight is 44, $\therefore$ the designation is R 744

Water: Molecular weight is 18, $\therefore$ the designation is R 718

(iii) Designation of Mixtures of Refrigerants Azeotropic mixtures are designated by 500 series, whereas, the zeotropic refrigerants are designated by 400 series.

7.12.4 Some Common Refrigerants

In early days, the commonly used refrigerants were ammonia, methyl chloride, sulphur dioxide, carbon dioxide, ethylchloride, propane, and butane. Some of them were toxic. Some had very high risk of explosion and were irritants. Thus they became obsolete except ammonia and carbon dioxide.

Apparently, the ideal refrigerants are in a chemical group called either fluorinated hydrocarbons or halocarbons (now known as *R*-12). They are being used since 1930, because of their excellent characteristics. They have good physical properties, are non-toxic, stable and inexpensive.

(*a*) *Air* It is designated as R-729. Dry air is used as a primary refrigerant in air refrigeration system, mainly used for aircraft refrigeration. It is also used as a secondary refrigerant in domestic refrigerators.

(*b*) *Water* It is designated as R-718. It is used in its solid form as ice for chilling of beverages and other things. It is also used as refrigerant in vapour absorption systems. Its high freezing-point temperature limits its use in a vapour compression system.

(*c*) *Carbon dioxide* It is designated as R-744. It is a non-toxic, non-irritating and non-flammable gas. It has extremely low freezing temperature, but high operating pressure and temperatures in refrigeration system. Its specific volume is also very low. Therefore, it requires a very small compressor. It has low *COP*, and thus is not preferred for commercial use. It is suitable for manufacturing of dry ice and marine purposes.

(*d*) *Ammonia* It is designated as R-717. Its chemical formula is NH_3. It is a highly toxic gas, irritating to the eyes, nose and throat. It is non-corrosive to ferrous metals and attacks brass and bronze. Its boiling point at atmospheric pressure is –33.3°C and its freezing point is –78°C. Its low boiling-point temperature makes it favourable for use at well below 0°C without lowering its pressure below atmospheric. It is cheap and therefore, widely used in large and commercial cold storages and ice plants. It is also used in vapour absorption systems.

(*e*) *Freon-12* It is a very popular refrigerant and designated as *R*-12. It is colourless, almost odourless, non-toxic, non-flammable and non-irritating. It has a low boiling point temperature as –29°C at atmospheric pressure. It is non-reactive with lubricating oils. It is relative costly and has low latent heat. Thus, refrigeration system requires large mass flow rate. It is used on small refrigerating machines.

(*f*) *Freon-22* It is also a popular refrigerant and designated as *R*-22. It has a low boiling point temperature and is able to maintain a temperature up to – 40°C. It is widely used in air-conditioning units and in household refrigerators. It can be used with reciprocating and centrifugal compressors.

7.12.5 Selection of Refrigerant

The important parameters to be considered in selection of a refrigerant are temperature of refrigerated space and type of equipment to be used.

The temperature of the refrigerant in the evaporator and the condenser are governed by the temperatures of the cold and warm regions, respectively. In order to have as effective heat transfer rate, a temperature difference of at least 10°C should be maintained between the refrigerant and medium, with which the system interacts thermally. These temperatures also fix the operating pressures in the evaporator and condenser. Therefore, the selection of a refrigerant is based partly on the suitability of pressure–temperature relationship in the range of the particular application.

It is generally avoided to have a very low pressure in the evaporator and an excessive high pressure in the condenser in order to minimize the leakages to or from the system.

7.12.6 Ozone Depletion

Ozone (O_3) gas consists of three atoms of oxygen per molecule. It is a vigorous oxidising agent and it cleans the air in the atmosphere. Ozone is present in a layer in the earth's atmosphere, known as the *stratosphere*, about 11–50 km above the earth's surface. The ozone layer blocks out most of harmful ultraviolet (UV) radiation coming from the sun. The ozone layer has been progressively depleting. One chlorine atom can destroy 100,000 ozone molecules by a continuous chain reaction.

The effects of depletion of ozone layer over the earth include the following:

1. The ultraviolet rays in the solar radiation will reach the earth surface and cause an increase in skin cancer (most deadly form of cancer).
2. An increase in eye diseases.
3. Reduction in immunity against disease.
4. Harmful effects on crops, timber, wild life and marine life.

The relative ability of a substance to deplete the ozone layer is called the *ozone depletion potential (ODP)*. *R*-11 and *R*-12 have the highest value, *ODP* = 1.0. Table 7.1 lists some of the refrigerants and their *ODP* values. HCFCs have relatively low *ODP*, while HFCs do not cause any harm to the ozone layer (*ODP* = 0).

Table 7.1 *Ozone Depletion Potential (ODP) of refrigerants*

Sub Group	*Refrigerant Designation*	*Chemical Formula*	*Refrigerant Name*	*ODP*
CFCs	*R*-11	CCl_3F	Trichlorofluoro methane	1.0
	R-12	CCl_2F_2	Dichlorodifluoro methane	1.0
	R-113	$CF_2Cl–CFCl_2$	Trifluoro-tirchlore ethane	0.8
	R-114	$CF_2Cl–CF_2Cl$	Tetrafluoro-dichloro ethane	1.0
HCFCs	*R*-22	$CHClF_2$	Chlorodifluoro methane	0.05
	R-123	$CHCl_2F_3$	Dichlorotrifluoro ethane	0.02
HFC	*R*-134a	CH_2FCF_3	Tetrafluoro ethane	0.0
Azeotrope	*R*-500	*R*-12/*R*-152	HCFC and CFC mixture	0.74
	R-717	NH_3	Ammonia	0

7.12.7 Eco-friendly Refrigerants

Refrigerants that are friendly to the ozone layer in the atmosphere that protects the earth from harmful ultraviolet rays are called *eco-friendly refrigerants*. For example, R-134*a* is a chlorine-free refrigerant and, thus it is an eco-friendly refrigerant.

The chlorine refrigerants react with the ozone layer and try to destroy the ozone layer in the upper atmosphere by continuous chain reaction. Hydrocarbons (HCs) and hydro fluorocarbon groups provide an alternative to chlorinated refrigerant. They contain no chlorine atom and therefore, have zero ozone depletion potential.

Two most common refrigerants, Freon-11 and Freon-12, are highly chlorinated and are very harmful for ozone depletion. Freon-11 is replaced by HCFC-123, which has 98% less ozone deletion potential and Freon-12 is replaced by R-134a (Tetrafluoroethane) with zero ODP.

7.12.8 Montreal Protocol

The major sources of chlorine in the atmosphere are the production of CFCs and HCFCs. The major developed nations have agreed to control the use and manufacturing of CFCs and HCFCs and the United Nations through its environment programme has persuaded many nations to sign the *Vienna Convention 1987*, a treaty specifically intended to control the production of substances which cause ozone depletion. *The Montreal Protocol* of this treaty outlines the phase reduction of compounds. Production and use of some compounds have been banned and some will be banned in future.

The viable solutions include alternate halocarbon refrigerants

1. For CFC-12 (*R*-12) used in automobile air conditioning and household refrigerator; HFC-134*a* (*R*-134*a*) is permanent substitute (ODP = 0)
2. For CFC-11 (*R*-11) used in centrifugal compressor, an interior substitute is HCFC-123 until this group is phased out
3. HCFC-22 (*R*-22) used in window and commercial air-conditioning and refrigeration; possible substitute is HFC mixture *R*-407*c* and *R*-410*a*.

Summary

- The removal of heat from a lower temperature region to a higher temperature region is called refrigeration. Devices that produce refrigeration are called refrigerators and cycles at which they operate are called the refrigeration cycle. The working fluid used in the cycle is called the *refrigerant*. The refrigerators used for heating purpose of a space by transferring heat from a cold medium are called the *heat pumps.*
- The performance of refrigerators and heat pumps is measured in terms of the coefficient of performance (*COP*).

$$(COP)_R = \frac{\text{Desired cooling effect}}{\text{Net work input}} = \frac{Q_L}{W_{in}}$$

$$(COP)_{HP} = \frac{\text{Desired heating effect}}{\text{Net work input}} = \frac{Q_H}{W_{in}}$$

- The most widely used refrigeration cycle is vapour compression cycle (VCC). In an ideal VCC, the refrigerant enters the compressor as a saturated vapour where it is compressed to high pressure. The superheated vapour is then cooled to saturate liquid state in the condenser before throttling to evaporator pressure, where the refrigerant evaporates, absorbs its enthalpy of evaporation (latent heat) from the refrigerated space.
- The power cycles can be used as refrigeration cycles by reversing the directions of processes involved. One of these is the reversed Brayton cycle, used for gas refrigeration systems. It is extremely used for aircraft cooling and liquefaction of gases. The *COP* of a gas refrigeration cycle is

$$(COP)_R = \frac{q_{in}}{w_{net,\,in}} = \frac{q_{in}}{w_{comp} - w_{Turbine}}$$

- One more attractive method of refrigeration is the vapour absorption refrigeration system, where the refrigeration is absorbed by a transport medium and is compressed in liquid form, the reduced compressor work to a pump work only. The used source energy is waste heat, solar energy, etc. The most commonly used vapour absorption refrigeration system is the ammonia–water system with ammonia as refrigerant and water as absorbent. The *COP* of such a system is

$$(COP)_R = \frac{\text{Refrigeration effect}}{\text{Heat input + Pump work}} = \frac{Q_L}{Q_H + W_p}$$

- The selection criteria of refrigerant for a refrigeration cycle involves normal operating pressure at very low and high temperatures, low specific volume at evaporator pressure, a high latent heat of evaporation at low temperature, chemical stability, non-toxic, non-corrosive and overall eco-friendly. CFCs and HCFCs contain large chlorine lebels and thus cause depletion to the ozone layer in the stratosphere of the earth's atmosphere. Thus, their use and production are gradually phased out.

Glossary

Air refrigeration A refrigeration system in which air is the working medium.

Heat pump A machine which keeps a space at higher temperature than its surroundings

Primary refrigerant A substance which undergoes refrigeration cycle

Refrigerant Working substance used in the refrigeration system

Refrigeration effect Cooling effect

Refrigeration Maintaining a space at low temperature than its surroundings

Refrigerator A machine which produces refrigeration effect

Secondary refrigerant A substance which acts as a heat carrier and exchanges heat with primary refrigerant only

Vapour absorption–Refrigeration A refrigeration system in which refrigerant and its absorbent are used for creating refrigeration effect without compressor

Vapour compression–Refrigeration A refrigeration system in which liquid and its vapour undergo refrigeration cycle

Review Questions

1. What are temperatures inside the fresh food and freezer compartments of a household refrigerator?
2. What is frosting? How can excessive frosting harm the performance of a refrigerator?
3. Define refrigerating effect. What is one tonne of refrigeration? What is the basic formula for calculating the tonnage of refrigeration?
4. What are the commonly used refrigerants for vapour compression refrigeration systems? Would water be a suitable refrigerant in a refrigerator?
5. What do you understand by primary and secondary refrigerants? Explain in brief.
6. How is the heat absorbed or removed from a low-temperature source and transferred to high-temperature source in a vapour compression system?
7. What is the difference between a refrigerator and a heat pump?
8. Why is the reversed Carnot cycle executed within a saturation curve and not a realistic model for refrigeration cycles?
9. Prove that $(COP)_{HP} = (COP)_R + 1$
10. Sketch the vapour compression cycle on a *T–s* diagram and derive an expression for its *COP*.
11. What is the function of a condenser in a refrigeration cycle?
12. What are the causes of irreversibilities in an actual refrigeration cycle ? Explain with the help of a *T–s* diagram.
13. What is the effect of lower evaporator pressure in a VCC on its performance?
14. Consider two vapour compression refrigeration cycles. The refrigerant enters the throttle valve as a saturated liquid at 30°C in one cycle and as a subcooled liquid at 30°C in another cycle. The evaporator pressure for both cycles is same. Which cycle do you think will have higher *COP*?
15. Why are CFC phased out? Which are the alternatives to CFCs?
16. What is ozone depletion? What are the remedies to save ozone in atmosphere?

Problems

1. If the power consumed by a reversible refrigeration compressor is 2 kW per tonne of refrigeration, what is it *COP*? *[1.75]*

2. What is the refrigeration load in *TR*, when 20 m^3/min of water is cooled from 13°C to 8°C? *[33.0 TR]*

3. Calculate the power required by a reversible refrigerator to produce 400 kg of ice per hour at –20°C from feed water at 25°C. Take latent heat of ice formation as 335 kJ/kg · K and specific heat of ice as 2.09 kJ/kg · K. *[9.92 kW]*

4. A refrigerator working on Bell Coleman cycle operates between atmospheric pressure and 7 bar. Air is drawn from a cold chamber at 13°C. Before air expander, it is cooled to 28°C. Calculate *COP*, if index of expansion and compression is 1.35 *[1.53]*

5. The *COP* of a vapour compression refrigeration system is 3.0. If the compressor motor draws a power of 10.5 kW at 91% motor efficiency, what is the refrigeration effect in *TR* of system? *[9.84 TR]*

6. (a) Ice is formed at 0°C from water at 30°C. In the refrigeration system the same water is used for condenser cooling and the temperature of brine is –15°C at evaporator. Considering the system operates on Carnot refrigeration system, calculate the *COP* of the system. *[5.7]*

7. Refrigerant 134*a* enters the compressor of an ideal vapour compression refrigeration system as saturated vapour at –16°C with a volumetric flow rate of 1 m^3/min. The refrigerant leaves the condenser at 36°C, 10 bar. Determine

(a) the compressor power, in kW

(b) the refrigerating capacity, in tonnes

(c) the coefficient of performance

8. The R-12 refrigerant is used in a refrigerator operating between 36°C and –7°C. The refrigerant is dry and saturated at the beginning of compression. If the actual *COP* is 0.65 times the theoretical *COP*, calculate the net cooling produced per hour for a flow rate of 5 kg/min. The properties of R-12 are given:

Temp. °C	h_f	h_{fg}	s_f	s_{fg}
36	456.4	585.3	4.74	0.67
–7	412.4	570.3	4.76	—

[1.77 TR]

9. The condenser and evaporator of a 15 *TR* ammonia refrigerating plant are maintained at 25°C and –10°C. The refrigerant is subcooled by 5°C in the condenser. The vapour leaving the evaporator is 0.97 dry. Calculate *COP* and power required. Use the following properties:

Temp. °C	h_f	h_{fg}	s_f	s_{fg}
25	317.687	1483.18	1.4084	5.3175
–10	154.056	1450.22	.8296	5.755

[6.86, 7.65 kW]

10. A refrigeration system of 15 tonnes capacity operates on a vapour compression cycle using R-22 at an evaporator temperature of 5°C and condensing temperature of 50°C. Calculate the refrigerant mass flow rate and compressor intake volume flow rate, if the volumetric efficiency is 72%. Use following properties:

Pressure	Temp.	h_f	h_{fg}	v_f	v_g
bar	°C	kJ/kg	kJ/kg	lit/kg	lit/kg
5.836	5	205.9	407.1	0.791	0.0404
19.423	50	263.3	417.7	.922	.0117

[0.365 kg/s, 0.01475 m^3/kg]

11. A 20 tonnes vapour compression refrigeration system using Freon-12 operates between an evaporator pressure of 1.004 bar and a condenser pressure of 13.663 bar. The system uses 10°C superheating. Calculate the mass flow rate, *COP*, degree of sub cooling and power input. The refrigerant leaving the condenser is dry saturated liquid and leaving the evaporator is dry saturated vapour. The compression is isentropic. The properties of Freon -12 are

Pressure	Temp.	h_f	h_{fg}	s_f	s_g	$C_{p,f}$	$C_{p,v}$
bar	°C	kJ/kg	kJ/kg	kJ/kg·K		kJ/kg·K	
1.004	–30	8.854	174.076	0.371	0.7165	–	0.579
13.663	55	90.201	207.766	0.3194	0.6777	1.074	–

[0.8346 kg/s, 1.8, 0, 38.9 kW]

12. In an aircraft cooling system, air enters the compressor at 0.1 MPa, 4°C and is compressed to 6 MPa with an isentropic efficiency of 72%. After being cooled to 55°C at constant pressure in a heat exchanger, the air then expands in a turbine to 0.1 MPa with an isentropic efficiency of 78%. The cooling load of the system is 3 tonnes of refrigeration at constant pressure before re-entering the compressor, which is driven by the turbine. Assuming that air is an ideal gas, determine the *COP* of refrigerator, driving power required and the mass flow rate of air.

13. An air refrigeration used for food storage provides 25 TR. The temperature of air entering the compressor at 7°C and temperature at exit of the cooler is 27°C. The quantity of air circulated in the system is 50 kg/min. The compression and expansion both follow the law $pv^{1.3}$ = constant. And g(gamma) = 1.4 and C_p = 1 kJ/kg·K. Find

(i) *COP* of the cycle, and

(ii) Power per ton of refrigeration required by compressor.

[Ans. (i) 1.125, (ii) 1.562 kW/TR]

Objective Questions

1. *COP* of a Carnot refrigeration cycle is greater than
   (a) vapour compression cycle
   (b) reversed Brayton cycle
   (c) vapour absorption cycle
   (d) all the above
2. In an ideal vapour compression refrigeration cycle, which process is irreversible?
   (a) Compression (b) Heat rejection
   (c) Throttling (d) Heat absorption
3. Subcooling of refrigerant in vapour compression refrigeration cycle
   (a) decreases *COP*
   (b) increases *COP*
   (c) decrease refrigerating effect
   (d) increases work input
4. Superheating of vapour refrigerant at evaporator in vapour compression cycle
   (a) decreases *COP*
   (b) increases *COP*
   (c) decrease refrigerating effect
   (d) increases work input
5. Heat is absorbed by a refrigerant during a refrigerant cycle
   (a) condenser (b) throttle valve
   (c) evaporator (d) compressor
6. Heat is rejected by a refrigerant during a refrigerant cycle
   (a) condenser (b) throttle valve
   (c) evaporator (d) compressor
7. In an ideal vapour compression refrigeration cycle, the refrigerant is in the form of superheated vapour before entering into
   (a) condenser (b) throttle valve
   (c) evaporator (d) compressor
8. In an ideal vapour compression refrigeration cycle, the refrigerant is in form of dry saturated vapour before entering into
   (a) condenser (b) throttle valve
   (c) evaporator (d) compressor
9. The refrigerant used in vapour absorption refrigeration system is
   (a) Freon-12 (b) CO_2
   (c) ammonia (d) R-134*a*
10. The *COP* of Carnot cycle when working as heat pump is
    (a) less than when it works as refrigerator
    (b) more than when it works as refrigerator
    (c) less than when it works as heat engine
    (d) equal to when it work as refrigerator
11. The secondary refrigerant acts to
    (a) undergo the refrigeration cycle
    (b) protect the ozone depletion
    (c) carry the heat within refrigerated space
    (d) cool the water
12. A zeotropes are mixture of
    (a) primary and secondary refrigerant
    (b) Ammonia and water
    (c) CFCs and HFCs
    (d) HCFCs and HFCs
13. Ozone depletion is caused due to
    (a) use of HFCs
    (b) use of HCs
    (c) use of CFC
    (d) use of Ammonia
14. Refrigerant R-134a is
    (a) an ecofriendly refrigerant
    (b) mixture of hydrofluoro carbon
    (c) Halo carbone refrigerant
    (d) all of above

Answers

1. (d) 2. (c) 3. (b) 4. (d) 5. (c) 6. (a) 7. (a) 8. (d)
9. (c) 10. (b) 11. (c) 12. (c) 13. (c) 14. (d)

8 CHAPTER

Psychrometry

Introduction

The *Psychrometry* is the study of the properties of moist air and is useful to engineers concerned with heating, cooling, and ventilation of buildings, ships and aircrafts.

In most of air-conditioning applications, the atmospheric air is used at low temperatures (below 30°C). The water vapour present in atmospheric air has very low partial pressure and it behaves as a perfect gas. Thus, the air-vapour mixture at low pressure and atmospheric temperature can be modeled as an ideal gas. That is,

(i) the equation of state is $pv = RT$,
(ii) the enthalpy of the gas mixture is the function of temperature only, and
(iii) the vapour behaves in all respects as it existed alone at its partial pressure and temperature of the mixture.

8.1 PSYCHROMETER

The psychrometer is an instrument, that is used to measure dry-bulb and wet-bulb temperatures in the laboratory. It is used in the sling method, and therefore, the equipment is called a *sling psychrometer* as shown in Fig. 8.1. It consists of a dry-bulb thermometer and a wet-bulb thermometer mounted side by side in a protective case that is attached to a handle by a swivel connection, so that the case can be rotated. The sling thermometer is rotated for one minute in air and then readings are taken from the thermometers.

The dry-bulb thermometer is directly exposed to the atmospheric air and measures the actual temperature of air. The bulb of a wet-bulb thermometer is covered with water soaked cotton wick. The temperature measured by a wick-covered bulb is the temperature of the liquid water in the wick and thus is called the *wet bulb temperature*.

8.2 DRY, MOIST AND SATURATED AIR

1. Dry Air The *dry air* includes all components of atmospheric air except water vapour (moisture). Thus, the dry air is the mixture of oxygen, nitrogen, carbon dioxide, argon, etc. At low temperature (cold conditions), its specific heat remains almost constant and is taken as 1.005 kJ/kg·K.

2. Moist Air The term *moist air* is the mixture of the dry air and water vapour, and each component of the mixture behaves as an ideal gas at states under considerations.

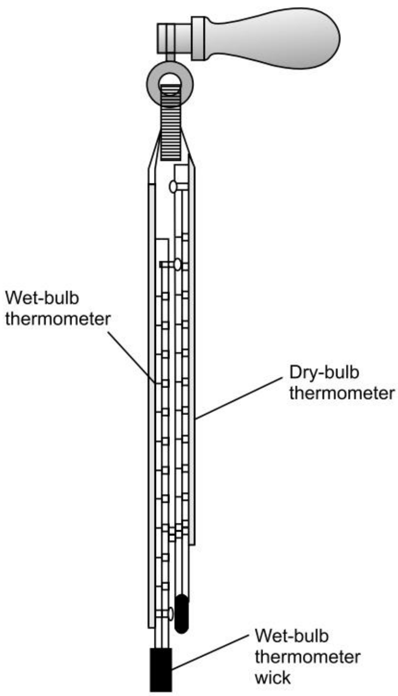


Fig. 8.1 *Sling psychrometer*

3. Unsaturated Air In atmospheric air, the moisture appears in the form of superheated vapour as invisible gas at atmospheric pressure. The mixture of superheated vapour and dry air is called *unsaturated air*.

4. Saturated Air The *saturated air* is moist air which contains maximum possible water vapour. The partial pressure of water vapour in saturated air is equal to the saturated pressure of steam corresponding to the temperature of moist air. Cooling of saturated air causes the separation of moisture and formation of fog.

Figure 8.2 illustrates the states of saturated and unsaturated air. At the state 1, the partial pressure p_v of water vapour is lower than the saturation pressure; thus the water vapour is in superheated state, the air is unsaturated at the mixture temperature T and total pressure p. At this state, partial pressure of air is $p_a = p - p_v$.

The state 2 is after isothermal cooling, when saturated water vapour is at the temperature T of

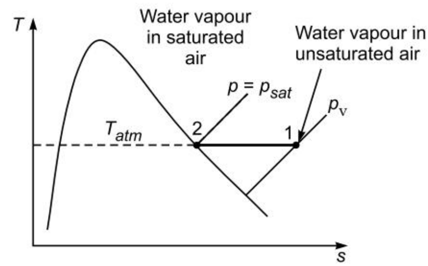


Fig. 8.2 *T–s diagram for unsaturated and saturated air*

the mixture. The corresponding partial pressure of air becomes p_{sat} equal to saturation pressure at temperature T and the partial pressure of air would be $p_a = p - p_{sat}$.

8.3 PROPERTIES OF MOIST AIR

Some of the important psychrometric properties of moist are defined below:

8.3.1 Dry-bulb Temperature

The *dry-bulb temperature* (DBT), designated as T (or sometimes T_{db}), of the mixture is the temperature measured by an ordinary thermometer placed in the air-vapour mixture.

8.3.2 Wet-bulb Temperature

The *wet-bulb temperature* (WBT), T_{wb}, is the temperature measured by a thermometer whose bulb is covered by a thoroughly wetted cotton wick.

When air passes over the wetted cotton wick, some of the water evaporates, cooling effect is produced at the bulb and the temperature recorded is lower than that of the air stream.

Figure 8.3 shows the method of measurement of wet and dry bulb temperatures. The two thermometers are located in the stream of unsaturated air. As the air stream passes the wet cotton wick, some water evaporates and produces a cooling effect in the wick. The heat is transferred from air to wick with corresponding cooling. An equilibrium condition is reached at which the wet bulb thermometer indicates a temperature, that is lower than the dry bulb temperature.

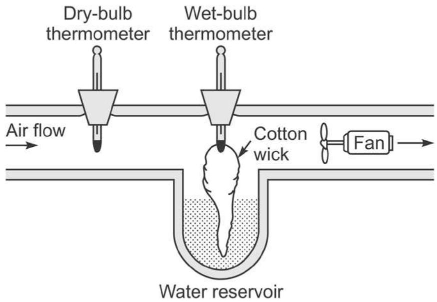


Fig. 8.3 *Steady-flow apparatus for measuring wet and dry-bulb temperature*

8.3.3 Wet-bulb Depression

The difference between the dry-bulb temperature and wet-bulb temperature is called as *wet-bulb depression*.

$$\text{Wet-bulb depression} = T - T_{wb} \quad ...(8.1)$$

The amount of wet-bulb depression depends on moisture content of air. For completely dry air, the wet-bulb depression is maximum, because dry air is capable of absorbing maximum amount of moisture. On the other hand, when air is saturated, it cannot absorb moisture any more and the wet bulb temperature reading equals the dry bulb temperature readings. It means, wet bulb depression approaches zero for saturated air.

8.3.4 Dew-point Temperature

The *dew-point temperature, T_{dp}*, of an air–vapour mixture is a temperature at which the vapour starts to condense, when it is cooled at constant pressure. The partial condensation of water vapour on window panes in winter, or on pipes carrying cold water encounters dew most often. The formation of dew on the grass is the best example. It occurs due to cooling of water vapour at constant pressure to its saturation temperature.

Figure 8.4 illustrates the dew point on a T–s diagram. The air–vapour mixture at the state 1 is superheated. It is cooled at constant pressure p_v to the state 2, where the saturation temperature reaches and condensation of vapour begins. The temperature at state 2 is dew point temperature.

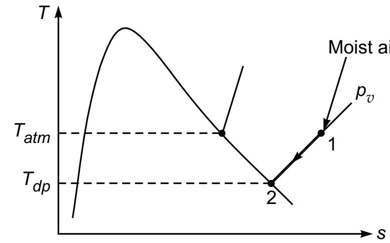


Fig. 8.4 *Cooling of vapour at constant pressure brings dew point temperature*

8.3.5 Dew-point Depression

The difference between the dry-bulb temperature and dew-point temperature is called as *dew-point depression*.

$$\text{Dew-point depression} = T - T_{dp} \quad ...(8.2)$$

8.3.6 Relative Humidity

It is defined as the *ratio of actual mass of water vapour to the mass of saturated vapour* produced in a mixture of air and water vapour at the same temperature and pressure. It is denoted by ϕ(phi). The water vapours are superheated and thus behave like an ideal gas,

thus $$\phi = \frac{m_v}{m_{sat}} = \frac{p_v V/R_v T}{p_{sat} V/R_v T} = \frac{p_v}{p_{sat}} \quad ...(8.3)$$

where p_v = partial pressure of vapour in the mixture

p_{sat} = saturation pressure of vapour at temperature of mixture

T = mixture temperature

Since mixture temperature and volume remain constant, thus it can also be expressed as

$$\phi = \frac{p_v}{p_{sat}} = \frac{v_{sat}}{v_v} = \frac{\rho_v}{\rho_{sat}} \quad ...(8.4)$$

8.3.7 Specific Humidity

The *specific humidity* is also called *humidity ratio* or *absolute humidity*. It is defined as *ratio of mass of water vapour to the mass of dry air* in a given volume of the mixture. It is denoted by ω (omega)

$$\omega = \frac{m_v}{m_a} \quad ...(8.5)$$

where the subscript a refers to dry air and v refers to vapour. Since both the vapour and mixture are considered as ideal gases, thus

$$m_v = \frac{p_v V}{R_v T} = \frac{p_v V \mathcal{M}_v}{R_u T}$$

and
$$m_a = \frac{p_a V}{R_a T} = \frac{p_a V \mathcal{M}_a}{R_u T}$$

Therefore, $$\omega = \frac{p_v \mathcal{M}_v}{p_a \mathcal{M}_a} \qquad ...(8.6)$$

where $\mathcal{M}$ is molecular weight and R_u is universal gas constant. Suffix a is used for air and v for vapour.

For an air–water vapour mixture($\mathcal{M}_v$ = 18 and $\mathcal{M}_a$ = 28.97), then Eqn. (8.6) reduces to

$$\omega = 0.622 \frac{p_v}{p_a} = 0.622 \frac{p_v}{p - p_v} \qquad ...(8.7)$$

where p is the total pressure of the mixture ($p_a + p_v$), thus $p_a = p - p_v$.

8.3.8 Degree of Saturation

It is defined as the ratio of *actual humidity ratio* to the *humidity ratio of saturated air* at the same temperature and total pressure. It is designated as μ and is expressed as

$$\mu = \frac{\omega}{\omega_{sat}} \qquad ...(8.8)$$

Using $\omega = 0.622 \dfrac{p_v}{p - p_v}$ from Eq. (8.7), and at saturation state

$$\omega_{sat} = 0.622 \frac{p_{sat}}{p - p_{sat}}$$

then
$$\mu = \frac{p_v (p - p_{sat})}{p_{sat} (p - p_v)} = \phi \frac{p - p_{sat}}{p - p_v} \qquad ...(8.9)$$

$$\mu p - \mu p_v = \phi (p - p_{sat})$$

Using $p_v = \phi p_{sat}$ from Eq. (8.3), then

or
$$\mu p = \phi (p - p_{sat} + \mu p_{sat})$$
$$= \phi [p - (1 - \mu) p_{sat}] \approx \phi p$$

Since $(1 - \mu) p_{sat} \approx 0$

$$\therefore \qquad \mu = \phi \qquad ...(8.10)$$

8.3.9 Enthalpy of Moist Air

The enthalpy of moist air is the sum of enthalpy of dry air and that of water vapour. Thus, the enthalpy h of moist air per kg of dry air is expressed as

or $$m_a h = m_a h_a + m_v h_g$$

or $$h = h_a + \frac{m_v}{m_a} h_g$$
$$= (h_a + \omega h_g) \text{ kJ/kg of dry air} \qquad ...(8.11)$$

where h_a = enthalpy of dry air
$= C_p T_{db} = 1.005\, T$
h_g = enthalpy of water vapour calculated as
$= 2500 + 1.88\, T$ kJ/kg of water vapour
T = dry-bulb temperature in °C

8.4 PARTIAL PRESSURE OF AIR AND VAPOUR

Figure 8.5 shows a closed system consisting of atmospheric air, a mixture of water vapour and dry air occupying volume V, at mixture pressure p and mixture temperature T. The mixture is assumed to follow ideal gas equation. Thus, the mixture pressure

$$p = \frac{n R_u T}{V}$$

Introducing $$n = \frac{m}{\mathcal{M}};$$

$$p = \frac{m R_u T}{\mathcal{M} V} \qquad ...(8.12)$$

where n, m, $\mathcal{M}$ and R_u denote the number of moles, mass, molecular weight and universal gas constant, respectively.

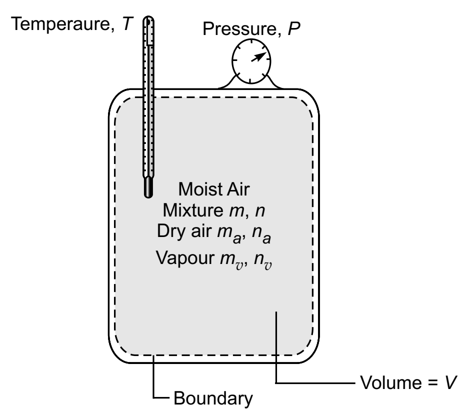


Fig. 8.5 *Mixture of moist air in a closed vessel*

The partial pressure of dry air p_a in the mixture;

$$p_a = \frac{m_a R_u T}{\mathcal{M}_a V} \qquad ...(8.13)$$

The partial pressure of water vapour p_v in the mixture

$$p_v = \frac{m_v R_u T}{\mathcal{M}_v V} \qquad ...(8.14)$$

where m_a, m_v denote the mass of dry air and water vapour, respectively and $\mathcal{M}_a$ and $\mathcal{M}_v$ are respective molecular weights in the mixture.

The total pressure of the mixture is the sum of partial pressure of water vapour and dry air present in the mixture.

$$p = p_a + p_v \qquad ...(8.15)$$

The saturation pressure of water corresponding to dew point can be obtained from steam tables, which is equal to the partial pressure of water vapour in air.

The wet-bulb temperature of the mixture is easily measured with a wetted wick thermometer. The partial pressure p_v of vapour in the mixture can be calculated by using Dr Willis H *Carrier's equation*

$$p_v = p_v' - \frac{(p - p_v')(T - T_{wb}) \times 1.8}{2800 - 1.3 \times (1.8T + 32)} \qquad ...(8.16)$$

where p_v = partial pressure of water vapour at dry bulb temperatutre in mixture,

p_v' = saturation water pressure at wet bulb temperature,

p = total pressure of moist air,

T = dry-bulb temperature,

T_{wb} = wet-bulb temperature.

Example 8.1 *Atmospheric air at 95 kPa, 30°C has a relative humidity of 70 per cent. Determine humidity ratio.*

Solution

Given Atmospheric air

$p = p_{atm} = 95$ kPa $\qquad T = 30°C = 303$ K

$\phi = 0.7$

To find Humidity ratio

Analysis From steam tables, the saturation pressure of water at 30°C

$$p_{sat} = 4.246 \text{ kPa}$$

The partial pressure of water vapour in mixture

$$p_v = \phi p_{sat} = 0.7 \times 4.246 = 2.97 \text{ kPa}$$

The partial pressure of air,

$$p_a = p - p_v = 95 - 2.97 = 92.03 \text{ kPa}$$

The specific humidity by using Eq. (8.7)

$$\omega = 0.622 \frac{p_v}{p_a} = 0.622 \times \frac{2.97}{92.03}$$

$$= \mathbf{0.02007 \text{ kg/kg/of dry air}}$$

Example 8.2 *A sample of 450 gram of moist air at 22°C, 101 kPa and 70% relative humidity is cooled to 5°C, while keeping the pressure constant. Determine (a) the initial humidty ratio (b) dew point temperature, and (c) amount of water vapour that condenses.*

Solution

Given A sample of moist air is cooled at constant pressure

$m = 0.45$ kg $\qquad T_1 = 22°C = 295$ K

$p = 101$ kPa $\qquad p$ = constant

$\phi = 70\%$ $\qquad T_2 = 5°C$

To find

(i) Initial humidity ratio,

(ii) Dew point temperature, and

(iii) Amount of water vapour that condenses.

