EIGHTH EDITION

Thermal Engineering

(In S.I. Units)

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Published by :

LAXMI PUBLICATIONS (P) LTD
113, Golden House, Daryaganj,
New Delhi-110002
Phone : 011-43 53 25 00
Fax : 011-43 53 25 28

www.laxmipublications.com
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Price : **Rs. 750.00** *Only.*

First Edition : 1993 ; *Second Edition* : 1999
Third Edition : 2001 ; *Fourth Edition* : 2004
Fifth Edition : 2005 ; *Sixth Edition* : 2006
Seventh Edition : 2009 ; *Eighth Edition* : 2010

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ETE-0609-750-THERMAL ENGG

C—900/12/09

Typeset at : Goswami Associates, Delhi

Printed at : Print Man (India)

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Foreword

It is a pleasure for me to write these few words on Shri R.K. Rajput's latest book on "Thermal Engineering".

Written by a highly experienced and gifted teacher of our pace-setting Polytechnic, this book deals with Thermodynamics and Heat Engines besides having an additional chapter on Non-Conventional Power Generation. I have no doubt that it will prove extremely useful for students of engineering institutions in our country.

Among the several distinctive features of this book, special mention may be made of the following : development of the subject in a logical and coherent manner, neat and simple diagrams to explain the text, large number of worked examples as well as unsolved examples for practice, numerous objective-type questions and the consistent use of SI units.

As is well known to his peers, the author of this volume has written a number of popular books earlier on various engineering subjects for the benefit of students in Polytechnics and Engineering Colleges. As is to be expected, a great deal of systematic and meticulous work has gone into the writing of this book.

I wish the author and his book all success.

Professor Dr. T.R. ANANTHARAMAN
Director, Thapar Institute of Engineering & Technology & Thapar Corporate Research & Development Centre
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Preface to The Eighth Edition

I am pleased to present the **Eighth Edition** (*Two Colours*) of this standard treatise. The warm reception which the previous editions and reprints of this book have enjoyed all over India and abroad, has been a matter of great satisfaction to me.

The entire book has been thoroughly revised ; besides adding a large number of **"Additional/Typical Worked Examples"** in various chapters.

It is hoped that the book in this comprehensive and complete form will prove to be of immense use for the students preparing various 'Universities' and Competitive Examinations (*e.g.,* UPSC, GATE etc.)

The author's thanks are due to his wife Ramesh Rajput for extending all cooperation during preparation and proof reading of this revised manuscript.

Any suggestions for the improvement of this book will be thankfully acknowledged and incorporated in the next edition.

—Author

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Preface to The First Edition

On the subject of "Thermal Engineering" several books are available in the market but most of them either lack in proper subject treatment or the solved examples are small in number and their solutions are too difficult to understand. The purpose of writing this book is to bridge these gaps and to present such a book to the students which should be easy to understand and assimilate comprehensive treatment of the subject matter in simple, lucid and direct language.

The book comprises of 26 Chapters, dealing with **'Thermodynamics'**, **'Heat Engines'** and **'Non-Conventional Power Generation'**.

Each chapter contains adequate text supported by simple and self explanatory figures, large number of Worked Examples both in SI and MKS units, Highlights, Theoretical Questions and several Unsolved Examples.

The book will prove to be of great help to the students preparing for engineering degree, A.M.I.E. (India), diploma and other competitive examinations.

The author's grateful thanks are due to Smt. Ramesh Rajput (wife) for extending full co-operation during the preparation of the manuscript.

The author is also thankful to Mr. Rakesh Syal for drawing neat diagrams for this book.

In the end, the author wishes to express his grateful thanks to M/s Laxmi Publications for taking extra pains in printing the book at a rapid pace and very systematically.

Although every care has been taken to make the book free of errors both in text as well as in solved examples, yet the author shall feel obliged if errors present are brought to his notice. Constructive criticism of the book will be warmly received.

—Author

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Introduction to SI Units and Conversion Factors

A. INTRODUCTION TO SI UNITS

SI, the international system of units are divided into three classes :

1. Base units

2. Derived units

3. Supplementary units.

From the scientific point of view division of SI units into these classes is to a certain extent arbitrary, because it is not essential to the physics of the subject. Nevertheless the General Conference, considering the advantages of a single, practical, world-wide system for international relations, for teaching and for scientific work, decided to base the international system on a choice of six well-defined units given in Table 1 below :

Table 1. SI Base Units

Quantity	*Name*	*Symbol*
length	metre	m
mass	kilogram	kg
time	second	s
electric current	ampere	A
thermodynamic temperature	kelvin	K
luminous intensity	candela	cd
amount of substance	mole	mol

The second class of SI units contains derived units, *i.e.*, units which can be formed by combining base units according to the algebraic relations linking the corresponding quantities. Several of these algebraic expressions in terms of base units can be replaced by special names and symbols can themselves be used to form other derived units.

Derived units may, therefore, be classified under three headings. Some of them are given in Tables 2, 3 and 4.

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Table 2. Examples of SI Derived Units Expressed in terms of Base Units

Quantity	*SI Units*	
	Name	*Symbol*
area	square metre	m^2
volume	cubic metre	m^3
speed, velocity	metre per second	m/s
acceleration	metre per second squared	m/s^2
wave number	1 per metre	m^{-1}
density, mass density	kilogram per cubic metre	kg/m^3
concentration (of amount of substance)	mole per cubic metre	mol/m^3
activity (radioactive)	1 per second	s^{-1}
specific volume	cubic metre per kilogram	m^3/kg
luminance	candela per square metre	cd/m^2

Table 3. SI Derived Units with Special Names

Quantity	*SI Units*			
	Name	*Symbol*	*Expression in terms of other units*	*Expression in terms of SI base units*
frequency	hertz	Hz	—	s^{-1}
force	newton	N	—	$m.kg.s^{-2}$
pressure	pascal	Pa	N/m^2	$m^{-1}.kg.s^{-2}$
energy, work, quantity of heat power	joule	J	N.m	$m^2.kg.s^{-2}$
radiant flux quantity of electricity	watt	W	J/S	$m^2.kg.s^{-3}$
electric charge	coloumb	C	A.s	s.A
electric tension, electric potential	volt	V	W/A	$m^2.kg.s^{-3}.A^{-1}$
capacitance	farad	F	C/V	$m^{-2}.kg^{-1}.s^4$
electric resistance	ohm	Ω	V/A	$m^2.kg.s^{-3}.A^{-2}$
conductance	siemens	S	A/V	$m^{-2}.kg^{-1}.s^3.A^2$
magnetic flux	weber	Wb	V.S.	$m^2.kg.s^{-2}.A^{-1}$
magnetic flux density	tesla	T	Wb/m^2	$kg.s^{-2}.A^{-1}$
inductance	henry	H	Wb/A	$m^2.kg.s^{-2}.A^{-2}$
luminous flux	lumen	lm	—	cd.sr
illuminance	lux	lx	—	$m^{-2}.cd.sr$

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Table 4. Examples of SI Derived Units Expressed by means of Special Names

Quantity	*SI Units*		
	Name	*Symbol*	*Expression in terms of SI base units*
dynamic viscosity	pascal second	Pa-s	$m^{-1}.kg.s^{-1}$
moment of force	metre newton	N.m	$m^2.kg.s^{-2}$
surface tension	newton per metre	N/m	$kg.s^{-2}$
heat flux density, irradiance	watt per square metre	W/m^2	$kg.s^{-3}$
heat capacity, entropy	joule per kelvin	J/K	$m^2.kg.s^{-2}.K^{-1}$
specific heat capacity, specific entropy	joule per kilogram kelvin	J/(kg.K)	$m^2.s^{-2}.K^{-1}$
specific energy	joule per kilogram	J/kg	$m^2.s^{-2}$
thermal conductivity	watt per metre kelvin	W/(m.K)	$m.kg.s^{-3}.K^{-1}$
energy density	joule per cubic metre	J/m^3	$m^{-1}.kg.s^{-2}$
electric field strength	volt per metre	V/m	$m.kg.s^{-3}.A^{-1}$
electric charge density	coloumb per cubic metre	C/m^3	$m^{-3}.s.A$
electric flux density	coloumb per square metre	C/m^2	$m^{-2}.s.A$
permitivity	farad per metre	F/m	$m^{-3}.kg^{-1}.s^4$
current density	ampere per square metre	A/m^2	—
magnetic field strength	ampere per metre	A/m	—
permeability	henry per metre	H/m	$m.kg.s^{-2}.A^{-2}$
molar energy	joule per mole	J/mol	$m^2.kg.s^{-2}mol^{-1}$
molar heat capacity	joule per mole kelvin	J/(mol.K)	$m^2.kg.s^{-2}.K^{-1}.mol^{-1}$

The SI units assigned to third class called "Supplementary units" may be regarded either as base units or as derived units. Refer Table 5 and Table 6.

Table 5. SI Supplementary Units

Quantity	*SI Units*	
	Name	*Symbol*
plane angle	radian	rad
solid angle	steradian	sr

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Table 6. Examples of SI Derived Units Formed by Using Supplementary Units

Quantity	*SI Units*	
	Name	*Symbol*
angular velocity	radian per second	rad/s
angular acceleration	radian per second squared	rad/s^2
radiant intensity	watt per steradian	W/sr
radiance	watt per square metre steradian	$W\text{-}m^{-2}.sr^{-1}$

Table 7. SI Prefixes

Factor	*Prefix*	*Symbol*	*Factor*	*Prefix*	*Symbol*
10^{12}	tera	T	10^{-1}	deci	d
10^{9}	giga	G	10^{-2}	centi	c
10^{6}	mega	M	10^{-3}	milli	m
10^{3}	kilo	k	10^{-6}	micro	μ
10^{2}	hecto	h	10^{-9}	nano	n
10^{1}	deca	da	10^{-12}	pico	p
			10^{-15}	fasnto	f
			10^{-18}	atto	a

B. CONVERSION FACTORS

1. Force :

$$1 \text{ newton} = \text{kg-m/sec}^2 = 0.012 \text{ kgf}$$

$$1 \text{ kgf} = 9.81 \text{ N}$$

2. Pressure :

$$1 \text{ bar} = 750.06 \text{ mm Hg} = 0.9869 \text{ atm} = 10^5 \text{ N/m}^2 = 10^3 \text{ kg/m-sec}^2$$

$$1 \text{ N/m}^2 = 1 \text{ pascal} = 10^{-5} \text{ bar} = 10^{-2} \text{ kg/m-sec}^2$$

$$1 \text{ atm} = 760 \text{ mm Hg} = 1.03 \text{ kgf/cm}^2 = 1.01325 \text{ bar}$$
$$= 1.01325 \times 10^5 \text{ N/m}^2$$

3. Work, Energy or Heat :

$$1 \text{ joule} = 1 \text{ newton metre} = 1 \text{ watt-sec}$$
$$= 2.7778 \times 10^{-7} \text{ kWh} = 0.239 \text{ cal}$$
$$= 0.239 \times 10^{-3} \text{ kcal}$$

$$1 \text{ cal} = 4.184 \text{ joule} = 1.1622 \times 10^{-6} \text{ kWh}$$

$$1 \text{ kcal} = 4.184 \times 10^3 \text{ joule} = 427 \text{ kgf-m}$$
$$= 1.1622 \times 10^{-3} \text{ kWh}$$

$$1 \text{ kWh} = 8.6042 \times 10^5 \text{ cal} = 860 \text{ kcal} = 3.6 \times 10^6 \text{ joule}$$

$$1 \text{ kgf-m} = \left(\frac{1}{427}\right) \text{ kcal} = 9.81 \text{ joules}$$

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4. Power :

1 watt = 1 joule/sec = 0.860 kcal/h

1 h.p. = 75 m kgf/sec = 0.1757 kcal/sec = 735.3 watt

1 kW = 1000 watts = 860 kcal/h

5. Specific heat :

1 kcal/kg-°K = 0.4184 joules/kg-K

6. Thermal conductivity :

1 watt/m-K = 0.8598 kcal/h-m-°C

1 kcal/h-m-°C = 1.163 watt/m-K = 1.163 joules/s-m-K.

7. Heat transfer co-efficient :

1 watt/m^2-K = 0.8598 kcal/m^2-h-°C

1 kcal/m^2-h-°C = 1.163 watt/m^2-K.

C. IMPORTANT ENGINEERING CONSTANTS AND EXPRESSIONS

Engineering constants and expressions	*M.K.S. system*	*SI Units*
1. Value of g_0	9.81 kg-m/kgf-sec^2	1 kg-m/N-sec^2
2. Universal gas constant	848 kgf-m/kg mole-°K	848 × 9.81 = 8314 J/kg-mole-K (∵ 1 kgf-m = 9.81 joules)
3. Gas constant (R)	29.27 kgf-m/kg-°K for air	$\frac{8314}{29}$ = 287 joules/kg-K for air
4. Specific heats (for air)	c_v = 0.17 kcal/kg-°K c_p = 0.24 kcal/kg-°K	c_v = 0.17 × 4.184 = 0.71128 kJ/kg-K c_p = 0.24 × 4.184 = 1 kJ/kg-K
5. Flow through nozzle-Exit velocity (C_2)	$91.5\sqrt{U}$, where U is in kcal	$44.7\sqrt{U}$, where U is in kJ
6. Refrigeration 1 ton	= 50 kcal/min	= 210 kJ/min
7. **Heat transfer** The Stefan Boltzman Law is given by :	$Q = \sigma T^4$ kcal/m^2-h where $\sigma = 4.9 \times 10^{-8}$ kcal/h-m^2-°K^4	$Q = \sigma T^4$ watts/m^2-h where $\sigma = 5.67 \times 10^{-8}$ W/m^2K^4

Nomenclature

A	area
b	steady-flow availability function
C	velocity
$°C$	temperature on the celsius (or centigrade) scale
c	specific heat
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
C_p	molar heat at constant pressure
C_v	molar heat at constant volume
D, d	bore ; diameter
E	emissive power ; total energy
e	base of natural logarithms
g	gravitational acceleration
H	enthalpy
h	specific enthalpy ; heat transfer co-efficient
h_f	specific enthalpy of saturated liquid (fluid)
h_{fg}	latent heat
h_g	specific enthalpy of saturated vapour ; gases
K	temperature on kelvin scale (*i.e.,* celsius absolute, compressibility)
k	thermal conductivity, blade velocity co-efficient
L	stroke
M	molecular weight
m	mass
$\dot{m}$	rate of mass flow
N	rotational speed
n	polytropic index, number of moles ; number of cylinders
P	power
p	absolute pressure
p_m	mean effective pressure
p_i	indicated mean effective pressure
p_b	brake mean effective pressure, back pressure

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Q	heat, rate of heat transfer
q	rate of heat transfer per unit area
R	gas constant ; thermal resistance ; radius ; total expansion ratio in compound steam engines
R_0	universal gas constant
r	radius, expansion ratio, compression ratio
S	entropy
s	specific entropy
T	absolute temperature ; torque
t	temperature
U	internal energy ; overall heat transfer co-efficient
u	specific internal energy
V	volume
v	specific volume
W	work ; rate of work transfer ; brake load ; weight
w	specific weight ; velocity of whirl
x	dryness fraction ; length

Greek Symbols

α	absorptivity
γ	ratio of specific heats, c_p/c_v
$\in$	emissivity ; effectiveness
η	efficiency
θ	temperature difference, angle
ρ	density
σ	Stefan-Boltzmann constant
ϕ	relative humidity, angle.

1 Kinetic Theory of Gases and Basic Concepts of Thermodynamics

1.1. Introduction to kinetic theory of gases. 1.2. Definition of thermodynamics. 1.3. Thermodynamic systems—System, boundary and surroundings—Closed system—Open system—Isolated system—Adiabatic system—Homogeneous system—Heterogeneous system. 1.4. Macroscopic and microscopic points of view. 1.5. Pure substance. 1.6. Thermodynamic equilibrium. 1.7. Properties of systems. 1.8. State. 1.9. Process. 1.10. Cycle. 1.11. Point function. 1.12. Path function. 1.13. Temperature. 1.14. Zeroth law of thermodynamics. 1.15. The thermometer and thermometric property—Introduction— Measurement of temperature—The international practical temperature scale—Ideal gas. 1.16. Pressure—Definition of pressure—Unit for pressure—Types of pressure measurement devices—Mechanical-type instruments. 1.17. Specific volume. 1.18. Reversible and irreversible processes. 1.19. Energy, work and heat—Energy—Work and heat. 1.20. Reversible work—Highlights—Objective Type Questions—Theoretical Questions— Unsolved Examples.

1.1. INTRODUCTION TO KINETIC THEORY OF GASES

The kinetic theory of gases deals with the behaviour of molecules constituting the gas. According to this theory, the molecules of all gases are in continuous motion. As a result of this they possess kinetic energy which is transferred from molecule to molecule during their collision. The energy so transferred produces a change in the velocity of individual molecules.

The complete phenomenon of molecular behaviour is quite complex. The *assumptions* are therefore made to simplify the application of theory of an ideal gas.

Assumptions :

1. The molecules of gases are assumed to be rigid, perfectly elastic solid spheres, identical in all respects such as mass, form etc.
2. The mean distance between molecules is very large compared to their own dimensions.
3. The molecules are in state of random motion moving in all directions with all possible velocities and gas is said to be in state of molecular chaos.
4. The collisions between the molecules are perfectly elastic and there are no intermolecular forces of attraction or repulsion. This means that energy of gas is all kinetic.
5. The number of molecules in a small volume is very large.
6. The time spent in collision is negligible, compared to the time during which the molecules are moving independently.
7. Between collisions, the molecules move in a straight line with uniform velocity because of frictionless motion between molecules. The distance between two collisions is called '*free path*' of the molecule, the average distance travelled by a molecule between successive collision is known as '*mean free path*'.
8. The volume of molecule is so small that it is negligible compared to total volume of the gas.

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Pressure exerted by an Ideal Gas

Let us consider a quantity of gas to be contained in a cubical vessel of side l with perfectly elastic wall and N represent the very large number of molecules in the vessel. Now let us consider a molecule which may be assumed to have a velocity C_1 in a certain direction. The velocity can be resolved into three components u_1, v_1, w_1 parallel to three co-ordinate axes X, Y and Z which are again assumed parallel to the sides of the cube as shown in Fig. 1.1.

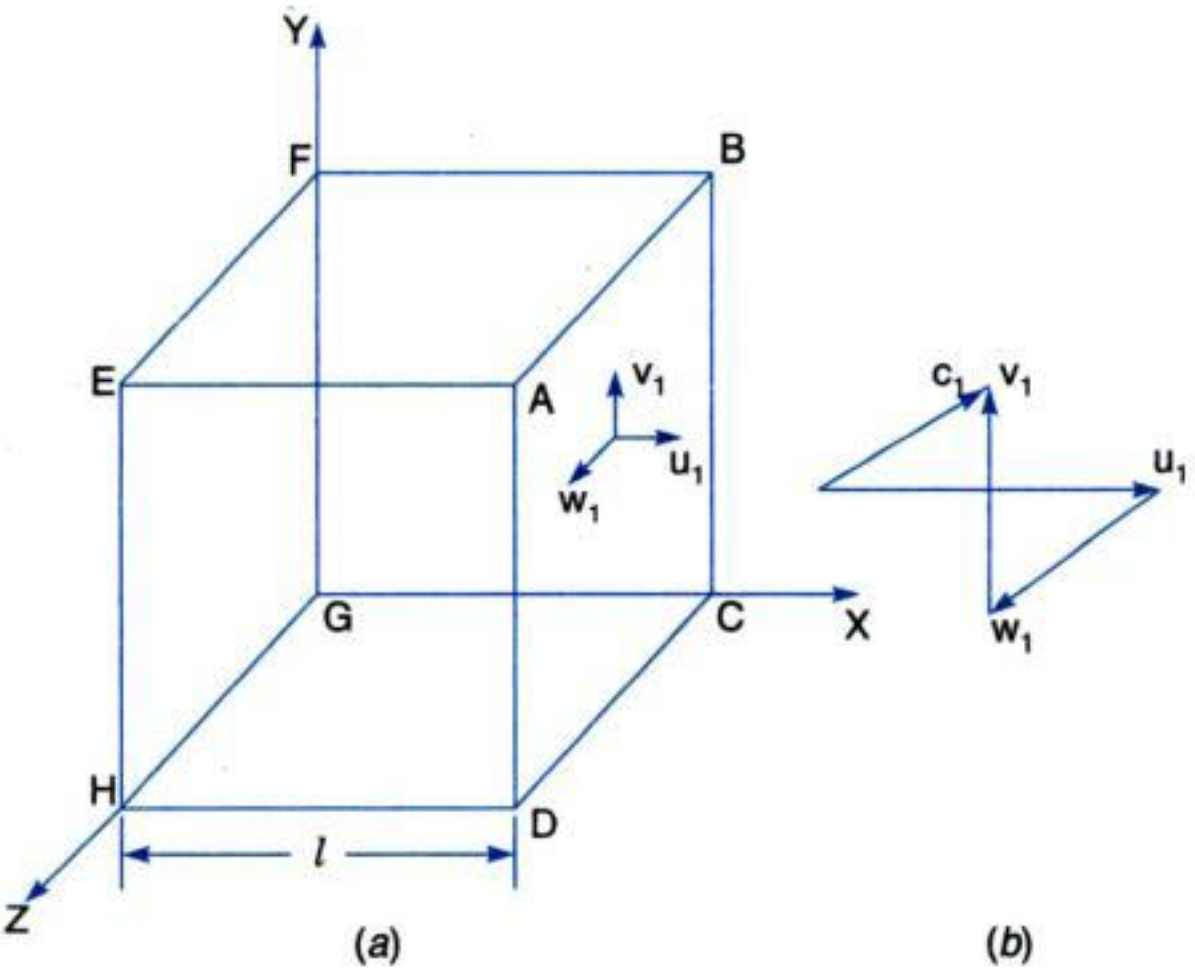

Fig. 1.1

Thus, $$C_1^2 = u_1^2 + v_1^2 + w_1^2 .$$

Let this molecule having mass m strike wall surface $ABCD$ of the cube with velocity u_1. Since the collision is perfectly elastic, the molecule will rebound from this surface with the same velocity u_1. Therefore,

The momentum of the molecule before it strikes the face $ABCD = mu_1$

The momentum of the molecule after impact $= -mu_1$.

Hence change of momentum at each impact in direction normal to the surface

$$ABCD = mu_1 - (-mu_1) = 2mu_1$$

After striking the surface $ABCD$, the molecule rebounds and travels back to the face $EFGH$, collides with it and travels back again to the face $ABCD$ covering $2l$ distance. This means molecule covers $2l$ distance to hit the same face again. Hence the time taken by the same molecule to strike the same face $ABCD$ again is $\frac{2l}{u_1}$.