Schematic with given data

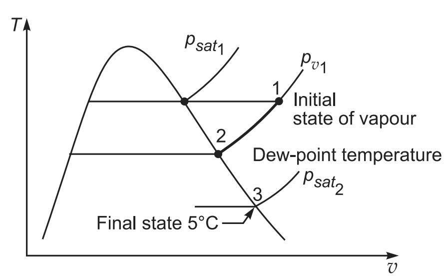


Fig. 8.6

Assumptions

(i) 1 kg of moist air sample as a closed system.

(ii) The gas phase of air and vapour can be treated as an ideal gas mixture.

Analysis From steam tables at 22°C;

$$p_{sat_1} = p_{sat} @ 22°C = 0.02645 \text{ bar}$$

The partial pressure of water vapour p_{v_1}

$$p_{v_1} = \phi\, p_{sat_1} = 0.7 \times 0.02645 = 0.0185 \text{ bar} = 1.85 \text{ kPa}$$

(i) The humidity ratio at initial state

$$\omega_1 = 0.622 \frac{p_{v_1}}{p - p_{v_1}} = 0.622 \times \frac{1.85}{101 - 1.85} = \mathbf{0.011\ kg/kg\ of\ dry\ air}$$

(ii) Dew-point temperature is the saturation temperature corresponding to partial pressure p_{v_1}. Interpolating between 16°C and 17°C, we get

$$T_{dp} = \mathbf{16.267°C}$$

(iii) The amount of condensate,

$$m_w = \text{Initial amount of water vapour} - \text{Final amount of water vapour} = m_{v_1} - m_{v_2}$$

For 1 kg of dry air and moisture,

$$1 \text{ kg} = m_a + m_{v_1}$$

Using $\omega_1 = \dfrac{m_{v_1}}{m_a}$;

$$1 \text{ kg} = \frac{m_{v_1}}{\omega_1} + m_{v_1} = m_{v_1}\left(\frac{1}{\omega} + 1\right)$$

or

$$m_{v_1} = \frac{\omega_1}{\omega_1 + 1} = \frac{0.011}{0.011 + 1} = 0.0109 \text{ kg/kg of dry air}$$

The mass of dry air/kg,

$$m_a = 1 - 0.0109 = 0.9891 \text{ kg}$$

Further, humidity ratio ω_2 at state 3, *i.e.,* at 5°C

$$p_{sat} = 0.00872 \text{ bar} = 0.872 \text{ kPa}$$

$$\omega_2 = \frac{0.622 \times 0.872}{101 - 0.872} = 0.0054 \text{ kg/kg of dry air}$$

The mass of water vapour

$$m_{v_2} = \omega_2\, m_a = 0.0054 \times 0.9891 = 0.00535 \text{ kg/kg of dry air}$$

The amount of water vapour condensed

$$m_w = m_{v_1} - m_{v_2} = 0.0109 - 0.00535 = 0.0055 \text{ kg/kg of dry air}$$

In the sample of 0.450 kg

$$m_w = 0.45 \times 0.0055 = \mathbf{0.0025\ kg}$$

Example 8.3 *An air–water vapour mixture is contained in a rigid closed vessel with a volume of 35 m^3 at 1.5 bar, 120°C, and ϕ = 10%. The mixture is cooled at constant volume with its temperature decrease to 22°C. Determine (a) the dew point temperature corresponding to initial state, (b) the temperature at which the condensation actually begins, and (c) amount of water condensed.*

Solution

Given An air–vapour mixture in a closed vessel,

$V = 35$ m^3 $\quad p_1 = 1.5$ bar

$T_1 = 120°C = 393$ K $\quad \phi_1 = 0.1$

$V_1 = V_2$ $\quad T_2 = 22°C$

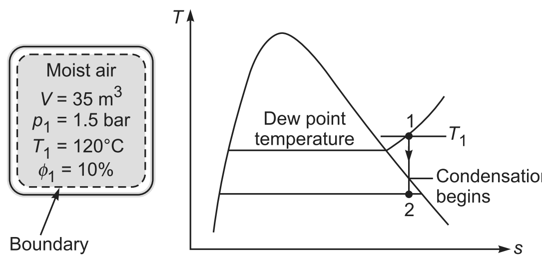


Fig. 8.7 *Schematic with given data and T–s diagram*

To find

(i) Dew point temperature,

(ii) The temperature, at which condensation starts,

(iii) Amount of water condensed.

Assumptions

(i) The contents of vessel are taken as closed system.

(ii) The gas phase of air and vapour can be treated as ideal gases.

(iii) When liquid water is present, the vapour exists at saturation temperature.

Analysis

(i) The partial pressure of vapour at the state 1;

$$p_{v_1} = \phi_1\, p_{sat_1}$$

$$p_{sat_1} = p_{sat} @ 120°C = 1.985 \text{ bar}$$

Thus $\quad p_{v_1} = 0.1 \times 1.985 = 0.1985$ bar

The saturation temperature corresponds to 0.1985 bar

$$T_{sat} = \mathbf{60°C}$$

It is the dew point temperature

(ii) The specific volume of vapour in the mixture

$$v_{v_1} = \frac{R_u T}{\mathcal{M}_{v_1} p_{v_1}}$$

$$= \left(\frac{8.314}{18}\right) \times \left(\frac{393 \text{ K}}{0.1985 \times 100 \text{ kPa}}\right)$$

$$= 9.145 \text{ m}^3/\text{kg}$$

Interpolating, $v_{v_1} = v_g$, gives $T = \mathbf{56°C}$

Thus the condensation will begin at 56°C.

(iii) The mass of water initially present in moist air

$$m_{v_1} = \frac{V}{v_{v_1}} = \frac{35}{9.145} = 3.827 \text{ kg}$$

The propetrties of steam at $T = 56°C$ at the state 2,

$$v_{f_2} = 0.001015 \text{ m}^3/\text{kg},$$

$$v_{fg_2} = 9.158 \text{ m}^3/\text{kg}$$

The vapour has a two-phase mixture having specific volume of 9.145 m^3/kg

$$v_{v_1} = v_{f_2} + x_2\,(v_{fg_2})$$

$$9.145 = 0.001015 + x_2\,(9.158)$$

or $$x_2 = 0.998$$

The mass of water vapour at state 2

$$m_{v_2} = x_2\, m_{v_1} = 0.998 \times 3.827 = 3.82 \text{ kg}$$

The mass of water vapour condensed

$$m_w = m_{v_1} - m_{v_2} = 3.827 - 3.821$$

$$= \mathbf{0.007\ kg}$$

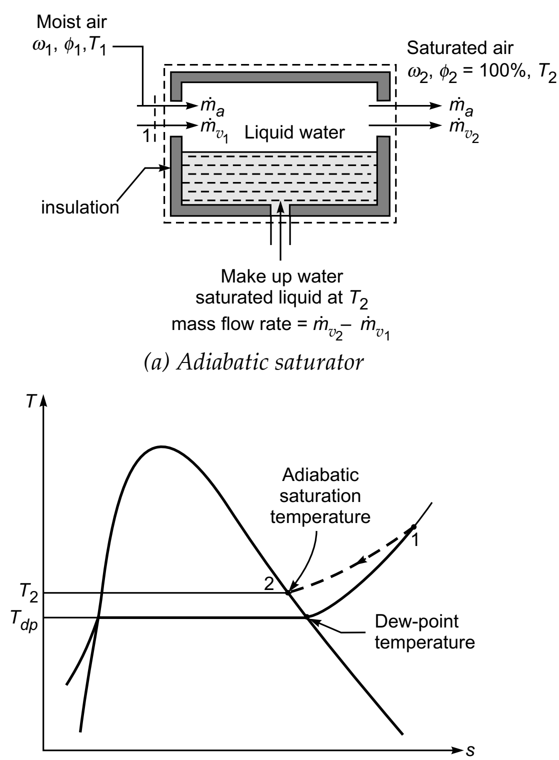
8.5 ADIABATIC SATURATION TEMPERATURE

In atmospheric air, the relative humidity is always less than 100%. The water-vapour pressure is lower than the saturation pressure. If this air is exposed to liquid water, some liquid evaporates and mixes with the air. The specific humidity of air increases. If such a process is carried in an insulated duct then the air temperature will also decrease due to latent heat of absorption during evaporation of water.

The *adiabatic saturation temperature* of atmospheric air is defined as the temperature which results from adding water vapour adiabatically to the atmospheric air in a steady flow manner until it becomes completely saturated.

A steady stream of unsaturated air with specific humidity ω_1 and temperature T_1 is passed through an insulated duct containing a pool of water as shown in Fig. 8.8(a). As air flows over the water, some water evaporates and its vapours mix with the air stream. The moisture content of air increases and its temperature decreases. If the duct is long enough, the air stream will come out completely saturated ($\phi = 100\%$) at the temperature T_2 which is called *adiabatic saturation temperature*.

If make up water is supplied to the duct at the rate at which it evaporates then the above process can be treated as a steady flow process. Figure 8.8(b) illustrates the adiabatic saturation process on the *T–s* diagram.



(a) Adiabatic saturator

(b) Process representation of T–s diagram

Fig. 8.8 *The adiabatic saturation process*

Analysis At steady state, the mass-flow rate of dry air entering the saturator $\dot{m}_a$ must be equal to the mass-flow rate of dry air leaving the saturator. The mass flow rate of make up water is the difference between the exiting and entering vapour flow rates denoted by $\dot{m}_{v_2} - \dot{m}_{v_1}$, respectively. These flow rates are labelled on Fig. 8.8(a). In absence of kinetic and potential energy changes, the steady-flow energy equation reduces to

Enthalpy of mixture in + enthalpy of liquid water added = Enthalpy of mixture out.

$$\dot{m}_a h_{a_1} + \dot{m}_{v_1} h_{g_1} + (\dot{m}_{v_2} - \dot{m}_{v_1})\, h_{f_2} = \dot{m}_a h_{a_2} + \dot{m}_{v_2} h_{g_2} \quad \ldots(8.17)$$

Dividing both sides by $\dot{m}_a$,

$$h_{a_1} + \omega_1 h_{g_1} + (\omega_2 - \omega_1) h_{f_2} = h_{a_2} + \omega_2 h_{g_2}$$

or
$$\omega_1 (h_{g_1} - h_{f_2}) = h_{a_2} - h_{a_1} + \omega_2 (h_{g_2} - h_{f_2})$$

For ideal gas behaviour

$$\omega_1 (h_{g_1} - h_{f_2}) = C_{pa} (T_2 - T_1) + \omega_2 h_{fg_2}$$

or
$$\omega_1 = \frac{C_{pa} (T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}} \quad \ldots(8.18)$$

where
$$\omega_2 = 0.622 \frac{p_{sat_2}}{p_2 - p_{sat_2}}$$

and
$$h_{fg_2} = h_{g_2} - h_{f_2}$$

The temperature of completely saturated air is called *wet bulb temperature* (*wbt*).

Example 8.4 *The air at 28°C and 1 bar has a specific humidity of 0.016 kg per kg of dry air. Determine (a) partial pressure of water vapour, (b) relative humidity, (c) dew point temperature, and (d) specific enthalpy.*

Solution The moist air with

p = 1 bar = 100 kPa

T = 28°C = 301 K

ω = 0.016 kg/kg of dry air

To find

(i) Partial pressure of water vapour, p_{v_1},

(ii) Relative humidity ϕ,

(iii) Dew point temperature, T_{dp},

(iv) Specific enthalpy.

Analysis

(i) Partial pressure of water vapour

The specific humidity is expressed as

$$\omega = 0.622 \frac{p_v}{p - p_v}$$

using numerical values

$$0.016 = 0.622 \times \frac{p_v}{100 - p_v}$$

or $100 - p_v = 38.875\, p_v$

or $p_v = $ **2.5 kPa**

(ii) Relative humidity ϕ;

From steam table, the saturation pressure p_{sat} corresponding to dry-bulb temperature of 28°C;

$$p_{sat} = 0.03778 \text{ bar} = 3.778 \text{ kPa}$$

$$\phi = \frac{p_v}{p_{sat}} = \frac{2.5}{3.778} = 0.661$$

= **66.1%**

(iii) Dew-point temperature

The dew-point temperature is the saturation temperature corresponding to partial pressure of vapour. Thus, saturation temperature corresponds to 2.5 kPa = **21.1°C**

(iv) Specific enthalpy

$$h = h_a + \omega h_g$$

where $h_a = C_p T_{db} = 1.005 \times (28°\text{C})$

$= 28.14$ kJ/kg.

$h_g = 2500 + 1.88\, T_{db} = 2500 + 1.88 \times 28$

$= 2552.64$ kJ/kg

$h = 28.14 + 0.016 \times 2552.64$

= **6898 kJ/kg**

Example 8.5 *The pressure of the air entering and leaving the adiabaltic saturator is 1 bar. The air enters at 30°C and leaves as saturated air at 20°C. Calculate the specific humidity, relative humidity of the air–vapour mixture entering.*

Solution

Given Adiabatic saturator with

$p_1 = p_2$ = 1 bar = 100 kPa

T_1 = 30°C

T_2 = 20°C

ϕ_2 = 100%

To find At entrance of adiabatic saturator

(i) Specific humidity, and

(ii) Relative humidity.

Assumptions

(i) Ideal gas mixture;

(ii) Constant specific heat of air; as C_{pa} = 1.0035 kJ/kg·K.

Analysis Properties of steam at 20°C

$$\text{At } 30°\text{C } h_{g_1} = 2556.3 \text{ kJ/kg (from steam table)}$$
$$p_{sat_1} = 4.246 \text{ kPa}$$
$$p_{sat_2} = 2.339 \text{ kPa}$$
$$h_{fg_2} = 2454.1 \text{ kJ/kg}$$
$$h_{f_2} = 83.96 \text{ kJ/kg}$$

(i) Since the water vapour leaving the saturator is completely saturated (ϕ_2 = 100%), thus

$$p_{v_2} = p_{sat_2}$$

The specific humidity of air leaving the saturator

$$\omega_2 = 0.622 \frac{p_{v_2}}{p - p_{v_2}} = 0.622 \times \frac{2.339}{100 - 2.339}$$
$$= 0.0149 \text{ kg/kg of dry air}$$

The specific humidity of air entering is calculated as

$$\omega_1 = \frac{C_{pa}(T_2 - T_1) + \omega_2 h_{fg_2}}{(h_{g_1} - h_{f_2})}$$

Using these values,

$$\omega_1 = \frac{1.0035 \times (20 - 30) + 0.0149 \times 2454.1}{2556.3 - 83.96}$$
$$= \mathbf{0.0107 \text{ kg/kg of dry air}}$$

(ii) The specific humidity of air entering can also be expressed as

$$\omega_1 = 0.622 \frac{p_{v_1}}{p - p_{v_1}}$$

$$\Rightarrow \quad 0.0107 = 0.622 \times \frac{p_{v_1}}{100 - p_{v_1}}$$

$$\text{or } 100 - p_{v_1} = 58.13\, p_{v_1}$$
$$\text{or } p_{v_1} = 1.69 \text{ kPa}$$

$$\text{Thus} \quad \phi_1 = \frac{p_{v_1}}{p_{sat_1}} = \frac{1.69}{4.246} = 0.398 = \mathbf{39.8\%}$$

8.6 PSYCHROMETRIC CHART

The psychrometric chart is a graphical presentation of properties of air–water vapour mixture. These charts are available in a number of different forms but the most commonly used chart is the ω–DBT chart. The typical layout of the chart is shown in Fig. 8.9. It has the following features:

1. The chart is based on the standard atmospheric pressure, i.e., 760 mm of Hg or 101.325 kPa.
2. The chart is a plot of *humidity ratio* and *vapour pressure*, on the vertical axis and *dry-bulb temperature* on the horizontal axis.
3. On the left end of the chart, the *saturation curve* represents the state of saturated air at different temperatures.
4. *Saturation curve* is a curve of 100% relative humidity. Other constant relative humidity curves also have the same shape.
5. The constant wet-bulb temperature lines have a downhill appearance to the right.
6. The constant specific-volume lines are steeper than constant wet-bulb temperature lines.
7. The constant-enthalpy lines are parallel to the constant wet-bulb temperature lines. Therefore, constant enthalpy and constant

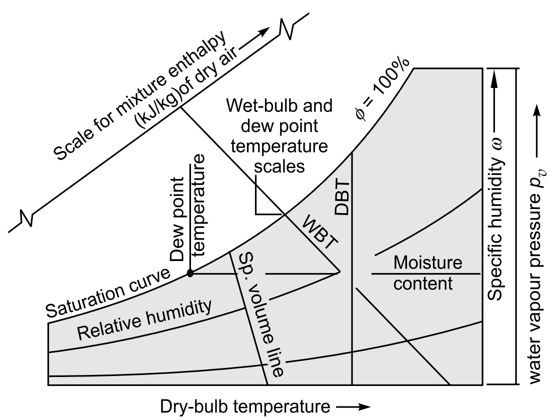


Fig. 8.9 *Psychrometric chart*

wet bulb temperature are represented on the same line.

8. As shown in Fig. 8.10, on the saturation line, the dry-bulb temperature T, wet-bulb temperature T_{wb} and dew-point temperature T_{dp} are identical.

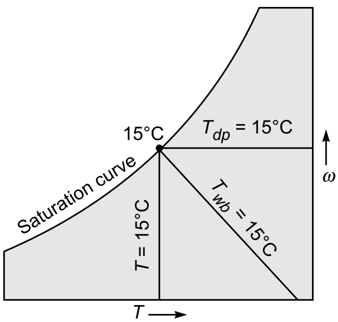


Fig. 8.10 *For saturated air, the dry-bulb, wet bulb and dew-point temperatures are identical*

The psychrometric chart is used extensively in air-conditioning applications. An ordinary cooling or heating process of air without change in its moisture content (ω = constant) is represented by the horizontal line. Any deviation from the horizontal line indicates that the moisture content of air is changed during the process.

Example 8.6 *The ambient conditions of air are 40°C DBT and 20°C WBT. Find the*

(a) Relative humidity, and

(b) Specific humidity.

Solution

Given Ambient air with

$$T = 40°C \quad \text{and} \quad T_{wb} = 20°C$$

To find

(i) Relative humidity, and

(ii) Specific humidity.

Analysis From the psychrometric chart at intersection point of 40°C DBT and 20°C WBT,

Relative humidity = **14.5%**

Specific humidity = **0.0066 kg/kg of dry air**

Example 8.7 *For the moist air at 30°C DBT and 15°C WBT, calculate*

(a) Specific humidity,

(b) Enthalpy,

(c) Relative humidity, and

(d) Dew-point temperature.

Solution

Given The moist air

$$T = 30°C \quad T_{wb} = 15°C$$

To find

(i) Specific humidity (ω),

(ii) Enthalpy (h),

(iii) Relative humidity (ϕ), and

(iv) DPT (T_{dp}).

Analysis From psychrometric chart at intersection point of T = 30°C and T_{wb} = 15°C

(i) Specific humidity ω = **0.00422 kg of dry air**

(ii) Enthalpy h = **42.2 kJ/kg of dry air**

(iii) Relative humidity ϕ = **17.5%**

(iv) Dew point temperature T_{dp} = **2.3°C**

Example 8.8 *Air at 40°C DBT and 25°C WBT is cooled down in an air-conditioning plant to 25°C DBT and 60% RH.*

Calculate the heat to be removed per kg of air if the COP of the unit is 3.5. Also, find the work required to cool 3 kg of air.

Solution

Given Moist air

State 1:	40°C DBT	and	25°C WBT
State 2:	25°C DBT	and	ϕ = 0.6
System	COP = 3.5	and	m = 3 kg

To find

(i) Heat removed per kg of air, and

(ii) Work input to system to cool 3 kg of air.

Analysis

(i) From the psychrometric chart,

At 40°C DBT and 25°C WBT, enthalpy of air;

$$h_1 = 76.5 \text{ kJ/kg of dry air}$$

At 25°C DBT and 60% RH, enthalpy of air;

$$h_2 = 55.2 \text{ kJ/kg of dry air}$$

Heat removed per kg of dry air

$$q_L = h_1 - h_2$$
$$= 76.5 - 55.2 = \mathbf{21.3\ kJ/kg}$$

(ii) The coefficient of performance of a refrigerator is given by

$$(COP)_R = \frac{RE}{W_{in}} = \frac{q_L}{w_{in}}$$

or
$$w_{in} = \frac{q_L}{(COP)_R} = \frac{21.3}{3.5} = 6.085 \text{ kJ/kg}$$

Work input for 3 kg of air

$$W_{in} = m\,w_{in} = 3 \times 6.085$$
$$= \mathbf{18.26\ kJ}$$

8.7 AIR-CONDITIONING PROCESSES

The basic thermodynamic processes of moist air are illustrated on the psychrometric chart in Fig. 8.11. These processes are

1. Sensible heating and cooling
2. Humidification and dehumidification
3. Humidification with heating/cooling
4. Dehumidification with heating/cooling
5. Mixing of two air streams.

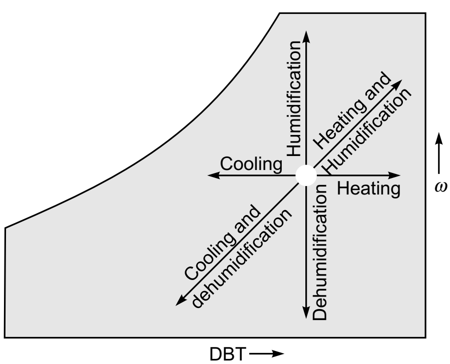
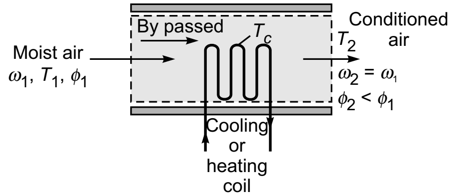


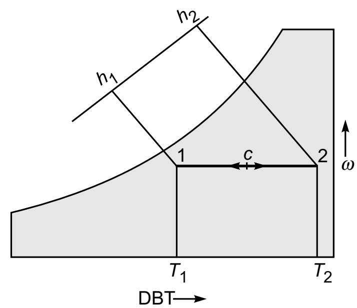
Fig. 8.11 *Basic air-conditioning processes*

8.7.1 Sensible Heating and Cooling

The sensible heating or sensible cooling process involves only change in dry bulb temperature, keeping specific humidity ω constant. Therefore, simple heating and cooling processes appear as a horizontal line on the psychrometric chart as shown in Fig. 8.12. The process 1–2 represents a sensible heating process while the process 2–1 represents a sensible cooling process.



(a) Schematic



(b) Representation on psychrometric chart

Fig. 8.12 *Sensible heating and cooling processes*

The sensible heat-transfer rate is given by

$$\dot{Q} = \dot{m}_a (h_2 - h_1)$$
$$= \dot{m}_a C_{pa} (T_2 - T_1) + \dot{m}_a \omega C_{pv} (T_2 - T_1)$$

or
$$\dot{Q} = \dot{m}_a (1.005 + 1.88\,\omega)(T_2 - T_1) \quad ...(8.19)$$

$\dot{m}_a$ = mass flow rate of dry air, kg/s

Where C_{pa} = specific heat of air, 1.005 kJ/kg·K

C_{pv} = specific heat of vapour, 1.88 kJ/kg·K

By-Pass Factor of Heating and Cooling Coils

As shown in Fig. 8.12(a), some mass of air flows over the coil, and remaining part of the air directly passes the duct without coming in contact of cooling or heating coil. Such a stream is referred as the *by-passed stream*. Therefore, the temperature of air coming out of the duct is not equal to the coil temperature. Thus, the by-pass factor (*BPF*) is defined as

$$BPF = \frac{\text{Temperature difference between coil and exit air}}{\text{Temperature difference between coil and entering air}}$$

$$= \frac{T_c - T_{exit}}{T_c - T_{inlet}} = \frac{T_c - T_2}{T_c - T_1} \quad ...(8.20)$$

where T_c is the temperature of the coil, and T_1 and T_2 are inlet and exit temperatures of air.

Efficiency of Heating and Cooling Coils

The efficiency of heating and cooling coils is the ratio between actual temperature change to the maximum possible temperature change.

For heating or cooling coil

$$\eta_h = \frac{\text{Actual temperature change}}{\text{Maximum possible temperature change}}$$

$$= \frac{T_2 - T_1}{T_c - T_1} \quad ...(8.21)$$

$$= \frac{T_2 - T_1 - T_c + T_c}{T_c - T_1}$$

$$= \frac{(T_c - T_1) - (T_c - T_2)}{T_c - T_1}$$

or $$\eta_h = 1 - \frac{(T_c - T_2)}{T_c - T_1} = 1 - BPF \quad ...(8.22)$$

Example 8.9 *Moist air enters a duct at 10 °C, 80% relative humidty, and a volumetric flow rate of 150 m³/min. The mixture is heated as it flows through the duct and leaves at 30°C. No moisture is added or removed and the mixture pressure remains approximately constant at 1 bar. For steady-state operation, determine (a) the rate of heat transfer, and (b) relative humidity at exit.*

Changes in kinetic and potential energy can be ignored and use a psychrometric chart for the solution.

Solution

Given Moist air enters a duct and is heated

At inlet: $T_1 = T_{db_1} = 10°C = 283\,K$

$\phi_1 = 80\% = 0.8$

$\dot{V} = 150\ m^3/min$

At exit: $T_2 = T_{db_2} = 30°C = 303\,K$

$\omega_1 = \omega_2$

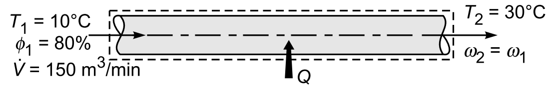
$p_1 = p_2 = 1$ bar

To find

(i) The rate of heat transfer, and

(ii) Relative humidity at the exit.

Schematic



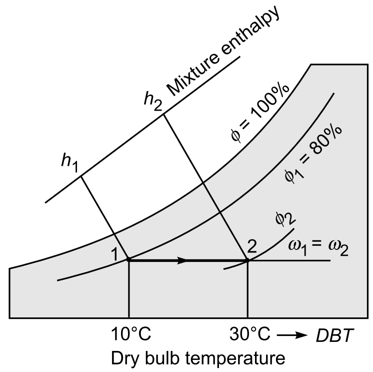


Fig. 8.13 *Sensible heating of moist air*

Assumption Specific gas constant for dry air as $R = 0.287$ kJ/kg·K

Analysis

(i) *Heat transfer rate*

From psychrometric chart

State 1: 10°C DBT and $\phi = 80\%$,

$h_1 = 25.7$ kJ/kg, $v_{g1} = 0.813\ m^3/kg$

State 2: $\omega_1 = \omega_2$ and 30°C DBT,

$h_2 = 45.9$ kJ/kg

The heat transfer per kg of dry air

$$q = h_2 - h_1$$
$$= 45.9 - 25.7 = 20.2 \text{ kJ/kg}$$

The mass-flow rate of moist air can be obtained as

$$\dot{m} = \frac{\dot{V}}{v_{g1}} = \frac{150}{0.813} = 184.5 \text{ kg/min}$$

Then the total heat transfer rate to air in the duct

$$\dot{Q} = \dot{m}\, q$$
$$= (184.5 \text{ kg/min}) \times (20.2 \text{ kJ/kg})$$
$$= \mathbf{3731.5 \text{ kJ/min}}$$

(ii) Relative humidity at exit

From psychrometric chart, at the state 2, 30° DBT, $\omega_2 = \omega_1$

$$\phi_2 = \mathbf{23.1\%}$$

Example 8.10 *A quantity of air having a volume of 300 m^3 at 30 °C DBT and 25°C WBT is heated to 40 °C DBT. Calculate the amount of heat added, and relative humidity at both the states.*

Solution

Given A quantity of air is heated sensibly;

At inlet: $T_1 = 30°C$

$T_{wb_1} = 25°C$

$V = 300 \text{ m}^3$

At exit: $T_2 = 40°C = 313$ K

$\omega_1 = \omega_2$

To find

(i) The rate of heat transfer, and

(ii) Relative humidity at inlet and exit.

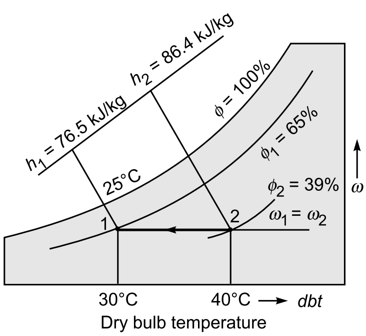


Fig. 8.14 *Sensible heating process*

Analysis From the psychrometric chart

State 1: 30°C *DBT* and 25°C *WBT*

$h_1 = 76.5$ kJ/kg

$v_g = 0.885 \text{ m}^3/\text{kg}$

$\phi_1 = \mathbf{65\%}$

State 2: $\omega_1 = \omega_2$ and 40°C *DBT*,

$h_2 = 86.4$ kJ/kg

$\phi_1 = \mathbf{39\%}$

Mass of moist air

$$m = \frac{V}{v_g} = \frac{300 \text{ m}^3}{0.885 \text{ m}^3/\text{kg}} = 339 \text{ kg}$$

The heat transfer to air

$$Q = m(h_2 - h_1)$$
$$= 339 \times (86.4 - 76.5) = \mathbf{3356 \text{ kJ}}$$

Example 8.11 *Air is cooled from 39°C DBT and 29% RH to 24°C at the rate of 5 m^3/s. Calculate the capacity of the cooling coil if the surface of the cooling coil is 20 °C. Also, calculate the by pass factor.*

Solution

Given A quantity of air is cooled sensibly;

At inlet: $T_1 = 39°C$

$\phi_1 = 29\%$

$\dot{V} = 5 \text{ m}^3/\text{s}$

At exit: $T_2 = 24°C$

$\omega_1 = \omega_2$

$T_c = 20°C$

To find

(i) Cooling capacity, and

(ii) By-pass factor.

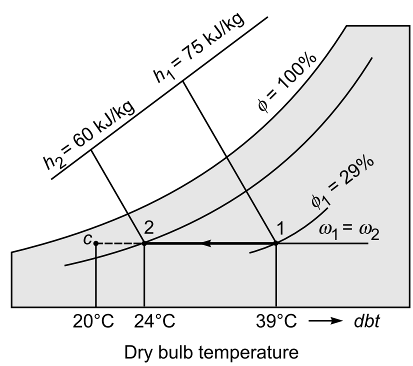


Fig. 8.15 *Sensible cooling process*

Analysis From the psychrometric chart

State 1: 39°C *DBT* and $\phi_1 = 29\%$

$h_1 = 75$ kJ/kg

$v_g = 0.9 \text{ m}^3/\text{kg}$

State 2: $\omega_1 = \omega_2$ and 24°C *DBT*

$h_2 = 60$ kJ/kg

Mass flow rate of air,

$$\dot{m}_a = \frac{\dot{V}}{v_g} = \frac{5 \text{ m}^3/\text{kg}}{0.9 \text{ m}^3/\text{kg}} = 5.56 \text{ kg/s}$$

(i) Capacity of cooling coil

$$\dot{Q} = \dot{m}_a (h_1 - h_2)$$
$$= 5.56 \times (75 - 60)$$
$$= \mathbf{83.4 \text{ kW} \quad \text{or} \quad 23.83 \text{ TR}}$$

(ii) By-pass factor

$$BPF = \frac{T_c - T_{exit}}{T_c - T_{inlet}} = \frac{T_c - T_2}{T_c - T_1}$$
$$= \frac{20 - 24}{20 - 39} = \mathbf{0.21}$$

Example 8.12 *Atmospheric air at 760 mm of Hg, 15 °C DBT and 11°C WBT enters a heating coil, whose temperature is 41°C. The by-pass factor of the coil is 0.5. Calculate DBT, WBT and the relative humidity of air leaving the coil. Also, calculate the sensible heat added to air per kg of dry air.*

Solution

Given A quantity of air is heated sensibly;

At inlet: $T_1 = 15°\text{C}$

$T_{wb_1} = 11°\text{C}$

$p_1 = 760$ mm of Hg = 101.3 kPa

Heating coil $T_c = 41°\text{C}$

BPF = 0.5

and $\omega_1 = \omega_2$

To find

(i) T_2, T_{wb_2}, ϕ_2 of air leaving the coil,

(ii) The rate of heat transfer.

Analysis The sensible heating process is shown schematically and on a psychrometric chart in Fig. 8.16.

(i) T_2, T_{wb_2}, ϕ_2 of air leaving the coil;

From psychrometric chart

State 1: 15°C *DBT* and 11°C *WBT*

$h_1 = 31.8$ kJ/kg and $\phi_1 = 62\%$

The by-pass factor is given by Eq. (8.20)

$$BPF = \frac{T_c - T_{exit}}{T_c - T_{inlet}} = \frac{T_c - T_2}{T_c - T_1}$$

$$0.5 = \frac{41 - T_2}{41 - 15}$$

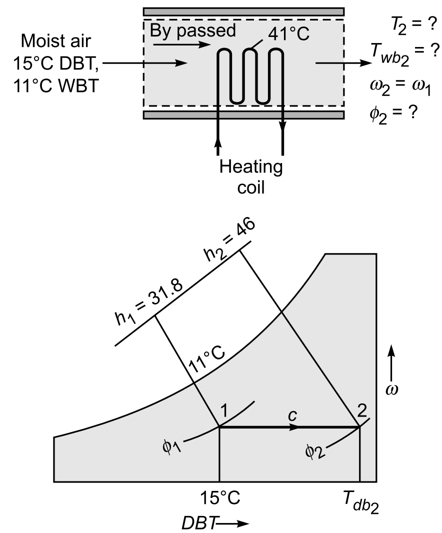


Fig. 8.16

$\Rightarrow \quad T_2 = \mathbf{28°C}$

State 2: Using psychrometric chart at coordinates of $\omega_1 = \omega_2$ and 28°C *DBT*, and wet bulb temperature and relative humidity are

$T_{wb_2} = \mathbf{16°C} \quad \phi_2 = \mathbf{29\%}$

and $h_2 = 46$ kJ/kg

Sensible heat transferred to 1 kg air

$$q = h_2 - h_1$$
$$= 46 - 31.8 = \mathbf{14.2 \text{ kJ/kg}}$$

8.7.2 Humidification and Dehumidification

When the moisture content of air is increased at constant temperature, the process is referred as *humidification* [process (1 – 2)], and when moisture is reduced from the moist air at constant temperature the process is referred as *dehumidification* [process (2 – 1)].

During humidification and dehumidification, the specific humidity ω and relative humidity ϕ, both change. The processes are shown in Fig. 8.17.

Actually, there is no method by which one can obtain simply humidification or dehumidification of air. These processes are accomplished by either simultaneous heating or cooling.

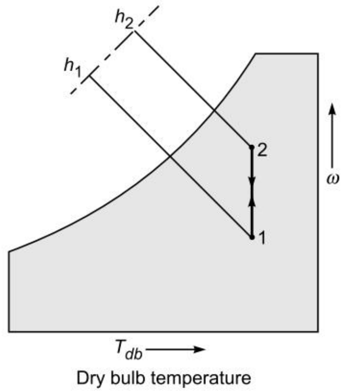


Fig. 8.17 *Humidification and dehumidification*

8.7.3 Heating and Humidification

This process is generally used in winter air-conditioning. It involves warming and humidifying of air simultaneously. In this process, the air passes through a humidifier where the hot water (or steam) is injected, both the specific humidity and dry-bulb temperature increase. The process of heating and humidification is shown by the line 1–2 on the psychometric chart in Fig. 8.18.

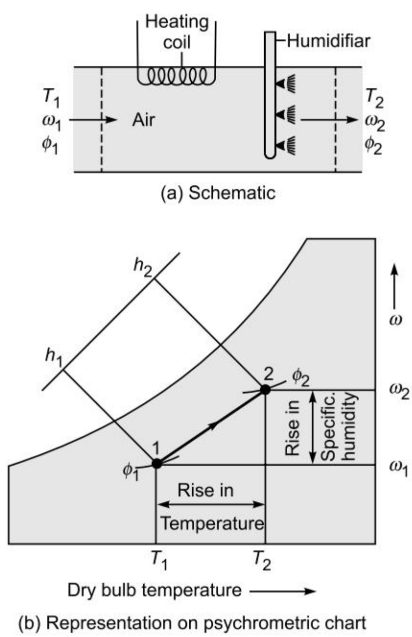


Fig. 8.18 *Heating with humidification*

Example 8.13 *Moist air at 22°C and a wet-bulb temperature of 9°C enters a steam spray humidifier. The mass flow rate of the dry air is 90 kg/min. Saturated water vapour at 110°C is injected into the mixture at a rate of 52 kg/h. There is no heat transfer with the surroundings and pressure is constant throughout at 1 bar. Determine (a) the humidity ratio, and (b) temperature of air leaving the humidifier.*

Solution

Given Humidification of moist air with the help of steam injection

$T_1 = 22°\text{C}$ $\quad$ $T_{wb_1} = 9°\text{C}$

$\dot{m}_a = 90$ kg/min $\quad$ $T_{sat} = 110°\text{C}$

$\dot{m}_s = 52$ kg/h $= 0.867$ kg/min

$p = 1$ bar $= 100$ kPa

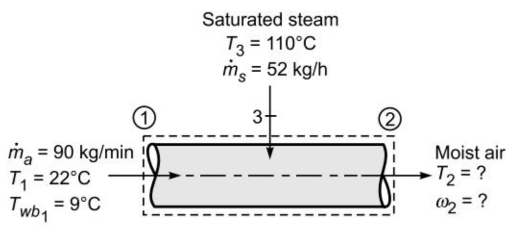


Fig. 8.19 *Schematic*

To find

(i) The specific humidity of air leaving the humidifier,

(ii) Exit temperature of moist air.

Analysis

(i) *The humidity ratio at the exit*

The mass flow rate balance during the flow through control volume between states 1–2

$$\dot{m}_{a_1} = \dot{m}_{a_2} \qquad \text{(dry air)}$$

$$\dot{m}_{v_1} + \dot{m}_s = \dot{m}_{v_2} \qquad \text{(water vapour)}$$

Dividing both sides by $\dot{m}_a$

$$\frac{\dot{m}_{v_1}}{\dot{m}_{a_1}} + \frac{\dot{m}_s}{\dot{m}_a} = \frac{\dot{m}_{v_2}}{\dot{m}_a}$$

or
$$\omega_2 = \omega_1 + \frac{\dot{m}_s}{\dot{m}_a}$$

From pychrometric chart against coordinates of 22°C *DBT* and 9°C *WBT*

$\omega_1 = 0.002$ kg/kg of dry air

$h_1 = 27.2$ kJ/kg of dry air

Thus, the specific humidity at the exit

$$\omega_2 = 0.002 + \frac{(0.867 \text{ kg/min})}{(90 \text{ kg/min})}$$

$$= \mathbf{0.0116 \text{ kg/kg of dry air}}$$

(ii) The exit temperature of moist air

From steam table,

$$h_s = h_g \text{ @ } 110°\text{C} = 2691.47 \text{ kJ/kg}$$

The enthalpy of leaving air

$$h_2 = h_1 + \frac{\dot{m}_s}{\dot{m}_a} h_s$$

$$= (27.2 \text{ kJ/kg}) + \frac{0.867 \text{ kg/min}}{90 \text{ kg/min}} \times (2691.47 \text{ kJ/kg})$$

or $h_2 = 53.11$ kJ/kg of dry air

Using psychrometric chart at coordinates of h_2 = 53.11 kJ/kg and ω_2 = 0.0116 kg/kg, the DBT and relative humidity are

$$T_2 = \mathbf{24°C}$$

$$\phi_2 = 61\%$$

Example 8.14 *The atmospheric air at 25°C DBT and 12°C WBT is flowing at a rate of 100 m³/min through a duct. The dry saturated steam at 100°C is injected into the air stream at a rate of 72 kg/h. Calculate the specific humidity, DBT, WBT, relative humidity and enthalpy of air leaving the duct.*

Solution

Given Humidification of atmospheric air with the help of steam injection

T_1 = 25°C $\quad T_{wb_1}$ = 12°C

$\dot{V}_a$ = 100 m³/min $\quad T_{sat}$ = 100°C

$\dot{m}_s$ = 72 kg/h = 1.2 kg/min

To find

(i) The specific humidity, DBT, WBT, ϕ_2 of air leaving the humidifier, and

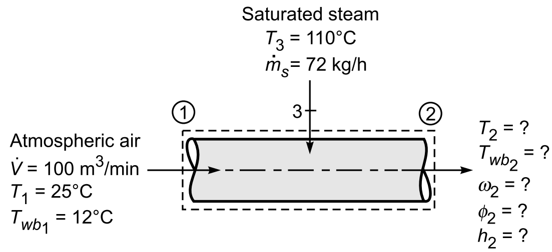
(ii) Specific enthalpy of air leaving.

Analysis From pychrometric chart against coordinate of 25°C *DBT* and 12°C *WBT*

ω_1 = 0.0034 kg/kg of dry air

h_1 = 33.2 kJ/kg of dry air

v_1 = 0.844 m³/kg



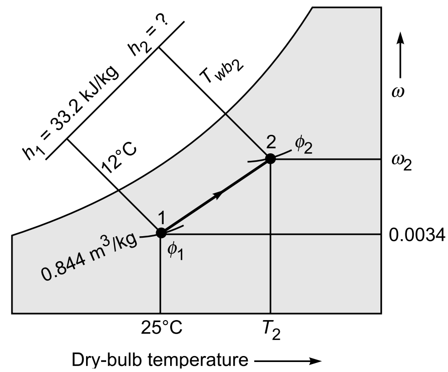


Fig. 8.20 *Process on psychrometric chart*

The mass flow rate of atmospheric air

$$\dot{m}_a = \frac{\dot{V}_a}{v_1} = \frac{100 \text{ m}^3/\text{min}}{0.844 \text{ m}^3/\text{kg}} = 118.48 \text{ kg/min}$$

(i) The specific humidity at the exit

$$\omega_2 = \omega_1 + \frac{\dot{m}_s}{\dot{m}_a} = 0.0034 + \frac{1.2}{118.48}$$

$$= \mathbf{0.0135 \text{ kg/kg of dry air}}$$

(ii) The exit temperature of moist air

From steam tables

$$h_s = h_g \text{ @ } 100°\text{C} = 2676 \text{ kJ/kg}$$

The enthalpy of leaving air

$$h_2 = h_1 + \frac{\dot{m}_s}{\dot{m}_a} h_s$$

$$= 33.2 + \frac{1.2}{118.48} \times 2676$$

$$= \mathbf{60.3 \text{ kJ/kg of dry air}}$$

Using psychrometric chart at coordinates of h_2 = 60.3 kJ/kg and ω_2 = 0.0135 kg/kg, and locate point 2. The properties of moist air at state 2 are

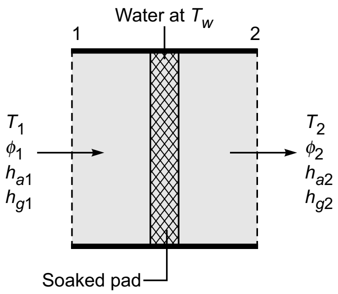
DBT $\quad T_2$ = **26°C**

WBT $\quad T_{wb_2}$ = **21°C**

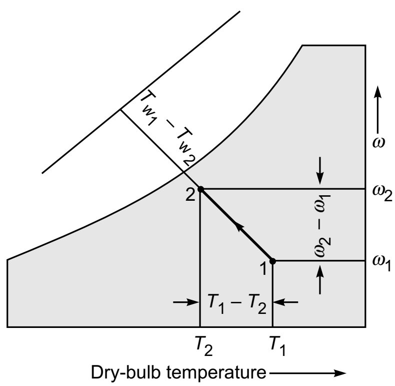
RH $\quad \phi_2$ = **65%**

8.7.4 Cooling and Humidification: Evaporative Cooling

Cooling and humidification, simultaneously can be accomplished by evaporative cooling. It involves either spraying of liquid water into air or forcing the air through a soaked pad that is kept saturated with water as in desert coolers, also known as evaporative coolers. A schematic of an evaporative cooler is shown in Fig. 8.21(a).



(a) Evaporative cooler



(b) Representation of procession psychrometric chart

Fig. 8.21 *Cooling and humidification process*

The low humidity air enters the humidifier at the state 1, and when it passes through a water soaked pad a part of water evaporates and gets mixed with air. The heat energy for evaporation is extracted from the water body and air stream, thus both water and air get cooled. Therefore, the stream of air leaving the humidifier at the state 2 is rich in moisture and low in temperature.

For negligible heat transfer to surroundings and in absence of work transfer, kinetic and potential energy losses, the steady flow energy equation yields to

$$(h_{a_2} + \omega_2 h_{g_2}) = (\omega_2 - \omega_1) h_{f_1} + (h_{a_1} + \omega_1 h_{g_1}) \quad ...(8.23)$$

where h_{f_1} is the specific enthalpy of the water entering the control volume.

The evaporative cooling follows a line of constant wet-bulb temperature on the psychrometric chart as shown in Fig. 8.21(b). Thus

$$T_{wb} = \text{constant} \quad ...(8.24)$$

$$h \approx \text{constant} \quad ...(8.25)$$

The evaporative cooling (humidification and cooling) takes place in cooling towers, evaporative condensers and desert coolers.

Example 8.15 *Air at 37°C and 10% relative humidity enters an evaporative cooler with a volumetric flow rate of 140 m^3/min. Moist air leaves the cooler at 20°C. The water is added to the soaked pad of the cooler at 20°C and evaporates fully into the moist air. There is no heat transfer with the surroundings and the pressure is constant throughout the process at 1 atm. Determine (a) the mass flow rate of water to the soaked pad, and (b) relative humidity of the moist air at the exit of the evaporative cooler.*

Solution

Given Humidification of moist air with the help of evaporative cooling.

$T_1 = 37°C \qquad \phi_1 = 10\% = 0.1$

$\dot{V} = 140 \text{ m}^3/\text{min} \qquad T_2 = 20°C$

$T_3 = 20°C \qquad p = 1 \text{ atm} = 101.3 \text{ kPa}$

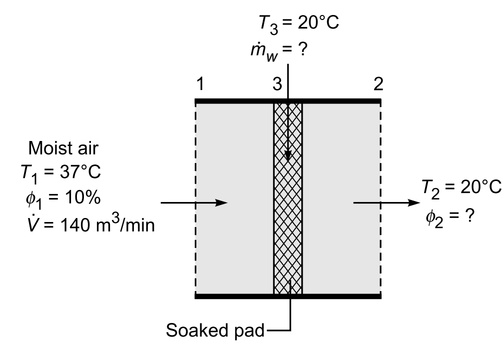


Fig. 8.22

To find

(i) The mass flow rate of water to the soaked pad.

(ii) The relative humidity of moist air leaving the evaporative cooler.

Assumptions

(i) Specific gas constant of air, $R = 0.287$ kJ/kg·K.

(ii) Specific heat at constant pressure of air, $C_p = 1.005$ kJ/kg·K.

Analysis

(i) The mass flow rate of water to the soaked pad

The mass conservation gives

$$\begin{aligned}\dot{m}_w &= \dot{m}_{v_2} - \dot{m}_{v_1}\\ &= \dot{m}_a(\omega_2 - \omega_1) \qquad \text{...(i)}\end{aligned}$$

From psychrometric chart at 37°C and 10% RH, the specific humidity at state 1

$$\omega_1 = 0.00388 \text{ kg/kg of dry air}$$

The mass flow rate of dry air at the state 1

$$\dot{m}_a = \frac{p\dot{V}}{RT} = \frac{101.3\times 140}{0.287\times 310} = 159.4 \text{ kg}$$

The specific humidity ω_2 can be obtained by using steady-flow energy Eq. (8.23)

$$h_{a_2} + \omega_2 h_{g_2} = (\omega_2 - \omega_1)h_{f_1} + h_{a_1} + \omega_1 h_{g_1}$$

or $$\omega_2(h_{g_2} - h_{f_1}) = h_{a_1} - h_{a_2} + \omega_1(h_{g_1} - h_{f_1})$$

or $$\omega_2 = \frac{C_p(T_1 - T_2) + \omega_1(h_{g_1} - h_{f_1})}{h_{g_2} - h_{f_1}} \qquad \text{...(ii)}$$

From steam table

$$\begin{aligned}h_{g_1} &= h_g @ 37°\text{C} = 2568.9 \text{ kJ/kg}\\ h_{g_2} &= h_g @ 20°\text{C} = 2538.1 \text{ kJ/kg}\\ h_{f_1} &= h_f @ 20°\text{C} = 83.96 \text{ kJ/kg}\end{aligned}$$

Thus from Eq. (ii)

$$\omega_2 = \frac{1.005\times(37-20) + 0.00388\times(2568.9 - 83.96)}{2538.1 - 83.96}$$

$$= 0.0109 \text{ kg/kg of dry air}$$

Substituting in Eq. (i)

$$\begin{aligned}\dot{m}_w &= 159.4\times(0.0109 - 0.00388)\\ &= \mathbf{1.12\ kg/min}\end{aligned}$$

(ii) The relative humidity of the moist air

The partial pressure of water vapour in air

$$\omega_2 = 0.622\frac{p_{v_2}}{p - p_{v_2}}$$

or $$p_{v_2} = \frac{\omega_2 p}{\omega_2 + 0.622} = \frac{0.0109\times 101.3}{0.0109 + 0.622}$$

$$= 1.744 \text{ kPa}$$

At 20°C, the saturation pressure of vapour,

$$p_{sat_2} = 0.02339 \text{ bar} = 2.339 \text{ kPa}$$

Thus $$\phi_2 = \frac{p_{v_2}}{p_{sat_2}} = \frac{1.744}{2.339} = 0.745 = \mathbf{74.5\%}$$

Alternatively, the relative humidity ϕ_2 can be obtained from psychrometric chart with $\omega_2 = 0.0109$ and $T_2 = 20$°C, the intersecting point gives $\phi_2 = 75\%$.

8.7.5 Dehumidification with Heating

It is based on the principle of adsorption, i.e., capillary action. Thermodynamically, an adsorption process is the nerve of the *adiabatic saturation process*. In this process, the air is passed over the adsorbing surface (chemicals like silica gel, activited alumina, etc. which have affinity for moisture). As the water vapour comes in contact of these substances, the moisture gets condensed out of air and gives up its latent heat, resulting into rise in temperature of air. The dehumidification and heating process is shown in Fig. 8.23.

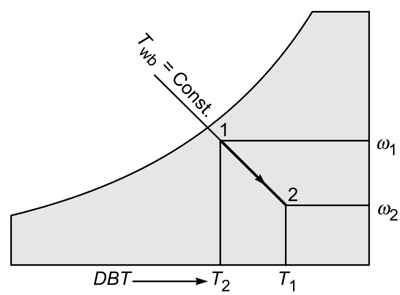
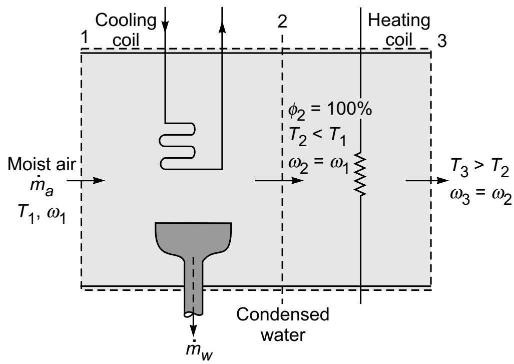


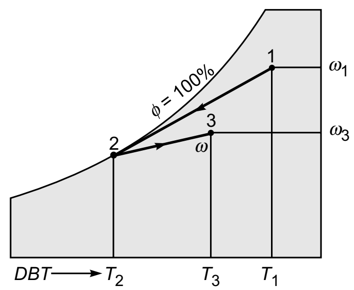
Fig. 8.23 *Adsorption dehumidification process*

8.7.6 Dehumidification with Cooling

When a moist air stream is cooled at constant pressure to a temperature below its dew point temperature, some of the water vapour initially present would condense. Fig. 8.24 shows the schematic of a dehumidifier and cooling process on psychrometric chart.



(a) schematic of dehumidifier



(b) Representation of dehumidification and cooling on psychrometic chart

Fig. 8.24

The moist air enters the state 1, flows across a cooling coil, some of the water vapour initially present in the moist air condenses and a saturated moist air mixture leaves the dehumidifier section at the state 2. Since the moist air leaving the cooling coil is saturated at temperature below the temperature of the moist air entering, the moist air stream might be unsuitable for direct use in occupied space. Thus this stream passes the heating section and is brough to the comfort condition at the state 3.

Example 8.16 *Air at 40°C DBT and 60% RH is cooled to 25°C DBT. It is achieved by cooling and dehumidification. Air flow rate is 40 m³/min. Using psychrometric chart, calculate:*

(a) the dew point temperature,

(b) mass of water drained out per hour,

(c) Capacity of cooling coil.

If the apparatus dew-point temperature (ADPT) is 20°C, find the by pass factor of coil.

Solution

Given Dehumidification and cooling of moist air:

T_1 = 40°C $\quad\quad$ ϕ_1 = 60%

T_2 = 25°C $\quad\quad$ $\dot{V}$ = 40 m³/min

$T_{ADP} = T_c$ = 20°C

To find

(i) Dew-point temperature,

(ii) Mass of water drained out per hour,

(iii) Capacity of cooling coil, and

(iv) By pass factor of coil.

Analysis The process of cooling and dehumidification is shown on psychrometric chart in Fig. 8.25.

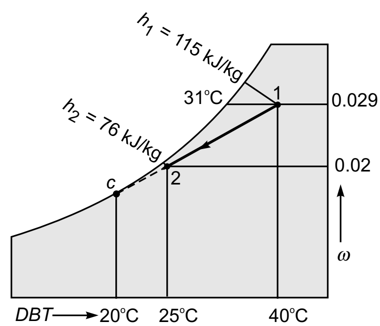


Fig. 8.25

Inlet conditions: ϕ_1 = 60%, T_1 = 40°C

From psychrometric chart;

v_1 = 0.925 m³/kg

ω_1 = 0.029 kg/kg of air

h_1 = 115 kJ/kg

Exit conditions: T_2 = 25°C

T_c = 20°C

ω_2 = 0.02 kg/kg of air

h_2 = 76 kJ/kg

(i) Dew-point temperature corresponding to the state 1 is **31°C**.

The mass flow rate of dry air,

$$\dot{m}_a = \frac{\dot{V}}{v_{a_1}} = \frac{40}{0.925}$$

= 43.24 kg/min or 2595.6 kg/h

(ii) Mass of moisture removed

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$$

= 2595.6 × (0.029 – 0.02)

= **23.36 kg/h**

(iii) Capacity of cooling coil

The heat transfer;

$$\dot{Q} = \dot{m}_a(h_1 - h_2) = 43.24 \times (115 - 76)$$
$$= 1686.36 \text{ kJ/min}$$

$$\text{Coil capacity} = \frac{\dot{Q}\text{ (kJ/min)}}{(211\text{ kJ/min/Ton})} = \frac{1686.36}{211}$$
$$= \mathbf{8.00\ Ton}$$

(iv) By pass factor

The by-pass factor is given by Eq. (8.33)

$$BPF = \frac{T_c - T_{exit}}{T_c - T_{inlet}} = \frac{T_c - T_2}{T_c - T_1}$$
$$= \frac{20 - 25}{20 - 40} = \mathbf{0.25}$$

Example 8.17 *Air enters a window air conditioner at 1 atm, 30°C and 80% RH at a rate of 10 m³/min and it leaves as saturated at 14°C. A part of moisture, which condenses during the process is also removed at 14°C. Determine the rate of heat and moisture removal from air.*

Solution

Given Dehumidification and cooling of moist air:

$p_1 = 1$ atm $\quad T_1 = 30°C$
$\phi_1 = 80\% \quad \dot{V} = 10\ m^3/min$
$T_2 = 14°C \quad \phi_2 = 100\%$

To find

(i) Rate of moisture removal,
(ii) Rate of heat transfer.

Analysis The process of cooling and dehumidification is shown on the psychrometric chart in Fig. 8.26.

Inlet conditions: $\phi_1 = 80\% \quad T_1 = 30°C$

From psychrometric chart;

$$v_1 = 0.86\ m^3/kg$$
$$\omega_1 = 0.022 \text{ kg/kg of air}$$
$$h_1 = 85.5 \text{ kJ/kg}$$

Exit conditions: $T_2 = 14°C \quad \phi_2 = 100\%$

$$\omega_2 = 0.0102 \text{ kg/kg of air}$$
$$h_2 = 39.5 \text{ kJ/kg}$$

The mass flow rate of dry air

$$\dot{m}_a = \frac{\dot{V}}{v_{a_1}} = \frac{10}{0.86} = 11.63 \text{ kg/min}$$

(i) Removal rate of moisture from air

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$$
$$= 11.63 \times (0.022 - 0.0102)$$
$$= 0.1372 \text{ kg/min or } \mathbf{8.23\ kg/h}$$

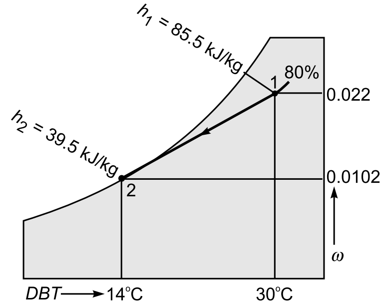


Fig. 8.26

(ii) Heat removal rate

$$\dot{Q} = \dot{m}_a(h_1 - h_2)$$
$$= 11.63 \times (85.5 - 39.5)$$
$$= 534.98 \text{ kJ/min or } \mathbf{8.916\ kW}$$

Example 8.18 *Moist air at 30°C and 50% relative humidity enters a dehumidifier operating at steady state with a volumetric flow rate of 280 m³/min. The moist air passes over a cooling coil and water vapour condenses. The condensate leaves the dehumidifier saturated at 10°C. Saturated moist air leaves in a separate stream at the same temperature. During the process, the pressure remains constant at 1.013 bar and heat transfer to the surroundings is negligble. Determine (a) the mass flow rate of dry air, (b) rate of water condensation in the dehumidifier in kg/kg of dry air flowing through the control volume, and (c) the required refrigerating capacity in tonnes.*

Solution

Given Dehumidification of moist air over a cooling coil.

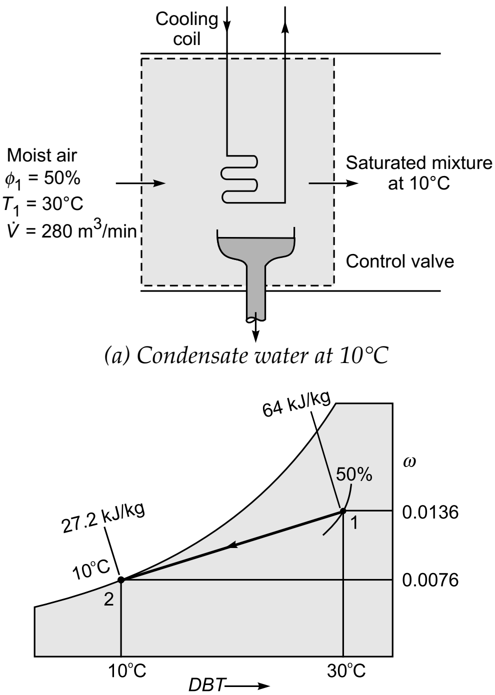
$\dot{V} = 280\ m^3/min \quad T_1 = 30°C = 303$ K
$\phi_1 = 50\% = 0.5 \quad T_2 = 10°C$
$p = 1.013$ bar $= 101.3$ kPa

To find

(i) The mass flow rate of dry air,
(ii) The rate of condensation of water vapour in dehumidifier in kg/kg of dry air,
(iii) Required refrigerating capacity in tonnes.

Assumptions

(i) The control volume is shown in Fig. 8.27.
(ii) Moist air leaves the cooling coil as saturated at 10°C.
(iii) Specific gas constant for dry air as $R = 0.287$ kJ/kg·K.



(a) Condensate water at 10°C

(b) Process on psychrometric chart

Fig. 8.27

Analysis

(i) The mass-flow rate of dry air

From psychrometric chart

The inlet conditions: p_1 = 101.3 kPa

ϕ_1 = 50% T_1 = 303 K

ω_1 = 0.0136 kg/kg of air

v_{g_1} = 0.875 m³/kg

h_1 = 64 kJ/kg

Exit conditions

T_2 = 10°C, ϕ_2 = 100%

h_2 = 27.2 kJ/kg

The mass flow rate of dry air

$$\dot{m}_a = \frac{\dot{V}}{v_{a_1}} = \frac{280}{0.875} = \mathbf{320\ kg/min}$$

(ii) Rate of moisture removal from air

$$\frac{\dot{m}_w}{\dot{m}_a} = (\omega_1 - \omega_2) = 0.0136 - 0.0076$$

$$= 0.006 \text{ kg/kg of dry air}$$

(iii) Refrigeration effect

Heat removal rate

$$\dot{Q} = \dot{m}_a(h_1 - h_2)$$

$$= 320 \times (64 - 27.2) = 11776 \text{ kJ/min}$$

The refrigeration effect in Tons of refrigeration (1 TR = 211 kJ/min)

$$TR = \frac{11776}{211} = \mathbf{55.81\ Tons}$$

8.8 AIR-CONDITIONING SYSTEMS

An air-conditioning system may provide heating, cooling or both. An air-conditioning system controls the condition of a confined space to produce a comfort effect over occupants. The size and complexity may range from a window unit for a small room to a large system for a building complex, but the basic principle is same. The air-conditioning systems can be classified into various types based on the following heads.

1. According to Season

(a) Summer air-conditioning system
(b) Winter air-conditioning system
(c) Year-round air-conditioning

2. According to Purpose

(a) Comfort air-conditioning system
(b) Industrial air-conditioning system

3. According to Arrangement of Equipment

(a) Unitary air-conditioning system
(b) Central air-conditioning system

8.9 AIR-CONDITIONING CYCLE

An air-conditioning cycle is shown in Fig. 8.28, consists of fans, filters, a refrigerating plant, ducting and control units. The working of an air-conditioning cycle can be divided into air circuit and refrigeration circuit separately.

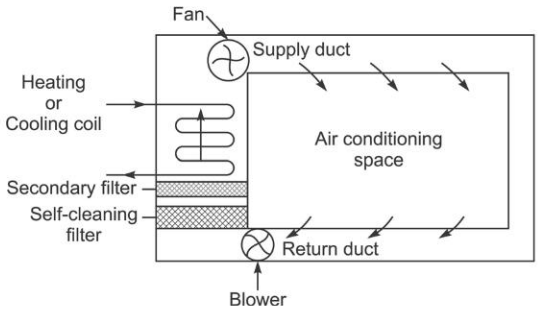


Fig. 8.28 *Air-conditioning cycle*

8.9.1 Air Circuit

The air circuit involves two air-flow ducts as

1. A supply duct for circulation of treated air inside the house
2. A return duct for collection of air from the house

The fan forces the conditioned air into the supply duct, which is connected to an opening in the room. The duct directs the air to the room through its outlets. The air enters the room and creates either a heating or cooling effect as required.

After use, the air is drawn by a blower from the room through the return duct and passes over the self-cleaning filter and secondary filter (electrostatic or fabric type) to remove the dust particles, and then passes over the evaporator (or heating coil).

The hot air comes in contact of the cooled fins and tubes of the evaporator, gets cooled, and the refrigerant in the evaporator evaporates. The clean cooled air is then again sent to the air-conditioned space through the supply ducts for the next cycle.

8.9.2 Refrigeration Circuit

The refrigeration cycle operates on vapour compression cycle and it circulates the refrigerant between the inside and outside units.The refrigerant vapour coming out of the evaporator enters the compressor, where it is compressed and delivered to the condenser. Another blower forces the atmospheric air over the condenser; thus the refrigerant is cooled and condensed. The hot air is exhausted outside and the condensed refrigerant passes through the expansion device to the evaporator.

8.10 SUMMER AIR-CONDITIONING SYSTEM

In summer, the comfort conditions are achieved by cooling and dehumidification of air. The dry bulb temperature of air entering the conditioner is quite high and thus it is essentially cooled by using cooling coils. The hot air also has high moisture contents; therefore, the removal of moisture is achieved by a dehumidifier.

A summer conditioning system is shown in Fig. 8.29. The fresh air from the surrounding atmosphere is mixed with re-circulated air in suitable proportion at state 1 and then the mixture is passed through the filter to remove dust, dirt and other unwanted materials from air to enter the system. The air is then cooled sensibly with the help of a cooling coil during process $1-x$. This cooled air is passed through a perforated membrane to remove the moisture from air (process x-y). The restricted passage of the perforated membrane condenses the water vapour into liquid water and is collected in

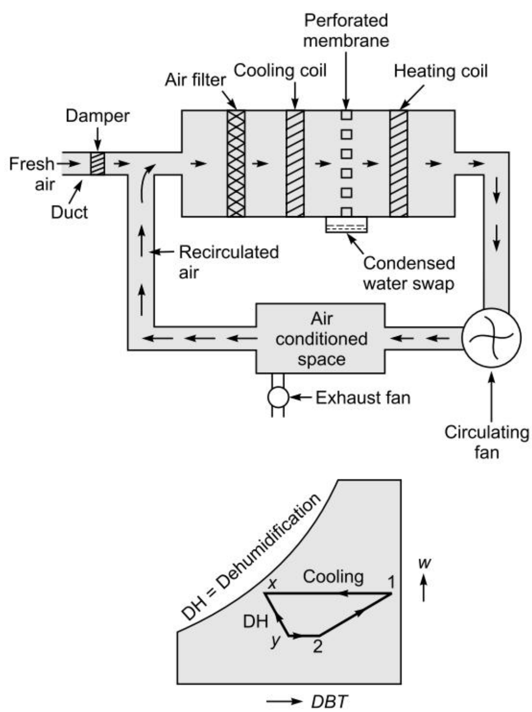


Fig. 8.29 *Summer air-conditioning system and its processes on psychrometric plot*

the swap. The air is then passed over heating coil for mild heating process $y - 2$ before entering into conditioned space by using a circulating fan.

The conditioned air provides required comfort conditions within confined space. A portion of air is exhausted to the atmosphere and the remaining air is used for recirculation with fresh air for the next cycle.

The processes of cooling, dehumidification and heating are shown on the psychrometic plot in Fig. 8.29(b). The process 2-1 is recirculation of air through return duct.

8.11 WINTER AIR-CONDITIONING SYSTEM

In winter, comfort conditions are achieved by heating and humidification of air. The dry-bulb temperature of air in the winter is quite low, and thus air is heated by using heating coils. The cold air is usually dry and has very low moisture content; therefore, moisure is added to improve relative humidity.

A winter air-conditioning system is shown in Fig. 8.30. The fresh air from the surrounding atmosphere is mixed with re-circulated air in suitable proportion at state 1 and then the mixture is passed through the filters to remove dust, dirt and other unwanted materials from air about to enter the system. The air is then passed over a preheater (process $1 - x$), which heats the air sensibly.

The preheated air at state x is then passed through the humidifier (process x-y) in which the required moisture is added to improve relative humidity of air. During flow of air through the humidifier, the air temperature gets reduced; therefore, the air is again heated to suitable temperature with the help of re-heater (process $y - 2$).

The air is then passed to the air-conditioned space through the ducting system to provide required comfort conditions within the confined space. A portion of the air is exhausted to the atmosphere using a suitable ventilator or exhaust fan and the remaining air is used for recirculation with fresh air for the next cycle.

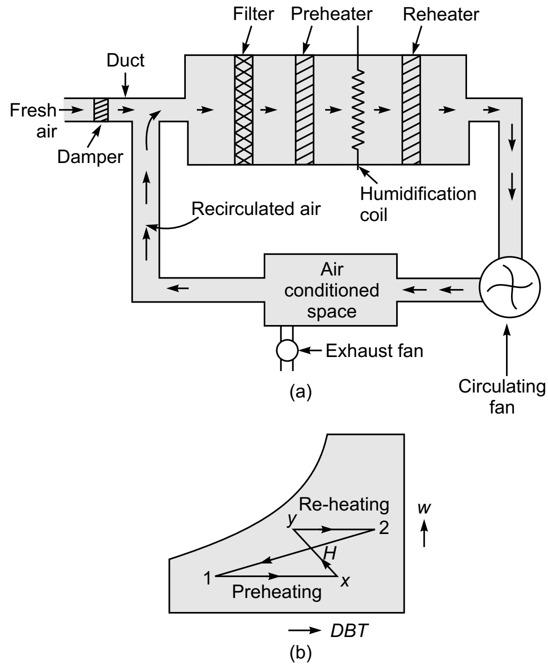


Fig. 8.30 *Winter air-conditioning system along with its processes on a psychrometric plot*

The processes of preheating, humidification and reheating are shown on the psychrometic plot in Fig. 8.29(b).

8.12 YEAR-ROUND AIR-CONDITIONING SYSTEM

In the tropical countries, the air-conditioning system design is primarily concerned with year-round cooling and humidification/dehumidification.

Thus, the year-round air-conditioning system is the combination of both summer and winter air-conditioning systems. The essential components of a year round air-conditioning system are shown in Fig. 8.31. The air-conditioner has a cooling coil, humidifier and heating coil.

The fresh air taken from the surrounding atmosphere is mixed with re-circulated air in suitable proportion and then the mixture is passed through the air filters to remove dust, dirt and other unwanted materials from air about to enter the system.

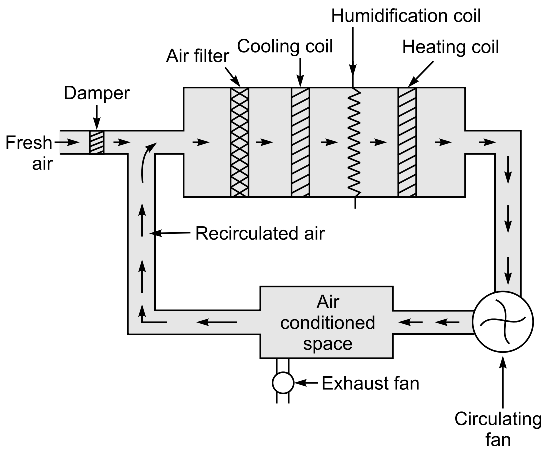


Fig. 8.31 *Year-round air-conditioning system*

For summer air-conditioning, the humidifier and heating coil are made inactive. The air is passed over a cooling coil. The temperature of the cooling coil is slightly kept lower than a summer air-conditioning system in order to allow condensation of some water particles from air to lower its relative humidity.

During the winter season, the heating coil and humidifier are made active. The heating coil heats the air and the humidifier adds the humidity to air, if it is dry. Thus, the year-round air-conditioning system can be used in any kind of ambient condition.

Types of Air-Conditioners

Air-conditioning systems may be classified as

(a) Unitary systems, and
(b) Central station systems.