Therefore, the rate of change of momentum for one molecule of the gas

$$= \frac{2mu_1}{\frac{2l}{u_1}} = \frac{mu_1^2}{l}$$

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According to Newton's second law of motion the rate of change of 'momentum is the force'. If F_1 is the force due to one molecule, then

$$F_1 = \frac{mu_1^2}{l}$$

Similarly, then force F_2 due to the impact of another molecule having velocity C_2 whose components are u_2, v_2, w_2 is given by

$$F_2 = \frac{mu_2^2}{l}$$

Hence total force F_x on the face $ABCD$ due to impact of N molecules is given by

$$F_x = \frac{m}{l}(u_1^2 + u_2^2 + \ldots\ldots u_N^2)$$

Since the pressure (p) is the force per unit area, hence pressure exerted on the wall $ABCD$ is given by

$$p_x = \frac{F_x}{l^2} = \frac{m}{l^3}(u_1^2 + u_2^2 + \ldots\ldots u_N^2)$$

Similarly, if p_y and p_z represent the pressures on other faces which are perpendicular to the Y and Z-axes respectively, we have

$$p_y = \frac{m}{l^3}(v_1^2 + v_2^2 + \ldots\ldots v_N^2)$$

and

$$p_z = \frac{m}{l^3}(w_1^2 + w_2^2 + \ldots\ldots w_N^2)$$

Since pressure exerted by the gas is the same in all directions, *i.e.*, $p_x = p_y = p_z$ the average pressure p of the gas is given by

$$p = \frac{p_x + p_y + p_z}{3}$$

or

$$p = \frac{m}{l^3}[(u_1^2 + v_1^2 + w_1^2) + (u_2^2 + v_2^2 + w_2^2) + \ldots\ldots(u_N^2 + v_N^2 + w_N^2)$$

But

$$C_1^2 = (u_1^2 + v_1^2 + w_1^2)$$

$$C_2^2 = (u_2^2 + v_2^2 + w_2^2) \text{ and so on}$$

$$l^3 = V = \text{volume of gas } (m^3)$$

$\therefore$

$$p = \frac{1}{3}\frac{m}{v}(C_1^2 + C_2^2 + C_3^2 + \ldots\ldots C_N^2)$$

or

$$p = \frac{1}{3}\frac{m}{v}N\overline{C}^2 \qquad \ldots(1.1)$$

where

$$\overline{C}^2 = \left(\frac{C_1^2 + C_2^2 + C_3^2 + \ldots\ldots C_N^2}{N}\right) \text{ known as mean square velocity}$$

or

$$\overline{C} = \sqrt{\frac{C_1^2 + C_2^2 + C_3^2 + \ldots\ldots C_N^2}{N}}$$

where $\overline{C}$ is called the root *mean square velocity* of the molecules and equal to the square root of the mean of square of velocities of individual molecules which is evidently not the same as mean of velocities of different molecules

$$\left[i.e., C_{mean} = \frac{C_1 + C_2 + C_3 + \ldots\ldots C_N}{N}\right]$$

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or $$pV = \frac{1}{3} m N\overline{C}^2 \qquad ...(1.2)$$

This *equation is the fundamental equation of kinetic theory of gases* and is often referred to as ***kinetic equation of gases.***

Equation (1.2) may be written as

$$pV = 2/3 \times 1/2\ m\ N\overline{C}^2$$

where $\frac{1}{2} mN\overline{C}^2$ is the *average transmission or linear kinetic energy of the system of particles.*

Equation (1.1) can be written as

$$p = 1/3\ \rho\ \overline{C}^2 \qquad ...(1.3)$$

where ρ is the density. $\left[\because \quad \rho = \frac{mN}{V}, i.e., \frac{\text{Total mass}}{\text{Total volume}}\right]$

This equation expresses the pressure which any volume of gas exerts in terms of its density under the prevailing conditions and its mean square molecular speed.

From equations (1.2) and (1.3),

$$\overline{C} = \sqrt{\frac{3p}{\rho}} = \sqrt{\frac{3pV}{mN}}$$

Kinetic interpretation of temperature :

If V_{mol} is the volume occupied by a gram molecule of a gas and N_0 is the number of moles in one gram molecule of gas,

M = molecular weight = mN_0. ...(*i*)

Since $p\ V_{mol} = R_0T$ .....Molar gas equation ...(*ii*)

From equations (1.2) and (*ii*),

$$1/3\ m\ N_0\ \overline{C} = R_0T$$

or, $$2/3 \times \tfrac{1}{2} m\ N_0\ \overline{C}^2 = R_0T$$

or, $$\tfrac{1}{2}\ m\overline{C}^2 = 3/2\ KT \qquad ...(1.4)$$

(*i.e.*, K.E. per molecule = 3/2 *KT*)

or, $$\overline{C} = \sqrt{\frac{3KT}{m}}$$

or, $$\overline{C} = \sqrt{\frac{3R_0T}{M}}$$

or, $$\overline{C} = \sqrt{3RT} \qquad ...(1.5)$$

R_0 = Universal gas constant

N_0 = Avogadro's number

$\frac{R_0}{N_0} = K$ (Boltzman's constant)

$\because \frac{K}{m} = \frac{R_0}{N_0\ m} = \frac{R_0}{M}$

$\because R = \frac{R_0}{M}$

where R is characteristic gas constant.

From equation (1.4) it is seen that *temperature is a measure* of the *average kinetic energy of translation possessed by molecule.* It is known as the ***kinetic interpretation of temperature.*** Hence, the absolute temperature of a gas is proportional to the mean translational kinetic energy of the molecules it consists. If the temperature is fixed, then the average K.E. of the molecules remains constant despite encounters.

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1.2. DEFINITION OF THERMODYNAMICS

Thermodynamics may be *defined* as follows :

Thermodynamics is an axiomatic science which deals with the relations among heat, work and properties of system which are in equilibrium. It describes state and changes in state of physical systems.

Or

Thermodynamics is *the science of the regularities governing processes of energy conversion.*

Or

Thermodynamics is *the science that deals with the interaction between energy and material systems.*

Thermodynamics, basicaily entails *four laws* or axioms known as Zeroth, First, Second and Third law of thermodynamics.

- The ***First law*** throws light on *concept of internal energy.*
- The ***Zeroth law*** deals with *thermal equilibrium* and establishes a *concept of temperature.*
- The ***Second law*** indicates the limit of *converting heat into work* and introduces the *principle of increase of entropy.*
- The ***Third law*** defines the *absolute zero of entropy.*

These laws are based on experimental observations and have no *mathematical proof.* Like all physical laws, these laws are based on *logical reasoning.*

1.3. THERMODYNAMIC SYSTEMS

1.3.1. System, Boundary and Surroundings

System. A system is *a finite quantity of matter or a prescribed region of space* (Refer Fig. 1.2)

Boundary. The *actual or hypothetical envelope enclosing the system* is the boundary of the system. The boundary may be fixed or it may move, as and when a system containing a gas is compressed or expanded. The boundary may be *real* or *imaginary.* It is not difficult to envisage a real boundary but an example of imaginary boundary would be one drawn around a system consisting of the fresh mixture about to enter the cylinder of an I.C. engine together with the remanants of the last cylinder charge after the exhaust process (Refer Fig. 1.3).

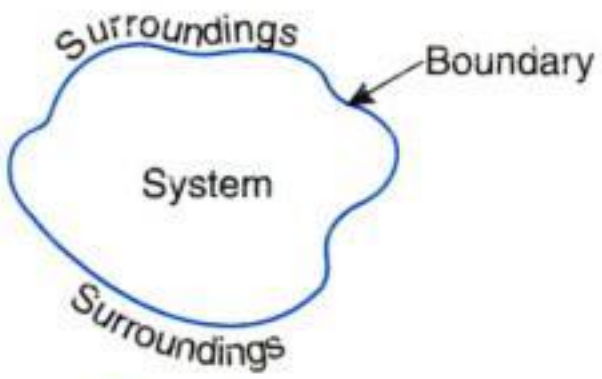

Fig. 1.2. The system.

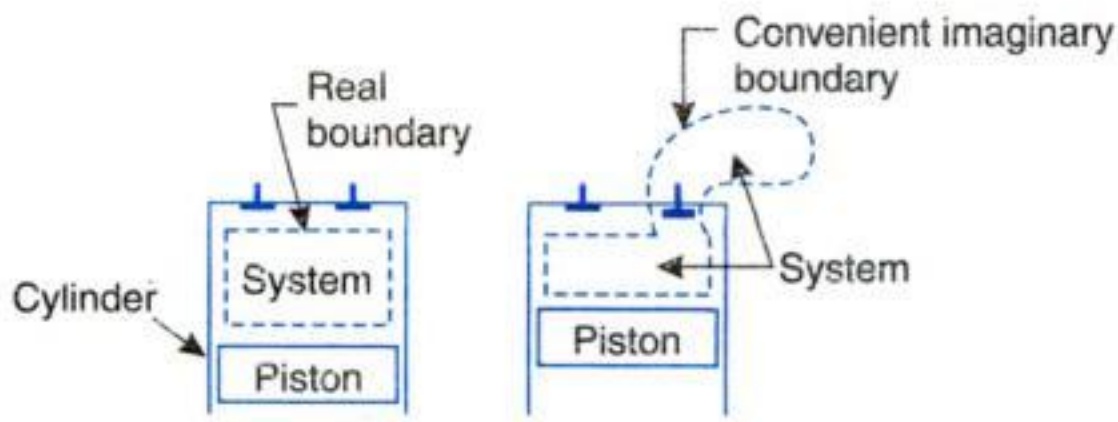

Fig. 1.3. The real and imaginary boundaries.

1.3.2. Closed System

Refer Fig. 1.4. *If the boundary of the system is impervious to the flow of matter, it is called* a ***closed system.*** An *example* of this system is mass of gas or vapour contained in an engine cylinder, the boundary of which is drawn by the cylinder walls, the cylinder head and piston crown. Here the *boundary is continuous and no matter may enter or leave.*

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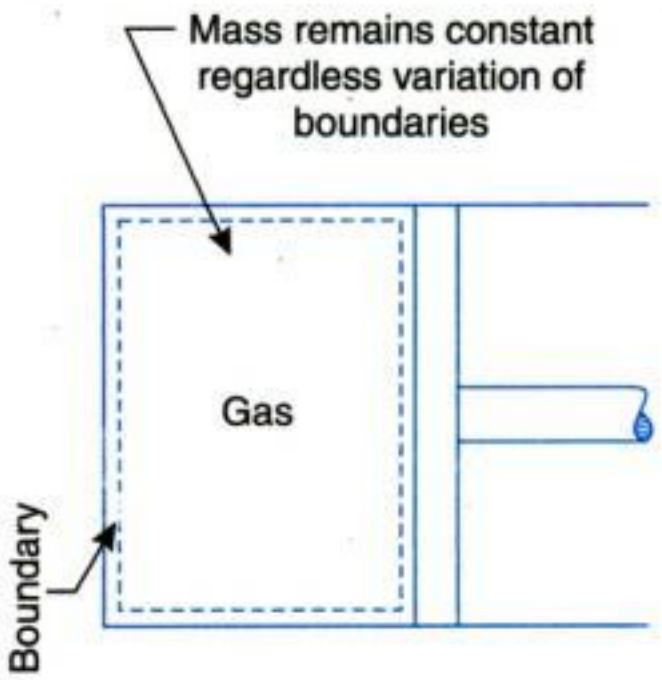

Fig. 1.4. Closed system.

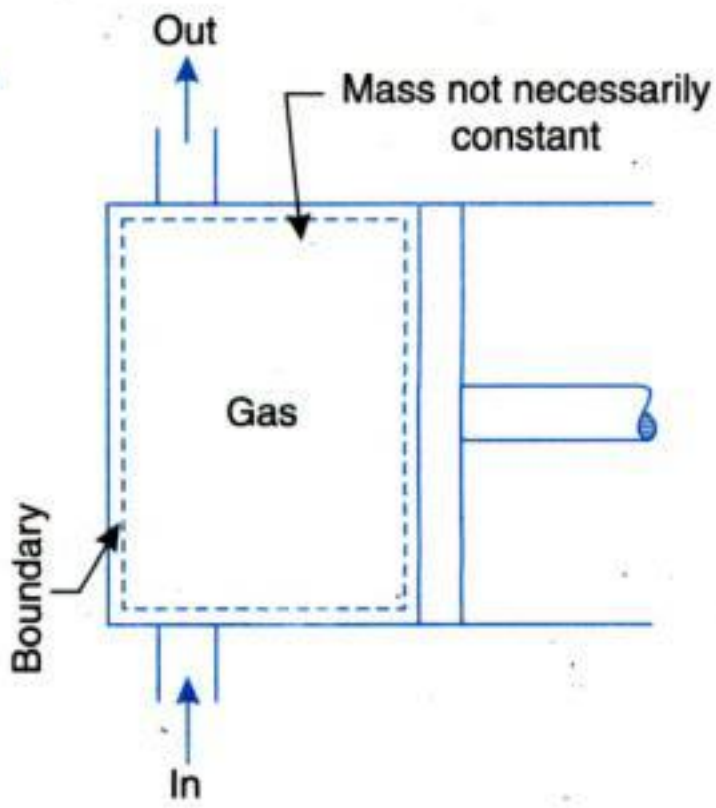

Fig. 1.5. Open system.

1.3.3. Open System

Refer to Fig. 1.5. An open system is one in which *matter flows into or out of the system*. Most of the engineering systems are open.

1.3.4. Isolated System

An isolated system is that system *which exchanges neither energy nor matter with any other system or with environment*.

1.3.5. Adiabatic System

An adiabatic system is one *which is thermally insulated from its surroundings*. It can, however, *exchange work with its surroundings*. If it does not, it becomes an isolated system.

Phase. A phase is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.

1.3.6. Homogeneous System

A system which consists of a single phase is termed as *homogeneous system*. Examples : Mixture of air and water vapour, water plus nitric acid and octane plus heptane.

1.3.7. Heterogeneous System

A system which consists of two or more phases is called a *heterogeneous system*. Examples : Water plus steam, ice plus water and water plus oil.

1.4. MACROSCOPIC AND MICROSCOPIC POINTS OF VIEW

Thermodynamic studies are undertaken by the following two different approaches :

1. Macroscopic approach—(*Macro* mean *big* or *total*)
2. Microscopic approach—(*Micro* means *small*)

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These approaches are discussed (in a comparative way) below :

S. No.	*Macroscopic approach*	*Microscopic approach*
1.	In this approach a certain quantity of matter is considered *without* taking into account the events occurring at molecular level. In other words this approach to thermodynamics is concerned with *gross or overall behaviour*. This is known as *classical thermodynamics*.	The approach considers that the system is made up of a very large number of discrete particles known as *molecules*. These molecules have different velocities and energies. The values of these energies are constantly changing with time. This approach to thermodynamics which is concerned directly with the *structure of the matter* is known as *statistical thermodynamics*.
2.	The analysis of macroscopic system requires simple mathematical formulae.	The behaviour of the system is found by using statistical methods as the number of molecules is very large. So advanced statistical and mathematical methods are needed to explain the changes in the system.
3.	The values of the properties of the system are their average values. For example, consider a sample of a gas in a closed container. The *pressure* of the gas is the average value of the pressure exerted by millions of individual molecules. Similarly the *temperature* of this gas is the average value of translational kinetic energies of millions of individual molecules. These properties like *pressure* and *temperature* can be measured very easily. *The changes in properties can be felt by our senses.*	The properties like *velocity, momentum*, impulse, kinetic energy, force of impact etc. which describe the molecule *cannot be easily measured by instruments. Our senses cannot feel them*.
4.	In order to describe a system only a few properties are needed.	Large number of variables are needed to describe a system. So the approach is complicated.

Note. Although the macroscopic approach seems to be different from microscopic one, there exists a relation between them. Hence when both the methods are applied to a particular system, they give the same result.

1.5. PURE SUBSTANCE

A pure substance is one that has a homogeneous and invariable chemical composition even though there is a change of phase. In other words, it is a system which is (*a*) homogeneous in composition, (*b*) homogeneous in chemical aggregation. Examples : Liquid, water, mixture of liquid water and steam, mixture of ice and water. The mixture of liquid air and gaseous air is not a pure substance.

1.6. THERMODYNAMIC EQUILIBRIUM

A system is in *thermodynamic equilibrium* if the temperature and pressure at all points are same ; there should be no velocity gradient ; the chemical equilibrium is also necessary. Systems under temperature and pressure equilibrium but not under chemical equilibrium are sometimes said to be in metastable equilibrium conditions. *It is only under thermodynamic equilibrium conditions that the properties of a system can be fixed*.

Thus for attaining a state of *thermodynamic equilibrium* the following *three* types of equilibrium states must be achieved :

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1. **Thermal equilibrium.** The temperature of the system does not change with time and has same value at all points of the system.

2. **Mechanical equilibrium.** There are no unbalanced forces within the system or between the surroundings. The pressure in the system is same at all points and does not change with respect to time.

3. **Chemical equilibrium.** No chemical reaction takes place in the system and the chemical composition which is same throughout the system does not vary with time.

1.7. PROPERTIES OF SYSTEMS

A property of a system is a characteristic of the system which depends upon its state, but not upon how the state is reached. There are two sorts of property :

1. **Intensive properties.** These properties *do not depend on the mass of the system. Examples* : Temperature and pressure.

2. **Extensive properties.** These properties *depend on the mass of the system. Example* : Volume. Extensive properties are often divided by mass associated with them to obtain the intensive properties. For example, if the volume of a system of mass m is V, then the specific volume of matter within the system is $\frac{V}{m} = v$ which is an intensive property.

1.8. STATE

State *is the condition of the system at an instant of time as described or measured by its properties. Or each unique condition of a system is called a* ***state***.

It follows from the definition of state that each property has a single value at each state. Stated differently, all properties are *state* or *point functions.* Therefore, all properties are identical for identical states.

On the basis of the above discussion, we can determine if a given variable is *property* or not by applying the following *tests* :

- *A variable is a property, if and only if, it has a single value at each equilibrium state.*
- A variable is a property, if and only if, the change in its value between any two prescribed equilibrium states is single-valued.

Therefore, *any variable whose change is fixed by the end states is a property.*

1.9. PROCESS

A process occurs when the system undergoes a change in a state or an energy transfer at a steady state. A process may be *non-flow* in which a fixed mass within the defined boundary is undergoing a change of state. *Example* : A substance which is being heated in a closed cylinder undergoes a ***non-flow process*** (Fig. 1.4). *Closed systems undergo non-flow processes.* A process may be a ***flow process*** *in which mass is entering and leaving through the boundary of an open system.* In a steady flow process (Fig. 1.5) mass is crossing the boundary from surroundings at entry, and an equal mass is crossing the boundary at the exit so that the total mass of the system remains constant. In an open system it is necessary to take account of the work delivered from the surroundings to the system at entry to cause the mass to enter, and also of the work delivered from the system at surroundings to cause the mass to leave, as well as any heat or work crossing the boundary of the system.

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Quasi-static process. Quasi means '*almost*'. A quasi-static process is also called a *reversible process. This process is a succession of equilibrium states and infinite slowness is its characteristic feature.*

1.10. CYCLE

Any process or series of processes whose end states are identical is termed a **cycle**. The processes through which the system has passed can be shown on a state diagram, but a complete section of the path requires in addition a statement of the heat and work crossing the boundary of the system. Fig. 1.6 shows such a cycle in which a system commencing at condition '1' changes in pressure and volume through a path 123 and returns to its initial condition '1'.

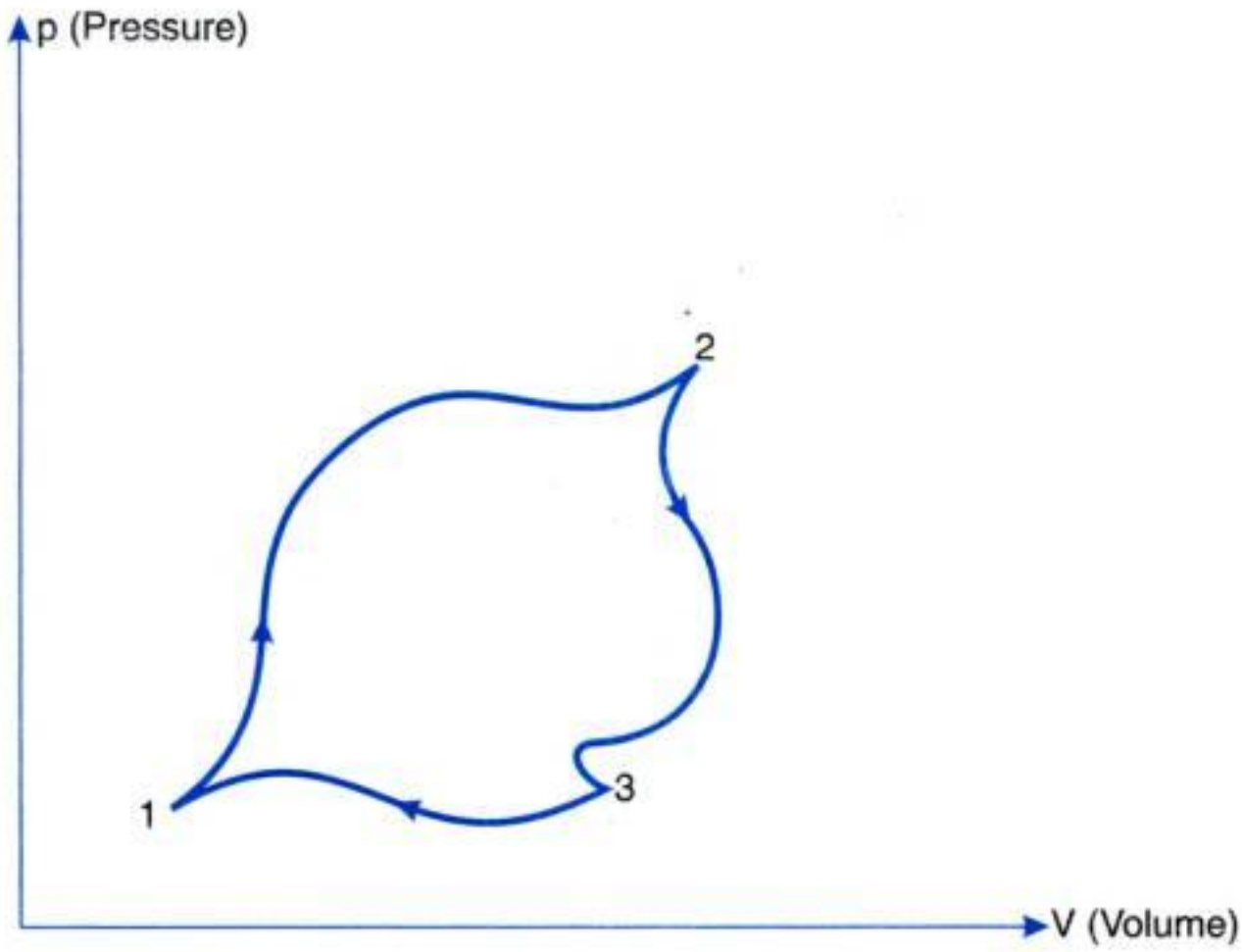

Fig. 1.6. Cycle of operations.

1.11. POINT FUNCTION

When two properties locate a point on the graph (co-ordinate axes) then those properties are called as **point function.**

Examples. Pressure, temperature, volume etc.

$$\int_1^2 dV = V_2 - V_1 \text{ (an } \textit{exact differential}\text{).}$$

1.12. PATH FUNCTION

There are certain quantities which cannot be located on a graph by a *point* but are given by the *area* or so, on that graph. In that case, the area on the graph, pertaining to the particular process, *is a function of the path of the process.* Such quantities are called **path functions.**

Examples. Heat, work etc.

Heat and work are *inexact differentials.* Their change cannot be written as difference between their end states.

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Thus $\int_1^2 \delta Q \neq Q_2 - Q_1$ and is shown as ${}_1Q_2$ or $Q_{1\text{-}2}$

Similarly $\int_1^2 \delta W \neq W_2 - W_1$, and is shown as ${}_1W_2$ or $W_{1\text{-}2}$

Note. The operator δ is used to denote inexact differentials and operator d is used to denote exact differentials.

1.13. TEMPERATURE

- *The temperature is a thermal state of a body which distinguishes a hot body from a cold body.* The temperature of a body is *proportional to the stored molecular energy i.e.,* the average molecular kinetic energy of the molecules in a system. (A particular molecule does not have a temperature, it has energy. The gas as a system has temperature).
- Instruments for measuring *ordinary temperatures* are known as ***thermometers*** and those for measuring *high temperatures* are known as ***pyrometers.***
- It has been found that a gas will not occupy any volume at a certain temperature. This temperature is known as *absolute zero temperature*. The temperatures measured with absolute zero as basis are called *absolute temperatures*. Absolute temperature is stated in degrees centigrade. The point of absolute temperature is found to occur at 273.15°C below the freezing point of water.

Then, Absolute temperature = Thermometer reading in °C + 273.15.

Absolute temperature can also be represented in degrees kelvin, denoted by K (SI unit).