8.13 UNITARY SYSTEM

The unitary systems are usually factory-assembled units including internal wiring, controls and piping and are available in wide ranges capacities. These units are installed within or near the conditioned space. These systems offer the advantage of ease of selection, low initial cost, ease of installation and removal. These systems are available as

1. Self-contained or single-package unit (window air-conditioner)
2. Split system or split-package unit
3. Roof-top air-conditioner

8.13.1 Window Air-Conditioner

It is a simple type of air-conditioner unit made as an enclosed assembly as shown in Fig. 8.32. It is designed as a unit for mounting in a window or through a wall. The function of a window-mounted air-conditioner is to provide comfort to the occupants in room by circulating clean, cool

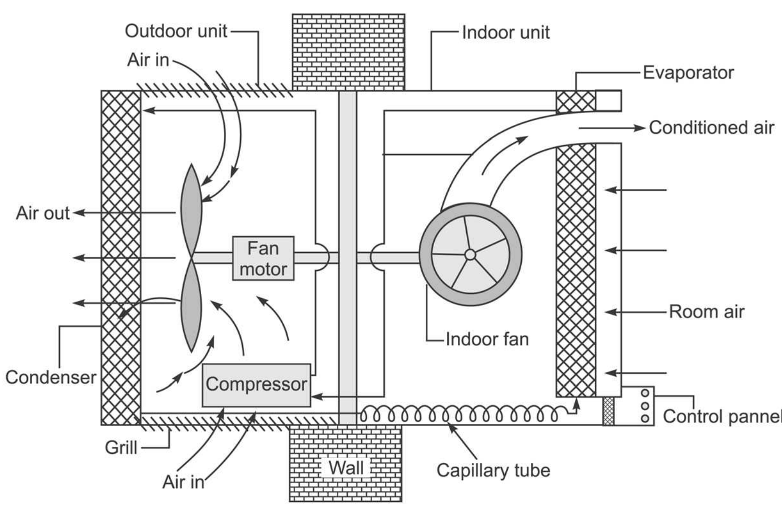


Fig. 8.32 *Window air-conditioner*

air. Their capacity is such that one unit is adequate to condition one room. Roughly 0.08 to 0.1 TR capacity is required for cooling of 1 m^2 room area. It does not require ducts for the free delivery of conditioned air to a space. The unit is divided into two parts:

(a) Indoor part
(b) Outdoor part

The indoor part includes a filter, evaporator, a motor-driven fan or a blower and an expansion device. The outdoor portion includes a hermetical sealed motor-driven compressor unit, a condenser and a fan.

A fan is used to force the outside air over the condenser coil to remove the heat from the compressed refrigerant. In order to draw air through the filter and force it over the evaporator coil to cool the atmospheric air, another fan is provided in the indoor portion. For driving the two fans, either the same motor or separate motors can be used for driving two fans.

A window air-conditioner works on the principle of vapour-compression refrigeration system.

8.13.2 Split Air-Conditioner

A split air-conditioner is also known as a *remote-mounted air-conditioner*. It is basically an air-conditioning system built in two distinct units: indoor unit and outdoor unit. The two units are connected by refrigerant piping lines.

The indoor unit consists of a fan and cooling coil. It is located in the space to be conditioned. It is a well-designed single casing, well insulated on the inside housing the evaporator coil, twin blower system with a motor, capillary tubes for refrigerant expansion, electronic controls and condensate drain provision.

The outdoor unit consists of a compressor, condenser coil and propeller fan with motor. The outdoor unit is connected to an indoor unit by extended suction and liquid pipelines.

Split air-conditioners are nowadays becoming popular because

1. They require less space in the room
2. They require no wall opening necessary
3. They have better air circulations inside, due to flexibility of proper location
4. The condenser, compressor and fan package can be located at any convenient place, where its noise is less objectionable and more accessible for maintainance.
5. They have flexibility of multiple evaporators in different rooms using single condenser–compressor unit.

8.13.3 Rooftop Unit

The rooftop unit is designed to be located on the roof of the building. The refrigeration, cooling and air-handling equipment are available in sections that are assembled together. The compressor and condenser are located at remote place.

The rooftop units are used with duct work and air outlets. These units do not use valuable building space and they are relatively low in cost.

8.14 CENTRAL AIR-CONDITIONING SYSTEM

A central air-conditioning system provides the coolant from one central location to different parts of a building. In this system, all the components are grouped together in one central room and conditioned air is distributed from the central room to the required places through insulated ducts as shown in Fig. 8.33. The central air-conditioner system is generally used for a refrigerating load above 25 TR and 2500 m^3/min of conditioned air.

The central air-conditioning systems provide fully controlled heating, cooling and ventilation. They are widely used in theaters, stores, restaurants and other public buildings. The system is complex and generally installed when the building is constructed.

The central air-conditioning system has the following advantages:

1. Automatic central control point

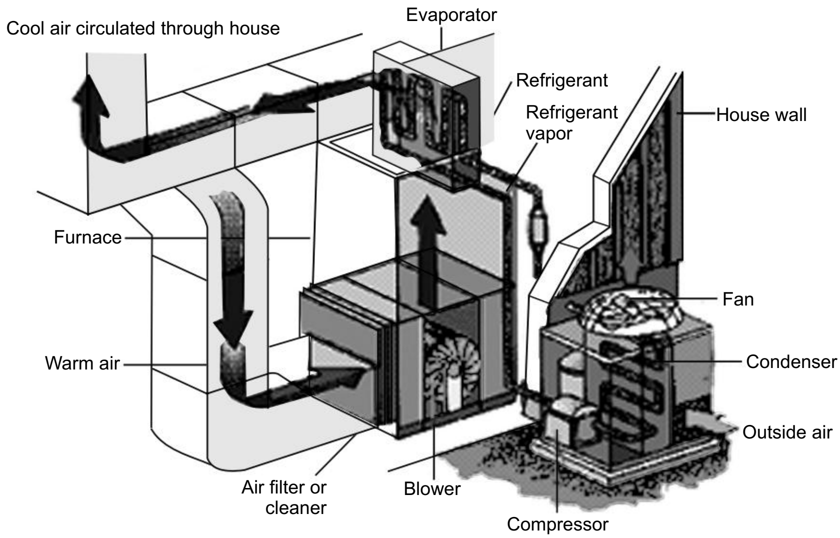


Fig. 8.33 *Central air-condition system*

2. Better air distribution
3. Easier maintenance, single cooling and heating equipments are installed in one location.

8.15 CLASSIFICATIONS OF CENTRAL AIR-CONDITIONING SYSTEM

The central station air-conditioning systems are classified according to cooling medium used to transport thermal energy to or from the air-conditioned space. The main types of cooling medium used in a central air-conditioning system are

1. All-air system
2. All-water system
3. Air–water system

8.15.1 All-Air System

In an all-air system, the sensible and latent cooling requirement of various parts of a building and equipments is provided by the conditioned cold air supplied by the central air-handling unit. The circulated air is then returned back to the air-handling unit, where it is further treated for the next cycle. The schematic arrangement of an all-air system is shown in Fig. 8.34.

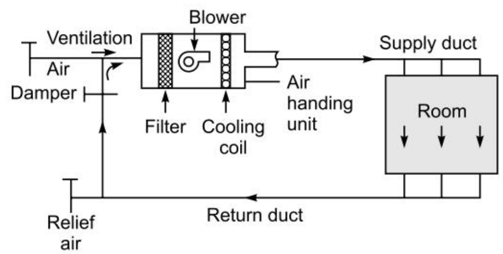


Fig. 8.34 *All-air-system central air-conditioning system*

8.15.2 All-Water System

In an all-water system, both sensible and latent space cooling are achieved by chilled water obtained from central refrigeration system through the cooling coils in terminal units located in various spaces in the building. Water has high specific heat, and thus it has the capacity to transport large thermal energy with lowest mass-flow rate. This characteristic of water lowers the size of piping work. It offers lowest initial and operating cost.

8.15.3 Air–Water System

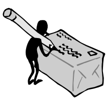
In an air–water system, both air and water are distributed to each air–conditioned space to create a cooling effect. In this system, the temperature of same space can be better regulated by varying the temperature of air or water (or both) during all seasons of the year. It requires less quantity of air circulation as compared to an all-air system. The reduced quantity of air can be supplied at high velocity to the conditioned space.

8.16 RATING OF AIR-CONDITIONING

Air-conditioning units are rated in terms of effective cooling capacity, which should be properly expressed in kilowatt units. But manufacturers of air-conditioners still rate them in terms of tonnes of refrigeration, which is equivalent to 3.517 kW. Window air-conditioners are available in 1 TR, 1.5 TR, 2 TR capacity.

The indicative TR load required for air-conditioning is presented below:

- Small office = 0.1 TR/m^2
- Medium size of office (10-30 persons occupancy) = 0.06 TR/m^2
- Large multistoried office/complex = 0.04 TR/m^2.



Summary

- The science of measuring properties of moist air is psychrometry.
- The temperature measured by an ordinary thermometer is termed as *dry-bulb temperature.*
- The *dew-point temperature* of dry air and water vapour mixture is defined as the saturation temperature corresponding to the partial pressure of the vapour in the mixture.
- The *adiabatic saturation temperature* of atmospheric air is defined as the temperature which results from adding water vapour adiabatically to the atmospheric air in steady flow, until it becomes saturated.
- The *wet-bulb temperature* is the temperature measured by a thermometer whose bulb is covered with water soaked cotton wick and placed in a stream of air.
- *The relative humidity* ϕ is defined as the ratio of actual mass of vapour to mass of vapour required to produce a saturated mixture of air at the same pressure and temperature:

$$\phi = \frac{m_v}{m_{sat}} = \frac{p_v}{p_{sat}}$$

- *The humidity ratio* ω is defined as the ratio of the mass of vapour in the atmospheric air to the mass of dry air:

$$\omega = \frac{m_v}{m_a} = 0.622\,\frac{p_v}{p - p_v}$$

where $p = p_a + p_v$ = total pressure of mixture.

- *The degree of saturation* μ is defined as the ratio of actual humidity ratio to the humidity ratio of saturated air at the same temperature and pressure:

$$\mu = \frac{\omega}{\omega_{sat}}$$

- The enthalpy of moist air is the sum of enthalpy of dry air and water vapour:

$$h = h_a + \omega h_g$$
$$= 2500 + 1.88\,T \text{ kJ/kg of water vapour.}$$

- The psychrometric chart is a graphical representation of properties of atmospheric air.
- Air washer is a system in which the air is passed through a spray of water.
- *Air-conditioning* is the process of treating air in an internal environment to achieve and maintain required standards of temperature, humidity, cleanliness and motion of air, regardless of surrounding conditions.
- An air-conditioning system may provide heating, cooling or both. An air-conditioning system controls the conditions of a confined space to produce a comfort effect over occupants. The indicative TR load required for air-conditioning is presented below:
  - Small office = 0.1 TR/m^2
  - Medium size of office (10–30 persons occupancy) = 0.06 TR/m^2
  - Large multistoried office/complex = 0.04 TR/m^2.

- In summer, the comfort conditions are achieved by cooling and dehumidification of air. In winter, the comfort conditions are achieved by heating and humidification of air and the year-round air conditioning system is the combination of both summer and winter air-conditioning systems.

Glossary

Air-conditioning Process of maintaining air at controlled temperature, humidity, purity and motion

Air refrigeration A refrigeration system in which air is the working medium

Air–conditioner Machine which creates air-conditioning

DBT Temperature indicated by an ordinary thermometer

Dehumidification Reduction of water vapour from moist air

DPT Temperature at which vapour starts to condense

Humidification Addition of water vapour in air

Psychrometry Study of moist air

Relative humidity Ratio of partial pressure of vapour to total pressure of mixture

Sensible heating/cooling Heat transfer with only change in DBT

Specific humidity Mass of water vapour per kg of dry air

WBT Temperature indicated by thermometer, when its bulb is covered by a wick thoroughly wetted.

Review Questions

1. What is the dew-point temperature?
2. List the properties of moist air that a psychrometric chart provides for an air-conditioning engineer.
3. Sketch a psychrometric chart and indicate the lines of constant wet-bulb temperature and constant relative humidity.
4. What is the difference between dry air and atmospheric air?
5. Define (a) specific humidity, and (b) relative humidity.
6. The moist air is passed through a cooling section where it is cooled and dehumidified. How do the specific humidity and relative humidity of air change during this process?
7. In the summer, the outer surface of a glass filled with iced water frequently 'sweats'. How can you explain this sweating?
8. When are the dry-bulb and dew-point temperatures identical?
9. What is an adiabatic saturation? When does the wet-bulb temperature equal the saturation temperature?
10. How is the enthalpy of air–water vapour mixture defined?
11. What do you understand by evaporative cooling? List three important characteristics of evaporative cooling systems.
12. What is sensible heating or cooling?
13. Explain the process of cooling and dehumidification.
14. Explain the process of heating and humidification.
15. Prove that the specific humidity is given by

$$\omega = 0.622 \frac{p_v}{p - p_v}$$

where p_v = partial pressure of water vapour
p = total pressure of air

16. Explain summer air-conditioning with the help of schematic.
17. Explain winter air-conditioning with the help of a schematic.
18. What is unitary air-conditioning system? Explain a window air-conditioning.
19. Explain the construction and working of a split air-conditioner.
20. State the advantages and disadvantages of central air conditioning system.

Problems

1. The moist air at 1 atm has 32°C DBT and 26°C WBT. Calculate
   (a) Partial pressure of water vapour,
   (b) Specific humidity,
   (c) Dew point temperature,
   (d) Relative humidity,
   (e) Degree of saturation, and
   (f) Enthalpy of mixture.

[(a) 0.03 bar (b) 0.0186 kg/kg of dry air, (c) 24.1°C, (d) 62.5% (e) 0.614, (f) 80.55 kJ/kg]

2. A sling psychrometer reads 40°C DBT and 36°C WBT. Calculate (a) specific humidity, (b) relative humidity, and (c) dew point. *[0.0238, 50%, 28.2°C]*

3. Calculate the amount of heat removed per kg of dry air, if the initial condition of air is 35°C DBT, 70% RH and final condition is 25°C DBT and 60% RH. *[46 kJ/kg]*

4. The moist air has 60% RH at 1 atm and 30°C. Calculate the specific humidity, dew point temperature, mass of water vapour and mass of dry air for 100 m^3 of moist air and partial pressure of water vapour and air.

[0.016 kg/kg of dry air, 21.25°C, 1.817 kg, 113.54 kg, 2.544 kPa, 98.78 kPa]

5. On a hot summer day, the ambient conditions are 40°C DBT, 20% RH. A desert cooler is used to increase the RH to 80%. Show the process on a psychrometric chart and by using it, calculate the temperature of exit air and minimum temperature to which the air can be cooled by a well-designed desert cooler. *[24.8°C, 22°C]*

6. DBT and WBT of atmospheric air at 1 atm are 25 and 15°C, respectively. Determine (a) specific humidity, (b) relative humidity, and (c) enthalpy of air.

[(a) 0.0065 kg/kg of dry air, (b) 33.2%, (c) 41.8 kJ/kg]

7. The air at 50°C has a relative humidity of 100%. What is its dew-point temperature? What mass of liquid water per kg of dry air will result if the mixture is cooled to 10°C at constant pressure of 90 kPa? *[50°C]*

8. Consider 100 m^3 of atmospheric air at 100 kPa, 15°C and 40% relative humidity. Find mass of water and humidity ratio. What is the dew point of the mixture? *[0.51 kg, 0.0043 kg, 1.4°C]*

9. Air at 10°C, 90% relative humidity and air at 30°C, 90% relative humidity are mixed steadily and adiabatically. The mass-flow rate of the colder stream is twice that of the other stream. What would be the state after mixing?

10. 10 m^3/min of air at 1 atm and 20°C with 90% RH is mixed with 20 m^3/min of air at 1 atm and 40°C with 20% RH. Calculate the resulting state of the mixture. *[h_3 = 61 kJ/kg, T_3 = 35.5°C]*

11. Consider a 500-litre rigid tank containing air–water vapour mixture at 100 kPa, 35°C with a 70%, RH. The system is cooled until the water just begins to condense. Determine the final temperature in the tank and the heat transfer for the process. *[28.2°C, –2.77 kJ]*

12. A stream of air at 1 atm, 30°C, 80% relative humidity flows at a rate of 28.0 kg/min. It is proposed to mix air at 1 atm, 15°C with a stream, steadily in an insulated mixing chamber so that the resulting temperature is 25°C. What is the rate of mass flow of the cooler air required and what would be the relative humidity of air leaving?

13. Air enters a dehumidifier at 34°C, 101 kPa and 60% RH. After being dehumidified, the air is heated to 20°C, with RH being 48%. Determine the temperature of air leaving the demudifier, if the air is saturated.

14. Saturated air at 20°C at a rate of 70 m^3/min is mixed adiabatically with the outside air at 35°C and 50% RH at a rate of 30 m^3/min. Assuming that the mixing process occurs at a pressure of 1 atm, determine the specific humidity, the relative humidity, and dry bulb temperature and volume flow rate of the mixture.

[0.0157 kg/kg of dry air, 80%, 26° C, 100.13 m^3/min]

15. Air at 1°C DBT and 80% relative humidity mixes adiabatically with an air stream at 18°C DBT, 40% relative humidity in the ratio of 1 to 3 by volume. Calculate the temperature and per cent saturation of the mixture. Take the barometric pressure 1.013 bar throughout constant.

[13.6°C and 48%]

Objective Questions

1. The psychrometry is
   (a) study of cooled air (b) study of hot air
   (c) study of moist air (d) None of the above
2. A psychrometer is an instrument, which measures
   (a) wet-bulb temperature
   (b) dry-bulb temperature
   (c) dew-point temperature
   (d) dry-bulb and wet-bulb temperatures
3. Temperature recorded by an ordinary thermocouple is known as
   (a) wet-bulb temperature
   (b) dry-bulb temperature
   (c) dew-point temperature
   (d) saturation temperature
4. Saturated air is
   (a) mixture of moisture and vapour
   (b) mixture of superheated vapour and air
   (c) moist air containing maximum possible moisture
   (d) None of the above
5. Temperature at which condensation of vapour begin is called
   (a) boiling temperature
   (b) saturation temperature
   (c) dew-point temperature
   (d) All of the above
6. Humidity ratio is defined as ratio of
   (a) mass of water vapour to mass of dry air in the mixture
   (b) mass of water vapour to mass of total mixture
   (c) mass of dry air to mass of water vapour
   (d) None of the above
7. The degree of saturation is defined as
   (a) mass of water vapour to mass of dry air in the mixture
   (b) ratio of actual humidity ratio to humidity ratio of saturated air
   (c) ratio of actual mass of water vapour to mass of saturated vapour
   (d) None of the above
8. Humidity ratio is called
   (a) relative humidity (b) specific humidity
   (c) degree of humidity (d) none of the above
9. Wet-bulb depression is equal to
   (a) sum of dry-bulb temperature and wet bulb temperature
   (b) average of dry-bulb temperature and wet-bulb temperature
   (c) difference of dry-bulb temperature and wet-bulb temperature
   (d) product of dry-bulb temperature and wet-bulb temperature, one of the above
10. On a psychrometric chart, the horizontal scale shows
    (a) wet-bulb temperature
    (b) dry-bulb temperature
    (c) dew-point temperature
    (d) absolute humidity
11. On a psychrometric chart, the vertical scale shows
    (a) wet-bulb temperature
    (b) dry-bulb temperature
    (c) dew-point temperature
    (d) absolute humidity
12. On a psychrometric chart, the wet-bulb temperature lines are
    (a) curved
    (b) horizontal but not uniform
    (c) vertical
    (d) straight and inclined
13. On a psychrometric chart, the relative humidity lines are
    (a) curved
    (b) horizontal and uniform
    (c) vertical
    (d) straight and inclined
14. On a psychrometric chart, the dew point temperature lines are
    (a) curved
    (b) horizontal and uniform

(c) vertical
(d) straight and inclined

15. On a psychrometric chart, the dry-bulb temperature lines are
 (a) curved
 (b) horizontal and uniform
 (c) vertical
 (d) straight and inclined

16. For a moist air, if DBT is 15°C, WBT is 15°C and DPT is also 15°C, the saturation temperature will be
 (a) 15°C (b) 25°C
 (c) 10°C (d) 35°C

17. The process of cooling of air at same humidity ratio is known as
 (a) sensible heating (b) sensible cooling
 (c) humidification (d) dehumidification

18. The process of adding moisture to air at constant dry-bulb temperature is known as
 (a) sensible heating (b) sensible cooling
 (c) humidification (d) dehumidification

19. The process of removing moisture from air at constant dry-bulb temperature is known as
 (a) sensible heating (b) sensible cooling
 (c) humidification (d) dehumidification

20. During the process of adiabatic saturation,
 (a) DBT decreases
 (b) specific humidity increase
 (c) RH increases
 (d) all of the above

Answers

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

A

Appendix

Table A.1 *Properties of Saturated Water (Liquid–Vapour): Temperature Table*

		Specific Volume m^3/kg		*Internal Energy* *kJ/kg*		*Enthalpy* *kJ/kg*			*Entropy* *kJ/kg.K*		
Temp. °*C*	*Sat. Presss.* bar	*Sat. Liquid* $v_f \times 10^3$	*Sat. Vapour* v_g	*Sat. Liquid* u_f	*Sat. Vapour* u_g	*Sat. Liquid* h_f	*Evap.* h_{fg}	*Sat. Vapour* h_g	*Sat. Liquid* s_f	*Sat. Vapour* s_g	*Temp.* °*C*
0.01	0.00611	1.0002	206.136	0.00	2375.3	0.01	2501.3	2501.4	0.0000	9.1562	0.01
4	0.00813	1.0001	157.232	16.77	2380.9	16.78	2491.9	2508.7	0.0610	9.0514	4
5	0.00872	1.0001	147.120	20.97	2382.3	20.98	2489.6	2510.6	0.0761	9.0257	5
6	0.00935	1.0001	137.734	25.19	2383.6	25.20	2487.2	2512.4	0.0912	9.0003	6
8	0.01072	1.0002	120.917	33.59	2386.4	33.60	2482.5	2516.1	0.1212	8.9501	8
10	0.01228	1.0004	106.379	42.00	2389.2	42.01	2477.7	2519.8	0.1510	8.9008	10
11	0.01312	1.0004	99.857	46.20	2390.5	46.20	2475.4	2521.6	0.1658	8.8765	11
12	0.01402	1.0005	93.784	50.41	2391.9	50.41	2473.0	2523.4	0.1806	8.8524	12
13	0.01497	1.0007	88.124	54.60	2393.3	54.60	2470.7	2525.3	0.1953	8.8285	13
14	0.01598	1.0008	82.848	58.79	2394.7	58.80	2468.3	2527.1	0.2099	8.8048	14
15	0.01705	1.0009	77.926	62.99	2396.1	62.99	2465.9	2528.9	0.2245	8.7814	15
16	0.01818	1.0011	73.333	67.18	2397.4	67.19	2463.6	2530.8	0.2390	8.7582	18
17	0.01938	1.0012	69.044	71.38	2398.8	71.38	2461.2	2532.6	0.2535	8.7351	17
18	0.02064	1.0014	65.038	75.57	2400.2	75.58	2458.8	2534.4	0.2679	8.7123	18
19	0.02198	1.0016	61.293	79.76	2401.6	79.77	2456.5	2536.2	0.2823	8.6897	19
20	0.02339	1.0018	57.791	83.95	2402.9	83.96	2454.1	2538.1	0.2966	8.6672	20
21	0.02487	1.0020	54.514	88.14	2404.3	88.14	2451.8	2539.9	0.3109	8.6450	21
22	0.02645	1.0022	51.447	92.32	2405.7	92.33	2449.4	2541.7	0.3251	8.6229	22
23	0.02810	1.0024	48.574	96.51	2407.0	96.52	2447.0	2543.5	0.3393	8.6011	23
24	0.02985	1.0027	45.883	100.70	2408.4	100.70	2444.7	2545.4	0.3534	8.5794	24
25	0.03169	1.0029	43.360	104.88	2409.8	104.89	2442.3	2547.2	0.3674	8.5580	25
26	0.03363	1.0032	40.994	109.06	2411.1	109.07	2439.9	2549.0	0.3814	8.5367	26
27	0.03567	1.0035	38.774	113.25	2412.5	113.25	2437.6	2550.8	0.3954	8.5156	27

Contd.

Table A.1 *Contd.*

Temp. °C	Sat. Presss. bar	Specific Volume m^3/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapour u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Entropy kJ/kg.K Sat. Liquid s_f	Sat. Vapour s_g	Temp. °C
28	0.03782	1.0037	36.690	117.42	2413.9	117.43	2435.2	2552.6	0.4093	8.4946	28
29	0.04008	1.0040	34.733	121.60	2415.2	121.61	2432.8	2554.5	0.4231	8.4739	29
30	0.04246	1.0043	32.894	125.78	2416.6	125.79	2430.5	2556.3	0.4369	8.4533	30
31	0.04496	1.0046	31.165	129.96	2418.0	129.97	2428.1	2558.1	0.4507	8.4329	31
32	0.04759	1.0050	29.540	134.14	2419.3	134.15	2425.7	2559.9	0.4644	8.4127	32
33	0.05034	1.0053	28.011	138.32	2420.7	138.33	2423.4	2561.7	0.4781	8.3927	33
34	0.05324	1.0056	26.571	142.50	2422.0	142.50	2421.0	2563.5	0.4917	8.3728	34
35	0.05628	1.0060	25.216	146.67	2423.4	146.68	2418.6	2565.3	0.5053	8.3531	35
36	0.05947	1.0063	23.940	150.85	2424.7	150.86	2416.2	2567.1	0.5188	8.3336	36
38	0.06632	1.0071	21.602	159.20	2427.4	159.21	2411.5	2570.7	0.5458	8.2950	38
40	0.07384	1.0078	19.523	167.56	2430.1	167.57	2406.7	2574.3	0.5725	8.2570	40
45	0.09593	1.0099	15.258	188.44	2436.8	188.45	2394.8	2583.2	0.6387	8.1648	45
50	0.1235	1.0121	12.032	209.32	2443.5	209.33	2382.7	2592.1	0.7038	8.0763	50
55	0.1576	1.0146	9.568	230.21	2450.1	230.23	2370.7	2600.9	0.7679	7.9913	55
60	0.1994	1.0172	7.671	251.11	2456.6	251.13	2358.5	2609.6	0.8312	7.9096	60
65	0.2503	1.0199	6.197	272.02	2463.1	272.06	2346.2	2618.3	0.8935	7.8310	65
70	0.3119	1.0228	5.042	292.95	2469.6	292.98	2333.8	2626.8	0.9549	7.7553	70
75	0.3858	1.0259	4.131	313.90	2475.9	313.93	2321.4	2635.3	1.0155	7.6824	75
80	0.4739	1.0291	3.407	334.86	2482.2	334.91	2308.8	2643.7	1.0753	7.6122	80
85	0.5783	1.0325	2.828	355.84	2488.4	355.90	2296.0	2651.9	1.1343	7.5445	85
90	0.7014	1.0360	2.361	376.85	2494.5	376.92	2283.2	2660.1	1.1925	7.4791	90
95	0.8455	1.0397	1.982	397.88	2500.6	397.96	2270.2	2668.1	1.2500	7.4159	95
100	1.014	1.0435	1.673	418.94	2506.5	419.04	2257.0	2676.1	1.3069	7.3549	100
110	1.433	1.0516	1.210	461.14	2518.1	461.30	2230.2	2691.5	1.4185	7.2387	110
120	1.985	1.0603	0.8919	503.50	2529.3	503.71	2202.6	2706.3	1.5276	7.1296	120
130	2.701	1.0697	0.6685	546.02	2539.9	546.31	2174.2	2720.5	1.6344	7.0269	130
140	3.6153	1.0797	0.5089	588.74	2550.0	589.13	2144.7	2733.9	1.7391	6.9299	140
150	4.758	1.0905	0.3928	631.68	2559.5	632.20	2114.3	2746.5	1.8418	6.8379	150
160	6.178	1.1020	0.3071	674.86	2568.4	675.55	2082.6	2758.1	1.9427	6.7502	160
170	7.917	1.1143	0.2428	718.33	2576.5	719.21	2049.5	2768.7	2.0419	6.6663	170
180	10.02	1.1274	0.1941	762.09	2583.7	763.22	2015.0	2778.2	2.1396	6.5857	180
190	12.54	1.1414	0.1565	806.19	2590.0	807.62	1978.8	2786.4	2.2359	6.5079	190
200	15.54	1.1565	0.1274	850.65	2595.3	852.45	1940.7	2793.2	2.3309	6.4323	200
210	19.06	1.1726	0.1044	895.53	2599.5	897.76	1900.7	2798.5	2.4248	6.3585	210
220	23.18	1.1900	0.08619	940.87	2602.4	943.62	1858.5	2802.1	2.5178	6.2861	220
230	27.95	1.2088	0.07158	986.74	2603.9	990.12	1813.8	2804.0	2.6099	6.2146	230
240	33.44	1.2291	0.05976	1033.2	2604.0	1037.3	1766.5	2803.8	2.7015	6.1437	240

Contd.

Table A.1 *Contd.*

Temp. °C	Sat. Presss. bar	Specific Volume m^3/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapour u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Entropy kJ/kg.K Sat. Liquid s_f	Sat. Vapour s_g	Temp. °C
250	39.73	1.2512	0.05013	1080.4	2602.4	1085.4	1716.2	2801.5	2.7927	6.0730	250
260	46.88	1.2755	0.04221	1128.4	2599.0	1134.4	1662.5	2796.6	2.8838	6.0019	260
270	54.99	1.3023	0.03564	1177.4	2593.7	1184.5	1605.2	2789.7	2.9751	5.9301	270
280	64.12	1.3321	0.03017	1227.5	2586.1	1236.0	1543.6	2779.6	3.0668	5.8571	280
290	74.36	1.3656	0.02557	1278.9	2576.0	1289.1	1477.1	2766.2	3.1594	5.7821	290
300	85.81	1.4036	0.02167	1332.0	2563.0	1344.0	1404.9	2749.0	3.2534	5.7045	300
320	112.7	1.4988	0.01549	1444.6	2525.5	1461.5	1238.6	2700.1	3.4480	5.5362	320
340	145.9	1.6379	0.01080	1570.3	2464.6	1594.2	1027.9	2622.0	3.6594	5.3357	340
360	186.5	1.8925	0.006945	1725.2	2351.5	1760.5	720.5	2481.0	3.9147	5.0526	360
374.14	220.9	3.155	0.003155	2029.6	2029.6	2099.3	0	2099.3	4.4298	4.4298	374.14

Table A.2 *Properties of* ***Saturated Water*** *(Liquid vapour): Pressure Table*

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy $kJ/kg \cdot K$		
Press. bar	Sat. Temp. °C	Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Sat. Vapour s_g	Press. bar
0.04	28.96	1.0040	34.800	121.45	2415.2	121.46	2432.9	2554.4	0.4226	8.4746	0.04
0.06	36.16	1.0064	23.739	151.53	2425.0	151.53	2415.9	2567.4	0.5210	8.3304	0.06
0.08	41.51	1.0084	18.103	173.87	2432.2	173.88	2403.1	2577.0	0.5926	8.2287	0.18
0.10	45.81	1.0102	14.674	181.82	2437.9	191.83	2392.8	2584.7	0.6493	8.1502	0.10
0.20	60.06	1.0172	7.649	251.38	2456.7	251.40	2358.3	2609.7	0.8320	7.9085	0.20
0.30	69.10	1.0223	5.229	289.20	2468.4	289.23	2336.1	2625.3	0.9439	7.7686	0.30
0.40	75.87	1.0265	3.993	317.53	2477.0	317.58	2319.2	2636.8	1.0259	7.6700	0.40
0.50	81.33	1.0300	3.240	340.44	2483.9	340.49	2305.4	2645.9	1.0910	7.5939	0.50
0.60	85.94	1.0331	2.732	359.79	2489.6	359.86	2293.6	2653.5	1.1453	7.5320	0.60
0.70	89.95	1.0360	2.365	376.63	2494.5	376.70	2283.3	2660.0	1.1919	7.4797	0.70
0.80	93.50	1.0380	2.087	391.58	2498.8	391.66	2274.1	2665.8	1.2329	7.4346	0.80
0.90	96.71	1.0410	1.869	405.06	2502.6	405.15	2265.7	2670.9	1.2695	7.3949	0.90
1.00	99.63	1.0432	1.694	417.36	2506.1	417.46	2258.0	2675.5	1.3026	7.3594	1.00
1.50	111.4	1.0528	1.159	466.94	2519.7	467.11	2226.5	2693.6	1.4336	7.2233	1.50
2.00	120.2	1.0605	0.8857	504.49	2529.5	504.70	2201.9	2706.7	1.5301	7.1271	2.00
2.50	127.4	1.0672	0.7187	535.10	2537.2	535.37	2181.5	2716.9	1.6072	7.0527	2.50
3.00	133.6	1.0732	0.6058	561.15	2543.6	561.47	2163.8	2725.3	1.6718	6.9919	3.00
3.50	138.9	1.0786	0.5243	583.95	2546.9	584.33	2148.1	2732.4	1.7275	6.9405	3.50
4.00	143.6	1.0836	0.4625	604.31	2553.6	604.74	2133.8	2738.6	1.7766	6.8959	4.00
4.50	147.9	1.0882	0.4140	622.25	2557.6	623.25	2120.7	2743.9	1.8207	6.8565	4.50
5.00	151.9	1.0926	0.3749	639.68	2561.2	640.23	2108.5	2748.7	1.8607	6.8212	5.00
6.00	158.9	1.1006	0.3157	669.90	2567.4	670.56	2086.3	2756.8	1.9312	6.7600	6.00
7.00	165.0	1.1080	0.2729	696.44	2572.5	697.22	2066.3	2763.5	1.9922	6.7080	7.00
8.00	170.4	1.1148	0.2404	720.22	2576.8	721.11	2048.0	2769.1	2.0462	6.6628	8.00
9.00	175.4	1.1212	0.2150	741.83	2580.5	742.83	2031.1	2773.9	2.0946	6.6226	9.00
10.0	179.9	1.1273	0.1944	761.68	2583.6	762.81	2015.3	2778.1	2.1387	6.5863	10.0
15.0	198.3	1.1539	0.1318	843.16	2594.5	844.84	1947.3	2792.2	2.3150	6.4448	15.0
20.0	212.4	1.1767	0.09963	906.44	2600.3	908.79	1890.7	2799.5	2.4474	6.3409	20.0
25.0	224.0	1.1983	0.07998	959.11	2603.1	962.11	1841.0	2803.1	2.5547	6.2575	25.0
30.0	233.9	1.2165	0.06668	1004.8	2604.1	1008.4	1795.7	2804.2	2.6457	6.1869	30.0
35.0	242.6	1.2347	0.05707	1045.4	2603.7	1049.8	1753.7	2803.4	2.7253	6.1253	35.0
40.0	250.4	1.2522	0.04978	1082.3	2602.3	1087.3	1714.1	2801.4	2.7964	6.0701	40.0
45.0	257.5	1.2692	0.04406	1116.2	2600.1	1121.9	1676.4	2798.3	2.8610	6.0199	45.0
50.0	264.0	1.2859	0.03944	1147.8	2597.1	1154.2	1640.1	2794.3	2.9202	5.9734	50.0

Contd.