1.14. ZEROTH LAW OF THERMODYNAMICS

- **'Zeroth law of thermodynamics'** *states that if two systems are each equal in temperature to a third, they are equal in temperature to each other.*

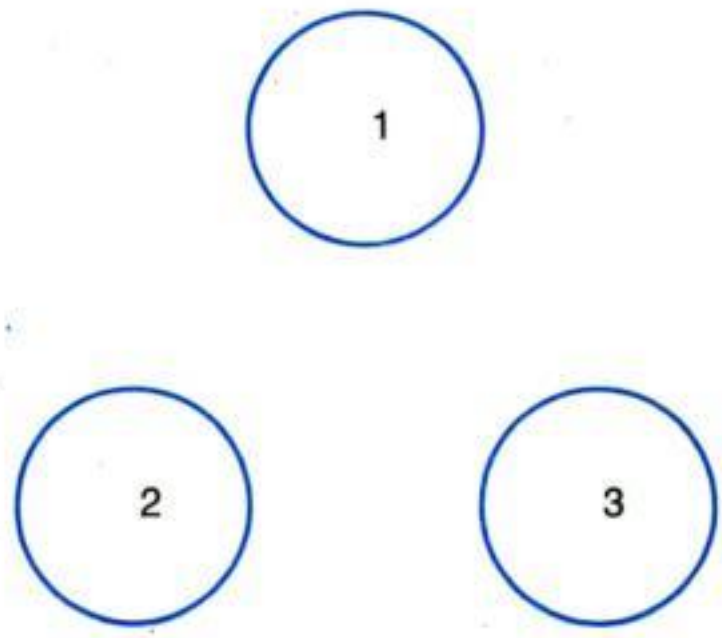

Fig. 1.7. Zeroth law of thermodynamics.

Example. Refer Fig. 1.7. System '1' may consist of a mass of gas enclosed in a rigid vessel fitted with a pressure gauge. If there is no change of pressure when this system is brought into contact with system '2' a block of iron, then the two systems are equal in temperature (assuming that the systems 1 and 2 do not react each other chemically or electrically). Experiment reveals that if system '1' is brought into contact with a third system '3' again with no change of properties then systems '2' and '3' will show no change in their properties when brought into contact provided they do not react with each other chemically or electrically. Therefore, '2' and '3' must be in equilibrium.

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This law was enunciated by R.H. Fowler in the year 1931. However, since the first and second laws already existed at that time, it was designated as ***zeroth law*** so that it *precedes* the first and second laws *to form a logical sequence*.

1.15. THE THERMOMETER AND THERMOMETRIC PROPERTY

1.15.1. Introduction

- The zeroth law of thermodynamics provides the basis for the measurement of temperature. It enables us to compare temperatures of two bodies '1' and '2' with the help of a third body '3' and say that the temperature of '1' is the same as the temperature of '2' without actually bringing '1' and '2' in thermal contact. In practice, body '3' in the zeroth law is called the **thermometer**. It is brought into thermal equilibrium with a set of standard temperature of a body '2', and is thus calibrated. Later, when any other body '1' is brought in thermal communication with the thermometer, we say that the body '1' has attained equality of temperature with the thermometer, and hence with body '2'. This way, the body '1' has the temperature of body '2' given for example by, say the height of mercury column in the thermometer '3'.
- The height of mercury column in a thermometer, therefore, becomes a **thermometric property**.

There are other methods of temperature measurement which utilize various other properties of materials, that are functions of temperature, as thermometric properties.

Six different kinds of thermometers, and the names of the corresponding thermometric properties employed are given below :

Thermometer	*Thermometric property*
1. Constant volumes gas	Pressure (p)
2. Constant pressure gas	Volume (V)
3. Alcohol or mercury-in-glass	Length (L)
4. Electric resistance	Resistance (R)
5. Thermocouple	Electromotive force (E)
6. Radiation (pyrometer)	Intensity of radiation (I or J)

1.15.2. Measurement of Temperature

Temperature can be depicted as a *thermal state which depends upon the internal or molecular energy of the body*.

1.15.2.1. Temperature Measuring Instruments

These instruments may be classified in two broad categories :

1. **Non-electrical methods :**

(i) By using change in volume of a liquid when its temperature is changed.

(ii) By using change in pressure of a gas when its temperature is changed.

(iii) By using changes in the vapour pressure when the temperature is changed.

2. **Electrical method :**

(i) By thermocouples.

(ii) By change in resistance of material with change in temperature.

(iii) By comparing the colours of filament and the object whose temperature is to be found out.

(iv) By ascertaining the energy received by radiation.

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The **thermometers** may also be classified as follows :

1. ***Expansion thermometers***

(*i*) Liquid-in-glass thermometers (*ii*) Bimetallic thermometers.

2. ***Pressure thermometers***

(*i*) Vapour pressure thermometers (*ii*) Liquid-filled thermometers

(*iii*) Gas-filled thermometers.

3. Thermocouple thermometers
4. Resistance thermometers
5. Radiation pyrometers
6. Optical pyrometers.

1. Expansion Thermometers :

The expansion thermometers make use of the differential expansion of two different substances. Thus in liquid-in-glass thermometers, it is the difference in expansion of liquid and the containing glass. And in bimetallic thermometers, the indication is due to the difference in expansion of the two solids. These thermometers are discussed below :

(*i*) **Liquid-in-glass thermometer.** This is a very familiar type of thermometer. The mercury or other liquid fills the glass bulb and extends into the bore of the glass stem. Mercury is the most suitable liquid and is used from – 38.9°C (melting point) to about 600°C. The thermometers employed in the laboratory have the scale engraved directly on the glass stem. A usual type of mercury-in-glass thermometer is shown in Fig. 1.8. An expansion bulb is usually provided at the top of the stem to allow room for expansion of mercury, in case the thermometer is subjected to temperature above its range. The upper limit for mercury-in-glass thermometers is about 600°C. As the upper limit is far above the boiling point of mercury, some inert gas *i.e.*, nitrogen is introduced above the mercury to prevent boiling.

Fig. 1.8. Mercury-in-glass thermometer.

Pentane, ethyl alcohol and toluene are the other liquids which can be used for liquid-in-glass thermometers. Since these liquids are normally colourless a dye is added to facilitate reading. These liquids have a low freezing point as shown below and are suitable for low temperature thermometers.

Liquid	*Boiling point*	*Freezing point*
Pentane	36°C	– 130°C
Ethyl alcohol	78°C	– 100°C
Toluene	110°C	– 92°C

(*ii*) **Bimetallic thermometers.** In a bimetallic thermometer differential expansion of bimetallic strips is used to indicate the temperature. It has the advantage over the liquid-in-glass thermometer, that it is *less fragile* and is *easier to read*. In this type of thermometer two flat strips of different metals are placed side by side and are welded together. Many different metals can be used for this purpose. Generally one is a *low expanding metal* and the other is *high expanding metal*. The bimetal strip is coiled in the form of a spiral or

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helix. *Due to rise in temperature, the curvature of the strip changes. The differential expansion of a strip causes the pointer to move on the dial of the thermometer.*

2. Pressure Thermometers :

In pressure thermometers liquids, gases and vapours can all be used. The principle on which they work is quite simple. The fluid is confined in a closed system. In this case the *pressure is a function of the temperature*, so that when the fluid is heated, the pressure will rise. And the temperature can be indicated by Bourdon type pressure gauge. In general, the thermometer consists of a bulb which contains bulk of the fluid. The bulb is placed in the region whose temperature is required. A capillary tube connects the bulb to a Bourdon tube, which is graduated with a temperature scale.

Pressure thermometers are discussed below :

(*i*) **Vapour pressure thermometer.** A schematic diagram of a vapour pressure thermometer is shown in Fig. 1.9. When the bulb containing the fluid is installed in the region whose temperature is required, some of the fluid vapourizes, and increases the vapour pressure. This change of pressure is indicated on the Bourdon tube. The relation between temperature and vapour pressure of a volatile liquid is of the *exponential form*. Therefore, the scale of a vapour pressure thermometer will not be linear.

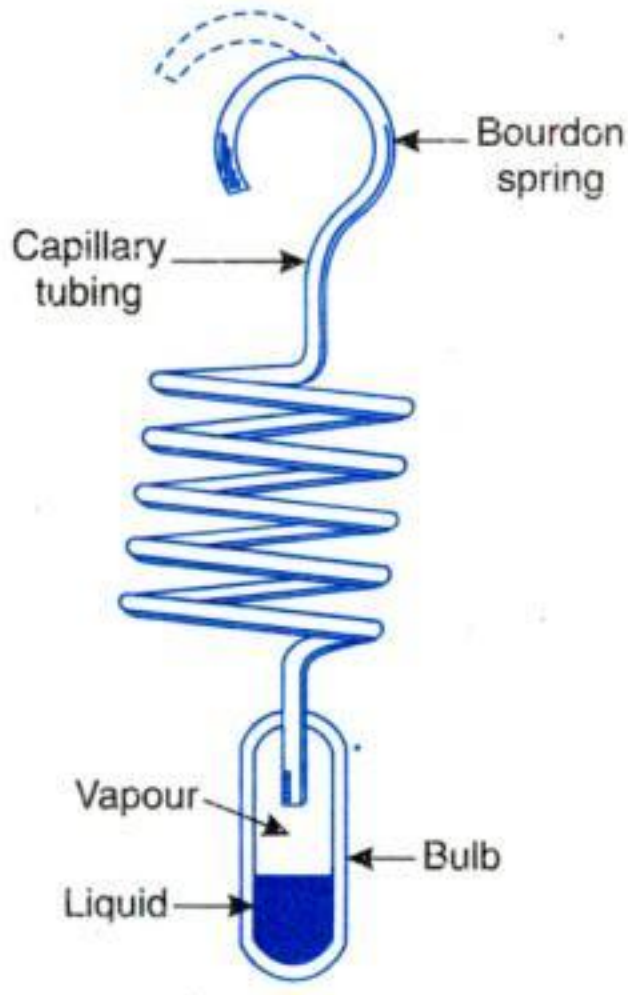

Fig. 1.9. Vapour pressure thermometer.

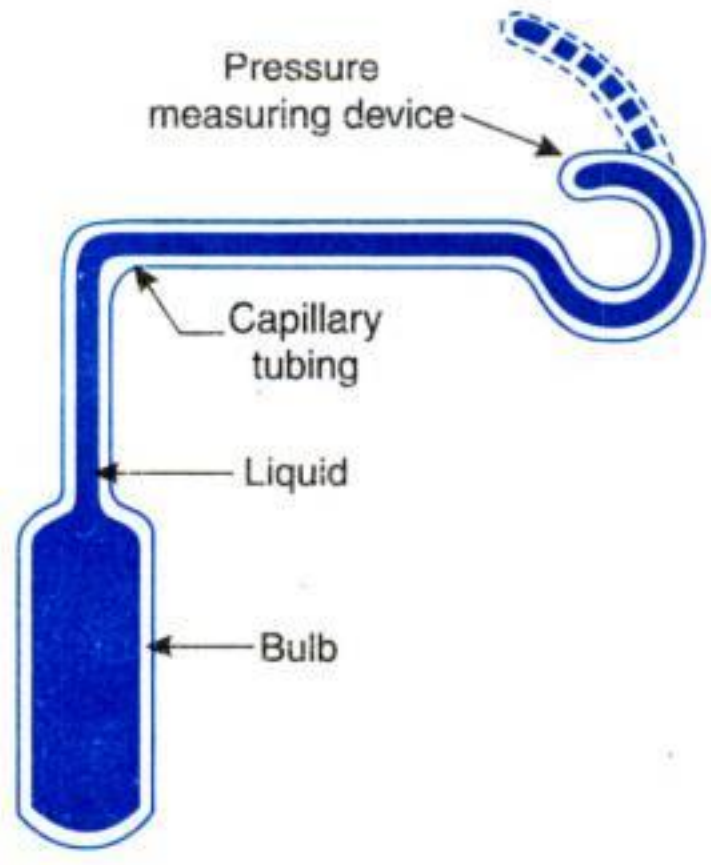

Fig. 1.10. Liquid-filled thermometer.

(*ii*) **Liquid-filled thermometer.** A liquid-filled thermometer is shown in Fig. 1.10. In this case, the *expansion of the liquid causes the pointer to move in the dial*. Therefore liquids having high co-efficient of expansion should be used. In practice many liquids *e.g.,* mercury, alcohol, toluene and glycerine have been successfully used. The operating pressure varies from about 3 to 100 bar. These type of thermometers could be used for a temperature upto 650°C in which mercury could be used as the liquid.

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In actual design, the internal diameter of the capillary tube and Bourdon tube is, made much smaller than that of the bulb. This is because the capillary tube is subjected to a temperature which is quite different from that of the bulb. Therefore, to minimise the effect of variation in temperature to which the capillary tube is subjected, the *volume of the bulb is made as large as possible as compared with the volume of the capillary*. However, large volume of bulb tends to increase time lag, therefore, a *compensating device is usually built into the recording or indicating mechanism, which compensates the variations in temperature of the capillary and Bourdon tubes.*

(*iii*) **Gas-filled thermometers.** The temperature range for gas thermometer is practically the same as that of liquid filled thermometer. The gases used in the gas thermometers are *nitrogen* and *helium*. Both these gases are chemically inert, have good values for their co-efficient of expansion and have low specific heats. The construction of this type of thermometer is more or less the same as mercury-thermometer in which Bourdon spring is used. The errors are also compensated likewise. The only difference in this case is that bulb is made much larger than used in liquid-filled thermometers. For good performance the volume of the bulb should be made at least 8 times than that of the rest of the system.

These thermometers are generally used for pressures below 35 bar.

3. Thermocouple Thermometers :

For higher range of temperature *i.e.*, above 650°C, filled thermometers are unsuitable. For higher range of temperature, thermocouples and pyrometers are used.

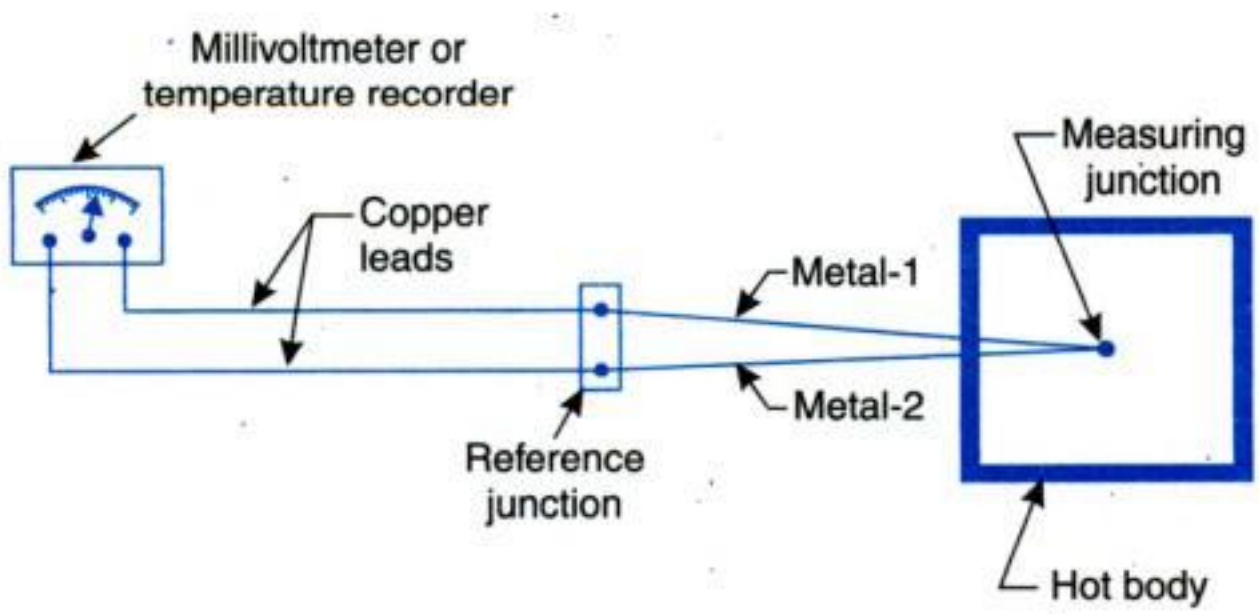

Fig. 1.11. Thermocouple.

In its simplest form a thermocouple consists of *two dissimilar metals or alloys which develop e.m.f. when the reference and measuring junctions are at different temperatures*. The reference junction or cold junction is usually maintained at some constant temperature, such as 0°C. Fig. 1.11, shows a simple circuit of a thermocouple and the temperature measuring device. In many industrial installations the instruments are equipped with automatic compensating devices for temperature changes of the reference junction, thus eliminating the necessity of maintaining this junction at constant temperature.

Table 1.1 gives the composition, useful temperatures range and temperature versus e.m.f. relationship for some commercial thermocouples.

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Table 1.1. Composition, Useful Temperature Range and e.m.f. Produced for Some Thermocouples

S.No.	*Thermocouple*	*Composition*	*Temperature (°C)*		*Thermoelectric power*		*Remarks*
			Useful range	*Max.*	*°C*	*Millivolt*	
1.	Platinum vs Platinum-rhodium	Pure platinum vs Pt + 10 or 13% Rh	400 to 1450	1700	0 500 1000 1500	0.0 4.219 9.569 15.498	Used for high temperature measurements
2.	Chromel vs alumel	90% Ni + 10% Cr vs 95% Ni + 5% (Al + Sn) Mn	– 200 to 1200	1450	– 200 0 300 600 900 1200	– 5.75 0.0 12.21 24.90 37.36 48.85	High resistance to oxidation
3.	Iron vs constantan	Pure iron vs 45-60% Cu + 55-40% Ni	– 200 to 750	1000	– 200 0 300 600 900	– 8.27 0.0 16.59 33.27 52.29	—
4.	Copper vs constantan	Pure copper vs Cu-Ni constantan	– 200 to 350	600	– 200 0 200 400	– 5.539 0.0 9.285 20.865	Not suitable in air due to excessive oxidation

4. Resistance Thermometers :

The fact that the electrical resistance of the metals increases with temperature is made use of in resistance thermometers which are purely electrical in nature. *A resistance thermometer is used for precision measurements below 150°C.*

A simple resistance thermometer consists of a *resistance element* or *bulb, electrical loads* and *a resistance measuring or recording instrument*. The resistance element (temperature sensitive element) is usually supplied by the manufacturers with its protecting tube and is ready for electrical connections. The resistance of the metal used as resistance element should be reproducible at any given temperature. *The resistance is reproducible if the composition or physical properties of the metal do not change with temperature. For this purpose platinum is preferred.* A *platinum resistance thermometer* can measure temperatures to within ± 0.01°C. However, because of high cost of platinum, nickel and copper are used as resistance elements for industrial purposes for low temperatures. The fine resistance wire is wound in a spiral form on a mica frame. The delicate coil is then enclosed in a porcelain or quartz tube. The change of resistance of this unit can be measured by instruments such as Wheatstone bridge, potentiometer or galv nometer.

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Advantages :

The resistance thermometers possess the following *advantages* over other devices :

1. A resistance thermometer is very accurate for low ranges below 150°C.
2. It requires no reference junction like thermocouples and as such is more effective at room temperature.
3. The distance between the resistance element and the recording element can be made much larger than is possible with pressure thermometers.
4. It resists corrosion and is physically stable.

Disadvantages :

1. The resistance thermometers cost more.
2. They suffer from time lag.

5. Radiation Pyrometers :

A device which measures the total intensity of radiation emitted from a body is called ***radiation pyrometer.***

The elements of a total radiation pyrometer are illustrated in Fig. 1.12. It collects the radiation from an object (hot body) whose temperature is required. A mirror is used to focus this radiation on a thermocouple. This energy which is concentrated on the thermocouple raises its temperature, and in turn generates an e.m.f. This e.m.f. is then measured either by the galvanometer or potentiometer method. Thus *rise of temperature is a function of the amount of radiation emitted from the object.*

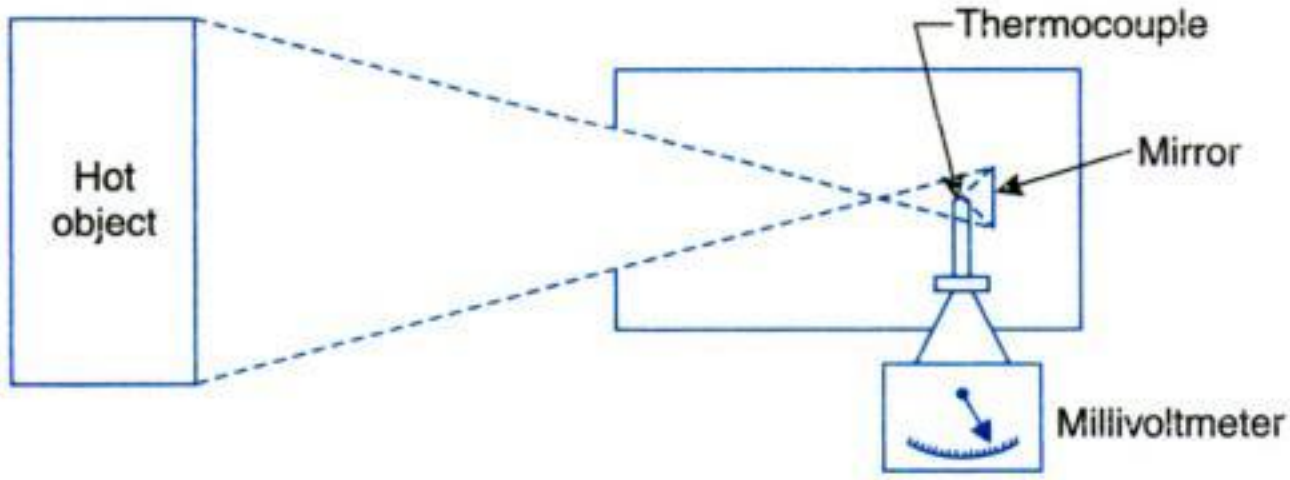

Fig. 1.12. A schematic diagram of radiation pyrometer.

Advantages of the pyrometers

1. The temperatures of moving objects can be measured.
2. A higher temperature measurement is possible than that possible by thermocouples etc.
3. The average temperatures of the extended surface can be measured.
4. The temperature of the objects which are not easily accessible can be measured.

6. Optical Pyrometers :

An optical pyrometer works on the principle *that matters glow above 480°C and the colour of visible radiation is proportional to the temperature of the glowing matter*. The amount of light radiated from the glowing matter (solid or liquid) is measured and employed to determine the temperature.

Fig. 1.13 shows a *disappearing filament pyrometer*.

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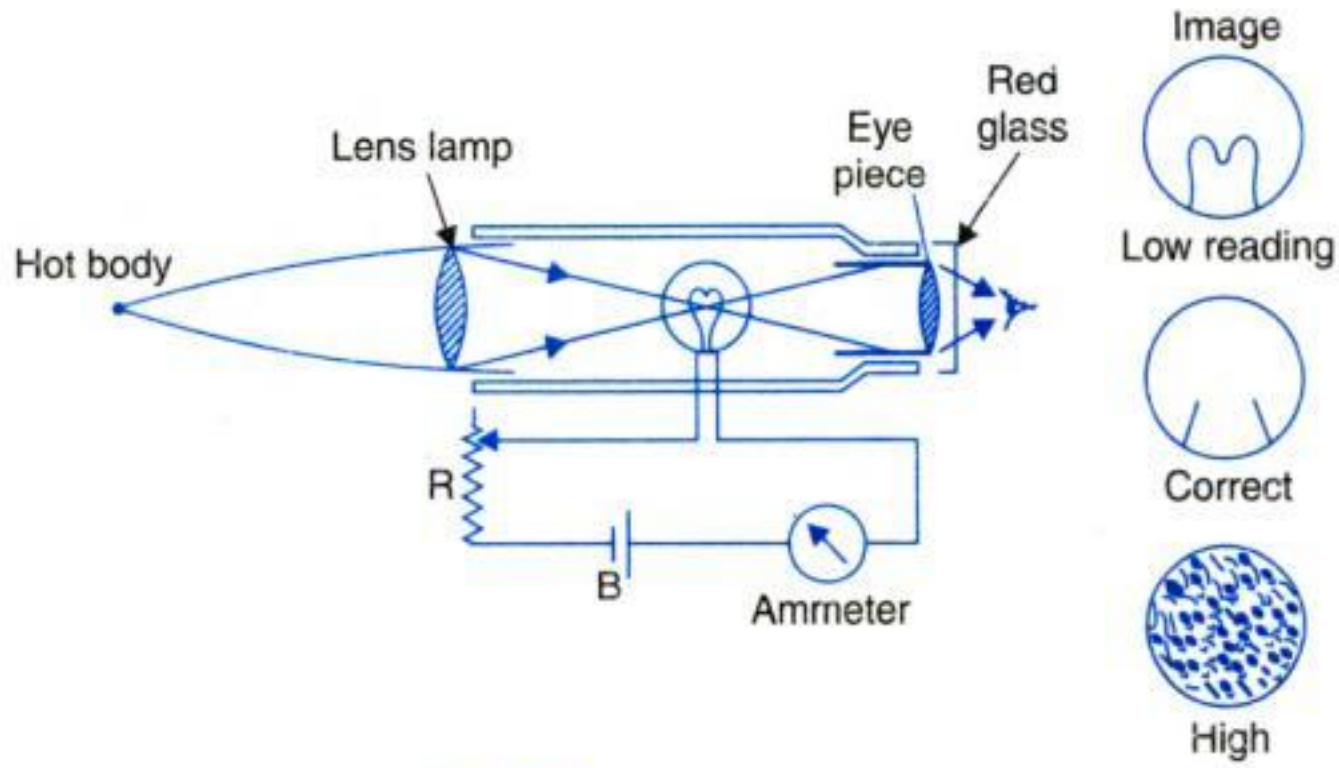

Fig. 1.13. Optical pyrometer.

Operation :

- The optical pyrometer is sighted at the hot body and focused.
- In the beginning filament will appear dark as compared to the background which is bright (being hot).
- By varying the resistance (R) in the filament circuit more and more current is fed into it, till filament becomes equally bright as the background and hence disappears.
- The current flowing in the filament at this stage is measured with the help of an ammeter which is calibrated directly in terms of temperature.
- If the filament current is further increased, the filament appears brighter as compared to the background which then looks dark.
- An optical pyrometer can measure temperatures ranging from 700 to 4000°C.

Table 1.2 gives the summary of temperature range of different instruments on next page.

1.15.3. The International Practical Temperature Scale

For the calibration of thermometric instruments the Seventh General Conference on Weight and Measures held in 1927 formulated a convenient scale known as the *International Practical Temperature Scale*. It was revised at Thirteenth General Conference in 1968. It consists of *reproducible reference temperatures or primary fixed points* defined by a number of pure substances with assigned values of temperatures determined with precision on ideal or perfect gas temperature scale as given in Table 1.3.

It is stated here that :

— The *triple point* represents an equilibrium state between solid, liquid and vapour phases of a substance.

— *Normal boiling point* is the temperature at which the substance boils at standard atmospheric pressure of 760 mm Hg.

— *Normal freezing point* is the solidification or the melting point temperature of the substance at standard atmospheric pressure.

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Table 1.2. Summary of Temperature Range of Different Instruments

Types of instruments	*Liquid in glass thermometer*	*Bimetallic thermometer*	*Pressure thermometers*			*Thermocouples*				*Resistance thermometers*			*Radiation pyrometers*	*Optical pyrometers*
	Mercury		Vapour pressure	Liquid filled	Gas filled	Iron-Constantan	Copper-Constantan	Chromel-Alumel	Platinum Platinum-Rhodium	Copper	Nickel	Platinum	–	–
Low temperature limit	– 39°C	– 75°C	– 250°C	– 87°C	– 269°C	– 196°C	– 250°C	–184°C	0°C	– 140°C	– 184°C	– 184°C	About room temp.	760°C
High temperature limit	600°C	540°C	316°C	650°C	540°C	760°C	350°C	1260°C	1540°C	120°C	316°C	760°C	As high as desired	As high as desired
Remark	Low price ; Fairly accurate ; Easily broken	Easier to read than Liquid-in-glass thermometers	Normally scale non-linear	To be compensated for variation in ambient temperatures	Requires large bulb ; Linear scale	Widely used in Industry	Constantan is 60% Cu 40% Ni	Chromel is 20% ; Cr 80% ; Ni-Alumel is 2% Al, 98% Ni,	Expensive	Very sensitive. Suitable for narrow range instruments			–	–

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Table 1.3. Fixed Points of the International Practical Temperature Scale of 1968

	Equilibrium state	*Assigned value of temperature*	
		T, K	*t°C*
1.	Triple point of hydrogen	13.81	– 259.34
2.	Boiling point of hydrogen at 33.306 kPa	17.042	– 266.108
3.	Normal boiling point of hydrogen	20.28	– 252.87
4.	Normal boiling point of neon	27.102	– 246.048
5.	Triple point of oxygen	54.361	– 218.789
6.	Normal boiling point of oxygen	90.188	– 182.962
7.	Triple point of water	273.16	0.01
8.	Normal boiling point of water	373.15	100.00
9.	Normal freezing point of antimony (antimony point)	630.74	357.59
10.	Normal freezing point of zinc (zinc point)	692.73	419.58
11.	Normal freezing point of silver (silver point)	1235.08	961.93
12.	Normal freezing point of gold (gold point)	1337.58	1064.43

Based on the available method of measurement, the whole temperature scale may be divided into four ranges. The equations for interpolation for each range are as follows :

1. **From – 259.34°C (triple point of hydrogen) to 0°C :**

A *platinum resistance thermometer of a standard design* is used and a polynomial of the following form is fitted between the resistance of the wire R_t and temperature t

$$R_t = R_0 (1 + At + Bt^2 + Ct^3) \qquad ...(1.6)$$

where R_0 = resistance at the ice point.

2. **From 0°C to 630.74°C (Antimony point) :**

- It is also based on *platinum resistance thermometer.*
- The diameter of the platinum wire must lie between 0.05 and 0.2 mm.

3. **From 630.74°C to 1064.43°C (Gold point) :**

- It is based on *standard platinum versus platinum-rhodium thermocouple.*
- Following equation between e.m.f. E and temperature t is employed :-

$$E = a + bt + ct^2 \qquad ...(1.7)$$

4. **Above 1064.43°C :**

- It is *based on the intensity of radiation J_T at temperature T emitted by a black body at a wavelength λ in the visible spectrum and by comparing this to the intensity of radiation J_{Au} at the same wavelength emitted by a black body at the gold point.*
- The temperature is calculated from Planck's equation for black body radiation

$$\frac{J_T}{J_{Au}} = \frac{\exp\left(\frac{C_2}{\lambda T_{Au}}\right) - 1}{\exp\left(\frac{C_2}{\lambda T}\right) - 1} \qquad ...(1.8)$$

where C_2 = 0.01438 in °C, and λ = wavelength in metres.

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Following points are worth noting for gas thermometers :

- *The gas thermometers are never used for the measurement of temperatures.* However, *they are ideal when used for calibration for establishing the ideal gas temperature scale, and for establishing a standard because of precision, reproducible results, and their reading being independent of the thermometric substance used.*
- *The gas thermometers can be used only for temperatures upto which gases do not liquify.*

Method in use before 1954 :

- Celsius and Fahrenheit scales are the two commonly used scales for the measurement of temperature. Symbols C and F are respectively used to denote the readings on these two scales. Until 1954 the temperature scales were based on two fixed points : (*i*) the *steam point* (boiling point of water at standard atmospheric pressure), and (*ii*) the *ice point* (freezing point of water).

The fixed points for these temperature scales are :

Temperature	Celsius scale	Fahrenheit scale
Steam point	100	212
Ice point	0	32
Interval	100	180

- The relation between a particular value C on celsius scale and F on Fahrenheit scale is found to be as mentioned below :

$$\frac{C}{100} = \frac{F-32}{180} \text{ or } \frac{C}{5} = \frac{F-32}{9} \qquad ...(1.9)$$

- Further the relation between a temperature difference Δt_F on Fahrenheit scale and Δt_C on celsius scale is

$$\Delta t_F = \frac{180}{100}\Delta t_C = \frac{9}{5}\Delta t_C = 1.8\,\Delta t_C \qquad ...(1.10)$$

- The use of two fixed points was found unsatisfactory and later *abandoned* because of the following *reasons* :

(*i*) It is difficult to achieve equilibrium between pure ice and air-saturated water (since when ice melts, it surrounds itself only with pure water and prevents intimate contact with air-saturated water).

(*ii*) There is extreme sensitiveness of the steam point to the change in pressure.

Method in use after 1954 :

It was suggested by Kelvin that a single fixed point only was necessary to establish a temperature. He pointed out that *triple point of water* (the state at which ice, liquid water and water vapour coexist in equilibrium) could be used as the single point. The tenth CGPM, in 1954, adopted this fixed point, and value was set at 0.01°C or 273.16 K in the Kelvin scale thus established. Correspondingly, the ice point of 0°C on the Celsius scale becomes equal to 273.15 K on the Kelvir scale. Celsius and Kelvin scales are distinguished by using distinct symbols t and T, the relatio between these two is then given by :

$$T(K) = t(°C) + 273.15 \qquad ...(1.11)$$

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1.15.4. Ideal Gas

From experimental observations it has been established that an ideal gas (to a good approximation) behaves according to the simple equation

$$pV = mRT \qquad ...(1.12)$$

where p, V and T are the pressure, volume and temperature of gas having mass m and R is a constant for the gas known as its ***gas constant.***

Eqn. (1.12) can be written as

$$pv = RT \qquad ...(1.13)$$

(where v = V/m)

In reality there is no gas which can be qualified as an ideal or perfect gas. However *all gases tend to ideal or perfect gas behaviour at all temperatures as their pressure approaches* ***zero pressure.***

For two states of the gas, Eqn. (1.12) can be written as,

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

or

$$\frac{T_2}{T_1} = \frac{p_2}{p_1} \times \frac{V_2}{V_1} \qquad ...(1.14)$$

With the help of this eqn. (1.14), the temperatures can be measured or compared.

1.16. PRESSURE

1.16.1. Definition of Pressure

Pressure is defined as a *force per unit area*. Pressures are exerted by gases, vapours and liquids. The instruments that we generally use, however, record pressure as the difference between two pressures. Thus, it is the *difference between the pressure exerted by a fluid of interest and the ambient atmospheric pressure.* Such devices indicate the pressure either above or below that of the atmosphere. When it is *above the atmospheric pressure*, it is termed *gauge pressure* and is *positive*. When it is *below* atmospheric, it is *negative* and is known as *vacuum*. Vacuum readings are given in millimetres of mercury or millimetres of water below the atmosphere.

It is necessary to establish an absolute pressure scale which is independent of the changes in atmospheric pressure. A pressure of absolute zero can exist only in complete vacuum. *Any pressure measured above the absolute zero of pressure* is termed an '*absolute pressure*'.

A schematic diagram showing the *gauge pressure, vacuum pressure* and the *absolute pressure* is given in Fig. 1.14.

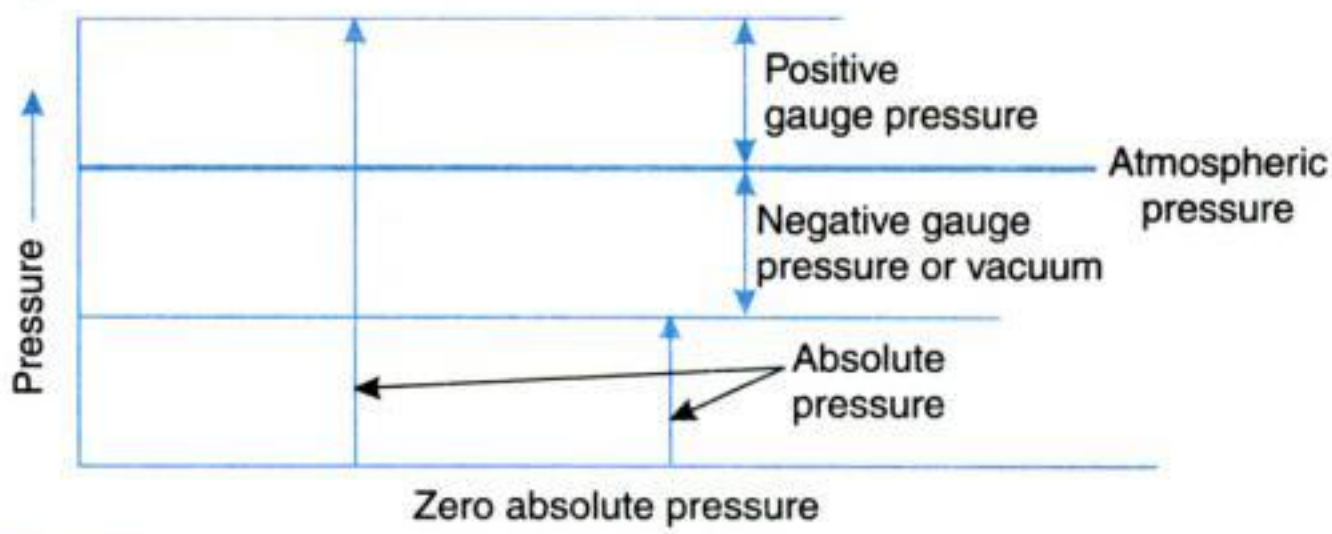

Fig. 1.14. Schematic diagram showing gauge, vacuum and absolute pressures.

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Mathematically :

(*i*) Absolute pressure = Atmospheric pressure + Gauge pressure

$$p_{abs.} = p_{atm.} + p_{gauge}.$$

(*ii*) Vacuum pressure = Atmospheric pressure – Absolute pressure.

Vacuum is defined as the *absence of pressure*. A *perfect vacuum* is obtained when *absolute pressure is zero*, at this instant *molecular momentum is zero*.

Atmospheric pressure is measured with the help of barometer.

1.16.2. Unit for Pressure

The fundamental SI unit of pressure is N/m^2 (sometimes called *pascal*, Pa) or bar. 1 bar = 10^5 N/m^2 = 10^5 Pa.

Standard atmospheric pressure = 1.01325 bar = 0.76 m (or 760 mm) Hg.

Low pressures are often expressed in terms of mm of water or mm of mercury. This is an abbreviated way of saying that the pressure is such that which will support a liquid column of stated height.

1.16.3. Types of Pressure Measurement Devices

The pressure may be measured by means of indicating gauges or recorders. These instruments may be mechanical, electro-mechanical, electrical or electronic in operation.

1. **Mechanical instruments.** These instruments may be classified into following two groups :

— The *first group* includes those instruments in which the *pressure* measurement is made by *balancing an unknown force with a known force*.

— The *second group* includes those employing *quantitative deformation of an elastic member for pressure measurement*.

2. **Electro-mechanical instruments.** These instruments usually *employ a mechanical means for detecting the pressure and electrical means for indicating or recording the detected pressure*.

3. **Electronic instruments.** Electronic pressure measuring instruments normally depend on some physical change that can be detected and indicated or recorded electronically.

1.16.4. Mechanical-type Instruments

The mechanical-type instruments are classified as follows :

1. **Manometer gauges**

(*i*) U-tube manometer (*ii*) Cistern manometer

(*iii*) Micro-manometer etc.

2. **Pressure gauges**

(*i*) Bourdon tube pressure gauge (*ii*) Diaphragm gauge

(*iii*) Vacuum gauge.

1.16.4.1. Liquid manometers

Low pressures are generally determined by *manometers* which employ liquid columns. It is difficult and costly to construct manometers to measure high pressures, as otherwise the liquid column will become unwieldy and temperature corrections will also be difficult. Their use is, therefore, *restricted to low pressures only*, and for such purposes they are *quite accurate*.

The liquids commonly employed for manometers are mercury and water. *Mercury is used for high and water for low pressures*. For this purpose a liquid is suitable if it has a low viscosity, so that it can adjust itself quickly, and also a low co-efficient of thermal expansion, so that density changes with temperature are minimum.

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1. *U-tube manometer :*

A U-tube manometer is in the form of U-tube and is made of glass. When no pressure is applied, the height of the liquid in the two legs is the same. The pressure is then applied to one leg, whilst the other is open to the atmosphere. Under this pressure the liquid will *sink* in this leg and will *rise* in the other. As the other leg is open to the air, therefore, the pressure on this side is known, and is barometric. Now the pressure applied to the first leg can be calculated. This is explained with reference to Fig. 1.15. This consists of a water manometer.

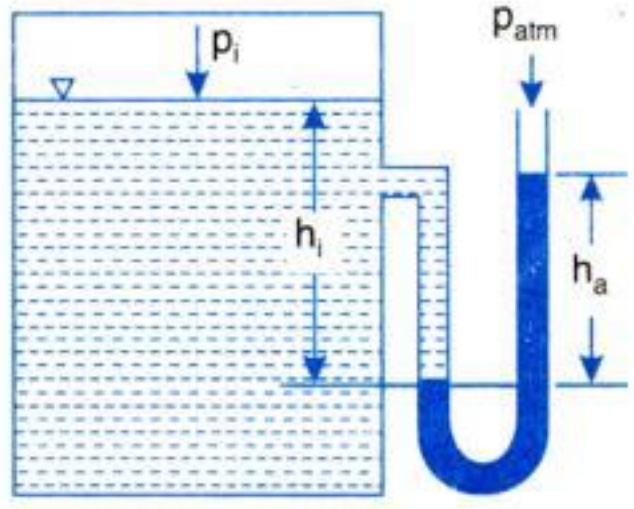

Fig. 1.15. Principle of U-tube manometer.

Considering equilibrium condition, we have

$$p_{atm} + w_a h_a = p_i + w_i h_i$$

$$\therefore \quad p_i = p_{atm} + w_a h_a - w_i h_i$$

where $p_{atm.}$ = Atmospheric pressure,

p_i = Pressure over water surface in the container,

h_a = Height of liquid in U-tube manometer,

h_i = Difference between water surface and lower surface of the liquid in manometer,

w_a = Specific weight of liquid,

w_i = Specific weight of water.

The U-tube manometer shown in Fig. 1.16 is of the simplest form. However, readings have to be taken *at two different places*. Moreover, the *deflection of the two columns may not be the same*. To avoid this difficulty cistern or well type manometer is used.

2. *Cistern manometer :*

Fig. 1.17 shows a cistern manometer. The mercury reservoir *A* is made large enough so that change of level in the reservoir is negligible. This form of manometer is generally used for measuring pressures above atmospheric. In this case, *only one reading of the level in the column is required*. However, a zero setting is necessary.

3. *Micro-manometer :*

The U-tube manometer (discussed above) is not very suitable for measuring very low pressures. Therefore, for such purposes, precision types are required. They are called *multiplying* or *micro-manometers*, because they multiply the movement of the level of the liquid. By far the most widely used type of multiplying manometer is the ***inclined manometer.*** If the tube is inclined as shown in Fig. 1.18 the sensitiveness of the U-tube manometer is increased. The inclined tube *causes a larger displacement of the liquid along the tube for a given pressure difference.*

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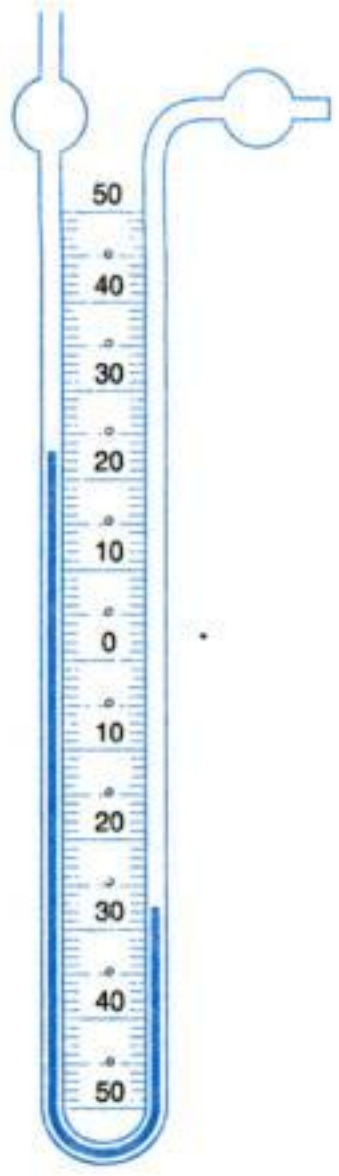

Fig. 1.16. U-tube manometer.

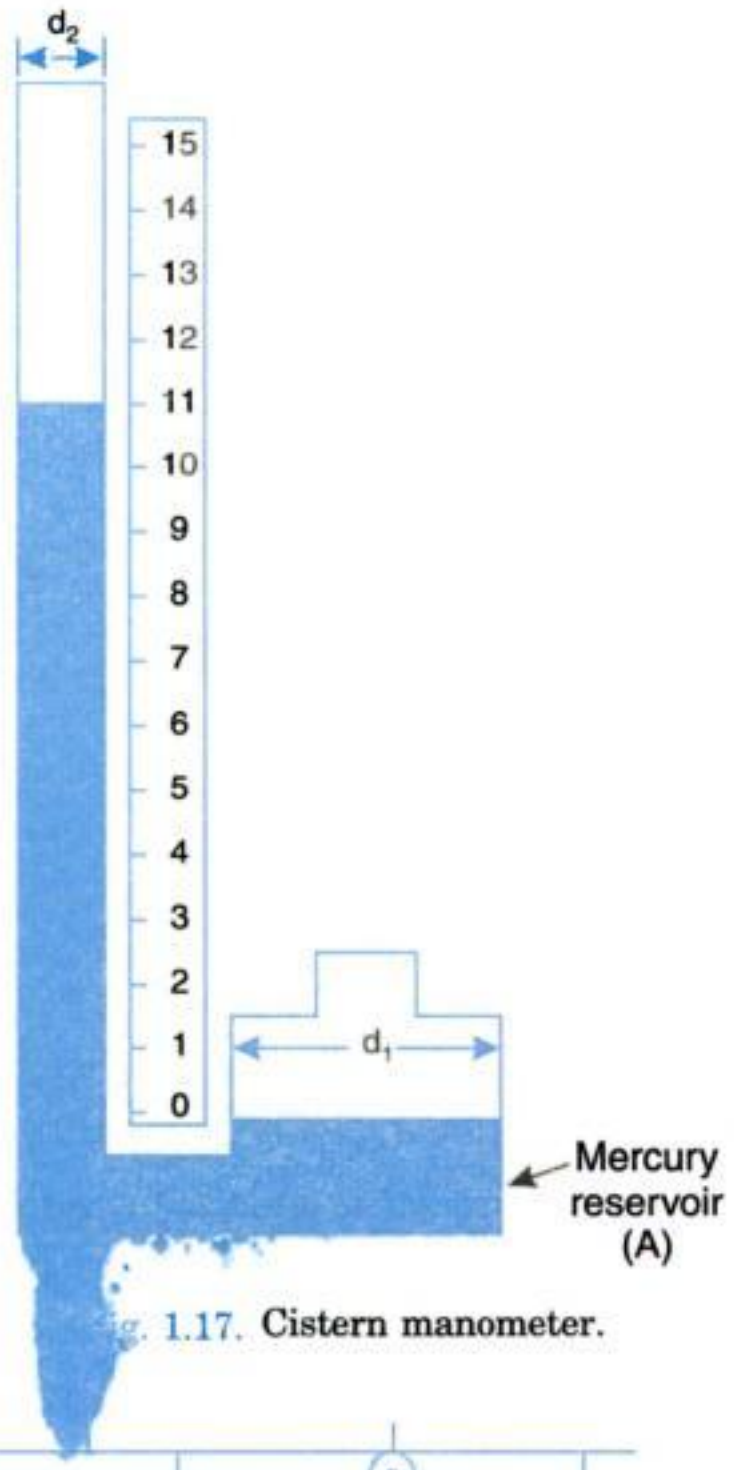

Fig. 1.17. Cistern manometer.