Table A.2 *Contd.*

Press. bar	Sat. Temp. °C	Specific Volume m^3/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapour u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Entropy $kJ/kg \cdot K$ Sat. Liquid s_f	Sat. Vapour s_g	Press. bar
60.0	275.6	1.3187	0.03244	1205.4	2589.7	1213.4	1571.0	2784.3	3.0267	5.8892	60.0
70.0	285.9	1.3513	0.02737	1257.6	2580.5	1267.0	1505.1	2772.1	3.1211	5.8133	70.0
80.0	295.1	1.3842	0.02352	1305.6	2569.8	1316.6	1441.3	2758.0	3.2068	5.7432	80.0
90.0	303.4	1.4178	0.02048	1350.5	2557.8	1363.3	1378.9	2742.1	3.2858	5.6772	90.0
100.0	311.1	1.4524	0.01803	1393.0	2544.4	1407.6	1317.1	2724.7	3.3596	5.6141	100.0
110.0	318.2	1.4886	0.01599	1433.7	2529.8	1450.1	1255.5	2705.6	3.4295	5.5527	110.0
120.0	324.8	1.5267	0.01426	1473.0	2513.7	1491.3	1193.6	2684.9	3.4962	5.4924	120.0
130.0	330.9	1.5671	0.01278	1511.0	2496.1	1531.5	1130.7	2662.2	3.5606	5.4323	130.0
140.0	336.8	1.6107	0.01149	1548.6	2476.8	1571.1	1066.5	2637.6	3.6232	5.3717	140.0
150.0	342.2	1.6581	0.01034	1585.6	2455.5	1610.5	1000.0	2610.5	3.6848	5.3098	150.0
160.0	347.4	1.7107	0.009306	1622.7	2431.7	1650.1	930.6	2580.6	3.7481	5.2455	160.0
170.0	352.4	1.7702	0.008364	1660.2	2405.0	1690.3	856.9	2547.2	3.8079	5.1777	170.0
180.0	357.1	1.8397	0.007489	1698.9	2374.3	1732.0	777.1	2509.1	3.8715	5.1044	180.0
190.0	361.5	1.9243	0.006657	1739.9	2338.1	1776.5	688.0	2464.5	3.9388	5.0228	190.0
200.0	365.8	2.036	0.005834	1785.6	2293.0	1826.3	583.4	2409.7	4.0139	4.9269	200.0
220.9	374.1	3.155	0.003155	2029.6	2029.6	2099.3	0	2099.3	4.4298	4.4298	220.9

Table A.3 *Properties of* ***Superheated Water Vapor***

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	p = 0.06 bar = 0.006 MPa (T_{sat} = 36.16°C)				*p = 0.35 bar = 0.035 MPa (T_{sat} = 72.69°C)*			
Sat.	23.739	2425.0	2567.4	8.3304	4.526	2473.0	2631.4	7.7158
80	27.132	2487.3	2650.1	8.5804	4.625	2483.7	2645.6	7.7564
120	30.219	2544.7	2726.0	8.7840	5.163	2542.4	2723.1	7.9644
160	33.302	2602.7	2802.5	8.9693	5.696	2601.2	2800.6	8.1519
200	36.383	2661.4	2879.7	9.1398	6.228	2660.4	2878.4	8.3237
240	39.462	2721.0	2957.8	9.2982	6.758	2720.3	2956.8	8.4828
280	42.540	2781.5	3036.8	9.4464	7.287	2780.9	3036.0	8.6314
320	45.618	2843.0	3116.7	9.5859	7.815	2842.5	3116.1	8.7712
360	48.696	2905.5	3197.7	9.7180	8.344	2905.1	3197.1	8.9034
400	51.774	2969.0	3279.6	9.8435	8.872	2968.6	3279.2	9.0291
440	54.851	3033.5	3362.6	9.9633	9.400	3033.2	3362.2	9.1490
500	59.467	3132.3	3489.1	10.1336	10.192	3132.1	3488.8	9.3194
	p = 0.70 bar = 0.07 MPa (T_{sat} = 89.95°C)				*p = 1.0 bar = 0.10 MPa (T_{sat} = 99.63°C)*			
Sat.	2.365	2494.5	2660.0	7.4797	1.694	2506.1	2675.5	7.3594
100	2.434	2509.7	2680.0	7.5341	1.696	2506.7	2676.2	7.3614
120	2.571	2539.7	2719.6	7.6375	1.793	2537.3	2716.6	7.4668
160	2.841	2599.4	2798.2	7.8279	1.984	2597.8	2796.2	7.6597
200	3.108	2659.1	2876.7	8.0012	2.172	2658.1	2875.3	7.8343
240	3.374	2719.3	2955.5	8.1611	2.359	2718.5	2954.5	7.9949
280	3.640	2780.2	3035.0	8.3162	2.546	2779.6	3034.2	8.1445
320	3.905	2842.0	3115.3	8.4504	2.732	2841.5	3114.6	8.2849
360	4.170	2904.6	3196.5	8.5828	2.917	2904.2	3195.9	8.4175
400	4.434	2968.2	3278.6	8.7086	3.103	2967.9	3278.2	8.5435
440	4.698	3032.9	3361.8	8.8286	3.288	3032.6	3361.4	8.6636
500	5.095	3131.8	3488.5	8.9991	3.565	3131.6	3488.1	8.8342
	p = 1.5 bar = 0.15 MPa (T_{sat} = 111.37°C)				*p = 3.0 bar = 0.30 MPa (T_{sat} = 133.55°C)*			
Sat.	1.159	2519.7	2693.6	7.2233	0.606	2543.6	2725.3	6.9919
120	1.188	2533.3	2711.4	7.2693				
160	1.317	2595.2	2792.8	7.4665	0.651	2587.1	2782.3	7.1276
200	1.444	2656.2	2872.9	7.6433	0.716	2650.7	2865.5	7.3115
240	1.570	2717.2	2952.7	7.8052	0.781	2713.1	2947.3	7.4774
280	1.695	2778.6	3032.8	7.9555	0.844	2775.4	3028.6	7.6299
320	1.819	2840.6	3113.5	8.0964	0.907	2838.1	3110.1	7.7722
360	1.943	2903.5	3195.0	8.2293	0.969	2901.4	3192.2	7.9061
400	2.067	2967.3	3277.4	8.3555	1.032	2965.6	3275.0	8.0330
440	2.191	3032.1	3360.7	8.4757	1.094	3030.6	3358.7	8.1538
500	2.376	3131.2	3487.6	8.6466	1.187	3130.0	3486.0	8.3251
600	2.685	3301.7	3704.3	8.9101	1.341	3300.8	3703.2	8.5892

Contd.

Table A.3 *Contd.*

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K
	p = 5.0 bar = 0.50 MPa (T_{sat} = 151.86°C)				p = 7.0 bar = 0.70 MPa (T_{sat} = 164.97°C)			
Sat.	0.3749	2561.2	2748.7	6.8213	0.2729	2572.5	2763.5	6.7080
180	0.4045	2609.7	2812.0	6.9656	0.2847	2599.8	2799.1	6.7880
200	0.4249	2642.9	2855.4	7.0592	0.2999	2634.8	2844.8	6.8865
240	0.4646	2707.6	2939.9	7.2307	0.3292	2701.8	2932.2	7.0641
280	0.5034	2771.2	3022.9	7.3865	0.3574	2766.9	3017.1	7.2233
320	0.5416	2834.7	3105.6	7.5308	0.3852	2831.3	3100.9	7.3697
360	0.5796	2898.7	3188.4	7.6660	0.4126	2895.8	3184.7	7.5063
400	0.6173	2963.2	3271.9	7.7938	0.4397	2960.9	3268.7	7.6350
440	0.6548	3028.6	3356.0	7.9152	0.4667	3026.6	3353.3	7.7571
500	0.7109	3128.4	3483.9	8.0873	0.5070	3126.8	3481.7	7.9299
600	0.8041	3299.6	3701.7	8.3522	0.5738	3298.5	3700.2	8.1956
700	0.8969	3477.5	3925.9	8.5952	0.6403	3476.6	3924.8	8.4391
	p = 10.0 bar = 1.0 MPa (T_{sat} = 179.91°C)				p = 15.0 bar = 1.5 MPa (T_{sat} = 198.32°C)			
Sat.	0.1944	2583.6	2778.1	6.5865	0.1318	2594.5	2792.2	6.4448
200	0.2060	2621.9	2827.9	6.6940	0.1325	2598.1	2796.8	6.4546
240	0.2275	2692.9	2920.4	6.8817	0.1483	2676.9	2899.3	6.6628
280	0.2480	2760.2	3008.2	7.0465	0.1627	2748.6	2992.7	6.8381
320	0.2678	2826.1	3093.9	7.1962	0.1765	2817.1	3081.9	6.9938
360	0.2873	2891.6	3178.9	7.3349	0.1899	2884.4	3169.2	7.1363
400	0.3066	2957.3	3263.9	7.4651	0.2030	2951.3	3255.8	7.2690
440	0.3257	3023.6	3349.3	7.5883	0.2160	3018.5	3342.5	7.3940
500	0.3541	3124.4	3478.5	7.7622	0.2352	3120.3	3473.1	7.5698
540	0.3729	3192.6	3565.6	7.8720	0.2478	3189.1	3560.9	7.6805
600	0.4011	3296.8	3697.9	8.0290	0.2668	3293.9	3694.0	7.8385
640	0.4198	3367.4	3787.2	8.1290	0.2793	3364.8	3783.8	7.9391
	p = 20.0 bar = 2.0 MPa (T_{sat} = 212.42°C)				p = 30.0 bar = 3.0 MPa (T_{sat} = 233.90°C)			
Sat.	0.0996	2600.3	2799.5	6.3409	0.0667	2604.1	2804.2	6.1869
240	0.1085	2659.6	2876.5	6.4952	0.0682	2619.7	2824.3	6.2265
280	0.1200	2736.4	2976.4	6.6828	0.0771	2709.9	2941.3	6.4462
320	0.1308	2807.9	3069.3	6.8452	0.0850	2788.4	3043.4	6.6245
360	0.1411	2877.0	3159.3	6.9917	0.0923	2861.7	3138.7	6.7801
400	0.1512	2945.2	3247.6	7.1271	0.0994	2932.8	3230.9	6.9212
440	0.1611	3013.4	3335.5	7.2540	0.1062	3002.9	3321.5	7.0520
500	0.1757	3116.2	3467.6	7.4317	0.1162	3108.0	3456.5	7.2338
540	0.1853	3185.6	3556.1	7.5434	0.1227	3178.4	3546.6	7.3474
600	0.1996	3290.9	3690.1	7.7024	0.1324	3285.0	3682.3	7.5085
640	0.2091	3362.2	2780.4	7.8035	0.1388	3357.0	3773.5	7.6106
700	0.2232	3470.9	3917.4	7.9487	0.1484	3466.5	3911.7	7.7571

Contd.

Table A.3 *Contd.*

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s $kJ/kg \cdot K$	v m^3/kg	u kJ/kg	h kJ/kg	s $kJ/kg \cdot K$
	p = 40 bar = 4.0 MPa (T_{sat} = 250.4°C)				p = 60 bar = 6.0 MPa (T_{sat} = 275.64°C)			
Sat.	0.04978	2602.3	2801.4	6.0701	0.03244	2589.7	2784.3	5.8892
280	0.05546	2680.0	2901.8	6.2568	0.03317	2605.2	2804.2	5.9252
320	0.06199	2767.4	3015.4	6.4553	0.03876	2720.0	2952.6	6.1846
360	0.06788	2845.7	3117.2	6.6215	0.04331	2811.2	3071.1	6.3782
400	0.07341	2919.9	3213.3	6.7690	0.04793	2892.9	3177.2	6.5408
440	0.07872	2992.2	3307.1	6.9041	0.05122	2970.0	3277.3	6.6853
500	0.08643	3099.5	3445.3	7.0901	0.05665	3082.2	3422.2	6.8803
540	0.09145	3171.1	3536.9	7.2056	0.06015	3156.1	3517.0	6.9999
600	0.09885	3279.1	3674.4	7.3688	0.06525	3266.9	3658.4	7.1677
640	0.1037	3351.8	3766.6	7.4720	0.06859	3341.0	3752.6	7.2731
700	0.1110	3462.1	3905.9	7.6198	0.07352	3453.1	3894.1	7.4234
740	0.1157	3536.6	3999.6	7.7141	0.07677	3528.3	3989.2	7.5190
	p = 80 bar = 8.0 MPa (T_{sat} = 295.06°C)				p = 100 bar = 10.0 MPa (T_{sat} = 311.06°C)			
Sat.	0.02352	2569.8	2758.0	5.7432	0.01803	2544.4	2724.7	5.6141
320	0.02682	2662.7	2877.2	5.9489	0.01925	2588.8	2781.3	5.7103
360	0.03089	2772.7	3019.8	6.1819	0.02331	2729.1	2962.1	6.0060
400	0.03432	2863.8	3138.3	6.3634	0.02641	2832.4	3096.5	6.2120
440	0.03742	2946.7	3246.1	6.5190	0.02911	2922.1	3213.2	6.3805
480	0.04034	3025.7	3348.4	6.6586	0.03160	3005.4	3321.4	6.5282
520	0.04313	3102.7	3447.7	6.7871	0.03394	3085.6	3425.1	6.6622
560	0.04582	3178.7	3545.3	6.9072	0.03619	3164.1	3526.0	6.7864
600	0.04845	3254.4	3642.0	7.0206	0.03837	3241.7	3625.3	6.9029
640	0.05102	3330.1	3738.3	7.1283	0.04048	3318.9	3723.7	7.0131
700	0.05481	3443.9	3882.4	7.2812	0.04358	3434.7	3870.5	7.1687
740	0.05729	3520.4	3978.7	7.3782	0.04560	3512.4	3968.1	7.2670
	p = 120 bar = 12.0 MPa (T_{sat} = 324.75°C)				p = 140 bar = 14.0 MPa (T_{sat} = 336.75°C)			
Sat.	0.01426	2513.7	2684.9	5.4924	0.01149	2476.8	2637.61	5.3717
360	0.01811	2678.4	2895.7	5.8361	0.01422	2617.4	2816.5	5.6602
400	0.02108	2798.3	3051.3	6.0747	0.01722	2760.9	3001.9	5.9448
440	0.02355	2896.1	3178.7	6.2586	0.01954	2868.6	3142.2	6.1474
480	0.02576	2984.4	3293.5	6.4154	0.02157	2962.5	3264.5	6.3143
520	0.02781	3068.0	3401.8	6.5555	0.02343	3049.8	3377.8	6.4610
560	0.02977	3149.0	3506.2	6.6840	0.02517	3133.6	3486.0	6.5941
600	0.03164	3228.7	3608.3	6.8037	0.02683	3215.4	3591.1	6.7172
640	0.03345	3307.5	3709.0	6.9164	0.02843	3296.0	3694.1	6.8326
700	0.03610	3425.2	3858.4	7.0749	0.03075	3415.7	3846.2	6.9939
740	0.03781	3503.7	3957.4	7.1746	0.03225	3495.2	3946.7	7.0952

Contd.

Table A.3 *Contd.*

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg · K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg · K
	p = 160 bar = 16.0 MPa (T_{sat} = 347.44°C)				*p* = 180 bar = 18.0 MPa (T_{sat} = 357.06°C)			
Sat.	0.00931	2431.7	2580.6	5.2455	0.00749	2374.3	2509.1	5.1044
360	0.01105	2539.0	2715.8	5.4614	0.00809	2418.9	2564.5	5.1922
400	0.01426	2719.4	2947.6	5.8175	0.01190	2672.8	2887.0	5.6887
440	0.01652	2839.4	3103.7	6.0429	0.01414	2808.2	3062.8	5.9428
480	0.01842	2939.7	3234.4	6.2215	0.01596	2915.9	3203.2	6.1345
520	0.02013	3031.1	3353.3	6.3752	0.01757	3011.8	3378.0	6.2960
560	0.02172	3117.8	3465.4	6.5132	0.01904	3101.7	3444.4	6.4392
600	0.02323	3201.8	3573.5	6.6399	0.02042	3188.0	3555.6	6.5696
640	0.02467	3284.2	3678.9	6.7580	0.02174	3272.3	3663.6	6.6905
700	0.02674	3406.0	3833.9	6.9224	0.02362	3396.3	3821.5	6.8580
740	0.02808	3486.7	3935.9	7.0251	0.02483	3478.0	3925.0	6.9623
	p = 200 bar = 20.0 MPa (T_{sat} = 365.81°C)				*p* = 240 bar = 24.0 MPa			
Sat.	0.00583	2293.0	2409.7	4.9269				
400	0.00994	2619.3	2818.1	5.5540	0.00673	2477.8	2639.4	5.2393
440	0.01222	2774.9	3019.4	5.8450	0.00929	2700.6	2923.4	5.6506
480	0.01399	2891.2	3170.8	6.0518	0.01100	2838.3	3102.3	5.8950
520	0.01551	2992.0	3302.2	6.2218	0.01241	2950.5	3248.5	6.0842
560	0.01689	3085.2	3423.0	6.3705	0.01366	3051.1	3379.0	6.2448
600	0.01818	3174.0	3537.6	6.5048	0.01481	3145.2	3500.7	6.3875
640	0.01940	3260.2	3648.1	6.6286	0.01588	3235.5	3616.7	6.5174
700	0.02113	3386.4	3809.0	6.7993	0.01739	3366.4	3783.8	6.6947
740	0.02224	3469.3	3914.1	6.9052	0.01835	3451.7	3892.1	6.8038
800	0.02385	3592.7	4069.7	7.0544	0.01974	3578.0	4051.6	6.9567
	p = 280 bar = 28.0 MPa				*p* = 320 bar = 32.0 MPa			
400	0.00383	2223.5	2330.7	4.7494	0.00236	1980.4	2055.9	4.3239
440	0.00712	2613.2	2812.6	5.4494	0.00544	2509.0	2683.0	5.2327
480	0.00885	2780.8	3028.5	5.7446	0.00722	2718.1	2949.2	5.5968
520	0.01020	2906.8	3192.3	5.9566	0.00853	2860.7	3133.7	5.8357
560	0.01136	3015.7	3333.7	6.1307	0.00963	2979.0	3287.2	6.0246
600	0.01241	3115.6	3463.0	6.2823	0.01061	3085.3	3424.6	6.1858
640	0.01338	3210.3	3584.8	6.4187	0.01150	3184.5	3552.5	6.3290
700	0.01473	3346.1	3758.4	6.6029	0.01273	3325.4	3732.8	6.5203
740	0.01558	3433.9	3870.0	6.7153	0.01350	3415.9	3847.8	6.6361
800	0.01680	3563.1	4033.4	6.8720	0.01460	3548.0	4015.1	6.7966
900	0.01873	3774.3	4298.8	7.1084	0.01633	3762.7	4285.1	7.0372

B

Appendix

Table B.1 *Thermodynamic Properities of* ***R-12***

Temp C T	Press. kPa p	*Specific Volume, m³/kg*				*Internal Energy, kJ/kg*					Temp °C T
		Sat. Liquid v_f	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Sat. Vapour s_g	
–90	2.8	0.000608	4.41555	–43.29	133.91	–43.28	189.75	146.46	–0.2086	0.8273	–90
–80	6.2	0.000617	2.13835	–34.73	137.82	–34.72	285.74	151.02	–0.1631	0.7984	–80
–70	12.3	0.000627	1.12728	–26.14	141.81	–26.13	181.76	155.64	–0.1198	0.7749	–70
–60	22.6	0.000637	0.63791	–17.50	145.86	–17.49	177.77	160.29	–0.0783	0.7557	–60
–50	39.1	0.000648	0.38310	–8.80	149.95	–8.78	173.73	164.95	–0.0384	0.7401	–50
–45	50.4	0.000654	0.30268	–4.43	152.01	–4.40	171.68	167.28	–0.0190	0.7334	–45
–40	64.2	0.000659	0.24191	–0.04	154.07	0	169.59	169.59	0	0.7274	–40
–35	80.7	0.000666	0.19540	4.37	156.13	4.42	167.48	171.90	0.0187	0.7219	–35
–30	100.4	0.000672	0.15937	8.79	158.28	8.86	165.34	174.09	0.0371	0.7170	–30
–29.8	101.3	0.000673	0.15803	8.98	158.30	9.05	165.24	174.20	0.0379	0.7168	–29.8
–25	123.7	0.000679	0.13117	13.24	160.25	13.33	163.15	176.48	0.0552	0.7126	–25
–20	150.9	0.000685	0.10885	17.71	162.31	17.82	160.92	178.74	0.0731	0.7087	–20
–15	182.6	0.000693	0.09102	22.20	164.35	22.33	158.64	180.97	0.0906	0.7051	–15
–10	219.1	0.000700	0.07665	26.72	166.39	26.87	156.31	183.19	0.1080	0.7019	–10
–5	261.0	0.000708	0.06496	31.26	168.42	31.45	153.93	185.37	0.1251	0.6991	–5
0	308.6	0.000716	0.05539	35.83	170.44	36.05	151.48	187.53	0.1420	0.6965	0
5	362.6	0.000724	0.04749	40.43	172.44	40.69	148.96	189.65	0.1587	0.6942	5
10	423.3	0.000733	0.04091	45.06	174.42	45.37	146.37	191.74	0.1752	0.6921	10
15	491.4	0.000743	0.03541	49.73	176.38	50.10	143.68	193.78	0.1915	0.6902	15
20	567.3	0.000752	0.03078	54.45	178.32	54.87	140.91	195.78	0.2078	0.6884	20
25	651.6	0.000763	0.02685	59.21	180.23	59.70	138.03	197.73	0.2239	0.6868	25
30	744.9	0.000774	0.02351	64.02	182.11	64.59	135.03	199.62	0.2399	0.6853	30
35	847.7	0.000786	0.02064	68.88	183.95	69.55	131.90	201.45	0.2559	0.6839	35
40	960.7	0.000798	0.01817	73.82	185.74	74.59	128.61	203.20	0.2718	0.6825	40
45	1084.3	0.000811	0.01603	78.83	187.49	79.71	125.16	204.87	0.2877	0.6811	45
50	1219.3	0.000826	0.01417	83.93	189.17	84.94	121.51	206.45	0.3037	0.6797	50

Table B.1 *Contd.*

Temp C T	*Press. kPa p*	*Specific Volume, m³/kg*				*Internal Energy, kJ/kg*					*Temp °C T*
		Sat. Liquid v_f	*Sat. Vapour* v_g	*Sat. Liquid* u_f	*Sat. Vapour* u_g	*Sat. Liquid* h_f	*Evap.* h_{fg}	*Sat. Vapour* h_g	*Sat. Liquid* s_f	*Sat. Vapour* s_g	
55	1366.3	0.000841	0.01254	89.12	190.78	90.27	117.65	107.92	0.3197	0.6782	55
60	1525.9	0.000858	0.01111	94.43	192.31	95.74	113.52	209.26	0.3358	0.6765	60
65	1698.8	0.000877	0.00985	99.87	193.73	101.36	109.10	210.46	0.3521	0.6747	65
70	1885.8	0.000997	0.00873	105.46	195.03	107.15	104.33	211.48	0.3686	0.6726	70
75	2087.5	0.000920	0.00772	111.23	196.17	113.15	99.14	212.29	0.3854	0.6702	75
80	2304.6	0.000946	0.00682	117.21	197.11	120.39	93.44	212.83	0.4027	0.6672	80
85	2538.0	0.000976	0.00600	123.45	197.80	125.93	87.11	213.04	0.4204	0.6636	85
90	2788.5	0.001012	0.00526	130.02	198.14	132.84	79.96	212.80	0.4389	0.6590	90
95	3056.9	0.001056	0.00456	137.01	197.99	140.23	71.71	211.94	0.4583	0.6531	95
100	3344.1	0.001113	0.00390	144.59	197.07	148.31	61.81	210.12	0.4793	0.6449	100
105	3650.9	0.001197	0.00324	153.15	194.73	157.52	49.05	206.57	0.5028	0.6325	105
110	3978.5	0.001364	0.00246	164.12	188.20	169.55	28.44	197.99	0.5533	0.6076	110
120.0	4116.8	0.001792	0.00179	176.06	176.06	183.43	0	183.43	0.5689	0.5689	120.0

Table B.2 *Properties of superheated Refrigerant,* ***R-12***

Temp. C	v m³/kg	h kJ/kg	s kJ/kg·K	v m³/kg	h kJ/kg	s kJ/kg·K	v m³/kg	h kJkg	s kJ/kg·K
	25 kPa (– 58.26)			*50 kPa (– 45.18)*			*100 kPa (– 30.10)*		
Sat.	0.58130	161.10	0.7527	0.30515	167.19	0.7336	0.15999	174.15	0.7171
–30	0.66179	176.19	0.8187	0.32738	175.55	0.7691	0.16006	174.21	0.7174
–20	0.69001	181.74	0.8410	0.34186	181.17	0.7917	0.16770	179.99	0.7406
–10	0.71811	187.40	0.8630	0.35623	186.89	0.8139	0.17522	185.84	0.7633
0	0.74613	193.17	0.8844	0.37051	192.70	0.8356	0.18265	191.77	0.7854
10	0.77409	199.03	0.9055	0.38472	198.61	0.8568	0.18999	197.77	0.8070
20	0.80198	204.99	0.9262	0.39886	204.62	0.8776	0.19728	203.85	0.8281
30	0.82982	211.05	0.9465	0.41296	210.71	0.8981	0.20451	210.02	0.8488
40	0.85762	217.20	0.9665	0.42701	216.89	0.9181	0.21169	216.26	0.8691
50	0.88538	223.45	0.9861	0.44103	223.16	0.9378	0.21884	222.58	0.8889
60	0.91312	229.77	1.0054	0.45502	229.51	0.9572	0.22596	228.98	0.9084
70	0.94083	236.19	1.0244	0.46898	235.95	0.9762	0.23305	235.46	0.9276
80	0.96852	242.68	1.0430	0.48292	242.46	0.9949	0.24011	242.01	0.9464
90	0.99618	249.26	1.0614	0.49684	249.05	1.0133	0.24716	248.63	0.9649
100	1.02384	255.91	1.0795	0.51074	255.71	1.0314	0.25419	255.32	0.9831
110	1.05148	262.63	1.0972	0.52463	262.45	1.0493	0.26121	262.08	1.0009
120	1.07910	269.43	1.1148	0.53851	269.26	1.0668	0.26821	268.91	1.0185
	200 kPa (– 12.53)			*300 kPa (– 0.86)*			*400 kPa (– 8.15)*		
Sat.	0.08354	182.07	0.7035	0.05690	187.16	0.6969	0.04321	190.97	0.6928
0	0.08861	189.80	0.7325	0.05715	187.72	0.6989	—	—	—
10	0.09255	196.02	0.7548	0.05998	194.17	0.7222	0.04363	192.21	0.6972
20	0.09642	202.28	0.7766	0.06273	200.64	0.7446	0.04584	198.91	0.7204
30	0.10023	208.60	0.7978	0.06542	207.12	0.7663	0.04797	205.58	0.7428
40	0.10399	214.97	0.8184	0.06805	213.64	0.7875	0.05005	212.25	0.7645
50	0.10771	221.41	0.8387	0.07064	220.19	0.8081	0.05207	218.94	0.7855
60	0.11140	227.90	0.8585	0.07319	226.79	0.8282	0.05406	225.65	0.8060
70	0.11506	234.46	0.8779	0.07571	233.44	0.8479	0.05601	232.40	0.8259
80	0.11869	241.09	0.8969	0.07820	240.15	0.8671	0.05794	239.19	0.8454
90	0.12230	247.77	0.9156	0.08067	246.90	0.8860	0.05985	246.02	0.8645
100	0.12590	254.53	0.9339	0.08313	253.72	0.9045	0.06173	252.89	0.8831
110	0.12948	261.34	0.9519	0.08557	260.58	0.9226	0.06360	259.81	0.9015
120	0.13305	268.21	0.9696	0.08799	267.50	0.9405	0.06546	266.79	0.9194
130	0.13661	275.15	0.9870	0.09041	274.48	0.9580	0.06730	273.81	0.9370
140	0.14016	282.14	1.0042	0.09281	281.51	0.9752	0.06913	280.88	0.9544
150	0.14370	289.19	1.0210	0.09520	288.59	0.9922	0.07095	287.99	0.9714

Table B.2 *Contd.*

Temp. C	*v m³/kg*	*h kJ/kg*	*s kJ/kg·K*	*v m³/kg*	*h kJ/kg*	*s kJ/kg·K*	*v m³/kg*	*h kJ/kg*	*s kJ/kg·K*
	500 kPa (15.60)			*750 kPa (30.26)*			*1000 kPa (41.64)*		
Sat.	0.03482	194.03	0.6899	0.02335	199.72	0.6852	0.01744	203.76	0.6820
30	0.03746	203.96	0.7235	–	–	–	–	–	–
40	0.03921	210.81	0.7457	0.02467	206.91	0.7086	–	–	–
50	0.04091	217.64	0.7672	0.02595	214.18	0.7314	0.01837	210.32	0.7026
60	0.04257	224.68	0.7881	0.02718	221.37	0.7547	0.01941	217.97	0.7259
70	0.04418	231.33	0.8083	0.02837	228.52	0.7745	0.02040	225.49	0.7481
80	0.04577	238.21	0.8281	0.02952	235.65	0.7949	0.02134	232.91	0.7695
90	0.04734	245.11	0.8473	0.03064	242.76	0.8148	0.02225	240.28	0.7900
100	0.04889	242.05	0.8662	0.03174	249.89	0.8342	0.02313	247.61	0.8100
110	0.05041	259.03	0.8847	0.03282	257.03	0.8530	0.02399	254.93	0.8293
120	0.05193	266.06	0.9028	0.03388	264.19	0.8715	0.02483	262.25	0.8482
130	0.05343	273.12	0.9205	0.03493	271.38	0.8895	0.02566	269.57	0.8665
140	0.05492	280.23	0.9379	0.03596	278.59	0.9072	0.02647	276.90	0.8845
150	0.05640	287.39	0.9550	0.03699	285.84	0.9246	0.02728	284.26	0.9021
160	0.05788	294.59	0.9718	0.03801	293.13	0.9416	0.02885	299.04	0.9193
170	0.05934	301.83	0.9884	0.03902	300.45	0.9583	0.02885	299.04	0.9362
180	0.06080	309.12	1.0046	0.04002	307.81	0.9747	0.02963	306.47	0.9528
	500 kPa (59.22)			*2000 kPa (72.88)*			*4000 kPa (110.32)*		
Sat	0.01132	209.06	0.6768	0.00813	211.97	0.6713	0.00239	196.90	0.6046
80	0.01305	226.73	0.7284	0.00870	219.02	0.6914	–	–	–
90	0.01377	234.77	0.7508	0.00941	228.23	0.7171	–	–	–
100	0.01446	242.65	0.7722	0.01003	236.94	0.7408	–	–	–
110	0.01512	250.41	0.7928	0.01061	245.34	0.7630	–	–	–
120	0.01575	258.10	0.8126	0.01116	253.53	0.7841	0.00374	225.18	0.6777
130	0.01636	265.74	0.8318	0.01168	261.58	0.8043	0.00433	238.69	0.7116
140	0.01696	273.35	0.8504	0.01217	269.53	0.8238	0.00478	249.93	0.7392
150	0.01754	280.94	0.8686	0.01265	277.41	0.8426	0.00517	260.12	0.7636
160	0.01811	288.52	0.8863	0.01312	285.24	0.8609	0.00552	269.71	0.7860
170	0.01867	296.11	0.9036	0.01357	293.04	0.8787	0.00585	278.90	0.8069
180	0.01922	303.70	0.9205	0.01401	300.82	0.8961	0.00615	287.82	0.8269
190	0.01977	311.31	0.9371	0.01445	308.59	0.9131	0.00643	296.55	0.8459
200	0.02031	318.93	0.9534	0.01488	316.36	0.9297	0.00671	305.14	0.8642
210	0.02084	326.58	0.9694	0.01530	324.14	0.9459	0.00697	313.61	0.8820
220	0.02137	334.24	0.9851	0.01572	331.92	0.9619	0.00723	322.01	0.8992

Table B.3 *Properties of Saturated* **Refrigerant 22** *(Liquid–Vapour): Temperature entry*

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg·K		
	Sat.	Sat.	Sat.	Sat.	Sat.	Sat.	Sat.	Sat.	Sat.	Sat.	
Temp. °C	Press. bar	Liquid $v_f \times 10^3$	Vapour v_g	Liquid u_f	Vapour u_g	Liquid h_f	Evap. h_{fg}	Vapour h_g	Liquid s_f	Vapour s_g	Temp. °C
–60	0.3749	0.6833	0.5370	–21.57	203.67	–21.55	245.35	223.81	–0.0964	1.0547	–60
–50	0.6451	0.6966	0.3239	–10.89	207.70	–10.85	239.44	228.60	–0.0474	1.0256	–50
–45	0.8290	0.7037	0.2564	–5.50	209.70	–5.44	236.39	230.95	–0.0235	1.0126	–45
–40	1.0522	0.7109	0.2052	–0.07	211.68	0.00	233.27	233.27	0.0000	1.0005	–40
–36	1.2627	0.7169	0.1730	4.29	213.25	4.38	230.71	235.09	0.0186	0.9914	–36
–32	1.5049	0.7231	0.1468	8.68	214.80	8.79	228.10	236.89	0.0369	0.9828	–32
–30	1.6389	0.7262	0.1355	10.88	215.58	11.00	226.77	237.78	0.0460	0.9787	–30
–28	1.7819	0.7294	0.1252	13.09	216.34	13.22	225.43	238.66	0.0551	0.9746	–28
–26	1.9345	0.7327	0.1159	15.31	217.11	15.45	224.08	239.53	0.0641	0.9707	–26
–22	2.2698	0.7383	0.0997	19.76	218.62	19.92	221.32	241.24	0.0819	0.9631	–22
–20	2.4534	0.7427	0.0926	21.99	219.37	22.17	219.91	242.09	0.0908	0.9595	–20
–18	2.6482	0.7462	0.0861	24.23	220.11	24.43	218.49	242.92	0.0996	0.9559	–18
–16	2.8547	0.7497	0.0802	26.48	220.85	26.69	217.05	243.74	0.1084	0.9525	–16
–14	3.0733	0.7533	0.0748	28.73	221.58	28.97	215.59	244.56	0.1171	0.9490	–14
–12	3.3044	0.7569	0.0698	31.00	222.30	31.25	214.11	245.36	0.1258	0.9457	–12
–10	3.5485	0.7606	0.0652	33.27	223.02	33.54	212.62	246.15	0.1345	0.9424	–10
–8	3.8062	0.7644	0.0610	35.54	223.73	35.83	211.10	246.93	0.1431	0.9392	–8
–6	4.0777	0.7683	0.0571	37.83	224.43	38.14	209.56	247.70	0.1517	0.9361	–6
–4	4.3638	0.7722	0.0535	40.12	225.13	40.46	208.00	248.45	0.1602	0.9330	–4
–2	4.6647	0.7762	0.0501	42.42	225.82	42.78	206.41	249.20	0.1688	0.9300	–2
0	4.9811	0.7803	0.0470	44.73	226.50	45.12	204.81	249.92	0.1773	0.9271	0
2	5.3133	0.7844	0.0442	47.04	227.17	47.46	203.18	250.64	0.1858	0.9241	2
4	5.6619	0.7887	0.0415	49.37	227.83	49.82	201.52	251.34	0.1941	0.9213	4
6	6.0275	0.7930	0.0391	51.71	228.48	52.18	199.84	252.02	0.2025	0.9184	6
8	6.4105	0.7974	0.0368	54.05	229.13	54.56	198.14	252.70	0.2109	0.9157	8
10	6.8113	0.8020	0.0346	56.40	229.76	56.95	196.40	253.35	0.2193	0.9129	10
12	7.2307	0.8066	0.0326	58.77	230.38	59.35	194.64	253.99	0.2276	0.9102	12
16	8.1268	0.8162	0.0291	63.53	231.59	64.19	191.02	255.21	0.2442	0.9048	16
20	9.1030	0.8263	0.0259	68.33	232.76	69.09	187.28	256.37	0.2607	0.8996	20
24	10.164	0.8369	0.0232	73.19	233.87	74.04	183.40	257.44	0.2772	0.8944	24
28	11.313	0.8480	0.0208	78.09	234.92	79.05	179.37	258.43	0.2936	0.8893	28
32	12.556	0.8599	0.0186	83.06	235.91	84.14	175.18	259.32	0.3101	0.8842	32
36	13.897	0.8724	0.0168	88.08	236.83	89.29	170.82	260.11	0.3265	0.8790	36
40	15.341	0.8858	0.0151	93.18	237.66	94.53	166.25	260.79	0.3429	0.8738	40
45	17.298	0.9039	0.0132	99.65	238.59	101.21	160.24	261.46	0.3635	0.8672	45
50	19.433	0.9238	0.0116	106.26	239.34	108.06	153.84	261.90	0.3842	0.8603	50
60	24.281	0.9705	0.0089	120.00	240.24	122.35	139.61	261.96	0.4264	0.8455	60