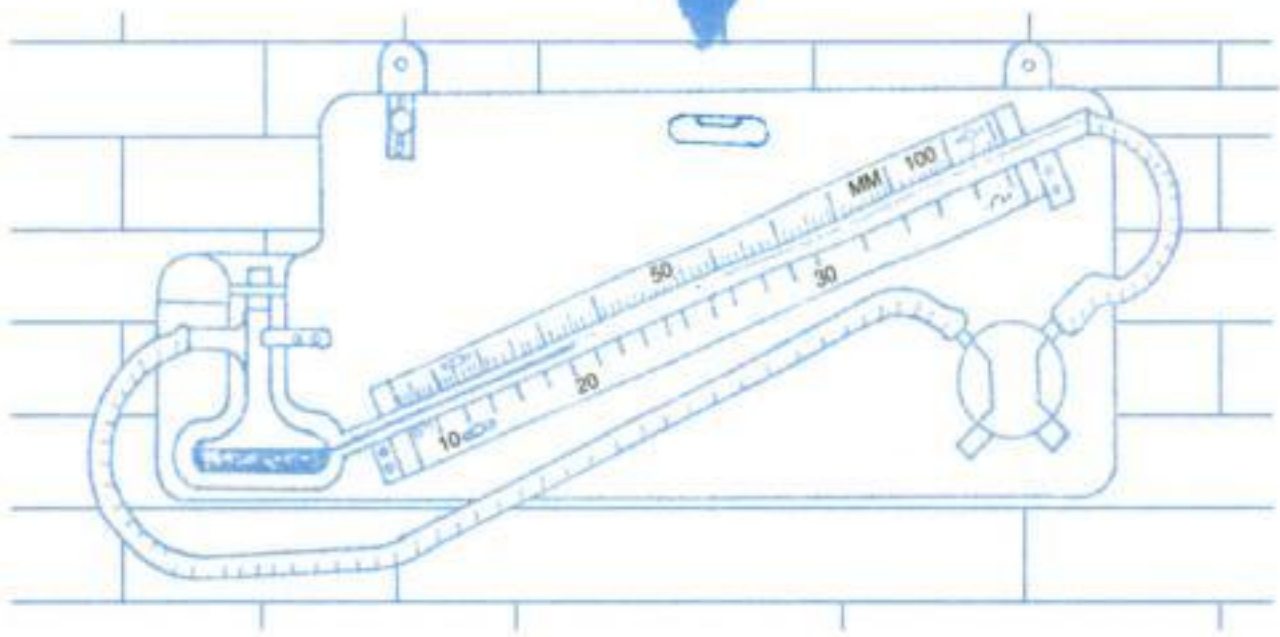

Fig. 1.18. Inclined manometer.

The principle of the inclined manometer is explained in Fig. 1.19. If pressure $p_1 = p_2$, then the level of liquid is shown by LM. However, when p_1 is slightly greater than p_2, the level is the reservoir sinks by h_2, whilst level in the tube rises by a greater distance h_1 as shown in the diagram. If h is the vertical distance between the two surfaces due to difference of pressure, then

$$h = h_1 + h_2$$

also

$$h_1 = d \sin \theta$$

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and $$h_2 \times A = d \times a$$

or $$h_2 = d \times \frac{a}{A}$$

where A = Area of cross-section of the reservoir, and
a = Area of cross-section of the inclined tube.

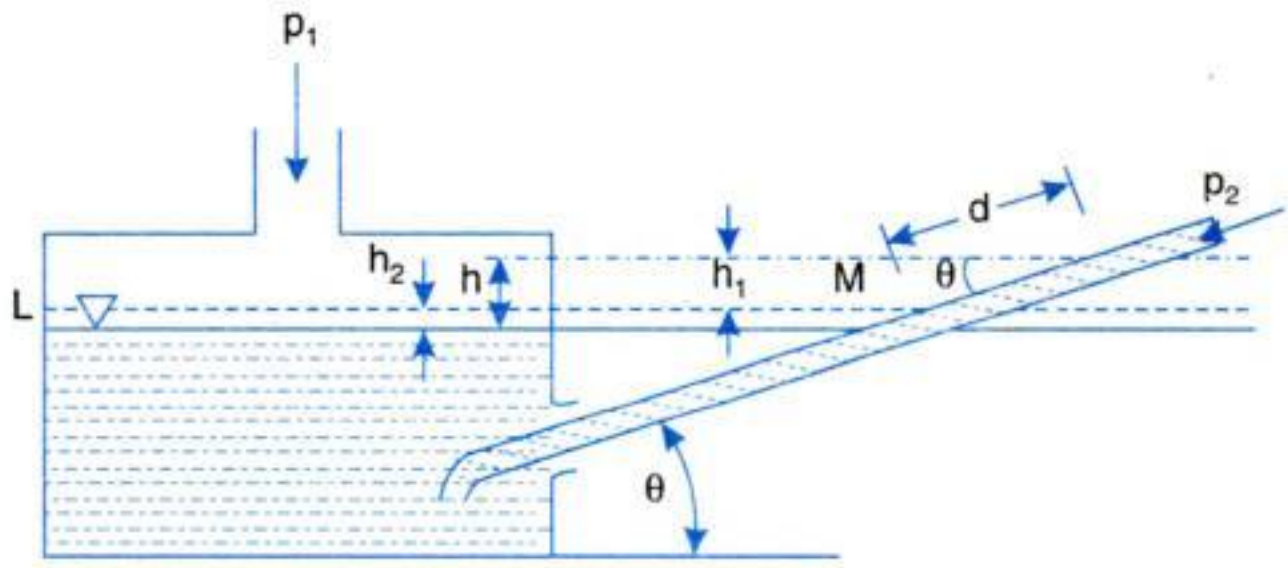

Fig. 1.19. Principle of inclined manometer.

Also the pressure difference Δp *i.e.*, $(p_1 - p_2)$

$$= hw = \left(d \times \sin\theta + d \times \frac{a}{A}\right) w$$

$$= dw\left(\sin\theta + \frac{a}{A}\right)$$

where w is the specific weight of the liquid.

The sensitiveness of the instrument can be varied by *changing the slope of the inclined tube.*

The position of the inclined tube is so arranged that $\left(\sin\theta + \frac{a}{A}\right)$ is round figure.

The multiplication factor of the gauge is $\frac{d}{h} = \frac{1}{\sin\theta + \frac{a}{A}}$

Thus the multiplication factor depends on θ and $\frac{a}{A}$. The *smaller the values of* θ *and* $\frac{a}{A}$, *the greater the multiplication factor.*

Advantages and disadvantages of manometers :

Advantages. The manometer claims the following *advantages :*

1. Relatively inexpensive.
2. Easy to fabricate.
3. Requires very little maintenance.
4. Good accuracy and sensitivity.
5. Their sensitivity can be changed by changing manometric fluids.
6. Particularly suitable to low pressures and low differential pressures.

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Disadvantages. The *disadvantages* of manometers are as follows :

1. Unsuitable for recording.
2. Generally large, bulky and fragile.
3. Their calibration is affected by changes in gravitational force and density of fluids and their calibration changes with altitude and temperature.
4. Surface tension of manometric fluid creates a capillary effect and possible hysteresis.
5. A particular manometer can be used only for measurement of a particular fluid/fluids.
6. Meniscus height has to be determined by accurate means to ensure improved accuracy.

Elastic pressure elements

Elastic pressure elements or mechanical type of transducers are used for measurement of very high pressures upto about 700 MN/m^2. There are *three* main types of pressure elements.

(*a*) Bourdon tube

(*b*) Diaphragms

(*c*) Bellows gauges

The action of these mechanical transducers depends upon the *displacement caused by the pressure*. The displacement produced may actuate a pointer whose deflection may be direct measure of the pressure applied or the displacement is measured with the help of a secondary transducer which is electrical in nature. The output of the secondary transducer which is electrical in format is a measure of the pressure applied.

(*a*) **Bourdon tubes/elements :**

Fig. 1.20 shows Bourdon tube configurations.

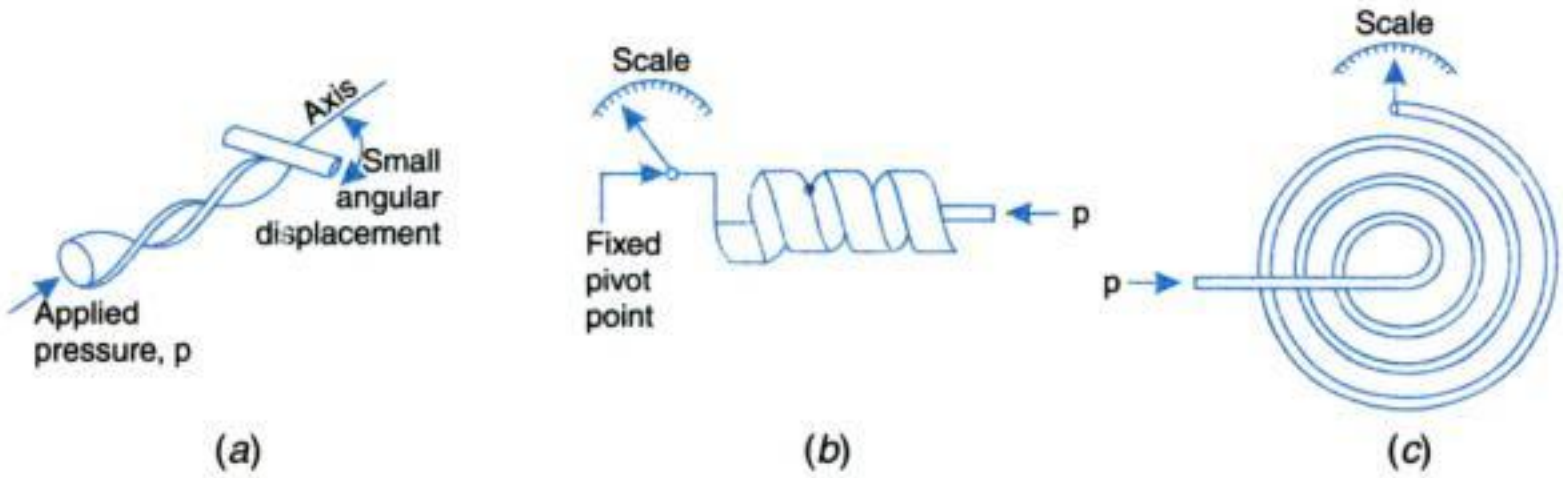

Fig. 1.20. (*a*) Twisted tube, (*b*) Helical, (*c*) Spiral.

Advantages. The Bourdon tube element has the following *advantages* :

1. Simple in construction and cheap.
2. Available in several different ranges.
3. Capability to measure gauge, absolute and differential pressures.
4. The sensitivities of Bourdon tube may be changed by changes in their dimensions.
5. Excellent sensitivity.
6. Simple and straight forward calibration with dead weight tester.
7. Easily adapted to strain, capacitance, magnetic and other electrical transducers.

Disadvantages :

1. Susceptibility to shock and vibration.
2. Inherent hysteresis and slow response to pressure changes.
3. Unsuitable for low pressure applications.

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(*b*) **Diaphragm elements :**

Fig. 1.21 shows basic diaphragm types.

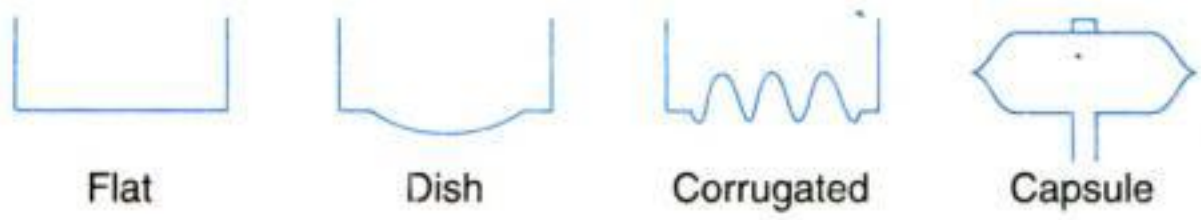

Fig. 1.21. Basic diaphragm types.

Advantages :

1. Small in size and moderately priced.
2. Wide linear range.
3. Can withstand high over pressures and under pressures.
4. Small hysteresis.
5. Can be used for measurement of absolute and differential pressures as also vacuum.

Disadvantages :

1. Need protection against shocks and vibrations.
2. Cannot be used to measure high pressures.
3. Difficult to repair.

(*c*) **Bellow gauges/elements :**

Fig. 1.22 shows some bellow gauges.

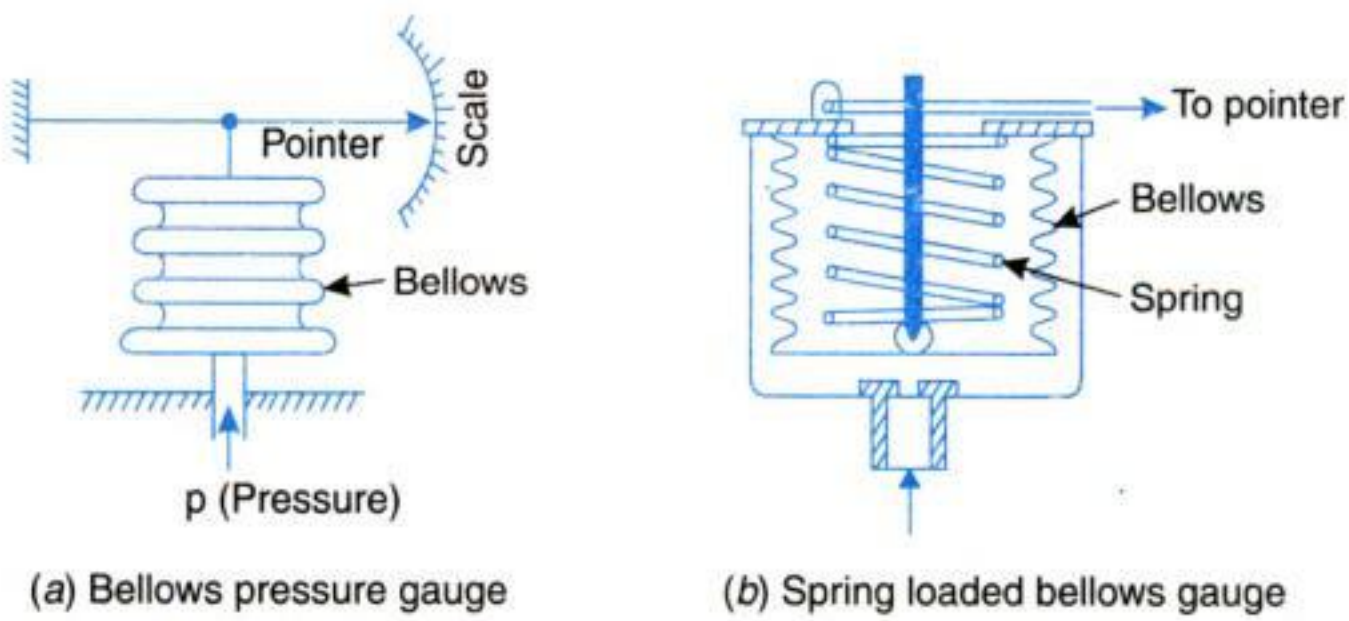

Fig. 1.22. Bellow gauges.

Advantages :

1. Simple and rugged construction.
2. Useful for measurement of low and medium pressures.
3. Moderate cost.
4. Can be used for measurement of absolute, gauge and differential pressures.

Disadvantages :

1. Need spring for accurate characterisation.
2. Greater hysteresis and zero drift problems.
3. Unsuitable for transient measurements due to longer relative motion and mass.
4. Requires compensation for ambient temperature changes.

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1.16.4.2. Important types of pressure gauges

The manometers and U-tubes (discussed earlier) are suitable for comparatively *low pressures.* For high pressures they become unnecessarily larger even when they are filled with heavy liquids. Therefore for measuring medium and high pressures, we make use of elastic pressure gauges. They employ different forms of elastic systems such as tubes, diaphragms or bellows etc. to measure the pressure. The elastic deformation of these elements is used to show the effect of pressure. Since these elements are deformed within the elastic limit only, therefore these gauges are sometimes called *elastic gauges.* Sometimes they are also called *secondary instruments,* which implies that they must be calibrated by comparison with primary instruments such as manometer etc.

Some of the important types of these gauges are enumerated and discussed below :

1. Bourdon tube type pressure gauge
2. Diaphragm gauge
3. Vacuum gauge.

1. *Bourdon tube type pressure gauge :*

A Bourdon type tube pressure gauge is used for measuring *high as well as low pressures.* A simple form of this gauge is shown in Fig. 1.23. In this case the pressure element consists of a metal tube of approximately *elliptical cross-section.* This tube is bent in the form of a segment of a circle and responds to pressure changes. When one end of the tube which is attached to the gauge case, is connected to the source of pressure, the internal pressure causes the tube to expand, whereby circumferential stress *i.e.,* hoop tension is set up. The free end of the tube moves and is in turn connected by suitable levers to a rack, which engages with a small pinion mounted on the same spindle as the pointer. Thus the pressure applied to the tube causes the rack and pinion to move. The pressure is indicated by the pointer over a dial which can be graduated in a suitable scale.

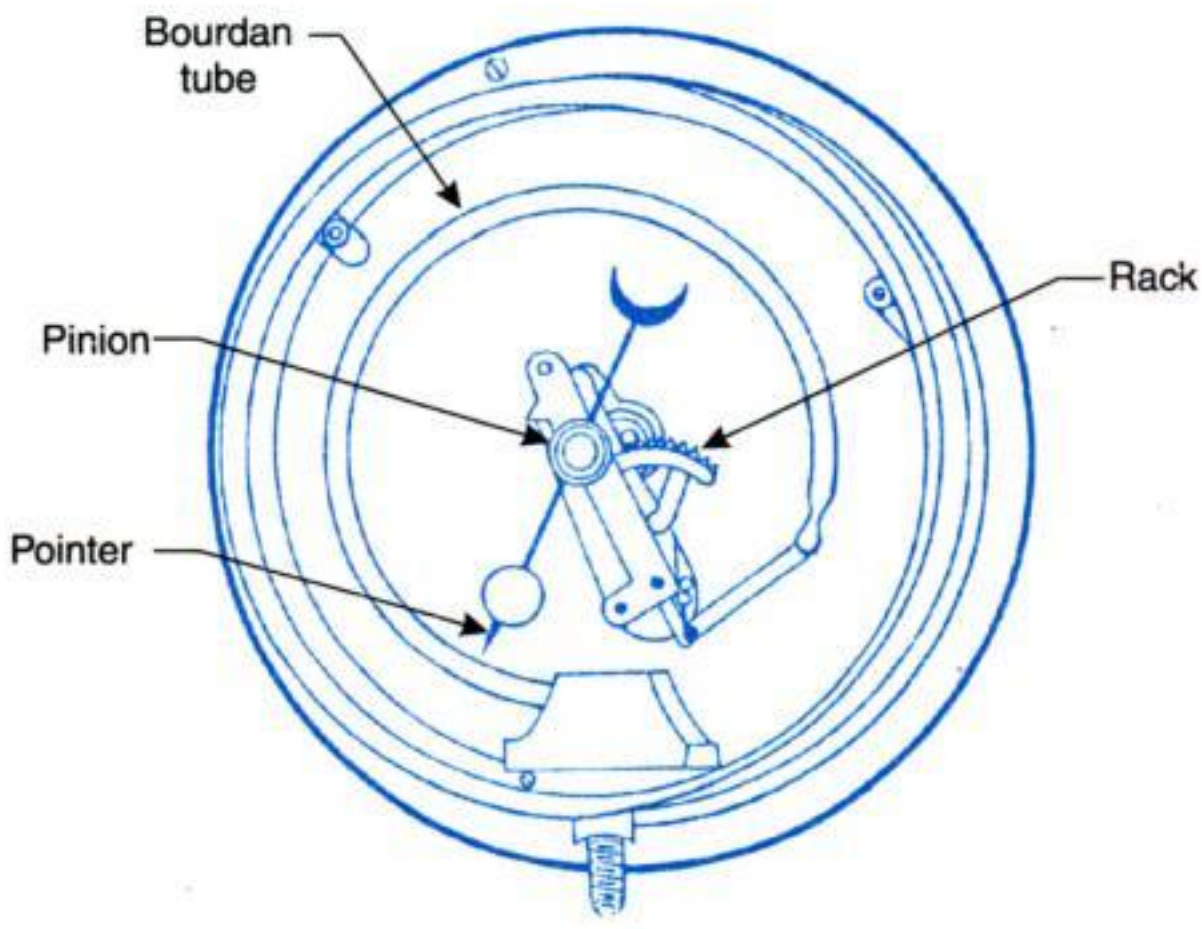

Fig. 1.23. Burdon tube pressure gauge.

The Bourdon tubes are generally made of *bronze* or *nickel steel*. The former is generally used for low pressures and the latter for high pressures.

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Depending upon the purpose for which they are required Bourdon tube gauges are made in different forms, some of them are :

(*i*) ***Compound Bourdon tube*** ..... used for measuring pressures both above and below atmospheric.

(*ii*) ***Double Bourdon tube*** ..... used where vibrations are encountered.

2. *Diaphragm gauge :*

This type of gauge employs a metallic disc or diaphragm instead of a bent tube. This disc or diaphragm is used for *actuating* the indicating device.

Refer Fig. 1.24. When pressure is applied on the lower side of the diaphragm, it is deflected upward. This movement of the diaphragm is transmitted to a rack and pinion. The latter is attached to the spindle of needle moving on a graduated dial. The dial can again be graduated in a suitable scale.

3. *Vacuum gauge :*

Bourdon gauges discussed earlier can be used to measure vacuum instead of pressure. Slight changes in the design are required for this purpose. Thus, in this case, the tube be *bent inward instead of outward* as in pressure gauge. *Vacuum gauges* are graduated in millimetres of mercury below atmospheric pressure. In such cases, therefore, absolute pressure in millimetres of mercury is the difference between barometer reading and vacuum gauge reading.

Vacuum gauges are used to measure the vacuum in the *condensers* etc. If there is leakage, the vacuum will drop.

The pressure gauge installation require the following *considerations* :

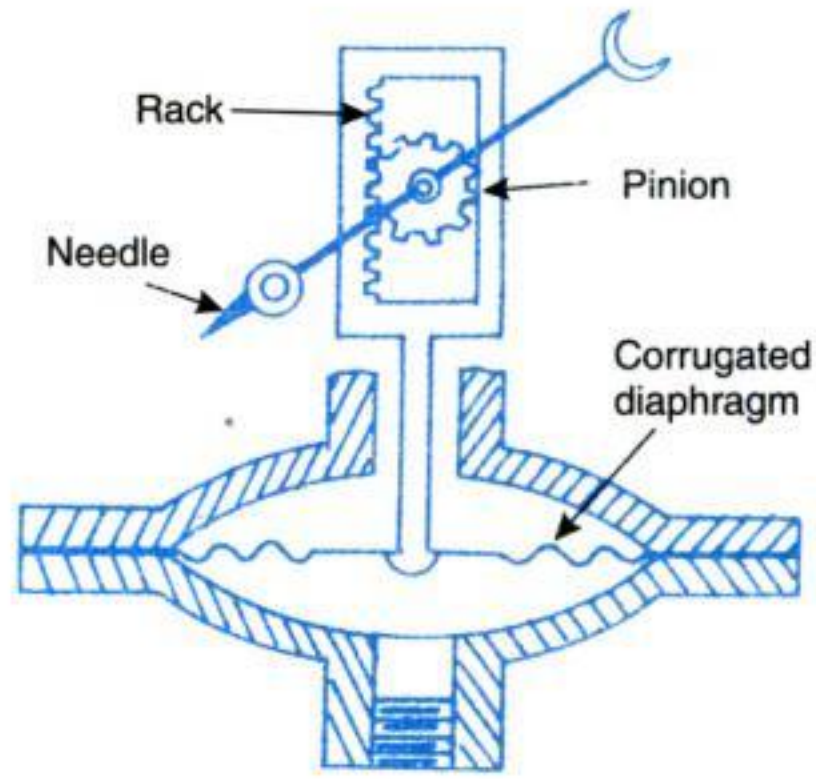

Fig. 1.24. Principle of diaphragm gauge.

(*i*) Flexible copper tubing and compression fittings are recommended for most installations.

(*ii*) The installation of a gauge cock and tee in the line close to the gauge is recommended because it permits the gauge to be removed for testing or replacement without having to shut down the system.

(*iii*) Pulsating pressures in the gauge line are not required.

(*iv*) The gauge and its connecting line is filled with an inert liquid and as such liquid seals are provided. Trapped air at any point of gauge lines may cause serious errors in pressure reading.

1.17. SPECIFIC VOLUME

The *specific volume* of a system is the volume occupied by the unit mass of the system. The symbol used is v and units are ; for example, m^3/kg. The symbol V will be used for volume. (Note that specific volume is *reciprocal of density*).

Example 1.1. *Convert the following readings of pressure to kPa assuming that barometer reads 760 mm of Hg.*

(*i*) *80 cm of Hg* (*ii*) *30 cm Hg vacuum*

(*iii*) *1.35 m H_2O gauge* (*iv*) *4.2 bar.*

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Solution. Assuming density of Hg, $\rho_{Hg} = 13.596 \times 1000$ kg/m^3
Pressure of 760 mm of Hg will be

$$= \rho \times g \times h = 13.596 \times 1000 \times 9.806 \times \frac{760}{1000}$$

$$= 101325 \text{ Pa} = 101.325 \text{ kPa}.$$

(*i*) **Pressure of 80 cm of Hg**

$$= \frac{800}{760} \times 101.325 = \mathbf{106.65\ kPa.} \quad \textbf{(Ans.)}$$

(*ii*) **30 cm Hg vacuum**

$$= 76 - 30 = 46 \text{ cm of Hg absolute.}$$

Pressure due to 46 cm of Hg

$$= \frac{460}{760} \times 101.325 = \mathbf{61.328\ kPa.} \quad \textbf{(Ans.)}$$

(*iii*) **Pressure due to 1.35 m H_2O gauge**

$$= 1000 \times 9.806 \times 1.35 = 13238 \text{ Pa} = \mathbf{13.238\ kPa.} \quad \textbf{(Ans.)}$$

(*iv*) **4.2 bar**

$$= 4.2 \times 10^2 \text{ kPa} = \mathbf{420\ kPa.} \quad \textbf{(Ans.)}$$

Note. Pressure of 1 atmosphere
= 760 mm of Hg
= 101325 N/m^2.

The above values are standard. To get this value we have to use $\rho_{Hg} = 13596$ kg/m^3 and $g = 9.806$ m/s^2. When we use $\rho_{Hg} = 13600$ kg/m^3 and $g = 9.81$ m/s^2, we get $p_{atm.} = 101396$ N/m^2 which is slightly different from 101325 N/m^2. It is recommended that for pressure of 1 atm. the value 101325 N/m^2 should be used.

Example 1.2. *On a piston of 10 cm diameter a force of 1000 N is uniformly applied. Find the pressure on the piston.*

Solution. Diameter of the piston, $d = 10$ cm (= 0.1 m)
Force applied on the piston, $F = 1000$ N

$$\therefore \text{ Pressure on the piston, } p = \frac{\text{Force}}{\text{Area}} = \frac{F}{A} = \frac{1000}{\pi / 4 \times (0.1)^2}$$

$$= 127307 \text{ N/m}^2 = \mathbf{127.307\ kN/m^2.} \quad \textbf{(Ans.)}$$

Example 1.3. *A tube contains an oil of specific gravity 0.9 to a depth of 120 cm. Find the gauge pressure at this depth (in kN/m^2).*

Solution. Specific gravity of oil = 0.9
Depth of oil in the tube, $h = 120$ cm = (1.2 m)
We know that $p = wh$

$= \rho.g.h$, ρ being the mass density
$= (0.9\ \rho_w) \times g \times h$, ρ_w being mass density of water

$$\left[\text{Specific gravity} = \frac{\rho}{\rho_w}\right]$$

$$= 0.9 \times 1000 \times 9.81 \times 1.2 \text{ N/m}^2$$

$$= 10594.8 \text{ N/m}^2 = \mathbf{10.595\ kN/m^2.} \quad \textbf{(Ans.)}$$

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Example 1.4. *A vacuum recorded in the condenser of a steam power plant is 740 mm of Hg. Find the absolute pressure in the condenser in Pa. The barometric reading is 760 mm of Hg.*

Solution. Vacuum recorded in the condenser = 740 mm of Hg

Barometric reading = 760 mm of Hg

We know that,

Absolute pressure in the condenser

= Barometric reading – vacuum in the condenser

= 760 – 740 = 20 mm of Hg

$= 20 \times 133.4 \text{ N/m}^2$ ($\because$ 1 mm of Hg = 133.4 N / m^2)

$= 2668 \text{ N/m}^2 =$ **2668 Pa.** (Ans.)

Example 1.5. *A vessel of cylindrical shape is 50 cm in diameter and 75 cm high. It contains 4 kg of a gas. The pressure measured with manometer indicates 620 mm of Hg above atmosphere when barometer reads 760 mm of Hg. Determine :*

(i) The absolute pressure of the gas in the vessel in bar.

(ii) Specific volume and density of the gas.

Solution. Diameter of the vessel, d = 50 cm (= 0.5 m)

Height of the vessel, h = 75 cm (= 0.75 m)

Mass of gas in the vessel, m = 4 kg

Manometer reading = 620 mm of Hg above atmosphere

Barometer reading = 760 mm of Hg

Now, volume of the vessel $= \frac{\pi}{4} d^2 \times h = \frac{\pi}{4} \times (0.5)^2 \times (0.75) = 0.147 \text{ m}^3$.

(*i*) **Total pressure in the vessel**

= 760 + 620 = 1380 mm of Hg

$= 1380 \times 133.4 \text{ N/m}^2$ [$\because$ 1 mm of Hg = 133.4 N/m^2]

$= 1.841 \times 10^5 \text{ N/m}^2 =$ **1.841 bar.** (Ans.) [$\because$ 1 bar = 10^5 N/m^2]

(*ii*) **Specific volume** $= \frac{0.147}{4} =$ **0.03675 m^3/kg.** (Ans.)

Density $= \frac{4}{0.147} =$ **27.21 kg/m^3.** (Ans.)

Example 1.6. *In a pipe line the pressure of gas is measured with a mercury manometer having one limb open to the atmosphere (Fig. 1.25). If the difference in the height of mercury in the two limbs is 550 mm, calculate the gas pressure.*

Given : Barometric reading = 761 mm of Hg

Acceleration due to gravity = 9.79 m/s^2

Density of mercury = 13640 kg/m^3.

Solution. At the plane LM, we have

$$p = p_0 + \rho g h$$

Now, $p_0 = \rho g h_0$

where h_0 = barometric height ; ρ = density of mercury ; p_0 = atmospheric pressure

Therefore, $p = \rho g h_0 + \rho g h = \rho g (h_0 + h)$

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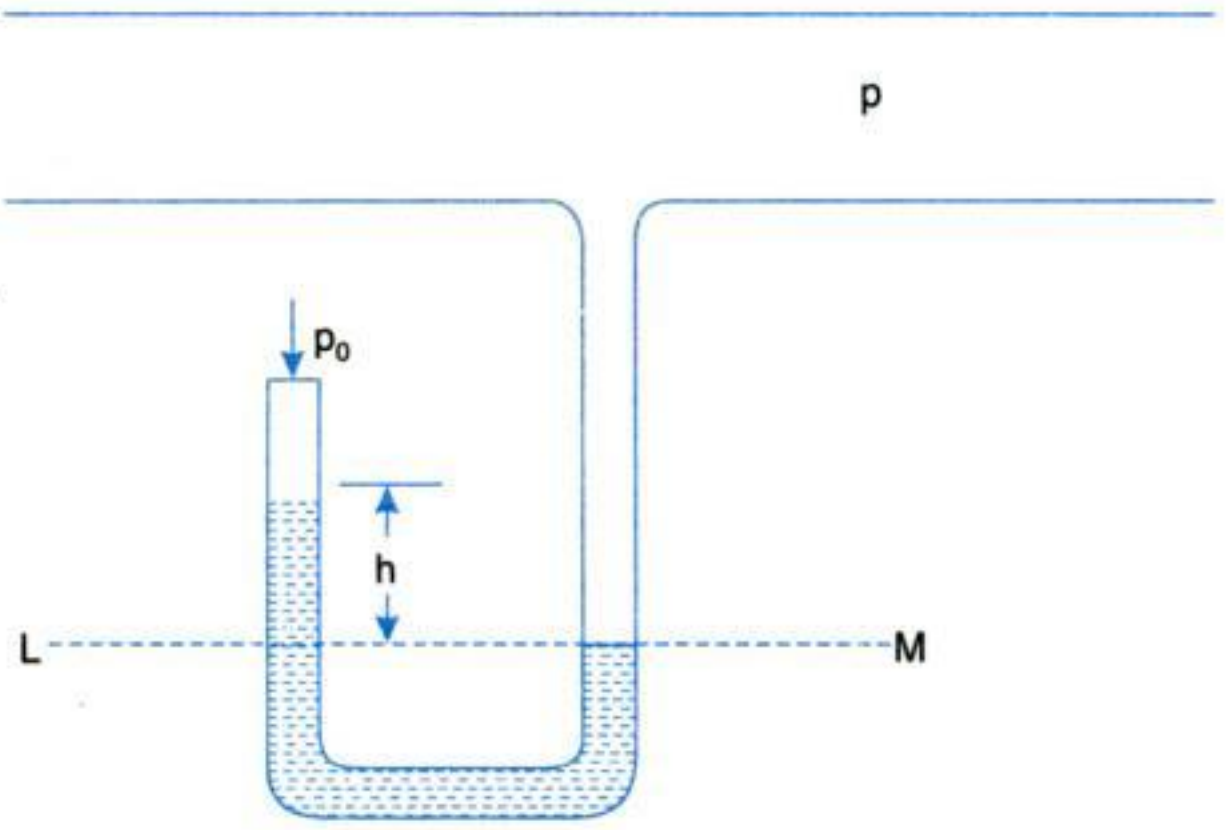

Fig. 1.25

$$= 13640 \times 9.79 \left(\frac{761}{1000} + \frac{550}{1000}\right) = 13640 \times 9.79\ (0.761 + 0.55)$$

$$= 175.065 \times 10^3\ \text{N/m}^2 = 175.065\ \text{kPa} = \textbf{1.75 bar.}\quad \textbf{(Ans.)}$$

Example 1.7. *A U-tube mercury manometer with one arm open to atmosphere is used to measure pressure in a steam pipe. The level of mercury in open arm is 97.5 mm greater than that in the arm connected to the pipe. Some of steam in the pipe condenses in the manometer arm connected to the pipe. The height of this column is 34 mm. The atmospheric pressure is 760 mm of Hg. Find the absolute pressure of steam.* **(Poona University, Nov. 1997)**

Solution. Equating the pressure in mm of Hg on both arms above the line *XX* (Fig. 1.26), we get

$$p_{\text{abs.}} + p_{\text{water}} = p_{\text{Hg}} + p_{\text{atm.}}$$

Now, $$p_{\text{water}} = \frac{34}{13.6} = 2.5 \text{ mm of Hg}$$

$\therefore$ $$p_{\text{abs}} + 2.5 = 97.5 + 760$$

or $$p_{\text{abs}} = 97.5 + 760 - 2.5$$

$$= 855 \text{ mm of Hg}$$

$$= 855 \times p_{\text{Hg}} \times g \times 10^{-5} \text{ bar}$$

$$= \frac{855}{1000}(m) \times (13.6 \times 1000)\ (\text{kg/m}^3)$$

$$\times 9.81 \times 10^{-5}$$

$$= \textbf{1.1407 bar.}\quad \textbf{(Ans.)}$$

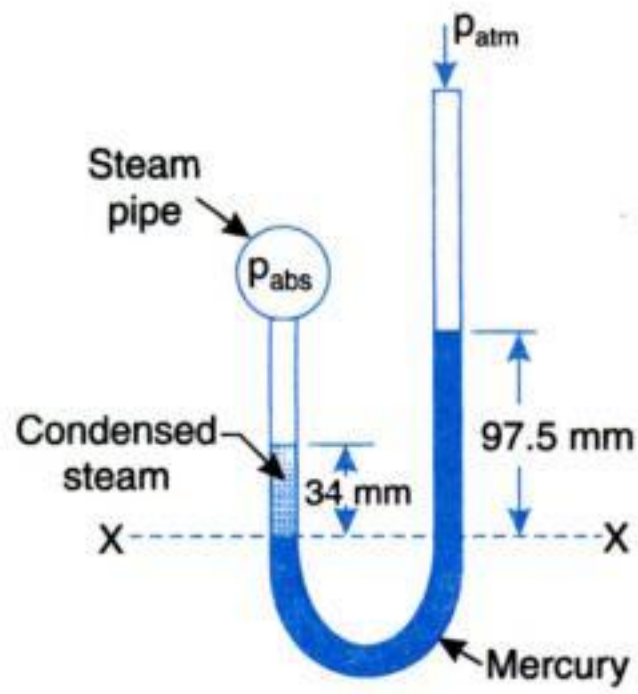

Fig. 1.26

Example 1.8. *A U-tube manometer is connected to a gas pipe. The level of the liquid in the manometer arm open to the atmosphere is 170 mm lower than the level of the liquid in the arm connected to the gas pipe. The liquid in the manometer has specific gravity of 0.8. Find the absolute pressure of the gas if the manometer reads 760 mm of Hg.* **(Poona University, Dec. 1996)**

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Solution. Equating pressure on both arms above the line *XX* (Fig. 1.27), we get

$$p_{gas} + p_{liquid} = p_{atm.} \qquad ...(i)$$

Now, $p_{liquid} = \rho.g.h$

$$= (0.8 \times 1000) \times 9.81 \times \frac{170}{1000}$$

$$= 1334.16 \text{ N/m}^2$$

$$= 0.0133416 \text{ bar}$$

$p_{atm.}$ = 760 mm of Hg = 1.01325 bar

Substituting these value is eqn. (*i*) above, we have

$$p_{gas} + 0.0133416 = 1.01325$$

$$\therefore \quad \boldsymbol{p_{gas} = 0.9999 \text{ bar.}} \quad \textbf{(Ans.)}$$

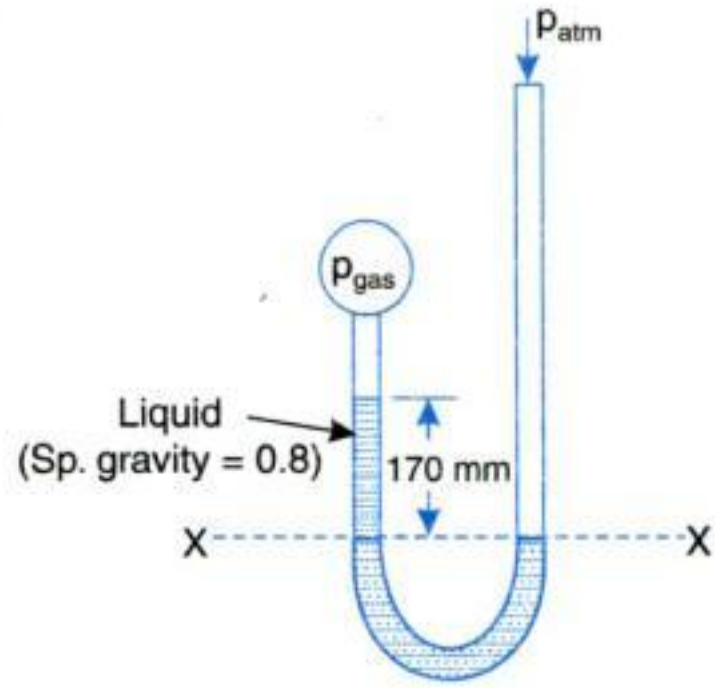

Fig. 1.27

Example 1.9. *Estimate the mass of a piston that can be supported by a gas entrapped under the piston in a 200 mm diameter vertical cylinder when a manometer indicates a difference of 117 mm of Hg column for the gas pressure.* **(Poona University, May 1996)**

Solution. Refer Fig. 1.28.

Let m = mass of the piston, kg.

p = pressure of the gas

= 117 mm of Hg column

Dia. of vertical cylinder, d = 200 mm

Now, downward force = $m.g$ ...(*i*)

and upward force = $p \times \pi/4\ d^2$ ...(*ii*)

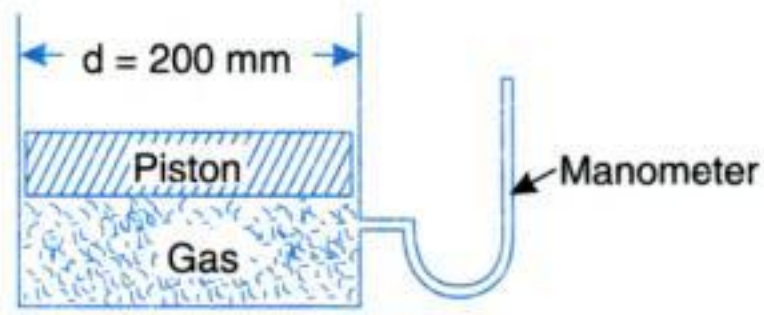

Fig. 1.28

Equating eqns. (*i*) and (*ii*), we get

$$m.g = p \times \pi/4\ d^2$$

$$m \times 9.81 = \left(13.6 \times 1000 \times 9.81 \times \frac{117}{1000}\right) \times \frac{\pi}{4} \times \left(\frac{200}{1000}\right)^2 \qquad (\because \quad p = \rho gh)$$

$$\therefore \quad \mathbf{m = 49.989 \text{ kg.}} \quad \textbf{(Ans.)}$$

1.18. REVERSIBLE AND IRREVERSIBLE PROCESSES

Reversible process. *A reversible process (also sometimes known as quasi-static process) is one which can be stopped at any stage and reversed so that the system and surroundings are exactly restored to their initial states.*

This process has the following *characteristics* :

1. It must pass through the same states on the reversed path as were initially visited on the forward path.

2. This process when undone will leave no history of events in the surroundings.

3. It must pass through a continuous series of equilibrium states.

No real process is truely reversible but some processes may approach reversibility, to close approximation.

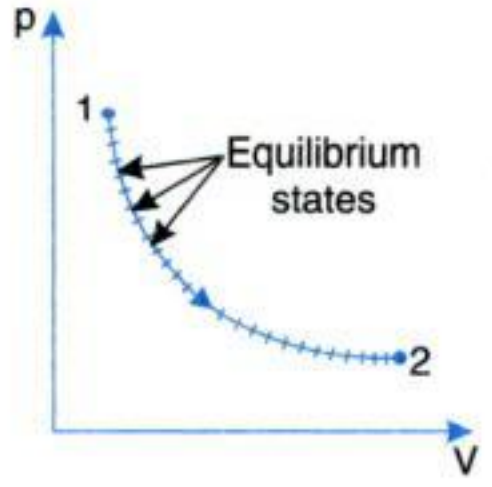

Fig. 1.29. Reversible process.

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Examples. Some examples of nearly reversible processes are :

(*i*) Frictionless relative motion.

(*ii*) Expansion and compression of spring.

(*iii*) Frictionless adiabatic expansion or compression of fluid.

(*iv*) Polytropic expansion or compression of fluid.

(*v*) Isothermal expansion or compression.

(*vi*) Electrolysis.

Irreversible process. An *irreversible process* is one *in which heat is transferred through a finite temperature.*

Examples.

(*i*) Relative motion with friction

(*ii*) Combustion

(*iii*) Diffusion

(*iv*) Free expansion

(*v*) Throttling

(*vi*) Electricity flow through a resistance

(*vii*) Heat transfer

(*viii*) Plastic deformation.

An irreversible process is usually represented by a dotted (or discontinuous) line joining the end states to indicate that the intermediate states are indeterminate (Fig. 1.30).

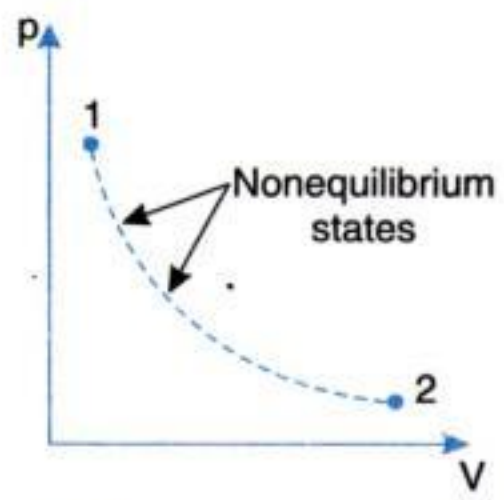

Fig. 1.30. Irreversible process.

Irreversibilities are of *two types* :

1. **External irreversibilities.** These are associated with *dissipating effects outside the working fluid.*

Example. *Mechanical friction occurring during a process due to some external source.*

2. **Internal irreversibilities.** These are associated with *dissipating effects within the working fluid.*

Example. *Unrestricted expansion of gas, viscosity and inertia of the gas.*

1.19. ENERGY, WORK AND HEAT

1.19.1. Energy

Energy is a general term embracing *energy in transition* and *stored energy*. The stored energy of a substance may be in the forms of *mechanical energy* and *internal energy* (other forms of stored energy may be chemical energy and electrical energy). Part of the stored energy may take the form of either potential energy (which is the gravitational energy due to height above a chosen datum line) or kinetic energy due to velocity. The balance part of the energy is known as *internal energy*. In a ***non-flow process*** *usually there is no change of potential or kinetic energy and hence change of mechanical energy will not enter the calculations.* In a ***flow process***, however, *there may be changes in both potential and kinetic energy and these must be taken into account while considering the changes of stored energy.* ***Heat and work*** are *the forms of energy in transition.* These are the only forms in which energy can cross the boundaries of a system. *Neither heat nor work can exist as stored energy.*

1.19.2. Work and Heat

Work

Work is said to be done when a *force moves through a distance*. If a part of the boundary of a system undergoes a displacement under the action of a pressure, the work done W is the product of the force (pressure × area), and the distance it moves in the direction of the force. Fig. 1.31 (*a*)

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illustrates this with the conventional piston and cylinder arrangement, the heavy line defining the boundary of the system. Fig. 1.31 (*b*) illustrates another way in which work might be applied to a system. A force is exerted by the paddle as it changes the momentum of the fluid, and since this force moves during rotation of the paddle work is done.

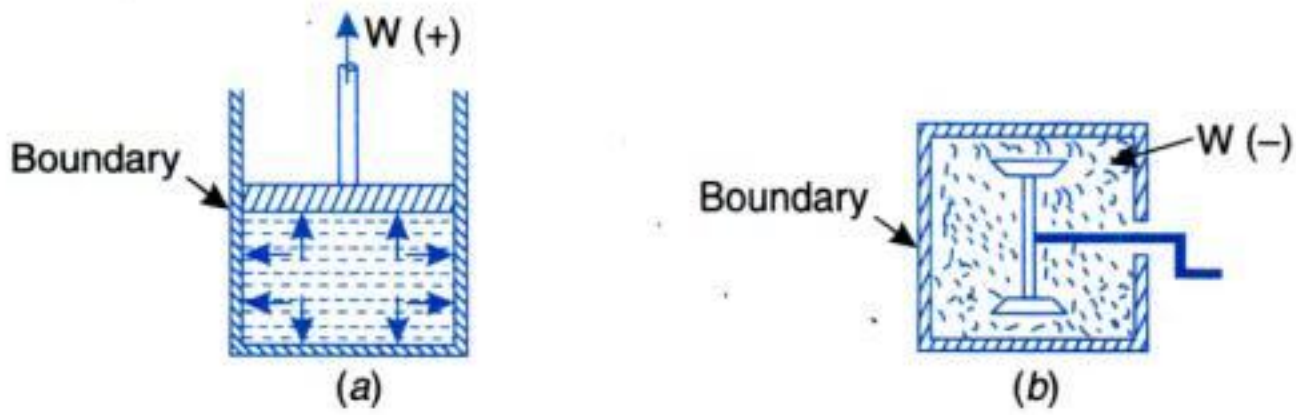

Fig. 1.31

Work is a transient quantity which only appears at the boundary while a change of state is taking place within a system. Work is 'something' which appears at the boundary when a system changes its state due to the movement of a part of the boundary under the action of a force.

Sign convention

- If the work is done *by* the system *on* the surroundings, *e.g.,* when a fluid expands pushing a piston outwards, the work is said to be *positive*.

  i.e., *Work output of the system* = + *W*

- If the work is done *on* the system *by* the surroundings, *e.g.*, when a force is applied to a rotating handle, or to a piston to compress a fluid, the work is said to be *negative*.

  i.e., *Work input to system* = – *W*

Heat

Heat (denoted by the symbol Q), may be, defined in an analogous way to work as follows :

"Heat is 'something' which appears at the boundary when a system changes its state due to a difference in temperature between the system and its surroundings".

Heat, like work, is a transient quantity which only appears at the boundary while a change is taking place within the system.

It is apparent that neither δW or δQ are exact differentials and therefore any integration of the elemental quantities of work or heat which appear during a change from state 1 to state 2 must be written as

$$\int_1^2 \delta W = W_{1-2} \text{ or } {}_1W_2 \text{ (or } W\text{), and}$$

$$\int_1^2 \delta Q = Q_{1-2} \text{ or } {}_1Q_2 \text{ (or } Q\text{)}$$

Sign convention :

If the heat flows *into* a system *from* the surroundings, the quantity is said to be *positive* and, conversely, if heat flows *from* the system to the surroundings it is said to be *negative*.