Table B.4 *Properties of Saturated **Refrigerant 22** (Liquid–Vapour): Pressure entry*

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg·K		
Press. bar	Sat. Temp. °C	Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Sat. Vapour s_g	Press. bar
0.40	–58.86	0.6847	0.5056	–20.36	204.13	–20.34	244.69	224.36	–0.0907	1.0512	0.40
0.50	–54.83	0.6901	0.4107	–16.07	205.76	–16.03	242.33	226.30	–0.0709	1.0391	0.50
0.60	–51.40	0.6947	0.3466	–12.39	207.14	–12.35	240.28	227.93	–0.0542	1.0294	0.60
0.70	–48.40	0.6989	0.3002	–9.17	208.34	–9.12	238.47	229.35	–0.0397	1.0213	0.70
0.80	–45.73	0.7026	0.2650	–6.28	209.41	– 6.23	236.84	230.61	–0.0270	1.0144	0.80
0.90	–43.30	0.7061	0.2374	–3.66	210.37	–3.60	235.34	231.74	–0.0155	1.0084	0.90
1.00	–41.09	0.7093	0.2152	–1.26	211.25	–1.19	233.95	232.77	–0.0051	1.0031	1.00
1.25	–36.23	0.7166	0.1746	4.04	213.16	4.14	230.86	234.99	0.0175	0.9919	1.25
1.50	–32.08	0.7230	0.1472	8.60	214.77	8.70	228.15	236.86	0.0366	0.9830	1.50
1.75	–28.44	0.7287	0.1274	12.61	216.18	12.74	225.73	238.47	0.0531	0.9755	1.75
2.00	–25.18	0.7340	0.1123	16.22	217.42	16.37	223.52	239.88	0.0678	0.9691	2.00
2.25	–22.22	0.7389	0.1005	19.51	218.53	19.67	221.47	241.15	0.0809	0.9636	2.25
2.50	–19.51	0.7436	0.0910	22.54	219.55	22.72	219.57	242.29	0.0930	0.9586	2.50
2.75	–17.00	0.7479	0.0831	25.36	220.48	25.56	217.77	243.33	0.1040	0.9542	2.75
3.00	– 14.66	0.7721	0.0765	27.99	221.34	28.22	216.07	244.29	0.1143	0.9502	3.00
3.25	–12.46	0.7561	0.0709	30.47	222.13	30.72	214.46	245.18	0.1238	0.9465	3.25
3.50	–10.39	0.7599	0.0661	32.82	222.88	33.09	212.91	246.00	0.1328	0.9431	3.50
3.75	–8.43	0.7636	0.0618	35.06	223.58	35.34	211.42	246.77	0.1423	0.9399	3.75
4.00	–6.56	0.7672	0.0581	37.18	224.24	37.49	209.99	247.48	0.1493	0.9370	4.00
4.25	–4.78	0.7706	0.0548	39.22	224.86	39.55	208.61	248.16	0.1569	0.9342	4.25
4.50	–3.08	0.7740	0.0519	41.17	225.45	41.52	207.27	248.80	0.1642	0.9316	4.50
4.75	–1.45	0.7773	0.0492	43.05	226.00	43.42	205.98	249.40	0.1711	0.9292	4.75
5.00	0.12	0.7805	0.0469	44.86	226.54	45.25	204.71	249.97	0.1777	0.9269	5.00
5.25	1.63	0.7836	0.0447	46.61	227.04	47.02	203.48	250.51	0.1841	0.9247	5.25
5.50	3.08	0.7867	0.0427	48.30	227.53	48.74	202.28	251.02	0.1903	0.9226	5.50
5.75	4.49	0.7897	0.0409	49.94	227.99	50.40	201.11	251.51	0.1962	0.9206	5.75
6.00	5.85	0.7927	0.0392	51.53	228.44	52.01	199.97	251.98	0.2019	0.9186	6.00
7.00	10.91	0.8041	0.0337	57.48	230.04	58.04	195.60	253.64	0.2231	0.9117	7.00
8.00	15.45	0.8149	0.2095	62.88	231.43	63.53	191.52	255.05	0.2419	0.9056	8.00
9.00	19.59	0.8252	0.0262	67.84	231.64	68.59	187.67	256.25	0.2591	0.9001	9.00
10.00	23.40	0.8352	0.0236	72.46	233.71	73.30	183.99	257.28	0.2748	0.8952	10.00
12.00	30.25	0.8546	0.0195	80.87	235.48	81.90	177.04	258.94	0.3029	0.8864	12.00
14.00	36.29	0.8734	0.0166	88.45	236.89	89.68	170.49	260.16	0.3277	0.8786	14.00
16.00	41.73	0.8919	0.0144	95.41	238.00	96.83	164.21	261.04	0.3500	0.8715	16.00
18.00	46.89	0.9104	0.0127	101.87	238.86	103.51	158.13	261.64	0.3705	0.8649	18.00
20.00	51.26	0.9291	0.0112	107.95	239.51	109.81	152.17	261.98	0.3895	0.8586	20.00
24.00	59.46	0.9677	0.0091	119.24	240.22	121.56	140.43	261.99	0.4241	0.8463	24.00

Table B.5 *Properties of supreheated* ***Refrigerant 22*** *Vapour*

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 0.4 bar = 0.04 MPa (T_{sat} = – 58.86 °C)				*p = 0.6 bar = 0.06 MPa* (T_{sat} = – 51.40 °C)			
Sat	0.50559	204.13	224.36	1.0512	0.34656	207.14	227.93	1.0294
–55	0.51532	205.92	226.53	1.0612				
–50	0.52787	208.26	229.38	1.0741	0.34895	207.80	228.74	1.0330
–45	0.54037	210.63	232.24	1.0868	0.35747	210.20	231.65	1.0459
–40	0.55284	213.02	235.13	1.0993	0.36594	212.62	234.58	1.0586
–35	0.56526	215.43	238.05	1.1117	0.37437	215.06	237.52	1.0711
–30	0.57766	217.88	240.99	1.1239	0.38277	217.53	240.49	1.0835
–25	0.59002	220.35	243.95	1.1360	0.39114	220.02	243.49	1.0956
–20	0.60236	222.85	246.95	1.1479	0.39948	222.54	246.51	1.1077
–15	0.61468	225.38	249.97	1.1597	0.40779	225.08	249.55	1.1196
–10	0.62697	227.93	253.01	1.1714	0.41608	227.65	252.62	1.1314
–5	0.63925	230.52	256.09	1.1830	0.42436	230.25	255.71	1.1430
0	0.65151	233.13	259.19	1.1944	0.43261	232.88	258.83	1.1545
	p = 0.8 bar = 0.08 MPa (T_{sat} = – 45.73°C)				*p = 1.0 bar = 0.10 MPa* (T_{sat} = – 41.9°C)			
Sat	0.26503	209.41	230.61	1.0144	0.21518	211.25	232.77	1.0031
–45	0.26597	209.76	231.04	1.0163				
–40	0.27245	212.21	234.01	1.0292	0.21633	211.79	233.42	1.0059
–35	0.27890	214.68	236.99	1.0418	0.22158	214.29	236.44	1.0187
–30	0.28530	217.17	239.99	1.0543	0.22679	216.80	239.48	1.0313
–25	0.29167	219.68	243.02	1.0666	0.23197	219.34	242.54	1.0438
–20	0.29801	222.22	246.06	1.0788	0.23712	221.90	245.61	1.0560
–15	0.30433	224.78	249.13	1.0908	0.24334	224.48	248.70	1.0681
–10	0.31062	227.37	252.13	1.1026	0.24734	227.08	251.82	1.0801
–5	0.31690	229.98	255.34	1.1143	0.25241	229.71	254.95	1.0919
0	0.32315	232.62	258.47	1.1259	0.25747	232.36	258.11	1.1035
5	0.32939	235.29	261.64	1.1374	0.26251	235.04	281.29	1.1151
10	0.33561	237.98	264.83	1.1488	0.26753	237.74	264.50	1.1265
	p = 1.5 bar = 0.15 MPa (T_{sat} = – 32.08°C)				*p = 2.0 bar = 0.20 MPa* (T_{sat} = – 25.18°C)			
Sat	0.14721	214.77	236.86	0.9830	0.11232	217.42	239.88	0.9691
–30	0.14872	215.85	238.16	0.9883				
–25	0.15232	218.45	241.30	1.0011	0.11242	217.51	240.00	0.9696
–20	0.15588	221.07	244.45	1.0137	0.11520	220.19	243.23	0.9825
–15	0.15941	223.70	247.61	1.0260	0.11795	222.88	246.47	0.9952
–10	0.16292	226.35	250.78	1.0382	0.12067	225.48	249.72	1.0076
–5	0.16640	229.02	253.98	1.0502	0.12336	228.30	252.97	1.0199
0	0.16987	231.70	257.18	1.0621	0.12603	231.03	256.23	1.0310
5	0.17331	234.42	260.41	1.0738	0.12868	233.78	259.51	1.0438
10	0.17674	237.15	263.66	1.0854	0.13132	236.54	262.81	1.0555
15	0.18015	239.91	266.93	1.0968	0.13393	239.33	266.12	1.0671
20	0.18355	242.69	270.22	1.1081	0.13653	242.14	269.44	1.0786
25	0.18693	245.49	273.53	1.1193	0.13912	244.97	272.79	1.0899

Table B.5 *Contd.*

T °C	*v* m³/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m³/kg	*u* kJ/kg	*h* kJkg	*s* kJ/kg·K
	p = 2.5 bar = 0.25 MPa (T_{sat} = – 19.51°C)				*p* = 3.0 bar = 0.30 MPa (T_{sat} = – 14.66°C)			
Sat	0.09097	219.55	242.29	0.9586	0.07651	221.34	244.29	0.9502
–15	0.09303	222.03	245.29	0.9703				
–10	0.09528	224.79	248.61	0.9831	0.07833	223.96	247.46	0.9623
–5	0.09751	227.55	251.93	0.9956	0.08025	226.78	250.86	0.9751
0	0.09971	230.33	255.26	1.0078	0.08214	229.61	254.25	0.9876
5	0.10189	233.12	258.59	1.0199	0.08400	232.44	257.64	0.9999
10	0.10405	235.92	261.93	1.0318	0.08585	235.28	261.04	1.0120
15	0.10619	238.74	265.29	1.0436	0.08767	238.14	264.44	1.0239
20	0.10831	241.58	268.66	1.0552	0.08949	241.01	267.85	1.0357
25	0.11043	244.44	272.04	1.0666	0.09128	243.89	271.28	1.0472
30	0.11253	247.31	275.44	1.0779	0.09307	246.80	274.72	1.0587
35	0.11461	250.21	278.86	1.0891	0.09484	249.72	278.17	1.0700
40	0.11669	253.13	282.30	1.1002	0.09660	252.66	281.64	1.0811
	p = 3.5 bar = 0.35 MPa (T_{sat} = – 10.39°C)				*p* = 4.0 bar = 0.40 MPa (T_{sat} = – 6.56°C)			
Sat	0.06605	222.88	246.00	0.9431	0.05812	224.24	247.48	0.9370
–10	0.06619	223.10	246.27	0.9441				
–5	0.06789	225.99	249.75	0.9572	0.05860	225.16	248.60	0.9411
0	0.06956	228.86	253.21	0.9700	0.06011	228.09	252.14	0.9542
5	0.07121	231.74	256.67	0.9825	0.06160	231.02	255.66	0.9670
10	0.07284	234.63	260.12	0.9948	0.06306	233.95	259.10	0.9795
15	0.07444	237.52	263.57	1.0069	0.06450	236.89	262.69	0.9918
20	0.07603	240.42	267.03	1.0188	0.06592	239.83	266.19	1.0039
25	0.07760	243.34	270.50	1.0305	0.06733	242.77	269.71	1.0158
30	0.07916	246.27	273.97	1.0421	0.06872	245.73	273.22	1.0274
35	0.08070	249.22	277.46	1.0535	0.07010	248.71	276.75	1.0390
40	0.08224	252.18	280.97	1.0648	0.07146	251.70	280.28	1.0504
45	0.08376	255.17	284.48	1.0759	0.07282	254.70	283.83	1.0616
	p = 4.5 bar = 0.45 MPa (T_{sat} = – 3.08°C)				*p* = 5.0 bar = 0.50 MPa (T_{sat} = –0.12°C)			
Sat	0.05189	225.45	248.80	0.9316	0.04686	226.54	249.97	0.9269
0	0.05275	227.29	251.03	0.9399				
5	0.05411	230.28	254.63	0.9529	0.04810	229.52	253.57	0.9399
10	0.05545	233.26	258.21	0.9657	0.04934	232.55	257.22	0.9530
15	0.05676	236.24	261.78	0.9782	0.05056	235.57	260.85	0.9657
20	0.05805	239.22	265.34	0.9904	0.05175	238.59	264.47	0.9781
25	0.05933	242.20	268.90	1.0025	0.05293	241.61	268.07	0.9903
30	0.06059	245.19	272.46	1.0143	0.05409	244.63	271.68	1.0023
35	0.06184	248.19	276.02	1.0259	0.05523	247.66	275.28	1.0141
40	0.06308	251.20	279.59	1.0374	0.05636	250.70	282.50	1.0257
45	0.06430	254.23	283.17	1.0488	0.05748	253.76	282.50	1.0371
50	0.06552	257.28	286.76	1.0600	0.05859	256.82	286.12	1.0484
55	0.06672	260.34	290.36	1.0710	0.05969	259.90	289.75	1.0595

Table B.5 *Contd.*

T °C	*v* m³/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m³/kg	*u* kJ/kg	*h* kJkg	*s* kJ/kg·K
	p = 5.5 bar = 0.55 MPa (T_{sat} = 3.08°C)				*p* = 6.0 bar = 0.60 MPa (T_{sat} = 5.8°C)			
Sat.	0.04271	227.53	251.02	0.9226	0.03923	228.44	251.98	0.9186
5	0.04317	228.72	252.46	0.9278				
10	0.04433	231.81	256.20	0.9411	0.04015	231.05	255.14	0.9299
15	0.04547	234.89	259.90	0.9540	0.04122	234.18	258.91	0.9431
20	0.04658	237.95	263.57	0.9667	0.04227	237.29	262.65	0.9560
25	0.04768	241.01	267.23	0.9790	0.04330	240.39	266.37	0.9685
30	0.04875	244.07	270.88	0.9912	0.04431	243.49	270.07	0.9808
35	0.04982	247.13	274.53	1.0031	0.04530	246.58	273.76	0.9929
40	0.05086	250.20	278.17	1.0148	0.04628	249.68	277.45	1.0048
45	0.05190	253.27	281.82	1.0264	0.04724	252.78	281.13	1.0164
50	0.05293	256.36	285.47	1.0378	0.04820	255.90	284.82	1.0279
55	0.05394	259.46	289.13	1.0490	0.04914	259.02	288.51	1.0393
60	0.05495	262.58	292.80	1.0601	0.05008	262.15	292.20	1.0504
	p = 7.0 bar = 0.70 MPa (T_{sat} = 10.91°C)				*p* = 8.0 bar = 0.80 MPa (T_{sat} = 15.45°C)			
Sat.	0.03371	230.04	253.64	0.9117	0.02953	231.43	255.05	0.9056
15	0.03451	232.70	256.86	0.9229				
20	0.03547	235.92	260.75	0.9363	0.03033	234.47	258.74	0.9182
25	0.03639	239.12	264.59	0.9493	0.03118	237.76	262.70	0.9315
30	0.03730	242.29	268.40	0.9619	0.03202	241.04	266.66	0.9448
35	0.03819	245.46	272.19	0.9743	0.03283	244.28	270.54	0.9574
40	0.03906	248.62	275.96	0.9865	0.03363	247.52	274.42	0.9700
45	0.03992	251.78	279.72	0.9984	0.03440	250.74	278.26	0.9821
50	0.04076	254.94	283.48	1.0101	0.03517	253.96	282.10	0.9941
55	0.04160	258.11	287.23	1.0216	0.03592	257.18	285.92	1.0058
60	0.04242	261.29	290.99	1.0330	0.03667	260.40	289.74	1.0174
65	0.04324	264.48	294.75	1.0442	0.03741	263.64	293.56	1.0287
70	0.04405	267.68	298.51	1.0552	0.03814	266.87	297.38	1.0400
	p = 9.0 bar = 0.90 MPa (T_{sat} = 19.59°C)				*p* = 10.0 bar = 1.00 MPa (T_{sat} = 23.40°C)			
Sat.	0.02623	232.64	256.25	0.9001	0.02358	233.71	257.28	0.8952
20	0.02630	232.92	256.59	0.9013				
30	0.02789	239.73	264.83	0.9289	0.02457	238.34	262.91	0.9139
40	0.02939	246.37	272.82	0.9549	0.02598	245.18	271.17	0.9407
50	0.03082	252.95	280.68	0.9795	0.02732	251.90	279.22	0.9660
60	0.03219	259.49	288.46	1.0033	0.02860	258.56	287.15	0.9902
70	0.03353	266.04	296.21	1.0262	0.02984	265.19	295.03	1.0135
80	0.03483	272.62	303.96	1.0484	0.03104	271.84	302.88	1.0361
90	0.03611	279.23	311.73	1.0701	0.03221	278.52	310.74	1.0580
100	0.03736	285.90	319.53	1.0913	0.03337	285.24	318.61	1.0794
110	0.03860	292.63	327.37	1.1120	0.03450	292.02	326.52	1.1003
120	0.03982	299.42	335.26	1.1323	0.03562	298.85	334.46	1.1207
130	0.04103	306.28	343.21	1.1523	0.03672	305.74	342.46	1.1408
140	0.04323	313.21	351.22	1.1719	0.03781	312.70	350.51	1.1605
150	0.04342	320.21	359.29	1.1912	0.03889	319.74	358.63	1.1790

Table B.5 *Contd.*

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m³/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 12.0 bar = 1.20 MPa (T_{sat} = 30.25°C)				p = 14.0 bar = 1.40 MPa (T_{sat} = 36.29°C)			
Sat.	0.01955	235.48	258.94	0.8864	0.01662	236.89	260.16	0.8786
40	0.02083	242.63	267.62	0.9146	0.01708	239.78	263.70	0.8900
50	0.02204	249.69	276.14	0.9413	0.01823	247.29	272.81	0.9186
60	0.02319	256.60	284.43	0.9666	0.01929	254.52	281.53	0.9452
70	0.02428	263.44	292.58	0.9907	0.02029	261.60	290.01	0.9703
80	0.02534	270.25	300.66	1.0139	0.02125	268.60	298.34	1.9942
90	0.02636	277.07	308.70	1.0363	0.02217	275.56	306.60	1.0172
100	0.02736	283.90	316.73	1.0582	0.02306	282.52	314.80	1.0395
110	0.02834	290.77	324.78	1.0794	0.02393	289.49	323.00	1.0612
120	0.02930	297.69	332.85	1.1002	0.02478	296.50	331.19	1.0823
130	0.03024	304.65	340.95	1.1205	0.02562	303.55	339.41	1.1029
140	0.03118	311.68	349.09	1.1405	0.02644	310.64	347.65	1.1231
150	0.03210	318.77	357.29	1.1601	0.02725	317.79	355.94	1.1429
160	0.03301	325.92	365.54	1.1793	0.02805	324.99	364.26	1.1624
170	0.03392	333.14	373.84	1.1983	0.02884	332.26	372.64	1.1815
	p = 16.0 bar = 1.60 MPa (T_{sat} = 41.73°C)				p = 18.0 bar = 1.80 MPa (T_{sat} = 46.69°C)			
Sat.	0.01440	238.00	261.04	0.8715	0.01265	238.86	261.64	0.8649
50	0.01533	244.66	269.18	0.8971	0.01301	241.72	265.14	0.8484
60	0.01634	252.29	278.43	0.9252	0.01401	249.86	275.09	0.9061
70	0.01728	259.65	287.30	0.9515	0.01492	257.57	284.43	0.9337
80	0.01817	266.86	275.93	0.9762	0.01576	265.04	293.40	0.9595
90	0.01901	274.00	304.42	0.9999	0.01655	272.37	302.16	0.9839
100	0.01983	281.09	312.82	1.0228	0.01731	279.62	310.77	1.0073
110	0.02062	288.18	321.17	1.0448	0.01804	286.83	319.30	1.0299
120	0.02139	295.28	329.51	1.0663	0.01874	294.04	327.78	1.0517
130	0.02214	302.41	337.84	1.0872	0.01943	301.26	336.24	1.0730
140	0.02288	309.58	346.19	1.1077	0.02011	308.50	344.70	1.0937
150	0.02361	316.79	354.56	1.1277	0.02077	315.78	353.17	1.1139
160	0.02432	324.05	362.97	1.1473	0.02142	323.10	361.66	1.1338
170	0.02503	331.37	371.42	1.1666	0.02207	330.47	370.19	1.1532
	p = 16.0 bar = 2.00 MPa (T_{sat} = 51.26°C)				p = 24.0 bar = 2.4 MPa (T_{sat} = 59.46°C)			
Sat.	0.01124	239.51	261.98	0.8586	0.00907	240.22	261.99	0.8463
60	0.01212	247.20	271.43	0.8873	0.00913	240.78	262.68	0.8484
70	0.01300	255.35	281.36	0.9167	0.01006	250.30	274.43	0.8831
80	0.01381	263.12	290.74	0.9436	0.01085	258.89	284.93	0.9133
90	0.01457	270.67	299.80	0.9689	0.01156	267.01	294.75	0.9407
100	0.01528	278.09	308.65	0.9929	0.01222	274.85	304.18	0.9663
110	0.01596	285.44	317.37	1.0160	0.01284	282.53	313.35	0.9906
120	0.01663	292.76	326.01	1.0383	0.01343	290.11	322.35	1.0137
130	0.01727	300.08	334.61	1.0598	0.01400	297.64	331.25	1.0361
140	0.01789	307.40	343.19	1.0808	0.01456	305.14	340.08	1.0577
150	0.01850	314.75	351.76	1.1013	0.01509	312.64	348.87	1.0787
160	0.01910	322.14	360.34	1.1214	0.01562	320.16	357.64	1.0992
170	0.01969	329.56	368.95	1.1410	0.01613	327.70	366.41	1.1192
180	0.02027	337.03	377.58	1.1603	0.01663	336.27	375.20	1.1388

Table B.6 *Properties of **Saturated Refrigerant 134a** (Liquid-Vapour): Temperature Entry*

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy $kJ/kg \cdot K$		
Temp. °C	Sat. Press. bar	Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Sat. Vapour s_g	Temp. °C
–40	0.5164	0.7055	0.3569	–0.04	204.45	0.00	222.88	222.88	0.0000	0.9560	–40
–36	0.6332	0.7113	0.2947	4.68	206.73	4.73	220.67	225.40	0.0201	0.9506	–36
–32	0.7704	0.7172	0.2451	9.47	209.01	9.52	218.37	227.90	0.0401	0.9456	–32
–28	0.9305	0.7233	0.2052	14.31	211.29	14.37	216.01	230.38	0.0600	0.9411	–28
–26	1.0199	0.7265	0.1882	16.75	212.43	16.82	214.80	231.62	0.0699	0.9390	–26
–24	1.1160	0.7296	0.1728	19.21	213.57	19.29	213.57	232.85	0.0798	0.9370	–24
–22	1.2192	0.7328	0.1590	21.68	214.70	21.77	212.32	234.08	0.0897	0.9351	–22
–20	1.3299	0.7361	0.1464	24.17	215.84	24.26	211.05	235.31	0.0996	0.9332	–20
–18	1.4483	0.7395	0.1350	26.67	216.97	26.77	209.76	236.53	0.1094	0.9315	–18
–16	1.5748	0.7428	0.1247	29.18	218.10	29.30	208.45	237,74	0.1192	0.9298	–16
–12	1.8540	0.7498	0.1068	34.25	220.36	34.39	205.77	240.15	0.1388	0.9267	–12
–8	2.1704	0.7569	0.0919	39.38	222.60	39.65	203.00	242.54	0.1583	0.9239	–8
–4	2.5274	0.7644	0.0794	44.56	224.84	44.75	200.15	244.90	0.1777	0.9213	–4
0	2.9282	0.7721	0.0689	49.79	227.06	50.02	197.21	247.23	0.1970	0.9190	0
4	3.3765	0.7801	0.0600	55.08	229.27	55.35	194.19	249.53	0.2162	0.9169	4
8	3.8756	0.7884	0.0525	60.43	231.46	60.73	191.07	251.80	0.2354	0.9150	8
12	4.4294	0.7971	0.0460	65.83	233.63	66.18	187.85	254.03	0.2505	0.9132	12
16	5.0416	0.8062	0.0405	71.29	235.78	71.69	184.52	256.22	0.2735	0.9116	16
20	5.7160	0.8157	0.0358	76.80	237.91	77.26	181.09	258.36	0.2924	0.9102	20
24	6.4566	0.8257	0.0317	82.37	240.01	82.90	177.55	260.45	0.3113	0.9089	24
26	6.8530	0.8309	0.0298	85.18	241.05	85.75	175.73	261.48	0.3208	0.9082	26
28	7.2675	0.8362	0.0281	88.00	242.08	88.61	173.89	262.50	0.3302	0.9076	28
30	7.7006	0.8417	0.0265	90.84	243.10	91.49	172.00	263.50	0.3396	0.9070	30
32	8.1528	0.8473	0.0250	93.70	244.12	94.39	170.09	264.48	0.3490	0.9064	32
34	8.6247	0.8530	0.0236	96.38	245.12	97.31	168.14	265.45	0.3584	0.9058	34
36	9.1168	0.8590	0.0223	99.47	246.11	100.25	166.15	266.40	0.3678	0.9053	36
38	9.6298	0.8651	0.0210	102.38	247.09	103.21	164.12	267.33	0.3772	0.9047	38
40	10.164	0.8714	0.0199	105.30	248.06	106.19	162.05	268.24	0.3866	0.9041	40
42	10.720	0.8780	0.0188	108.25	249.02	109.19	159.94	269.14	0.3960	0.9035	42
44	11.299	0.8847	0.0177	111.22	249.96	112.22	157.79	270.01	0.4054	0.9030	44
48	12.526	0.8989	0.0159	117.22	251.79	118.35	153.33	271.68	0.4243	0.9017	48
52	13.851	0.9142	0.0142	123.31	253.55	124.58	148.66	273.24	0.4432	0.9004	52
56	15.278	0.9308	0.0127	129.51	255.23	130.93	143.75	271.68	0.4622	0.8990	56
60	16.813	0.9488	0.0114	135.83	256.81	137.42	138.57	275.99	0.4814	0.8973	60
70	21.162	1.0026	0.0086	152.22	260.15	154.34	124.08	278.43	0.5302	0.8918	70
80	26.324	1.0766	0.0054	169.88	262.14	172.71	106.41	279.12	0.5814	0.8827	80
90	32.435	1.1949	0.0046	189.82	261.34	193.69	82.63	276.32	0.6380	0.8655	90
100	39.742	1.5443	0.0027	218.60	248.49	224.74	34.40	259.13	0.7196	0.8117	100

Table B.7 *Properties of* ***Saturated refrigerant 134a*** *(Liquid–Vapour): Pressure Entry*

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy $kJ/kg \cdot K$		
Press. bar	Sat. Temp. °C	Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Sat. Vapour s_g	Press. bar
0.6	–37.07	0.7097	0.3100	3.41	206.12	3.46	221.27	224.72	0.0147	0.9520	0.6
0.8	–31.21	0.7184	0.2366	10.41	209.46	10.47	217.92	228.39	0.0440	0.9447	0.8
1.0	–26.43	0.7258	0.1917	16.22	212.18	16.29	215.06	231.35	0.0678	0.9395	1.0
1.2	–22.36	0.7323	0.1614	21.23	214.50	21.32	212.54	233.86	0.0879	0.9354	1.2
1.4	–18.80	0.7381	0.1395	25.66	216.52	25.77	210.27	236.04	0.1055	0.9322	1.4
1.6	–15.62	0.7435	0.1229	29.66	218.32	29.78	208.19	237.97	0.1211	0.9295	1.6
1.8	–12.73	0.7485	0.1098	33.31	219.94	33.45	206.26	239.71	0.1352	0.9273	1.8
2.0	–10.09	0.7532	0.0993	36.69	221.43	36.84	204.46	241.30	0.1481	0.9253	2.0
2.4	–5.37	0.7618	0.0834	42.77	224.07	42.95	201.14	244.09	0.1710	0.9222	2.4
2.8	–1.23	0.7697	0.0719	48.18	226.38	48.39	198.13	246.52	0.1911	0.9197	2.8
3.2	2.48	0.7701	0.0632	53.06	228.43	53.31	195.35	248.66	0.2089	0.9177	3.2
3.6	5.84	0.7839	0.0564	57.54	230.28	57.82	192.76	250.58	0.2251	0.9160	3.6
4.0	8.93	0.7904	0.0509	61.69	231.97	62.00	190.32	252.32	0.2399	0.9145	4.0
5.0	15.74	0.8056	0.0409	70.93	235.64	71.33	184.74	256.07	0.2723	0.9117	5.0
6.0	21.58	0.8196	0.0341	78.99	238.74	79.48	179.71	259.19	0.2999	0.9097	6.0
7.0	26.72	0.8328	0.0292	86.19	241.42	86.78	175.07	261.85	0.3242	0.9080	7.0
8.0	31.33	0.8454	0.0255	92.75	243.78	93.42	170.73	264.15	0.3459	0.9066	8.0
9.0	35.53	0.8576	0.0226	98.79	245.88	99.56	166.62	266.18	0.3656	0.9054	9.0
10.0	39.39	0.8695	0.0202	104.42	247.77	105.29	162.68	267.97	0.3838	0.9043	10.0
12.0	46.32	0.8928	0.0166	114.69	251.03	115.76	155.23	270.89	0.4164	0.9023	12.0
14.0	52.43	0.9159	0.0140	123.98	253.74	125.26	148.14	273.40	0.4453	0.9003	14.0
16.0	57.92	0.9392	0.0121	132.52	256.00	134.02	141.31	275.53	0.4714	0.8982	16.0
18.0	62.91	0.9631	0.0105	140.49	257.88	142.22	134.60	276.83	0.4954	0.8959	18.0
20.0	67.49	0.9878	0.0093	148.02	259.41	149.99	127.95	277.94	0.5178	0.8934	20.0
25.0	77.59	1.0562	0.0069	165.48	261.84	168.22	111.06	279.17	0.5686	0.8854	25.0
30.0	86.22	1.1416	0.0053	181.88	262.16	185.30	92.71	278.01	0.6156	0.8735	30.0

Table B.8 *Properties of* ***Superheated Refrigerant 134a*** *Vapour*

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 0.6 bar = 0.06 MPa (T_{sat} = – 37.07°C)				p = 1.0 bar = 0.10 MPa (T_{sat} = – 26.43°C)			
Sat.	0.31003	206.12	224.72	0.9520	0.19170	212.18	231.35	0.9395
–20	0.33536	217.86	237.98	1.0062	0.19770	216.77	236.54	0.9602
–10	0.34992	224.97	245.96	1.0371	0.20686	224.70	244.70	0.9918
0	0.36433	232.24	254.10	1.0675	0.21587	231.41	252.99	1.0227
10	0.37861	239.69	262.41	1.0973	0.22473	238.96	261.43	1.0531
20	0.39279	247.32	270.89	1.1267	0.23349	246.67	270.02	1.0829
30	0.40688	255.12	279.53	1.1557	0.24216	254.54	278.76	1.1122
40	0.42091	263.10	288.35	1.1844	0.25076	262.58	287.66	1.1411
50	0.43487	271.25	297.34	1.2126	0.25930	270.79	296.72	1.1696
60	0.44879	279.58	306.51	1.2405	0.26779	279.16	305.94	1.1977
70	0.46266	288.08	315.84	1.2681	0.27623	287.70	315.32	1.2254
80	0.47650	296.75	325.34	1.2954	0.28464	296.40	324.87	1.2528
90	0.49031	305.58	335.00	1.3224	0.29302	305.27	334.57	1.2799
	p = 1.4 bar = 0.14 MPa (T_{sat} = – 18.80°C)				p = 1.8 bar = 0.18 MPa (T_{sat} = – 12.75°C)			
Sat	0.13945	216.52	236.04	0.9322	0.10983	219.94	239.71	0.9273
–10	0.14549	223.03	243.40	0.9606	0.11135	222.02	242.06	0.9362
0	0.15219	230.55	251.86	0.9922	0.11678	229.67	250.69	0.9684
10	0.15875	238.21	260.43	1.0230	0.12207	237.44	259.41	0.9998
20	0.16520	246.01	269.13	1.0532	0.12723	244.33	268.23	1.0304
30	0.17155	253.96	277.97	1.0828	0.13230	253.36	277.17	1.0604
40	0.17783	262.06	286.96	1.1120	0.13730	261.53	286.24	1.0898
50	0.18404	270.32	296.09	1.1407	0.14222	269.85	295.45	1.1187
60	0.19020	278.74	305.37	1.1690	0.14710	278.31	304.79	1.1472
70	0.19633	287.32	314.80	1.1969	0.15183	286.93	314.28	1.1753
80	0.20241	296.06	324.39	1.2244	0.15672	295.71	323.92	1.2030
90	0.20846	304.95	334.14	1.2516	0.16148	304.63	333.70	1.2303
100	0.21449	314.01	344.04	1.2785	0.16622	313.72	343.63	1.2573
	p = 2 bar = 0.20 MPa (T_{sat} = – 5.42°C)				p = 2.4 bar = 0.24 MPa (T_{sat} = –5.42°C)			
Sat.	0.09933	221.43	241.30	0.9253	0.08343	224.07	244.09	0.9222
–10	0.09938	221.50	241.38	0.9256				
0	0.10438	229.23	250.10	0.9582	0.08574	228.31	248.89	0.9399
10	0.10922	237.05	258.89	0.9898	0.08993	236.26	257.84	0.9721
20	0.11394	244.99	267.78	1.0206	0.09399	244.30	266.85	1.0034
30	0.11856	253.06	276.77	1.0508	0.09794	252.45	275.95	1.0339
40	0.12311	261.26	285.88	1.0804	0.10181	260.72	285.16	1.0930
50	0.12758	269.61	295.12	1.1094	0.10562	269.12	294.47	1.0930
60	0.13201	278.10	304.50	1.1380	0.10937	277.87	303.91	1.1218
70	0.13639	286.74	314.02	1.1661	0.11307	286.35	313.49	1.1501
80	0.14073	295.53	323.68	1.1939	0.11674	295.18	323.19	1.1780
90	0.14504	304.47	333.48	1.2212	0.12037	304.15	333.04	1.2055
100	0.14932	313.57	343.43	1.2483	0.12398	313.27	343.03	1.2326