In other words :

Heat received by the system = + *Q*

Heat rejected or given up by the system = – *Q*.

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Comparison of Work and Heat

Similarities :

(*i*) Both are *path functions and inexact differentials.*

(*ii*) Both are boundary phenomenon *i.e.,* both are recognized at the boundaries of the system as they cross them.

(*iii*) Both are associated with a process, not a state. Unlike properties, work or heat has no meaning at a state.

(*iv*) Systems possess energy, but not work or heat.

Dissimilarities :

(*i*) In heat transfer temperature difference is required.

(*ii*) In a stable system there cannot be work transfer, however, there is no restriction for the transfer of heat.

(*iii*) The sole effect external to the system could be reduced to rise of a weight but in the case of a heat transfer other effects are also observed.

1.20. REVERSIBLE WORK

Let us consider an ideal frictionless fluid contained in a cylinder above a piston as shown in Fig. 1.32 (*a*). *Assume that the pressure and temperature of the fluid are uniform and that there is no friction between the piston and the cylinder walls.*

Let A = Cross-sectional area of the piston,

p = Pressure of the fluid at any instant,

$(p - dp)\,A$ = Restraining *force exerted* by the surroundings on the piston, and

dl = The distance moved by the piston under the action of the *force exerted.*

Then work done by the fluid on the piston is given by force times the distance moved,

i.e., Work done by the fluid

$$= (pA) \times dl = pdV$$

(where dV = a small increase in volume)

Or considering unit mass

Work done = pdv (where v = specific volume)

This is only *true* when (*a*) the *process is frictionless and (b) the difference in pressure between the fluid and its surroundings during the process is infinitely small.* Hence when a reversible process takes place between state 1 and state 2, we have

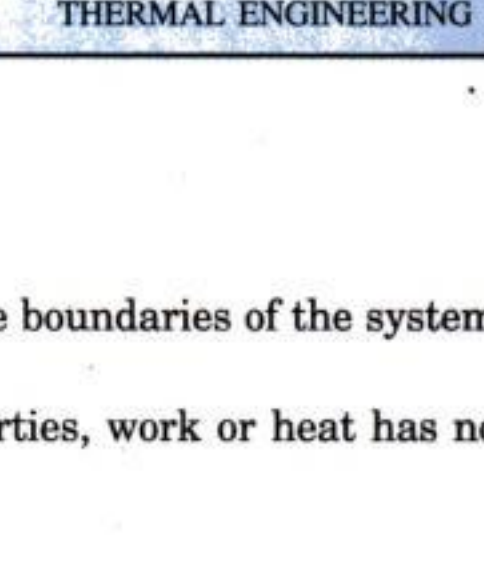

Fig. 1.32 (*a*)

$$\text{Work done by the unit mass of fluid} = \int_1^2 pdv$$

When a fluid undergoes a reversible process a series of state points can be joined up to form a line on a diagram of properties. The work done by the fluid during any reversible process is therefore given by the area under the line of process plotted on a p-v diagram [Fig. 1.32 (*b*)].

i.e., Work done = Shaded area on Fig. 1.32 (*b*)

$$= \int_1^2 pdv\,.$$

When p can be expressed in terms of v then the integral, $\int_1^2 pdv$, can be evaluated.

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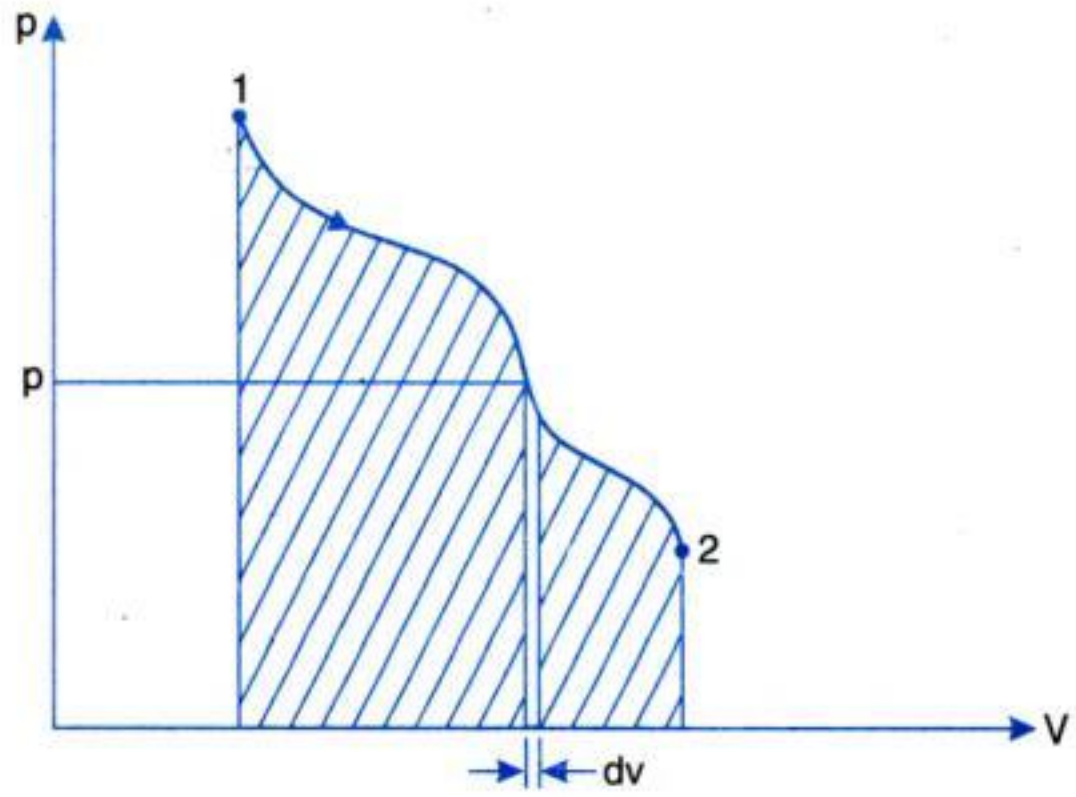

Fig. 1.32 (*b*)

Example 1.10. *An artificial satellite revolves round the earth with a relative velocity of 800 m/s. If acceleration due to gravity is 9 m/s² and gravitational force is 3600 N, calculate its kinetic energy.*

Solution. Relatively velocity of satellite, v = 800 m/s

Acceleration due to gravity, $g = 9 \text{ m/s}^2$

Gravitational force, $m.g = 3600 \text{ N}$

$\therefore$ Mass, $m = \dfrac{3600}{g} = \dfrac{3600}{9} = 400 \text{ kg}.$

Kinetic energy $= \dfrac{1}{2} mv^2 = \dfrac{1}{2} \times 400 \times (800)^2 \text{ J} = 128 \times 10^6 \text{ J}$ or **128 MJ.** **(Ans.)**

Example 1.11. *The specific heat capacity of the system during a certain process is given by*

$$c_n = (0.4 + 0.004\ T)\ kJ/kg°C.$$

If the mass of the gas is 6 kg and its temperature changes from 25°C to 125°C find :

(*i*) *Heat transferred ;* (*ii*) *Mean specific heat of the gas.*

Solution. Mass of the gas, m = 6 kg

Change in temperature of the gas = 25°C to 125°C

(*i*) **Heat transferred, Q :**

We know that heat transferred is given by,

$$Q = \int m\ c_n\ dT = 6 \int_{25}^{125} (0.4 + 0.004\ T)\ dT$$

$$= 6 \left[0.4\ T + 0.04 \left(\frac{T^2}{2} \right) \right]_{25}^{125}$$

$$= 6[0.4\ (125 - 25) + 0.002\ (125^2 - 25^2)]$$

$$= 6(40 + 30) = \textbf{420 kJ.} \quad \textbf{(Ans.)}$$

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(*ii*) **Mean specific heat of the gas, c_n :**

$$Q = m.c_n.dT$$

i.e., $$420 = 6 \times c_n \times (125 - 25)$$

$$\therefore \quad \mathbf{c_n = \frac{420}{6 \times 100} = 0.7\ kJ/kg°C.} \quad \textbf{(Ans.)}$$

Example 1.12. *A temperature scale of certain thermometer is given by the relation*

$$t = a\ ln\ p + b$$

where a and b are constants and p is the thermometric property of the fluid in the thermometer. If at the ice point and steam point the thermometric properties are found to be 1.5 and 7.5 respectively what will be the temperature corresponding to the thermometric property of 3.5 on Celsius scale.

(Poona University, Nov. 1997)

Solution. $t = a \ln p + b$ ...(Given)

On Celsius scale :

Ice point = 0°C, and

Steam point = 100°C

∴ From given conditions, we have

$$0 = a \ln 1.5 + b \quad ...(i)$$

and $$100 = a \ln 7.5 + b \quad ...(ii)$$

i.e., $$0 = a \times 0.4054 + b \quad ...(iii)$$

and $$100 = a \times 2.015 + b \quad ...(iv)$$

Subtracting (*iii*) from (*iv*), we get

$$100 = 1.61a$$

or $$a = 62.112$$

Substituting this value in eqn. (*iii*), we get

$$b = -0.4054 \times 62.112 = -25.18$$

∴ When $p = 3.5$ the value of temperature is given by

$$t = 62.112 \ln (3.5) - 25.18 = \mathbf{52.63°C.} \quad \textbf{(Ans.)}$$

Example 1.13. *A thermocouple with test junction at t°C on gas thermometer scale and reference junction at ice point gives the e.m.f. as*

$$e = 0.20\ t - 5 \times 10^{-4} t^2\ mV.$$

The millivoltmeter is calibrated at ice and steam points. What will be the reading on this thermometer where the gas thermometer reads 70°C ?

Solution. $e = 0.20\ t - 5 \times 10^{-4} t^2$ mV ...(Given)

At ice point : When $t = 0°C$, $e = 0$

At steam point : When $t = 100°C$,

$$e = 0.20 \times 100 - 5 \times 10^{-4} \times (100)^2 = 15 \text{ mV}$$

Now, when $t = 70°C$

$$e = 0.20 \times 70 - 5 \times 10^{-4} \times (70)^2 = 11.55 \text{ mV}$$

∴ When the gas thermometer reads 70°C the thermocouple will read

$$t = \frac{100 \times 11.55}{15} = \mathbf{77°C.} \quad \textbf{(Ans.)}$$

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☞ **Example 1.14.** *Comment whether the following quantities can be called as properties or not :*

$(i) \int pdV$, $(ii) \int Vdp$, *and* $(iii) \int pdV + \int Vdp$.

Solution. $(i) \int pdV$:

p is a function of V and integral can only be evaluated if relation between p and V is known. It is thus an *inexact differential* and hence **not a property.**

$(ii) \int Vdp$:

It is **not a property** for the *same reason* as mentioned in (i).

$(iii) \int pdV + \int Vdp$:

$$\int pdV + \int Vdp = \int pdV + Vdp = \int d(pV) = pV.$$

Thus the integral can be evaluated without knowing the relation between p and V. It is an *exact differential* and hence **it is a property.**

Example 1.15. *Gas from a cylinder of compressed helium is used to inflate an inelastic flexible balloon, originally folded completely flat, to a volume 0.6 m^3. If the barometer reads 760 mm Hg, what is the amount of work done upon the atmosphere by the balloon ? Sketch the system before and after the process.*

Solution. Refer Fig. 1.33. The firm line B_1 shows the boundary of the system before the process, and dotted line B_2 shows the boundary after the process.

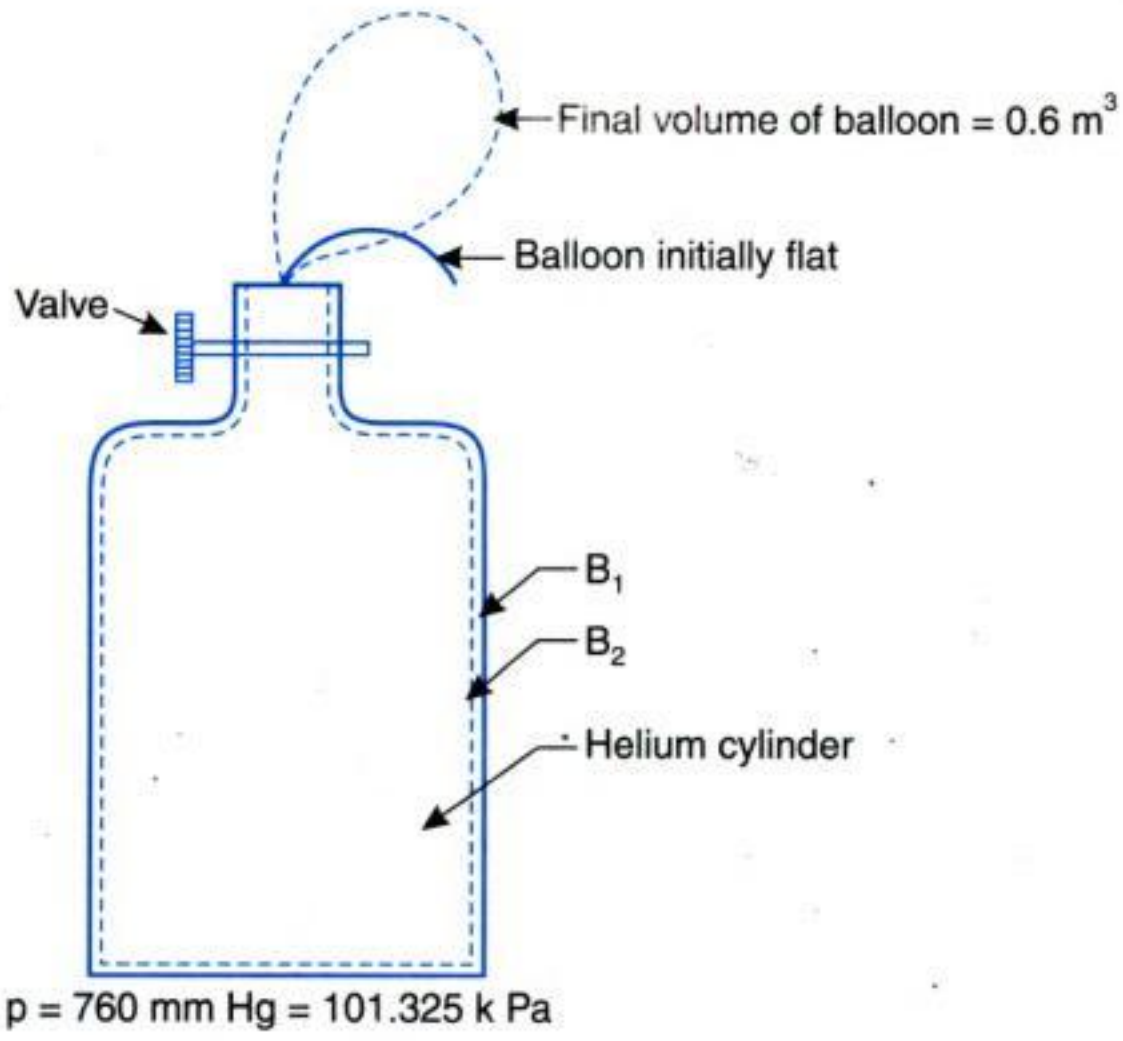

Fig. 1.33

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The displacement work,

$$W_d = \int_{\text{cylinder}} pdV + \int_{\text{balloon}} pdV = 0 + \int_{\text{balloon}} pdV$$

$$= 101.325 \times 0.6 \qquad [\because \quad dV = 0.6 \text{ m}^3]$$

$$= \mathbf{60.795 \text{ kJ.}} \quad \textbf{(Ans.)}$$

This is a positive work, because the *work is done by the system*. Work done by the atmosphere is – 60.795 kJ. Since the wall of the cylinder is rigid there is no *pdV*-work involved in it.

It is assumed that the pressure in the balloon is atmospheric at all times, since the balloon fabric is light, inelastic and unstressed. If the balloon were elastic and stressed during the filling process, the work done by the gas would be *greater* than 60.795 kJ by an amount equal to the work done in stretching the balloon, although the displacement work done by atmosphere is still – 60.795 kJ. However, if the system includes both the gas and the balloon, the displacement work should be 60.795 kJ, as estimated above.

Example 1.16. *Determine the work done by the air which enters into an evacuated vessel from atmosphere when the valve is opened. The atmospheric pressure is 1.013 bar and 1.5 m^3 of air at atmospheric condition enters into the vessel.*

Solution. Fig. 1.34 shows the initial and final condition of the system.

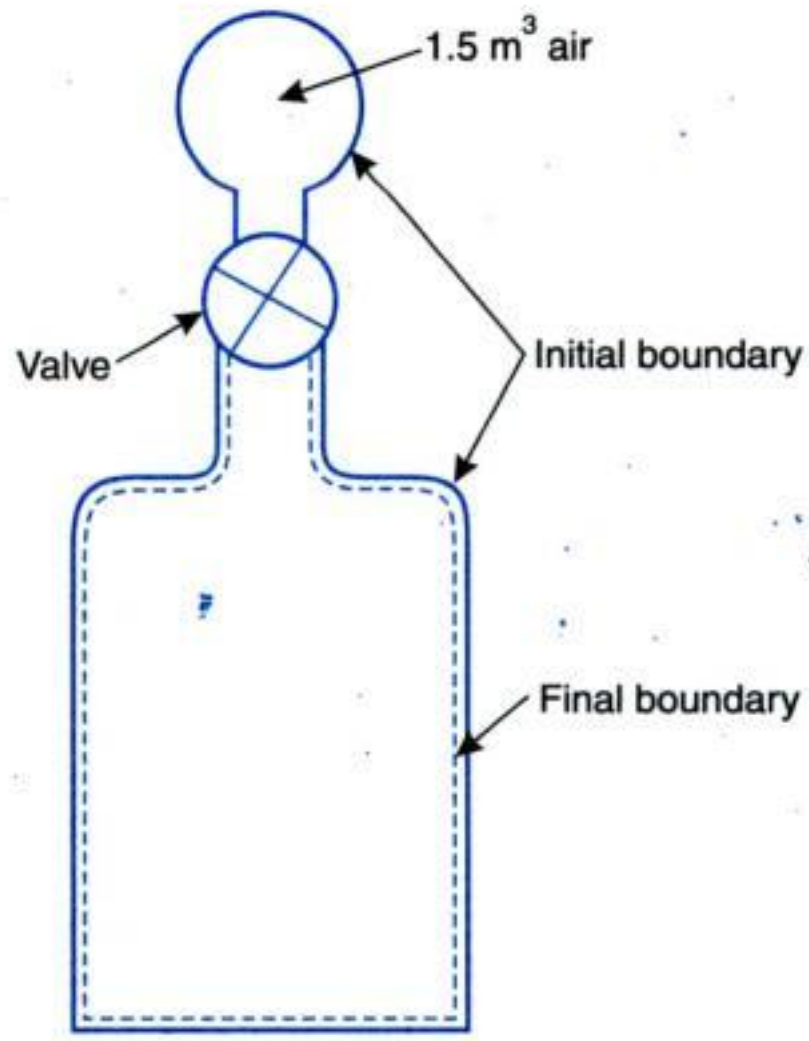

Fig. 1.34

No work is done by the boundary in contact with the vessel as the boundary does not move. Work is done by the external boundary at constant pressure.

$$\therefore \qquad W = \int_{V_1}^{V_2} p\,dV = \int_{1.5}^{0} p\,dV \qquad [\because \ V_1 = 1.5 \text{ m}^3 \text{ and } V_2 = 0]$$

$$= p(0 - 1.5) = 1.013 \times 10^5 \times (-1.5)$$

$$= -1.5195 \times 10^5 \text{ J} = \mathbf{-151.95 \text{ kJ.}} \quad \textbf{(Ans.)}$$

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Since the free air boundary is contracting, the work done by the system is negative, and the surroundings do positive work upon the system.

Example 1.17. *A piston and cylinder machine containing a fluid system has a stirring device as shown in Fig. 1.35. The piston is frictionless, and it is held down against the fluid due to atmospheric pressure of 101.3 kPa. The stirring device is turned 9500 revolutions with an average torque against the fluid of 1.25 Nm. Meanwhile the piston of 0.65 m diameter moves out 0.6 m. Find the net work transfer for the system.*

Solution. Refer Fig. 1.35.

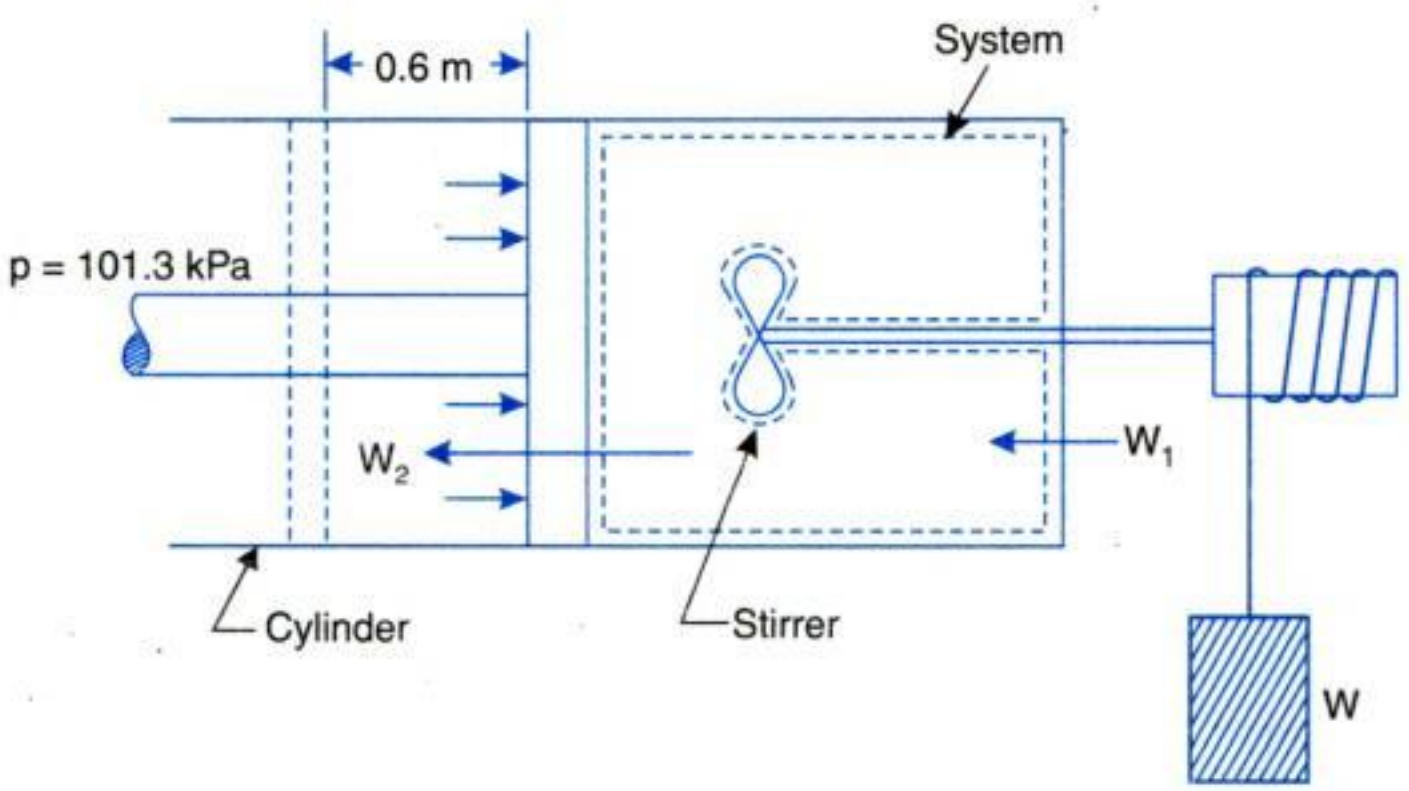

Fig. 1.35

Work done by *the stirring device* upon the system,

$$W_1 = 2\pi NT$$

where, T = Torque = 1.25 Nm

N = Number of revolutions = 9500

$$W_1 = 2\pi \times 9500 \times 1.25 = 74622 \text{ Nm} = 74.622 \text{ kJ}$$

This is *negative work* for the system.