Table B.8 *Contd.*

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJkg	*s* kJ/kg·K
	p = 2.8 bar = 0.28 MPa (T_{sat} = – 1.23°C)				p = 3.2 bar = 0.32 MPa (T_{sat} = 2.48°C)			
Sat.	0.07193	226.38	246.52	0.9197	0.06322	228.43	248.66	0.9177
0	0.07240	227.37	247.64	0.9238				
10	0.07613	235.44	256.76	0.9566	0.06576	234.61	255.65	0.9427
20	0.07972	243.59	265.91	0.9883	0.06901	242.87	264.95	0.9749
30	0.08320	251.83	275.12	1.0192	0.07214	251.19	274.28	1.0062
40	0.08660	260.17	284.42	1.0494	0.07518	259.61	283.67	1.0367
50	0.08992	268.64	293.81	1.0789	0.07815	268.14	293.15	1.0665
60	0.09319	277.23	303.32	1.1079	0.08106	276.79	302.72	1.0957
70	0.09641	285.96	312.95	1.1364	0.08392	285.56	312.41	1.1243
80	0.09960	294.82	322.71	1.1644	0.08674	294.46	322.22	1.1525
90	0.10275	303.83	332.60	1.1920	0.08953	303.50	332.15	1.1802
100	0.10587	312.98	342.62	1.2193	0.09229	312.68	342.21	1.2076
110	0.10897	322.27	352.78	1.2461	0.09503	322.00	352.40	1.2345
120	0.11205	331.71	363.08	1.2727	0.09774	331.45	362.73	1.2611
	p = 4.0 bar = 0.40 MPa (T_{sat} = 8.93°C)				p = 5.0 bar = 0.50 MPa (T_{sat} = 15.74°C)			
Sat.	0.05089	231.97	252.32	0.9145	0.04086	235.64	256.07	0.9117
10	0.05119	232.87	253.35	0.9182				
20	0.05397	241.37	262.96	0.9515	0.04188	239.40	260.34	0.9264
30	0.05662	249.89	272.54	0.9837	0.04416	248.20	270.28	0.9597
40	0.05917	258.47	282.14	1.0148	0.04633	256.99	280.16	0.9918
50	0.06164	267.13	291.79	1.0452	0.04842	265.83	290.04	1.0229
60	0.06405	275.89	301.51	1.0748	0.05043	274.73	299.05	1.0531
70	0.06641	284.75	311.32	1.1038	0.05240	283.72	309.92	1.0825
80	0.06873	293.73	321.23	1.1322	0.05432	292.80	319.96	1.1114
90	0.07102	302.84	331.25	1.1602	0.05620	302.00	330.10	1.1397
100	0.07327	312.07	341.38	1.1878	0.05805	311.31	340.33	1.1675
110	0.07550	321.44	351.64	1.2149	0.05988	320.74	350.68	1.1949
120	0.07771	330.94	362.03	1.2417	0.06168	330.30	361.14	1.2218
130	0.07991	340.58	372.54	1.2681	0.06347	339.98	371.72	1.2484
140	0.08208	350.35	383.18	1.2941	0.06524	349.79	382.42	1.2746
	p = 6.0 bar = 0.60 MPa (T_{sat} = 21.58°C)				p = 7.0 bar = 0.70 MPa (T_{sat} = 26.72°C)			
Sat.	0.03408	238.74	259.19	0.9097	0.02918	241.42	261.85	0.9080
30	0.03581	246.41	267.89	0.9388	0.02979	244.51	265.37	0.9197
40	0.03774	255.45	378.90	0.9719	0.03157	253.83	275.93	0.9539
50	0.03958	264.48	288.23	1.0037	0.03324	263.08	286.35	0.9867
60	0.04134	273.54	298.35	1.0346	0.03482	272.31	296.69	1.0182
70	0.04304	282.66	208.48	1.0645	0.03634	281.57	307.01	1.0487
80	0.04469	291.86	318.67	1.0938	0.03781	290.88	317.35	1.0784
90	0.04631	301.14	328.93	1.1225	0.03924	300.27	327.74	1.1074
100	0.04790	310.53	339.27	1.1505	0.04064	309.74	338.19	1.1358
110	0.04946	320.03	349.70	1.1781	0.04201	319.31	348.71	1.1637
120	0.05099	329.64	360.24	1.2053	0.04335	328.98	359.33	1.1910
130	0.05251	339.38	370.88	1.2320	0.04468	338.76	370.04	1.2179
140	0.05402	349.23	381.64	1.2584	0.04599	348.66	380.86	1.2444
150	0.05550	359.21	392.52	1.2844	0.04729	358.68	391.79	1.2706
160	0.05698	369.32	403.51	1.3100	0.04857	368.82	402.82	1.2963

Table B.8 *Contd.*

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s $kJ/kg \cdot K$	v m^3/kg	u kJ/kg	h $kJkg$	s $kJ/kg \cdot K$
	p = 8 bar = 0.8 MPa (T_{sat} = 31.33°C)				*p = 9.0 bar = 0.90 MPa* (T_{sat} = 35.53°C)			
Sat.	0.02547	243.78	264.15	0.9066	0.02255	245.88	266.18	0.9054
40	0.02691	252.13	273.66	0.9374	0.02325	250.32	271.25	0.9217
50	0.02846	261.62	284.39	0.9711	0.02472	260.09	282.34	0.9566
60	0.02992	271.04	294.98	1.0034	0.02609	269.72	293.21	0.9897
70	0.03131	280.45	305.50	1.0345	0.02738	279.30	303.94	1.0214
80	0.03264	289.89	316.00	1.0647	0.02861	288.87	314.62	1.0521
90	0.03393	299.37	326.52	1.0940	0.02980	298.46	325.28	1.0819
100	0.03519	308.93	337.08	1.1227	0.03095	308.11	335.96	1.1109
110	0.03642	318.57	347.71	1.1508	0.03207	317.82	346.68	1.1392
120	0.03762	328.31	358.40	1.1784	0.03316	327.62	357.47	1.1670
130	0.03881	338.14	369.19	1.2055	0.03423	337.52	368.33	1.1943
140	0.03997	348.09	380.07	1.2321	0.03529	347.51	379.27	1.2211
150	0.04113	358.15	391.05	1.2584	0.03633	357.61	390.31	1.2475
160	0.04227	368.32	402.14	1.2843	0.03736	367.82	401.44	1.2735
170	0.04340	378.61	413.33	1.3098	0.03838	378.14	412.68	1.2992
180	0.04452	389.02	424.63	1.3351	0.03939	388.57	424.02	1.3245
	p = 10.0 bar = 1.00 MPa (T_{sat} = 39.39°C)				*p = 12.0 bar = 1.20 MPa* (T_{sat} = 46.32°C)			
Sat.	0.02020	247.77	267.97	0.9043	0.01663	251.03	270.99	0.9023
40	0.02029	248.39	268.68	0.9066				
50	0.02171	258.48	280.19	0.9428	0.01712	254.98	275.52	0.9164
60	0.02301	268.35	291.36	0.9768	0.01835	265.42	287.44	0.9527
70	0.02423	278.11	302.34	1.0093	0.01947	275.59	298.96	0.9868
80	0.02538	287.82	313.20	1.0405	0.02051	285.62	310.24	1.0192
90	0.02649	297.53	324.01	1.0707	0.02150	295.59	321.39	1.0503
100	0.02755	307.27	334.82	1.1000	0.02244	305.54	332.47	1.0804
110	0.02858	317.06	345.65	1.1286	0.02335	315.50	343.52	1.1096
120	0.02959	326.93	356.52	1.1567	0.02423	325.51	354.58	1.1381
130	0.03058	336.88	367.46	1.1841	0.02508	335.58	365.68	1.1600
140	0.03154	346.92	378.46	1.2111	0.02592	345.73	376.83	1.1933
150	0.03250	357.06	389.56	1.2376	0.02674	355.95	388.04	1.2201
160	0.03344	367.31	400.74	1.2638	0.02754	366.27	399.33	1.2465
170	0.03436	377.66	412.02	1.2895	0.02834	376.69	410.70	1.2724
180	0.03528	388.12	423.40	1.3149	0.02912	387.21	422.16	1.2980
	p = 14.0 bar = 1.40 MPa (T_{sat} = 52.43°C)				*p = 12.0 bar = 1.20 MPa* (T_{sat} = 57.92°C)			
Sat.	0.01405	253.74	273.40	0.9003	0.01208	256.00	275.33	0.8982
60	0.01495	262.17	283.10	0.9297	0.01233	258.48	278.20	0.9069
70	0.10603	272.87	295.31	0.9658	0.01340	269.89	291.33	0.9457
80	0.01701	283.29	307.10	0.9997	0.01435	280.78	303.74	0.9813
90	0.01792	293.55	318.63	1.0319	0.01521	291.39	315.72	1.0148
100	0.01878	303.73	330.02	1.0628	0.01601	301.84	327.46	1.0467
110	0.01960	313.88	341.32	1.0927	0.01677	312.20	339.04	1.0773
120	0.02039	324.05	352.59	1.1218	0.01750	322.53	350.53	1.1069
130	0.02115	334.25	363.86	1.1501	0.01820	332.87	361.99	1.1357
140	0.02189	344.50	375.15	1.1777	0.01887	343.24	373.44	1.1638
150	0.02262	354.82	386.49	1.2048	0.01953	353.66	384.91	1.1912
160	0.02333	365.22	397.89	1.2315	0.02017	364.15	396.43	1.2181
170	0.02403	375.71	409.36	1.2576	0.02080	374.71	407.99	1.2445
180	0.02472	386.29	420.90	1.2834	0.02142	385.35	419.62	1.2704
190	0.02541	396.96	432.53	1.3088	0.02203	396.08	431.33	1.2960
200	0.02608	407.73	444.24	1.3338	0.02263	406.90	443.11	1.3212

Table B.9 *Properties of R-717: **Saturated Ammonia** (Liquid–Vapour): Temperature Entry*

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K		
Temp. °C	Press. bar	Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Sat. Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g	Temp. °C
–50	0.4086	1.4245	2.6265	–43.94	1264.99	–43.88	1416.20	1372.32	–0.1922	6.1543	–50
–45	0.5453	1.4367	2.0060	–22.03	1271.19	–21.95	1402.52	1380.57	–0.0951	6.0523	–45
–40	0.7174	1.4493	1.5524	–0.10	1277.20	0.00	1388.56	1388.56	0.0000	5.9557	–40
–36	0.8850	1.4597	1.2757	17.47	1281.87	17.60	1377.17	1394.77	0.0747	5.8819	–36
–32	1.0832	1.4703	1.0561	35.09	1286.41	35.25	1365.55	1400.81	0.1484	5.8111	–32
–30	1.1950	1.4757	0.9634	43.93	1288.63	44.10	1359.65	1403.75	0.1849	5.7765	–30
–28	1.3159	1.4812	0.8803	52.78	1290.82	52.97	1353.68	1406.66	0.2212	5.7430	–28
–26	1.4465	1.4867	0.8056	61.65	1292.97	61.86	1347.65	1409.51	0.2572	5.7100	–26
–22	1.7390	1.4980	0.6780	79.46	1297.18	79.72	1335.36	1415.08	0.3287	5.6457	–22
–20	1.9019	1.5038	0.6233	88.40	1299.23	88.68	1329.10	1417.79	0.3642	5.6144	–20
–18	2.0769	1.5096	0.5739	97.36	1301.25	97.68	1322.77	1420.45	0.3994	5.5837	–18
–16	2.2644	1.5155	0.5291	106.36	1303.23	106.70	1316.35	1423.05	0.4346	5.5536	–16
–14	2.4652	1.5215	0.4885	115.37	1305.17	115.75	1309.86	1425.61	0.4695	5.5239	–14
–12	2.6798	1.5276	0.4516	124.42	1307.08	124.83	1303.28	1428.11	0.5043	5.4948	–12
–10	2.9089	1.5338	0.4180	133.50	1308.95	133.94	1296.61	1430.55	0.5389	5.4662	–10
–8	3.1532	1.5400	0.3874	142.60	1310.78	143.09	1289.86	1432.95	0.5734	5.4380	–8
–6	3.4134	1.5464	0.3595	151.74	1312.57	152.26	1283.02	1435.28	0.6077	5.4103	–6
–4	3.6901	1.5528	0.3340	160.88	1314.32	161.46	1276.10	1437.56	0.6418	5.3831	–4
–2	3.9842	1.5594	0.3106	170.07	1316.04	170.69	1269.08	1439.78	0.6759	5.3562	–2
0	4.2962	1.5660	0.2892	179.29	1317.71	179.96	1261.97	1441.94	0.7097	5.3298	0
2	4.6270	1.5727	0.2695	188.53	1319.34	189.20	1254.77	1444.03	0.7435	5.3038	2
4	4.9773	1.5796	0.2514	197.80	1320.92	198.59	1247.48	1446.07	0.7770	5.2781	4
6	5.3479	1.5866	0.2348	207.10	1322.47	207.95	1240.09	1448.04	0.8105	5.2529	6
8	5.7395	1.5936	0.2195	216.42	1323.96	217.34	1232.61	1449.94	0.8438	5.2279	8
10	6.1529	1.6008	0.2054	225.77	1325.42	226.75	1225.03	1451.78	0.8769	5.2033	10
12	6.5890	1.6081	0.1923	235.14	1326.82	236.20	1217.35	1453.55	0.9099	5.1791	12
16	7.5324	1.6231	0.1691	253.95	1329.48	255.18	1201.70	1456.87	0.9755	5.1314	16
20	8.5762	1.6386	0.1492	272.86	1331.94	274.26	1185.64	1459.90	1.0404	5.0849	20
24	9.7274	1.6547	0.1320	291.84	1334.19	293.45	1169.16	1462.61	1.1048	5.0394	24
28	10.993	1.6714	0.1172	310.92	1336.20	312.75	1152.24	1465.00	1.1686	4.9948	28
32	12.380	1.6887	0.1043	330.07	1337.97	332.17	1134.87	1467.03	1.2319	4.9509	32
36	13.896	1.7068	0.0930	349.32	1339.47	351.69	1117.00	1468.70	1.2946	4.9078	36
40	15.549	1.7256	0.0831	368.67	1340.70	371.35	1098.62	1469.97	1.3569	4.8652	40
45	17.819	1.7503	0.0725	393.01	1341.81	396.31	1074.84	1470.96	1.4341	4.8125	45
50	20.331	1.7765	0.0634	417.56	1342.42	421.17	1050.09	1471.26	1.5109	4.7604	50

Table B.10 *Properties of R-717:* ***Saturated Ammonia*** *(Liquid–Vapour): Pressure Entry*

Press. bar	Temp. °C	Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy $kJ/kg \cdot K$		Press. bar
		Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Sat. Liquid u_f	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Sat. Vapour s_g	
0.40	–50.36	1.4236	2.6795	–45.52	1264.54	–45.46	1417.18	1371.72	–0.1992	6.1618	0.40
0.50	–46.53	1.4330	2.1752	–28.73	1269.31	–28.66	1406.73	1378.07	–0.1245	6.0829	0.50
0.60	–43.28	1.4410	1.8345	–14.51	1273.27	–14.42	1397.76	1383.34	–0.0622	6.0186	0.60
0.70	–40.46	1.4482	1.5884	–2.11	1276.66	–2.01	1389.85	1387.84	–0.0086	5.9643	0.70
0.80	–37.94	1.4546	1.4020	8.93	1279.61	9.04	1382.73	1391.78	0.0386	5.9174	0.80
0.90	–35.67	1.4605	1.2559	18.91	1282.24	19.04	1376.23	1395.27	0.0808	5.8760	0.90
1.00	–33.60	1.4660	1.1381	28.03	1284.61	28.18	1370.23	1398.41	0.1191	5.8391	1.00
1.25	–29.07	1.4782	0.9237	48.03	1289.65	48.22	1356.89	1405.11	0.2018	5.7610	1.25
1.50	–25.22	1.4889	0.7787	65.10	1293.80	65.32	1345.28	1410.61	0.2712	5.6973	1.50
1.75	–21.86	1.4984	0.6740	80.08	1297.33	80.35	1334.92	1415.27	0.3312	5.6435	1.75
2.00	–18.86	1.5071	0.5946	93.50	1300.39	93.80	1325.51	1419.31	0.3843	5.5969	2.00
2.25	–16.15	1.5151	0.5323	105.68	1303.08	106.03	1316.83	1422.86	0.4319	5.5558	2.25
2.50	–13.67	1.5225	0.4821	116.88	1305.49	117.26	1308.76	1426.03	0.4753	5.5190	2.50
2.75	–11.37	1.5295	0.4408	127.26	1307.67	127.68	1301.20	1428.88	0.5152	5.4858	2.75
3.00	–9.24	1.5361	0.4061	136.96	1309.65	137.42	1294.05	1431.47	0.5520	5.4554	3.00
3.25	–7.24	1.5424	0.3765	146.06	1311.46	146.57	1287.27	1433.84	0.5864	5.4275	3.25
3.50	–5.36	1.5484	0.3511	154.66	1313.14	155.20	1280.81	1436.01	0.6186	5.4016	3.50
3.75	–3.58	1.5542	0.3289	162.80	1314.68	166.38	1274.64	1438.03	0.6489	5.3774	3.75
4.00	–1.90	1.5597	0.3094	170.55	1316.12	171.18	1268.71	1439.89	0.6776	5.3548	4.00
4.25	–0.29	1.5650	0.2921	177.96	1317.47	178.62	1263.01	1441.63	0.7048	5.3336	4.25
4.50	1.25	1.5702	0.2767	185.04	1318.73	185.75	1257.50	1443.25	0.7308	5.3135	4.50
4.75	2.72	1.5752	0.2629	191.84	1319.91	192.59	1252.18	1444.77	0.7555	5.2946	4.75
5.00	4.13	1.5800	0.2503	198.39	1321.02	199.18	1247.02	1446.19	0.7791	5.2765	5.00
5.25	5.48	1.5847	0.2390	204.69	1322.07	205.52	1242.01	1447.53	0.8018	5.2594	5.22
5.50	6.79	1.5893	0.2286	210.78	1323.06	211.65	1237.15	1448.80	0.8236	5.2430	5.50
5.75	8.05	1.5938	0.2191	216.66	1324.00	217.58	1232.41	1449.99	0.8446	5.2273	5.75
6.00	9.27	1.5982	0.2104	222.37	1324.89	223.32	1227.79	1451.12	0.8649	5.2122	6.00
7.00	13.79	1.6148	0.1815	243.56	1328.04	244.69	1210.38	1455.07	0.9394	5.1576	7.00
8.00	17.84	1.6302	0.1596	262.64	1330.64	263.95	1194.36	1458.30	1.0054	5.1099	8.00
9.00	21.52	1.6446	0.1424	280.05	1332.82	281.53	1179.44	1460.97	1.0649	5.0675	9.00
10.00	24.89	1.6584	0.1285	296.10	1334.66	297.76	1165.42	1463.18	1.1191	5.0294	10.00
12.00	30.94	1.6841	0.1075	324.99	1337.52	327.10	1139.52	1466.53	1.2152	4.9625	12.00
14.00	36.26	1.7080	0.0923	350.58	1339.56	352.97	1115.82	1468.79	1.2987	4.9050	14.00
16.00	41.03	1.7306	0.0808	373.69	1340.97	376.46	1093.77	1470.23	1.3729	4.8542	16.00
18.00	45.38	1.7522	0.0717	394.85	1341.88	398.00	1073.01	1471.01	1.4399	4.8086	18.00
20.00	49.37	1.7731	0.0644	414.44	1342.37	417.99	1053.27	1471.26	1.5012	4.7670	20.00

Table B.11 *Properties of R-717:* ***Superheated Refrigerant Ammonia Vapour***

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJkg	*s* kJ/kg·K
	p = 0.4 bar = 0.04 MPa (T_{sat} = – 50.30°C)				*p* = 0.6 bar = 0.06 MPa (T_{sat} = – 43.28°C)			
Sat.	2.6795	1264.54	1371.72	6.1618	1.8345	1273.27	1383.34	6.0186
–50	2.6841	1265.11	1372.48	6.1652				
–45	2.7481	1273.05	1382.98	6.2118				
–40	2.8118	1281.01	1393.48	6.2573	1.8630	1278.62	1390.40	6.0490
–35	2.8753	1288.96	1403.98	6.3018	1.9061	1286.75	1401.12	6.0946
–30	2.9385	1296.93	1414.47	6.3455	1.9491	1294.88	1411.83	6.1390
–25	3.0015	1304.90	1424.96	6.3882	1.9918	1303.01	1422.52	6.1826
–20	3.0644	1312.88	1435.46	6.4300	20343	1311.13	1433.19	6.2251
–15	3.1271	1320.86	1445.95	6.4711	2.0766	1319.25	1443.85	6.2668
–10	3.1896	1328.87	1456.45	6.5114	2.1188	1327.37	1454.50	6.3077
–5	3.2520	1336.88	1466.95	6.5509	2.1609	1335.49	1465.14	6.3478
0	3.3142	1344.90	1477.17	6.5898	2.2028	1343.61	1475.78	6.3871
5	3.3764	1352.95	1488.00	6.6280	2.2446	1351.75	1486.43	6.4257
	p = 0.8 bar = 0.08 MPa (T_{sat} = – 37.94°)				*p* = 1.0 bar = 0.10 MPa (T_{sat} = – 33.60°C)			
Sat.	1.4021	1279.61	1391.78	5.9174	1.1381	1284.61	1398.41	5.8391
–35	1.4215	1284.51	1398.23	5.9446				
–30	1.4543	1292.81	1409.15	5.9900	1.1573	1290.71	1406.71	5.8723
–25	1.4868	1301.09	1420.04	6.0343	1.1838	1299.15	1417.53	5.9175
–20	1.5192	1309.36	1430.90	6.0777	1.2101	1307.57	1428.58	5.9616
–15	1.5514	1317.61	1441.72	6.1200	1.2362	1315.96	1439.58	6.0046
–10	1.5834	1325.85	1452.53	6.1615	1.2621	1324.33	1450.54	6.0467
–5	1.6153	1334.09	1463.31	6.2021	1.2880	1332.67	1461.46	6.0878
0	1.6471	1342.31	1474.08	6.2419	1.3136	1341.00	1472.37	6.1281
5	1.6788	1350.54	1484.84	6.2809	1.3392	1349.33	1483.25	6.1676
10	1.7103	1358.77	1495.60	6.3192	1.3647	1357.64	1494.11	6.2063
15	1.7418	1367.01	1506.35	6.3568	1.3900	1365.95	1504.96	6.2442
20	1.7732	1375.25	1517.10	6.3539	1.4153	1374.27	1515.80	6.2816
	p = 1.5 bar = 0.15 MPa (T_{sat} = – 25.22°C)				*p* = 2.0 bar = 0.20 MPa (T_{sat} = – 18.86°C)			
Sat.	0.7787	1293.80	1410.61	5.6973	0.59460	1300.39	1419.31	5.5969
–25	0.7795	1294.20	1411.13	5.6994				
–20	0.7978	1303.00	1422.67	5.7454				
–15	0.8158	1311.75	1434.12	5.7902	0.60542	1307.43	1428.51	5.6328
–10	0.8336	1320.44	1445.49	5.8338	0.61926	1316.46	1440.31	5.6781
–5	0.8514	1329.08	1456.79	5.8764	0.63294	1325.41	1452.00	5.7221
0	0.8689	1337.68	1468.02	5.9179	0.64648	1334.29	1463.59	5.7645
5	0.8864	1346.25	1479.20	5.9585	0.65989	1343.11	1475.09	5.8066
10	0.9037	1354.78	1490.34	5.9981	0.67320	1351.87	1486.51	5.8473
15	0.9210	1363.29	1501.44	6.0370	0.68640	1360.59	1497.87	5.8871
20	0.9382	1371.79	1512.51	6.0751	0.69952	1369.28	1509.10	5.9260
25	0.9553	1380.28	1523.56	6.1125	0.71256	1377.93	1520.44	5.9641
30	0.9723	1388.76	1534.60	6.1492	0.72553	1386.56	1531.67	6.0014

Table B.11 *Contd.*

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 2.5 bar = 0.25 MPa (T_{sat} = – 13.67°C)				*p = 3.0 bar = 0.30 MPa* (T_{sat} = – 9.24°C)			
Sat.	0.48213	1305.49	1426.03	5.5190	0.40607	1309.65	1431.47	5.4554
–10	0.49051	1312.37	1435.00	5.5534				
–5	0.50180	1321.65	1447.10	5.5989	0.41428	1317.80	1442.08	5.4953
0	0.51293	1330.83	1459.06	5.6431	0.42382	1327.28	1454.43	5.5409
5	0.52393	1339.91	1470.89	5.6860	0.43323	1336.64	1466.61	5.5851
10	0.53482	1348.91	1482.61	5.7278	0.44251	1345.89	1478.65	5.6280
15	0.54560	1357.84	1494.25	5.7685	0.45169	1355.05	1490.56	5.6697
20	0.55630	1366.72	1505.80	5.8083	0.46078	1364.13	1502.36	5.7103
25	0.56691	1375.55	1517.28	5.8471	0.46978	1373.14	1514.07	5.7499
30	0.57745	1384.34	1528.70	5.8851	0.47870	1382.09	1525.70	5.7886
35	0.58793	1393.10	1540.08	5.9223	0.48756	1391.00	1357.26	5.8264
40	0.59835	1401.84	1551.42	5.9589	0.49637	1399.86	1548.77	5.8635
45	0.60872	1410.56	1562.74	5.9947	0.50512	1408.70	1560.24	5.8998
	p = 3.5 bar = 0.35 MPa (T_{sat} = – 5.36°C)				*p = 4.0 bar = 0.40 MPa* (T_{sat} = – 1.90°C)			
Sat.	0.35108	1313.14	1436.01	5.4016	0.30942	1316.12	1439.89	5.3548
0	0.36011	1323.66	1449.70	5.4522	0.31227	1319.95	1444.86	5.3731
10	0.37654	1342.82	1474.61	5.5417	0.32701	1339.68	1470.49	5.4652
20	0.39251	1361.49	1498.87	5.6259	0.34129	1358.81	1495.33	5.5515
30	0.40814	1379.81	1522.66	5.7057	0.35520	1377.49	1519.57	5.6328
40	0.42350	1397.87	1546.09	5.7818	0.36884	1395.85	1543.38	5.7101
60	0.45363	1433.55	1592.32	5.9249	0.39550	1431.97	1590.17	5.8549
80	0.48320	1469.06	1638.18	6.0586	0.42160	1467.77	1636.41	5.9867
100	0.51240	1504.73	1684.07	6.1850	0.44733	1503.64	1682.58	6.1169
120	0.54136	1540.79	1730.26	6.3056	0.47280	1539.85	1728.97	6.2380
140	0.57013	1577.38	1776.92	6.4213	0.49808	1576.55	1775.79	6.3541
160	0.59876	1616.60	1824.16	6.5330	0.52323	1613.86	1823.16	6.4661
180	0.62728	1652.51	1872.06	6.6411	0.54827	1651.85	1871.16	6.5744
200	0.65572	1691.15	1920.65	6.7460	0.57322	1690.56	1919.85	6.6796
	p = 4.5 bar = 0.45 MPa (T_{sat} = 1.25°C)				*p = 5.0 bar 0.50 MPa* (T_{sat} = 4.13°C)			
Sat.	0.27671	1318.73	1443.25	5.3135	0.25034	1321.02	1446.19	5.2765
10	0.28846	1336.48	1466.29	5.3962	0.25757	1333.22	1462.00	5.3330
20	0.30142	1356.09	1491.72	5.4845	0.26949	1353.32	1488.06	5.4234
30	0.31401	1375.15	1516.45	5.5674	0.28103	1372.76	1513.28	5.5080
40	0.32631	1393.80	1540.64	5.6460	0.29227	1391.74	1537.87	5.5878
60	0.35029	1430.37	1588.00	5.7926	0.31410	1428.76	1585.81	5.7362
80	0.37369	1466.47	1634.63	5.9285	0.33535	1465.16	1632.84	5.8733
100	0.39671	1502.55	1681.07	6.0564	0.35621	1501.46	1679.56	6.0020
120	0.41947	1538.91	1727.67	6.1781	0.37681	1537.97	1726.37	6.1242
140	0.44205	1575.73	1774.65	6.2946	0.39722	1574.90	1773.51	6.2412
160	0.46448	1613.13	1822.15	6.4069	0.41749	1612.40	1821.14	6.3537
180	0.48681	1651.20	1870.26	6.5155	0.43765	1650.54	1869.36	6.4626
200	0.50905	1689.97	1919.04	6.6208	0.45771	1689.38	1918.24	6.5681

Table B.11 *Contd.*

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m³/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 5.5 bar = 0.55 MPa (T_{sat} = 6.79°C)				*p = 6.0 bar = 0.60 MPa* (T_{sat} = 9.27°C)			
Sat.	0.22861	1323.06	1448.80	5.2430	0.21038	1324.89	1451.12	5.2122
10	0.23227	1329.88	1457.63	5.2743	0.21115	1326.47	1453.16	5.2195
20	0.24335	1350.50	1484.34	5.3671	0.22155	1347.62	1480.55	5.3145
30	0.25403	1370.35	1510.07	5.4534	0.23152	1367.90	1506.81	5.4026
40	0.26441	1389.64	1535.07	5.5345	0.24118	1387.52	1532.23	5.4851
50	0.27454	1408.53	1559.53	5.6114	0.25059	1406.67	1557.03	5.5631
60	0.28449	1427.13	1583.60	5.6848	0.25981	1425.49	1581.38	5.6373
80	0.30398	1463.85	1631.04	5.8230	0.27783	1462.52	1629.22	5.7768
100	0.32307	1500.36	1678.05	5.9525	0.29546	1499.25	1676.52	5.9071
120	0.34190	1537.02	1725.07	6.0753	0.31281	1536.07	1723.76	6.0304
140	0.36054	1574.07	1772.37	6.1926	0.32997	1573.24	1771.22	6.1481
160	0.37903	1611.66	1820.13	6.3055	0.34699	1610.92	1819.12	6.2613
180	0.39742	1649.88	1868.46	6.4146	0.36390	1649.22	1867.56	6.3707
200	0.41571	1688.79	1917.43	6.5203	0.38071	1688.20	1916.63	6.4766
	p = 7.0 bar = 0.70 MPa (T_{sat} = 13.79°C)				*p = 8.0 bar = 0.80 MPa* (T_{sat} = 17.84°C)			
Sat.	0.18148	1328.04	1455.07	5.1576	0.15958	1330.64	1458.30	5.1099
20	0.18721	1341.72	1472.77	5.2186	0.16138	1335.59	1464.70	5.1318
30	0.19610	1362.88	1500.15	5.3104	0.16948	1357.71	1493.29	5.2277
40	0.20464	1383.20	1526.45	5.3958	0.17720	1378.77	1520.53	5.3161
50	0.21293	1402.90	1551.95	5.4760	0.18465	1399.05	1546.77	5.3986
60	0.22101	1422.16	1576.87	5.5519	0.19189	1418.77	1572.28	5.4763
80	0.23674	1459.85	1625.56	5.6939	0.20590	1457.14	1621.86	5.6209
100	0.25205	1497.02	1673.46	5.8258	0.21949	1494.77	1670.37	5.7545
120	0.26709	1534.12	1721.12	5.9502	0.23280	1532.24	1718.48	5.8801
140	0.28193	1571.57	1768.92	6.0688	0.24590	1569.89	1766.61	5.9995
160	0.29663	1609.44	1817.08	6.1826	0.25886	1607.96	1815.04	6.1140
180	0.31121	1647.90	1865.75	6.2925	0.27170	1646.57	1683.94	6.2243
200	0.32571	1687.02	1915.01	6.3988	0.28445	1685.83	1913.39	6.3311
	p = 9.0 bar = 0.90 MPa (T_{sat} = 21.52°C)				*p = 10.0 bar = 1.00 MPa* (T_{sat} = 24.89°C)			
Sat.	0.14239	1332.82	1460.97	5.0675	0.12852	1334.66	1463.18	5.0294
30	0.14872	1352.36	1486.20	5.1520	0.13206	1346.82	1478.88	5.0816
40	0.15582	1374.21	1514.45	5.2436	0.13868	1369.52	1508.20	5.1768
50	0.16263	1395.11	1541.47	5.3286	0.14499	1391.07	1536.06	5.2644
60	0.16922	1415.32	1567.61	5.4083	0.15106	1411.79	1562.86	5.3460
80	0.18191	1454.39	1618.11	5.5555	0.16270	1451.60	1614.31	5.4960
100	0.19416	1492.50	1667.24	5.6908	0.17389	1490.20	1664.10	5.6332
120	0.20612	1530.30	1715.81	5.8176	0.18478	1528.35	1713.13	5.7612
140	0.21788	1568.20	1764.29	5.9379	0.19545	1566.51	1761.96	5.8803
160	0.22948	1606.46	1813.00	6.0530	0.20598	1604.97	1810.94	5.9981
180	0.24097	1645.24	1862.12	6.1639	0.21638	1643.91	1860.29	6.1095
200	0.25237	1684.64	1911.77	6.2711	0.22670	1683.44	1910.14	6.2171

Table B.11 *Contd.*

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m³/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 12.0 bar = 1.20 MPa (T_{sat} = 30.94°C)				*p = 14.0 bar = 1.40 MPa* (T_{sat} = 36.26°C)			
Sat.	0.10751	1337.52	1466.53	4.9625	0.09231	1339.56	1468.79	4.9050
40	0.11287	1359.73	1495.18	5.0553	0.09432	1349.29	1481.33	4.9453
60	0.12378	1404.54	1553.07	5.2347	0.10423	1396.97	1542.98	5.1360
80	0.13387	1445.91	1606.56	5.3906	0.11324	1440.06	1598.59	5.2984
100	0.14347	1485.55	1657.71	5.5315	0.12172	1480.79	1651.20	5.4433
120	0.15275	1524.41	1707.71	5.6620	0.12986	1520.41	1702.21	5.5765
140	0.16181	1563.09	1757.26	5.7850	0.13777	1559.63	1752.52	5.7013
160	0.17072	1601.95	1806.81	5.9021	0.14552	1598.92	1802.65	5.8198
180	0.17950	1641.23	1856.63	6.0145	0.15315	1638.53	1852.94	5.9333
200	0.18819	1681.05	1906.87	6.1230	0.16068	1678.64	1903.59	6.0427
220	0.19680	1721.50	1957.66	6.2282	0.16813	1719.35	1954.73	6.1485
240	0.20534	1762.63	2009.04	6.3303	0.17551	1760.72	2006.43	6.2513
260	0.21382	1804.48	2061.06	6.4297	0.18283	1802.78	2058.75	6.3513
280	0.22225	1847.04	2113.74	6.5267	0.19010	1845.55	2111.69	6.4488
	p = 16.0 bar = 1.60 MPa (T_{sat} = 41.03°C)				*p = 18.0 bar = 1.80 MPa* (T_{sat} = 45.38°C)			
Sat.	0.08079	1340.97	1470.23	4.8542	0.07174	1341.88	1471.01	4.8086
60	0.08951	1389.06	1532.28	5.0461	0.07801	1380.77	1521.10	4.9627
80	0.09774	1434.02	1590.40	5.2156	0.08565	1427.79	1581.97	5.1399
100	0.10539	1475.93	1644.56	5.3648	0.09267	1470.96	1637.78	5.2937
120	0.11268	1516.34	1696.64	5.5008	0.09931	1512.22	1690.98	5.4326
140	0.11974	1556.14	1747.72	5.6276	0.10570	1552.61	1742.88	5.5614
160	0.12663	1595.85	1798.45	5.7475	0.11192	1592.76	1794.23	5.6828
180	0.13339	1635.81	1849.23	5.8621	0.11801	1633.08	1845.50	5.7985
200	0.14005	1676.21	1900.29	5.9723	0.12400	1673.78	1896.98	5.9096
220	0.14663	1717.18	1951.79	6.0789	0.12991	1715.00	1948.83	6.0170
240	0.15314	1758.79	2003.81	6.1823	0.13574	1756.85	2001.18	6.1210
260	0.15959	1801.07	2056.42	6.2829	0.14152	1799.35	2054.08	6.2222
280	0.16599	1844.05	2109.64	6.3809	0.14724	1842.55	2107.58	6.3207
	p = 20.0 bar = 2.00 MPa (T_{sat} = 49.37°C)							
Sat.	0.06445	1342.37	1471.26	4.7670				
60	0.06445	1372.05	1509.54	4.8838				
80	0.07596	1421.36	1573.27	5.0696				
100	0.08248	1465.89	1630.86	5.2283				
120	0.08861	1508.03	1685.24	5.3703				
140	0.09447	1549.03	1737.98	5.5012				
160	0.10016	1589.65	1789.97	5.6241				
180	0.10571	1630.82	1841.74	5.7409				
200	0.11116	1671.33	1893.64	5.8530				
220	0.11652	1712.82	1945.87	5.9611				
240	0.12182	1754.90	1998.54	6.0658				
260	0.12706	1797.63	2051.74	6.1675				
280	0.13224	1841.03	2105.50	6.2665				

Table B.12 *Properties of Saturated* ***Propane*** *(Liquid–Vapour): Temperature Entry*

Temp. °C	Press. bar	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapour v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapour u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Entropy kJ/kg · K Sat. Liquid s_f	Sat. Vapour s_g	Temp. °C
–100	0.02888	1.553	11.27	–128.4	319.5	–128.4	480.4	352.0	–0.634	2.140	–100
–90	0.06426	1.578	5.345	–107.8	329.3	–107.8	471.4	363.6	–0.519	2.055	–90
–80	0.1301	1.605	2.7704	–87.0	339.3	–87.0	462.4	375.4	–0.408	1.986	–80
–70	0.2434	1.633	1.551	–65.8	349.5	–65.8	453.1	387.3	–0.301	1.929	–70
–60	0.4261	1.663	0.9234	–44.4	359.9	–44.3	443.5	399.2	–0.198	1.883	–60
–50	0.7046	1.694	0.5793	–22.5	370.4	–22.4	433.6	411.2	–0.098	1.845	–50
–40	1.110	1.728	0.3798	–0.2	381.0	0.0	423.2	423.2	0.000	1.815	–40
–30	1.677	1.763	0.2585	22.6	391.6	22.9	412.1	435.0	0.096	1.791	–30
–20	2.444	1.802	0.1815	45.9	402.4	46.3	400.5	446.8	0.190	1.772	–20
–10	3.451	1.844	0.1309	69.8	413.2	70.4	388.0	458.4	0.282	1.757	–10
0	4.743	1.890	0.09653	94.2	423.8	95.1	374.5	469.6	0.374	1.745	0
4	5.349	1.910	0.08591	104.2	428.1	105.3	368.8	474.1	0.410	1.741	4
8	6.011	1.931	0.07666	114.3	432.3	115.5	362.9	478.4	0.446	1.737	8
12	6.732	1.952	0.06858	124.6	436.5	125.9	356.8	482.1	0.482	1.734	12
16	7.515	1.975	0.06149	135.0	440.7	136.4	350.5	486.9	0.519	1.731	16
20	8.362	1.999	0.05525	145.4	444.8	147.1	343.9	491.0	0.555	1.728	20
24	9.278	2.024	0.04973	156.1	448.9	158.0	337.0	495.0	0.591	1.725	24
28	10.27	2.050	0.04483	166.9	452.9	169.0	329.9	498.9	0.627	1.722	28
32	11.33	2.078	0.04048	177.8	456.7	180.2	322.4	502.6	0.663	1.720	32
36	12.47	2.108	0.03659	188.9	460.6	191.6	314.6	506.2	0.699	1.717	36
40	13.69	2.140	0.03310	200.2	464.3	203.1	306.5	509.6	0.736	1.715	40
44	15.00	2.174	0.02997	211.7	467.9	214.9	298.0	512.9	0.772	1.712	44
48	16.40	2.211	0.02714	223.4	471.4	227.0	288.9	515.9	0.809	1.709	48
52	17.89	2.250	0.02459	235.3	474.6	239.3	279.3	518.6	0.846	1.705	52
56	19.47	2.293	0.02227	247.4	477.7	251.9	269.2	521.1	0.884	1.701	56
60	21.16	2.340	0.02015	259.8	480.6	264.8	258.4	523.2	0.921	1.697	60
65	23.42	2.406	0.01776	275.7	483.6	281.4	243.8	525.2	0.969	1.690	65
70	25.86	2.483	0.01560	292.3	486.1	298.7	227.2	526.4	1.018	1.682	70
75	28.49	2.573	0.01363	309.5	487.8	316.8	209.8	526.6	1.069	1.671	75
80	31.31	2.683	0.01182	327.6	488.2	336.0	189.2	525.2	1.122	1.657	80
85	34.36	2.827	0.01011	347.2	486.9	356.9	164.7	521.6	1.178	1.638	85
90	37.64	3.038	0.008415	369.4	482.2	380.8	133.1	513.9	1.242	1.608	90
95	41.19	3.488	0.006395	399.8	467.4	414.2	79.5	493.7	1.330	1.546	95
96.7	42.48	4.535	0.004535	434.9	434.9	454.2	0.0	457.2	1.437	1.437	96.7

Table B.13 *Properties of Saturated* ***Propane*** *(Liquid–Vapour): Pressure Entry*

		Specific Volume m^3/kg		*Internal Energy* kJ/kg		*Enthalpy* kJ/kg			*Entropy* $kJ/kg \cdot K$		
Press. *bar*	*Temp.* °*C*	*Sat. Liquid* $v_f \times 10^3$	*Sat. Vapour* v_g	*Sat. Liquid* u_f	*Sat. Vapour* u_g	*Sat. Liquid* h_f	*Evap.* h_{fg}	*Sat. Vapour* h_g	*Sat. Liquid* s_f	*Sat. Vapour* s_g	*Press.* *bar*
0.05	–93.28	1.570	6.752	–114.6	326.0	–114.6	474.4	359.8	–0.556	2.081	0.05
0.10	–83.87	1.594	3.542	–95.1	335.4	–95.1	465.9	370.8	–0.450	2.011	0.10
0.25	–69.55	1.634	1.513	–64.9	350.0	–64.9	452.7	387.8	–0.297	1.927	0.25
0.50	–56.93	1.672	0.7962	–37.7	363.1	–37.6	440.5	402.9	–0.167	1.871	0.50
0.75	–48.68	1.698	0.5467	–19.6	371.8	–19.5	432.3	412.8	–0.085	1.841	0.75
1.00	–42.38	1.719	0.4185	–5.6	378.5	–5.4	425.7	420.3	–0.023	1.822	1.00
2.00	–25.43	1.781	0.2192	33.1	396.6	33.5	406.9	440.4	0.139	1.782	2.00
3.00	–14.16	1.826	0.1496	59.8	408.7	60.3	393.3	453.6	0.244	1.762	3.00
4.00	–5.46	1.865	0.1137	80.8	418.0	81.5	382.0	463.5	0.324	1.751	4.00
5.00	1.74	1.899	0.09172	98.6	425.7	99.5	372.1	471.6	0.389	1.743	5.00
6.00	7.93	1.931	0.07680	114.2	432.2	115.3	363.0	478.3	0.446	1.737	6.00
7.00	13.41	1.960	0.06598	128.2	438.0	129.6	354.6	484.2	0.495	1.733	7.00
8.00	18.33	1.989	0.05776	141.0	443.1	142.6	346.7	489.3	0.540	1.729	8.00
9.00	22.82	2.016	0.05129	152.9	447.6	154.7	339.1	493.8	0.580	1.726	9.00
10.00	26.95	2.043	0.04606	164.0	451.8	166.1	331.8	497.9	0.618	1.723	10.00
11.00	30.80	2.070	0.04174	174.5	455.6	176.8	324.7	501.5	0.652	1.721	11.00
12.00	34.39	2.096	0.03810	184.4	459.1	187.0	317.8	504.8	0.685	1.718	12.00
13.00	37.77	2.122	0.03499	193.9	462.2	196.7	311.0	507.7	0.716	1.716	13.00
14.00	40.97	2.148	0.03231	203.0	465.2	206.0	304.4	510.4	0.745	1.714	14.00
15.00	44.01	2.174	0.02997	211.7	467.9	215.0	297.9	512.9	0.772	1.712	15.00
16.00	46.89	2.200	0.02790	220.1	470.4	223.6	291.4	515.0	0.799	1.710	16.00
17.00	49.65	2.227	0.02606	228.3	472.7	232.0	285.0	517.0	0.824	1.707	17.00
18.00	52.30	2.253	0.02441	236.2	474.9	240.2	278.6	518.8	0.849	1.705	18.00
19.00	54.83	2.280	0.02292	243.8	476.9	248.2	272.2	520.4	0.873	1.703	19.00
20.00	57.27	2.308	0.02157	251.3	478.7	255.9	265.9	521.8	0.896	1.700	20.00
22.00	61.90	2.364	0.01921	265.8	481.7	271.0	253.0	524.0	0.939	1.695	22.00
24.00	66.21	2.424	0.01721	279.7	484.3	285.5	240.1	525.6	0.981	1.688	24.00
26.00	70.27	2.487	0.01549	293.1	486.2	299.6	226.9	526.5	1.021	1.681	26.00
28.00	74.10	2.555	0.01398	306.2	487.5	313.4	213.2	526.6	1.060	1.673	28.00
30.00	77.72	2.630	0.01263	319.2	488.1	327.1	198.9	526.0	1.097	1.664	30.00
35.00	86.01	2.862	0.009771	351.4	486.3	361.4	159.1	520.5	1.190	1.633	35.00
40.00	93.38	3.279	0.007151	387.9	474.7	401.0	102.3	503.3	1.295	1.574	40.00
42.48	96.70	4.535	0.004535	434.9	434.9	454.2	0.0	454.2	1.437	1.437	42.48

Table B.14 *Properties of Superheated* ***Propane***

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 0.05 bar = 0.005 MPa (T_{sat} = – 93.28°C)				p = 0.1 bar = 0.01 MPa (T_{sat} = – 83.87°C)			
Sat.	6.752	326.0	359.8	2.081	3.542	367.3	370.8	2.011
–90	6.877	329.4	363.8	2.103				
–80	7.258	339.8	376.1	2.169	3.617	339.5	375.7	2.037
–70	7.639	350.6	388.8	2.233	3.808	350.3	388.4	2.101
–60	8.018	361.8	401.9	2.296	3.999	361.5	401.5	2.164
–50	8.397	373.3	415.3	2.357	4.190	373.1	415.0	2.226
–40	8.776	385.1	429.0	2.418	4.380	385.0	428.8	2.286
–30	9.155	397.4	443.2	2.477	4.570	397.3	433.0	2.346
–20	9.533	410.1	457.8	2.536	4.760	410.0	457.6	2.405
–10	9.911	423.2	472.8	2.594	4.950	423.1	472.6	2.463
0	10.29	436.8	488.2	2.652	5.139	436.7	488.1	2.520
10	10.67	450.8	504.1	2.709	5.329	450.6	503.9	2.578
20	11.05	570.6	520.4	2.765	5.518	465.1	520.3	2.634
	p = 0.5 bar = 0.05 MPa (T_{sat} = – 56.93°C)				p = 1.0 bar = 0.1 MPa (T_{sat} = – 42.38°C)			
Sat.	0.796	363.1	402.9	1.871	0.4185	378.5	420.3	1.822
–50	0.824	371.3	412.5	1.914				
–40	0.863	383.4	426.6	1.976	0.4234	381.5	423.8	1.837
–30	0.903	396.0	441.1	2.037	0.4439	394.2	438.6	1.899
–20	0.942	408.8	455.9	2.096	0.4641	407.3	453.7	1.960
–10	0.981	422.1	471.1	2.155	0.4842	420.7	469.1	2.019
0	1.019	435.8	486.7	2.213	0.5040	434.4	484.8	2.078
10	1.058	449.8	502.7	2.271	0.5238	448.6	501.0	2.136
20	1.096	464.3	519.1	2.328	0.5434	463.3	517.6	2.194
30	1.135	479.2	535.9	2.384	0.5629	478.2	534.5	2.251
40	1.173	494.6	553.2	2.440	0.5824	493.7	551.9	2.307
50	1.211	510.4	570.9	2.496	0.6018	509.5	569.7	2.363
60	1.249	526.7	589.1	2.551	0.6211	525.8	587.9	2.419
	p = 2.0 bar = 0.2 MPa (T_{sat} = – 25.43°C)				p = 3.0 bar = 0.3 MPa (T_{sat} = – 14.16°C)			
Sat.	0.2192	396.6	440.4	1.782	0.1496	408.7	453.6	1.762
–20	0.2251	404.0	449.0	1.816				
–10	0.2358	417.7	464.9	1.877	0.1527	414.7	460.5	1.789
0	0.2463	431.8	481.1	1.938	0.1602	429.0	477.1	1.851
10	0.2566	446.3	497.6	1.997	0.1674	443.8	494.0	1.912
20	0.2669	461.1	514.5	2.056	0.1746	458.8	511.2	1.971
30	0.2770	476.3	531.7	2.113	0.1816	474.2	528.7	2.030
40	0.2871	491.9	549.3	2.170	0.1885	490.1	546.6	2.088
50	0.2970	507.9	567.3	2.227	0.1954	506.2	564.8	2.145
60	0.3070	524.3	585.7	2.283	0.2022	522.7	583.4	2.202
70	0.3169	541.1	604.5	2.339	0.2090	539.6	602.3	2.258
80	0.3267	558.4	623.7	2.394	0.2157	557.0	621.7	2.314
90	0.3365	576.1	643.4	2.449	0.2223	574.8	641.5	2.369

Table B.14 *Properties of Superheated* ***Propane***

T °C	*v* m³/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m³/kg	*u* kJ/kg	*h* kJkg	*s* kJ/kg·K
	p = 4.0 bar = 0.4 MPa (T_{sat} = – 5.46°C)				*p = 5.0 bar = 0.5 MPa* (T_{sat} = 1.74°C)			
Sat.	0.1137	418.0	463.5	1.751	0.09172	425.7	471.6	1.743
0	0.1169	426.1	472.9	1.786				
10	0.1227	441.2	490.3	1.848	0.09577	438.4	486.3	1.796
20	0.1283	456.6	507.9	1.909	0.1005	454.1	504.3	1.858
30	0.1338	472.2	525.7	1.969	0.1051	470.0	522.5	1.919
40	0.1392	488.1	543.8	2.027	0.1096	486.1	540.9	1.979
50	0.1445	504.4	562.2	2.085	0.1140	502.5	559.5	2.038
60	0.1498	521.1	581.0	2.143	0.1183	519.4	578.5	2.095
70	0.1550	538.1	600.1	2.199	0.1226	536.6	597.9	2.153
80	0.1601	555.7	619.7	2.255	0.1268	554.1	617.5	2.209
90	0.1652	573.5	639.6	2.311	0.1310	572.1	637.6	2.265
100	0.1703	591.8	659.9	2.366	0.1351	590.5	658.0	2.321
110	0.1754	610.4	680.6	2.421	0.1392	609.3	678.9	2.376
	p = 6.0 bar = 0.6 MPa (T_{sat} = 7.93°C)				*p = 7.0 bar = 0.7 MPa* (T_{sat} = 13.41°C)			
Sat.	0.07680	432.2	478.3	1.737	0.06598	438.0	484.2	1.733
10	0.07769	435.6	482.2	1.751				
20	0.08187	451.5	500.6	1.815	0.06847	448.8	496.7	1.776
30	0.08588	467.7	519.2	1.877	0.07210	465.2	515.7	1.840
40	0.08978	484.0	537.9	1.938	0.07558	481.9	534.8	1.901
50	0.09357	500.7	556.8	1.997	0.07896	498.7	554.0	1.962
60	0.09729	517.6	576.0	2.056	0.08225	515.9	573.5	2.021
70	0.1009	535.0	595.5	2.113	0.08547	533.4	593.2	2.079
80	0.1045	552.7	615.4	2.170	0.08863	551.2	613.2	2.137
90	0.1081	570.7	635.6	2.227	0.09175	569.4	633.6	2.194
100	0.1116	589.2	656.2	2.283	0.09482	587.9	654.3	2.250
110	0.1151	608.0	677.1	2.338	0.09786	606.8	675.3	2.306
120	0.1185	627.3	698.4	2.393	0.1009	626.2	696.8	2.361
	p = 8.0 bar = 0.8 MPa (T_{sat} = 18.33°C)				*p = 9.0 bar = 0.9 MPa* (T_{sat} = 22.82°C)			
Sat.	0.05776	443.1	489.3	1.729	0.05129	447.2	493.8	1.726
20	0.05834	445.9	492.6	1.740				
30	0.06170	462.7	512.1	1.806	0.05355	460.0	508.2	1.774
40	0.06489	479.6	531.5	1.869	0.05633	477.2	528.1	1.839
50	0.06796	496.7	551.1	1.930	0.05938	494.7	548.1	1.901
60	0.07094	514.0	570.8	1.990	0.06213	512.2	568.1	1.962
70	0.07385	531.6	590.7	2.049	0.06479	530.0	588.3	2.022
80	0.07669	549.6	611.0	2.107	0.06738	548.1	608.7	2.081
90	0.07948	567.9	631.5	2.165	0.06992	566.5	629.4	2.138
100	0.08222	586.5	652.3	2.221	0.07241	585.2	650.4	2.195
110	0.08493	605.6	673.5	2.277	0.07487	604.3	671.7	2.252
120	0.08761	625.0	695.1	2.333	0.07729	623.7	693.3	2.307
130	0.09026	644.8	717.0	2.388	0.07969	643.6	715.3	2.363
140	0.09289	665.0	739.3	2.442	0.08206	663.8	737.7	2.418

Table B.14 *Contd.*

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJkg	*s* kJ/kg·K
	p = 10.0 bar = 1.0 MPa (T_{sat} = 26.95°C)				*p* = 12.0 bar = 1.2 MPa (T_{sat} = 34.39°C)			
Sat.	0.04606	451.8	497.9	1.723	0.03810	459.1	504.8	1.718
30	0.04696	457.1	504.1	1.744				
40	0.04980	474.8	524.6	1.810	0.03957	469.4	516.9	1.757
50	0.05248	492.4	544.9	1.874	0.04204	487.8	538.2	1.824
60	0.05505	510.2	565.2	1.936	0.04436	506.1	559.3	1.889
70	0.05752	528.2	585.7	1.997	0.04657	524.4	580.3	1.951
80	0.05995	546.4	606.3	2.056	0.04869	543.1	601.5	2.012
90	0.06226	564.9	627.2	2.114	0.05075	561.8	622.7	2.071
100	0.06456	583.7	648.3	2.172	0.05275	580.9	644.2	2.129
110	0.06681	603.0	669.8	2.228	0.05470	600.4	666.0	2.187
120	0.06903	622.6	691.6	2.284	0.05662	620.1	688.0	2.244
130	0.07122	642.5	713.7	2.340	0.05851	640.1	710.3	2.300
140	0.07338	662.8	736.2	2.395	0.06037	660.16	733.0	2.355
	p = 14.0 bar = 1.4 MPa (T_{sat} = 40.97°C)				*p* = 16.0 bar = 1.6 MPa (T_{sat} = 46.89°C)			
Sat.	0.03231	465.2	510.4	1.714	0.02790	470.4	515.0	1.710
50	0.03446	482.6	530.8	1.778	0.02861	476.7	522.5	1.733
60	0.03664	501.6	552.9	1.845	0.03075	496.6	545.8	1.804
70	0.03869	520.4	574.6	1.909	0.03270	516.2	568.5	1.871
80	0.04063	539.4	596.3	1.972	0.03453	535.7	590.9	1.935
90	0.04249	558.6	618.1	2.033	0.03626	555.2	613.2	1.997
100	0.04429	577.9	639.9	2.092	0.03792	574.8	635.5	2.058
110	0.04604	597.5	662.0	2.150	0.03952	594.7	657.9	2.117
120	0.04774	617.5	684.3	2.208	0.04107	614.8	680.5	2.176
130	0.04942	637.7	706.9	2.265	0.04259	635.3	703.4	2.233
140	0.05107	658.3	729.8	2.321	0.04407	656.0	726.5	2.290
150	0.05268	679.2	753.0	2.376	0.04553	677.1	749.9	2.346
160	0.05428	700.5	776.5	2.431	0.04696	698.5	773.6	2.401
	p = 18.0 bar = 1.8 MPa (T_{sat} = 52.30°C)				*p* = 20.0 bar = 2.0 MPa (T_{sat} = 57.27°C)			
Sat.	0.02441	474.9	518.8	1.705	0.02157	478.7	521.8	1.700
60	0.02606	491.1	538.0	1.763	0.02216	484.8	529.1	1.722
70	0.02798	511.4	561.8	1.834	0.02412	506.3	554.5	1.797
80	0.02974	531.6	585.1	1.901	0.02585	527.1	578.8	1.867
90	0.03138	551.5	608.0	1.965	0.02744	547.6	602.5	1.933
100	0.03293	571.5	630.8	2.027	0.02892	568.1	625.9	1.997
110	0.03443	591.7	653.7	2.087	0.03033	588.4	649.2	2.059
120	0.03586	612.1	676.6	2.146	0.3169	609.2	672.6	2.119
130	0.03726	632.7	699.8	2.204	0.03299	630.0	696.0	2.178
140	0.03863	653.6	723.1	2.262	0.03426	651.2	719.7	2.236
150	0.03996	674.8	746.7	2.318	0.03550	672.5	743.5	2.293
160	0.04127	696.3	770.6	2.374	0.03671	694.2	767.6	2.349
170	0.04256	718.2	794.8	2.429	0.03790	716.2	792.0	2.404
180	0.04383	740.4	819.3	2.484	0.03907	738.5	816.6	2.459

Table B.14 *Contd.*

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJkg	s kJ/kg·K
	p = 22.0 bar = 2.2 MPa (T_{sat} = 61.90°C)				p = 24.0 bar = 2.4 MPa (T_{sat} = 66.21°C)			
Sat.	0.01921	481.8	524.0	1.695	0.01721	484.3	525.6	1.688
70	0.02086	500.5	546.4	1.761	0.01802	493.7	536.9	1.722
80	0.02261	522.4	572.1	1.834	0.01984	517.0	564.6	1.801
90	0.02417	543.5	596.7	1.903	0.02141	539.0	590.4	1.873
100	0.02561	564.5	620.8	1.969	0.02283	560.6	615.4	1.941
110	0.02697	585.3	644.6	2.032	0.02414	581.9	639.8	2.006
120	0.02826	606.2	668.4	2.093	0.02538	603.2	664.1	2.068
130	0.02949	627.3	692.2	2.153	0.02656	624.6	688.3	2.129
140	0.03069	648.6	716.1	2.211	0.02770	646.0	712.5	2.188
150	0.03185	670.1	740.2	2.269	0.02880	667.8	736.9	2.247
160	0.03298	691.9	764.5	2.326	0.02986	689.7	761.4	2.304
170	0.03409	714.1	789.1	2.382	0.03091	711.9	786.1	2.360
180	0.03517	736.5	813.9	2.437	0.03193	734.5	811.1	2.416
	p = 26.0 bar = 2.6 MPa (T_{sat} = 70.27°C)				p = 30.0 bar = 3.0 MPa (T_{sat} = 77.72°C)			
Sat.	0.01549	486.2	526.5	1.681	0.01263	488.2	526.0	1.664
80	0.01742	511.0	556.3	1.767	0.01318	495.4	534.9	1.689
90	0.01903	534.2	583.7	1.844	0.01506	522.8	568.0	1.782
100	0.02045	556.4	609.6	1.914	0.01654	547.2	596.8	1.860
110	0.02174	578.3	634.8	1.981	0.01783	570.4	623.9	1.932
120	0.02294	600.0	659.6	2.045	0.01899	593.0	650.0	1.999
130	0.02408	621.6	684.2	2.106	0.02007	615.4	675.6	2.063
140	0.02516	643.4	708.8	2.167	0.02109	637.7	701.0	2.126
150	0.02621	665.3	733.4	2.226	0.02206	660.1	726.3	2.186
160	0.02723	687.4	758.2	2.283	0.02300	682.6	751.6	2.245
170	0.02821	709.9	783.2	2.340	0.02390	705.4	777.1	2.303
180	0.02918	732.5	808.4	2.397	0.02478	728.3	802.6	2.360
190	0.03012	755.5	833.8	2.452	0.02563	751.5	828.4	2.417
	p = 35.0 bar = 3.5 MPa (T_{sat} = 86.01°C)				p = 40.0 bar = 4.0 MPa (T_{sat} = 93.38°C)			
Sat.	0.00977	486.3	520.5	1.633	0.00715	474.7	503.3	1.574
90	0.01086	502.4	540.5	1.688				
100	0.01270	532.9	577.3	1.788	0.00940	512.1	549.7	1.700
110	0.01408	558.9	608.2	1.870	0.01110	544.7	589.1	1.804
120	0.01526	583.4	636.8	1.944	0.01237	572.1	621.6	1.887
130	0.01631	607.0	664.1	2.012	0.01344	597.4	651.2	1.962
140	0.01728	630.2	690.7	2.077	0.01439	621.9	679.5	2.031
150	0.01819	653.3	717.0	2.140	0.01527	645.9	707.0	2.097
160	0.01906	676.4	743.1	2.201	0.01609	669.7	734.1	2.160
170	0.01989	699.6	769.2	2.261	0.01687	693.4	760.9	2.222
180	0.02068	722.9	795.3	2.319	0.01761	717.3	787.7	2.281
190	0.02146	746.5	821.6	2.376	0.01833	741.2	814.5	2.340
200	0.02221	770.3	848.0	2.433	0.01902	765.3	841.4	2.397

Appendix C

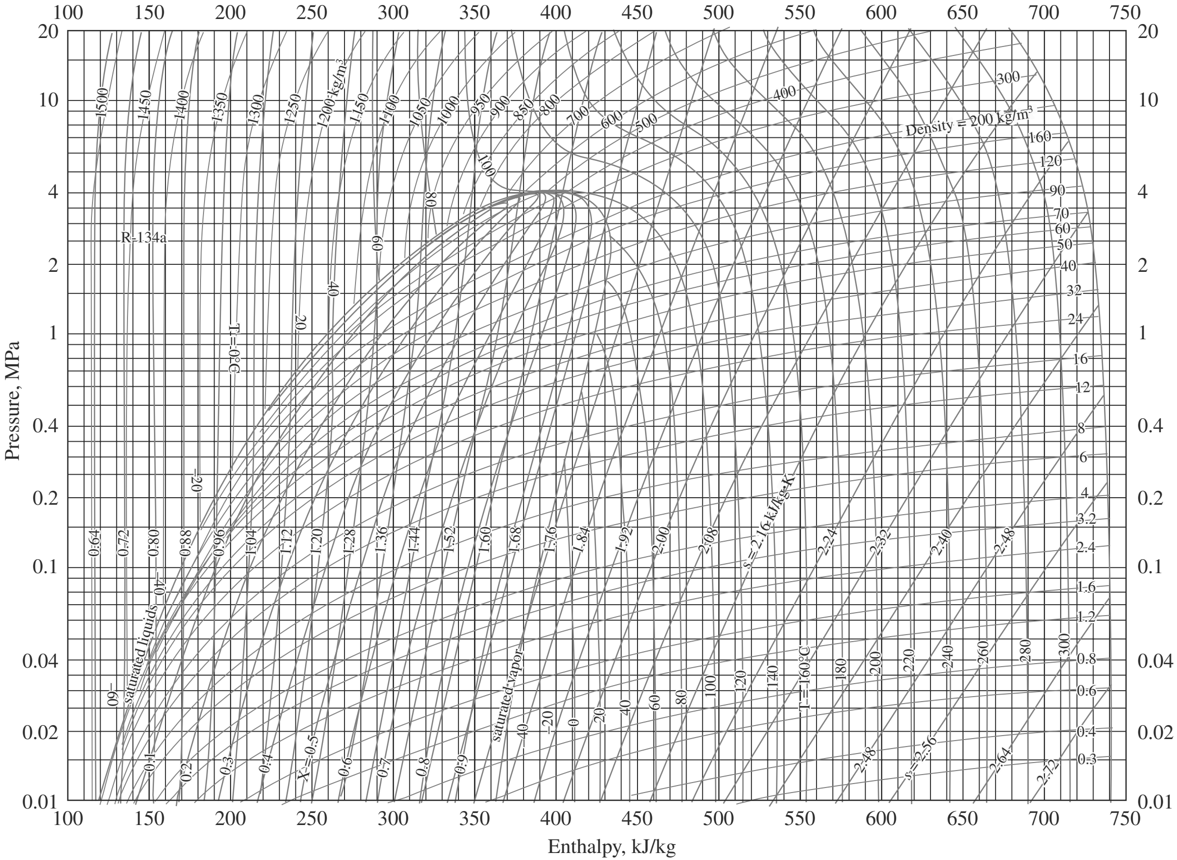


Fig. C.1 *Pressure-enthalpy diagram for refrigerant R134a*

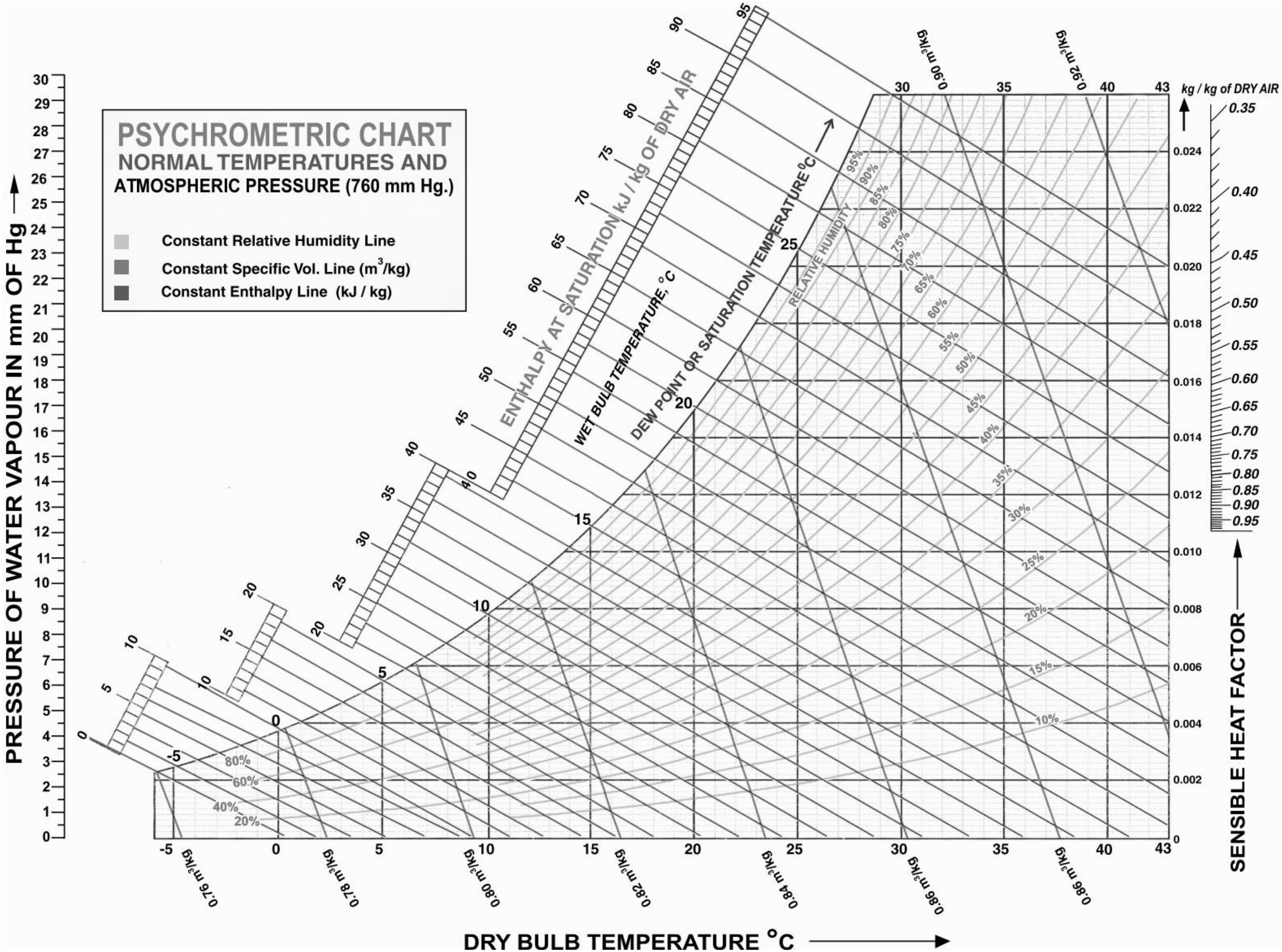

PSYCHROMETRIC CHART
NORMAL TEMPERATURES AND
ATMOSPHERIC PRESSURE (760 mm Hg.)
Constant Relative Humidity Line
Constant Specific Vol. Line (m³/kg)
Constant Enthalpy Line (kJ / kg)
ENTHALPY AT SATURATION kJ / kg OF DRY AIR
WET BULB TEMPERATURE, °C
DEW POINT OR SATURATION TEMPERATURE °C
RELATIVE HUMIDITY
PRESSURE OF WATER VAPOUR IN mm OF Hg
DRY BULB TEMPERATURE °C
kg / kg of DRY AIR
SENSIBLE HEAT FACTOR
0.76 m³/kg
0.78 m³/kg
0.80 m³/kg
0.82 m³/kg
0.84 m³/kg
0.86 m³/kg
0.88 m³/kg
0.90 m³/kg
0.92 m³/kg
-5 0 5 10 15 20 25 30 35 40 43
0 0.002 0.004 0.006 0.008 0.010 0.012 0.014 0.016 0.018 0.020 0.022 0.024
0.35 0.40 0.45 0.50 0.55 0.60 0.65 0.70 0.75 0.80 0.85 0.90 0.95
10% 15% 20% 25% 30% 35% 40% 45% 50% 55% 60% 65% 70% 75% 80% 85% 90% 95%

Fig. C.2