Work done *by the system* upon the surroundings

$$W_2 = (pA).\ L$$

where, p = Pressure = 101.3 kPa

A = Area of the piston = $\pi/4 \times (0.65)^2 = 0.3318$ m^2, and

L = Distance moved by the piston = 0.6 m

$$W_2 = 101.3 \times 0.3318 \times 0.6 = 20.167 \text{ kJ}$$

This is a *positive work for the system.*

Hence, the net work transfer for the system

$$W_{net} = W_1 + W_2 = -74.622 + 20.167 = \mathbf{-54.455\ kJ.}\quad \textbf{(Ans.)}$$

Example 1.18. *A diesel engine piston which has an area of 45 cm^2 moves 5 cm during part of suction stroke. 300 cm^3 of fresh air is drawn in from the atmosphere. The pressure in the cylinder during suction stroke is 0.9×10^5 N/m^2 and the atmospheric pressure is 1.013×10^5 N/m^2. The difference between the suction and atmospheric pressure is accounted for flow resistance in the suction pipe and inlet valve. Find the net work done during the process.*

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Solution. Area of diesel engine piston

$$= 45 \text{ cm}^2 = 45 \times 10^{-4} \text{ m}^2$$

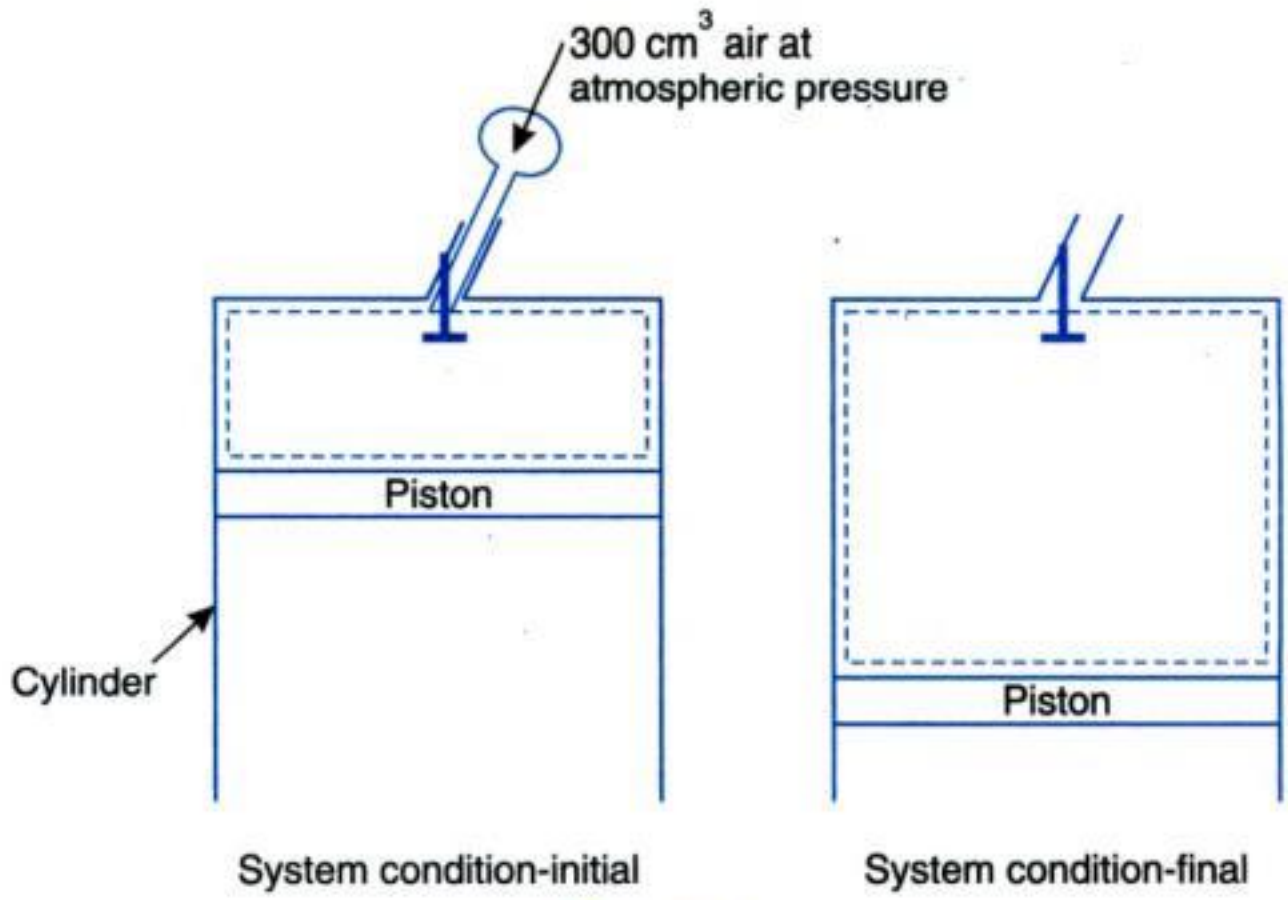

Fig. 1.36

Amount of fresh air drawn in from atmosphere

$$= 300 \text{ cm}^3 = 300 \times 10^{-6} \text{ m}^3$$

The pressure inside the cylinder during suction stroke

$$= 0.9 \times 10^5 \text{ N/m}^2$$

Atmospheric pressure $= 1.013 \times 10^5 \text{ N/m}^2$

Initial and final conditions of the system are shown in Fig. 1.36.

Net work done = Work done by free air boundary + work done on the piston

The work done by the free air = –ve because *boundary contracts*

The work done by the cylinder on the piston = +ve because the *boundary expands*

$$\therefore \quad \textbf{Net work done} = \int_{\text{Piston}} pdV + \int_{\substack{\text{Free air} \\ \text{boundary}}} pdV$$

$$= \left[0.9 \times 10^5 \times 45 \times 10^{-4} \times \frac{5}{100} - 1.013 \times 10^5 \times 300 \times 10^{-6}\right]$$

$$= [20.25 - 30.39] = \textbf{– 10.14 Nm or J.} \quad \textbf{(Ans.)}$$

Example 1.19. *The properties of a closed system change following the relation between pressure and volume as $pV = 3.0$ where p is in bar V is in m^3. Calculate the work done when the pressure increases from 1.5 bar to 7.5 bar.*

Solution. Initial pressure, $p_1 = 1.5$ bar

Final pressure, $p_2 = 7.5$ bar

Relation between p and V, $pV = 3.0$

Work done, W :

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The work done during the process is given by

$$W = \int_{V_1}^{V_2} p dV$$

$$V_1 = \frac{3.0}{p_1} = \frac{3.0}{1.5} = 2 \text{ m}^3$$

$$V_2 = \frac{3.0}{p_1} = \frac{3.0}{1.5} = 0.4 \text{ m}^3$$

$\therefore$ $$W = 10^5 \int_2^{0.4} \frac{3.0}{V} dV \text{ Nm} \qquad [\because \text{ 1 bar} = 10^5 \text{ N/m}^2]$$

$$= 10^5 \times 3.0 \left[\log_e V\right]_2^{0.4} = 10^5 \times 3.0(\log_e 0.4 - \log_e 2)$$

$$= -3 \times 10^5 \log_e (2/0.4) = -3 \times 10^5 \times \log_e 5 = -3 \times 10^5 \times 1.61 \text{ Nm}$$

$$= -4.83 \times 10^5 \text{ Nm} = -4.83 \times 10^5 \text{ J} = -\textbf{483 kJ.} \quad \textbf{(Ans.)}$$

Example 1.20. *To a closed system 150 kJ of work is supplied. If the initial volume is 0.6 m^3 and pressure of the system changes as $p = 8 - 4V$, where p is in bar and V is in m^3, determine the final volume and pressure of the system.*

Solution. Amount of work supplied to a closed system = 150 kJ

Initial volume = 0.6 m^3

Pressure-volume relationship, $p = 8 - 4V$

The work done during the process is given by

$$W = \int_{V_1}^{V_2} p dV$$

$$= 10^5 \int_{0.6}^{V_2} (8 - 4V) dV = 10^5 \left[8V - 4 \times \frac{V^2}{2}\right]_{0.6}^{V}$$

$$= 10^5[8(V_2 - 0.6) - 2(V_2^2 - 0{,}6^2)]$$

$$= 10^5[8V_2 - 4.8 - 2V_2^2 + 0.72]$$

$$= 10^5[8V_2 - 2V_2^2 - 4.08] \text{ Nm or J}$$

But this work is equal to -150×10^3 J as this work is supplied to the system.

$\therefore$ $$-150 \times 10^3 = 10^5[8V_2 - 2V_2^2 - 4.08]$$

or $$2V_2^2 - 8V_2 + 2.58 = 0$$

$$V_2 = \frac{8 \pm \sqrt{64 - 4 \times 2 \times 2.58}}{4} = \frac{8 \pm 6.585}{4} = 0.354 \text{ m}^3$$

Positive sign is incompatible with the present problem, therefore it is not considered.

$\therefore$ **Final volume, V_2 = 0.354 m^3.** **(Ans.)**

and, **final pressure,** $p_2 = 8 - 4V = 8 - 4 \times 0.354$

$= 6.584$ bar = **6.584×10^5 N/m^2 or Pa.** **(Ans.)**

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REVERSIBLE WORK

Example 1.21. *A fluid at a pressure of 3 bar, and with specific volume of 0.18 m^3/kg, contained in a cylinder behind a piston exapnds reversibly to a pressure of 0.6 bar according to a law, $p = \frac{C}{v^2}$, where C is a constant. Calculate the work done by the fluid on the piston.*

Solution. Refer Fig. 1.37.

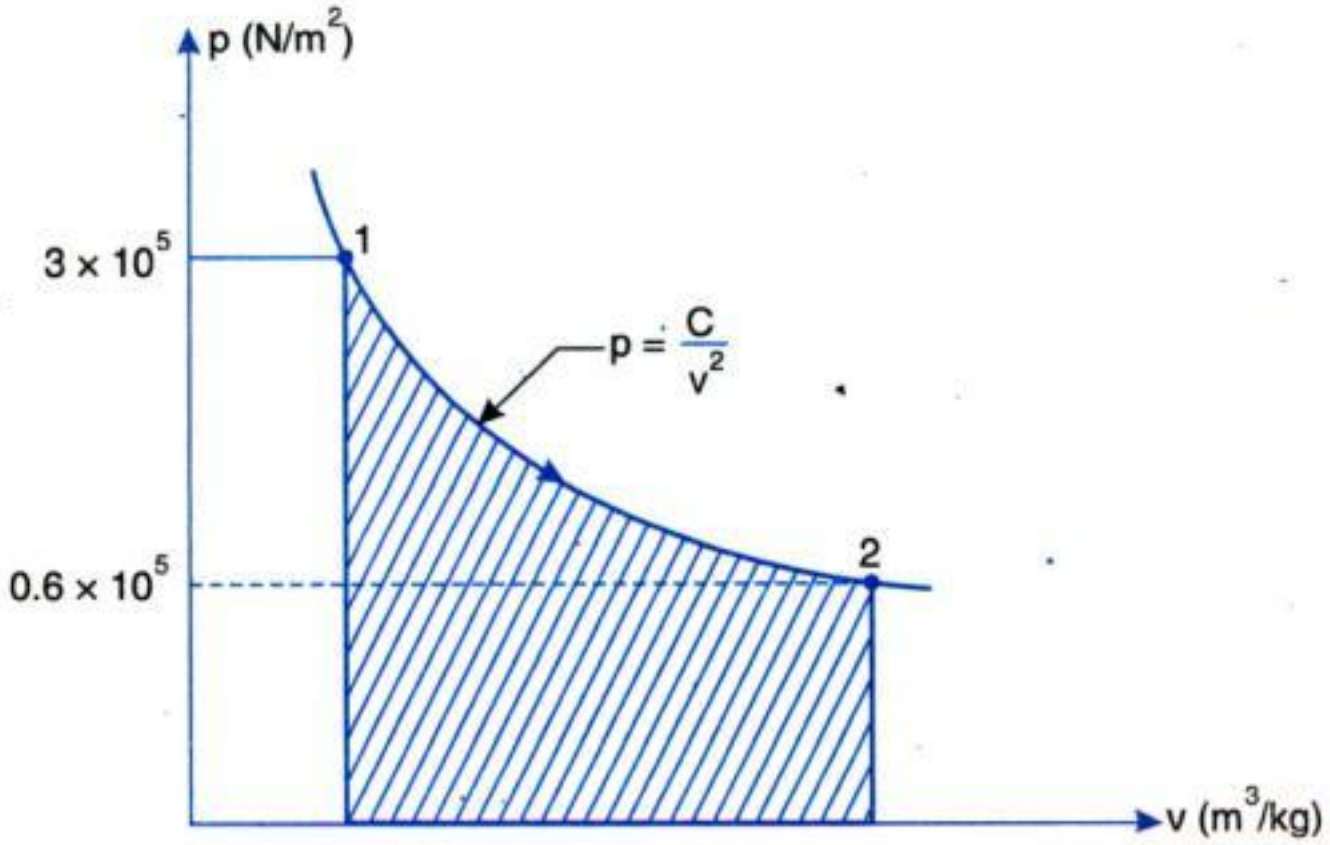

Fig. 1.37

$$p_1 = 3 \text{ bar} = 3 \times 10^5 \text{ N/m}^2$$
$$v_1 = 0.18 \text{ m}^3/\text{kg}$$

Work done = Shaded area $= \int_1^2 p\,dv$

i.e., Work done, $W = \int_1^2 \frac{C}{v^2}\,dv = C\int_1^2 \frac{dv}{v^2} = C\left|\frac{v^{-2+1}}{-2+1}\right|_{v_1}^{v_2}$

$$= C\left[-v^{-1}\right]_{v_1}^{v_2} = C\left[-\frac{1}{v}\right]_{v_1}^{v_2} = C\left[\frac{1}{v_1} - \frac{1}{v_2}\right] \quad ...(i)$$

Also $C = pv^2 = p_1 {v_1}^2 = 3 \times 0.18^2 = 0.0972 \text{ bar } (\text{m}^3/\text{kg})^2$

and $v_2 = \sqrt{\frac{C}{p_2}} = \sqrt{\frac{0.0972}{0.6}} = 0.402 \text{ m}^3/\text{kg}$

Substituting the values of C, v_1 and v_2 in eqn. (i), we get

Work done, $W = 0.0972 \times 10^5 \left[\frac{1}{0.18} - \frac{1}{0.402}\right] \text{ Nm/kg}$

= 29840 Nm/kg. **(Ans.)**

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Example 1.22. *A cylinder contains 1 kg of a certain fluid at an initial pressure of 20 bar. The fluid is allowed to expand reversibly behind a piston according to a law pV^2 = constant until the volume is doubled. The fluid is then cooled reversibly at constant pressure until the piston regains its original position ; heat is then supplied reversibly with the piston firmly locked in position until the pressure rises to the original value of 20 bar. Calculate the net work done by the fluid, for an initial volume of 0.05 m^3.*

Solution. Refer Fig. 1.38.

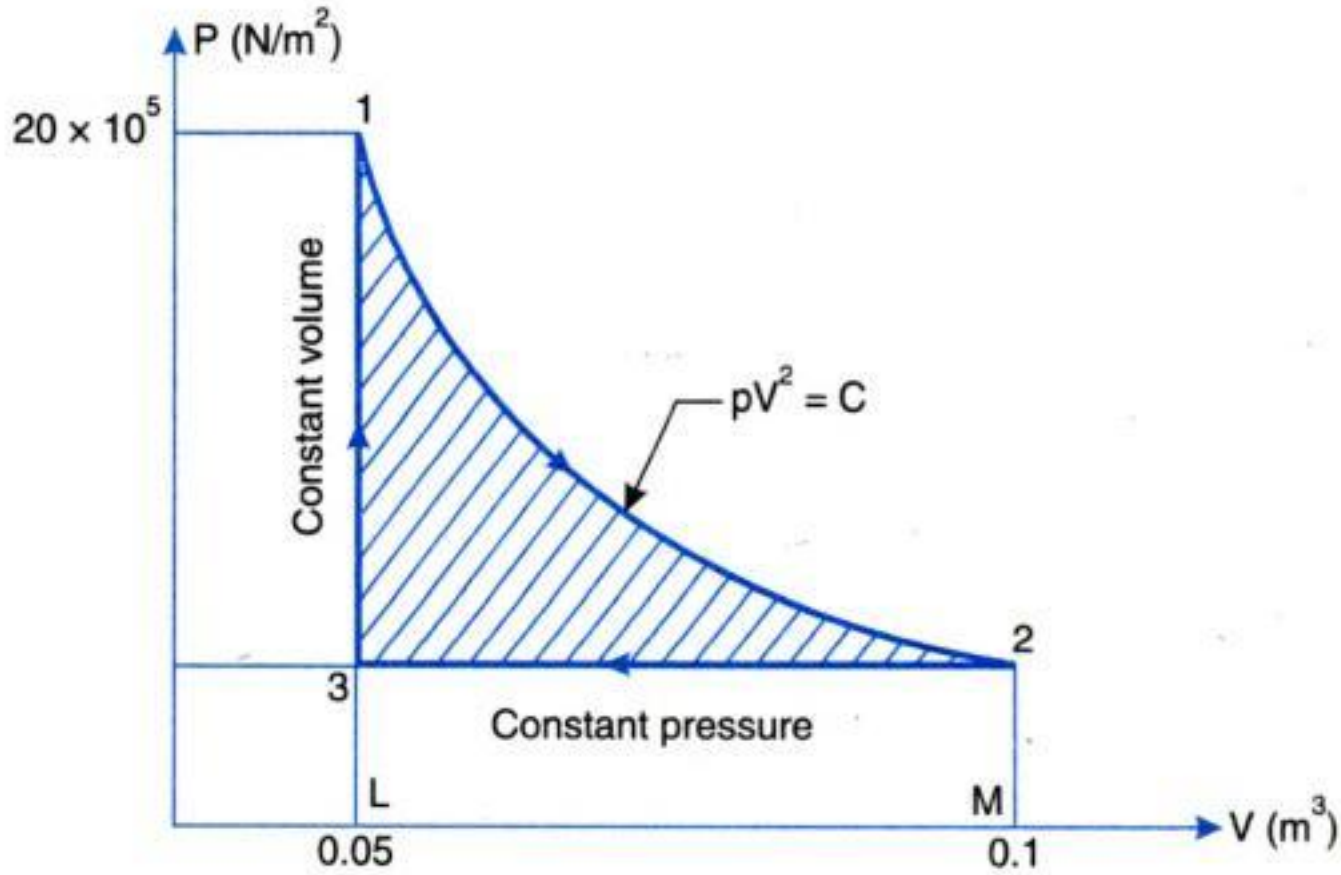

Fig. 1.38

Mass of fluid, $m = 1$ kg

$p_1 = 20$ bar $= 20 \times 10^5$ N/m^2

$V_1 = 0.05$ m^3

Considering the process 1–2

$$p_1 V_1^2 = p_2 V_2^2$$

$$\therefore \quad p_2 = p_1 \left(\frac{V_1}{V_2}\right)^2 = 20 \left(\frac{V_1}{2V_1}\right)^2 \qquad [\because \quad V_2 = 2V_1 \text{ (given)}]$$

$$= \frac{20}{4} = 5 \text{ bar}$$

Work done by the fluid from 1 to 2 = Area 12 ML1 = $\int_1^2 p\,dV$

i.e., $$W_{1-2} = \int_{v_1}^{v_2} \frac{C}{V^2}\,dV \text{ , where } C = p_1 V_1^2 = 20 \times 0.05^2 \text{ bar m}^6$$

$$\therefore \quad W_{1-2} = 10^5 \times 20 \times 0.0025 \left[-\frac{1}{V}\right]_{0.05}^{0.1}$$

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$$= 10^5 \times 20 \times 0.0025 \left(\frac{1}{0.05} - \frac{1}{0.1}\right) = 50000 \text{ Nm}$$

Work done on fluid from 2 to 3

$$= \text{Area } 32ML3 = p_2 (V_2 - V_3) = 10^5 \times 5 \times (0.1 - 0.05) = 25000 \text{ Nm}$$

Work done during the process 3-1

= 0, because piston is locked in position *(i.e., Volume remains constant)*

$\therefore$ **Net work done by the fluid**

= Enclosed area 1231 = 50000 – 25000

= **25000 Nm.** **(Ans.)**

HIGHLIGHTS

1. *Thermodynamics* is an axiomatic science which deals with the relations among heat, work and properties of systems which are in equilibrium. It basically entails four laws or axioms known as *Zeroth, First, Second* and *Third* law of thermodynamics.
2. A *system* is a finite quantity of matter or a prescribed region of space.

   A system may be a *closed, open* or *isolated* system.
3. A *phase* is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.
4. A *homogeneous system* is one which consists of a *single phase*.
5. A *heterogeneous system* is one which consists of *two or more phases*.
6. A *pure substance* is one that has a homogeneous and invariable chemical composition even though there is a change of phase.
7. A system is in *thermodynamic equilibrium* if temperature and pressure at all points are same ; there should be no *velocity gradient*.
8. A *property of a system* is a characteristic of the system which depends upon its state, but not upon how the state is reached.

   Intensive properties do not depend on the mass of the system.

   Extensive properties depend on the mass of the system.
9. *State* is the condition of the system at an instant of time as described or measured by its properties. Or each unique condition of a system is called a state.
10. A *process* occurs when the system undergoes a change in state or an energy transfer takes place at a steady state.
11. Any process or series of processes whose end states are identical is termed a *cycle*.
12. The *pressure* of a system is the force exerted by the system on unit area of boundaries. Vacuum is defined as the absence of pressure.
13. A *reversible process* is one which can be stopped at any stage and reversed so that the system and surroundings are exactly restored to their initial states.

    An *irreversible process* is one in which heat is transferred through a finite temperature.
14. Zeroth law of termodynamics states that if two systems are each equal in temperature to a third, they are equal in temperature to each other.
15. Infinite slowness is the characteristic feature of a quasi-static process. A quasi-static process is a succession of equilibrium states. It is also called a reversible process.

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OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. A definite area or space where some thermodynamic process takes place is known as
   (*a*) thermodynamic system (*b*) thermodynamic cycle
   (*c*) thermodynamic process (*d*) thermodynamic law.
2. An open system is one in which
   (*a*) heat and work cross the boundary of the system, but the mass of the working substance does not
   (*b*) mass of working substance crosses the boundary of the system but the heat and work do not
   (*c*) both the heat and work as well as mass of the working substances cross the boundary of the system
   (*d*) neither the heat and work nor the mass of the working substances cross the boundary of the system.
3. An isolated system
   (*a*) is a specified region where transfer of energy and/or mass take place
   (*b*) is a region of constant mass and only energy is allowed to cross the boundaries
   (*c*) cannot transfer either energy or mass to or from the surroundings
   (*d*) is one in which mass within the system is not necessarily constant
   (*e*) none of the above.
4. In an extensive property of a thermodynamic system
   (*a*) extensive heat is transferred (*b*) extensive work is done
   (*c*) extensive energy is utilised (*d*) all of the above
   (*e*) none of the above.
5. Which of the following is an intensive property of a thermodynamic system ?
   (*a*) Volume (*b*) Temperature
   (*c*) Mass (*d*) Energy.
6. Which of the following is the extensive property of a thermodynamic system ?
   (*a*) Pressure (*b*) Volume
   (*c*) Temperature (*d*) Density.
7. When two bodies are in thermal equilibrium with a third body they are also in thermal equilibrium with each other. This statement is called
   (*a*) Zeroth law of thermodyamics (*b*) First law of thermodynamics
   (*c*) Second law of thermodynamics (*d*) Kelvin Planck's law.
8. The temperature at which the volume of a gas becomes zero is called
   (*a*) absolute scale of temperature (*b*) absolute zero temperature
   (*c*) absolute temperature (*d*) none of the above.
9. The value of one bar (in SI units) is equal to
   (*a*) 100 N/m^2 (*b*) 1000 N/m^2
   (*c*) 1×10^4 N/m^2 (*d*) 1×10^5 N/m^2
   (*e*) 1×10^6 N/m^2.
10. The absolute zero pressure will be
    (*a*) when molecular momentum of the system becomes zero
    (*b*) at sea level (*c*) at the temperature of – 273 K
    (*d*) under vacuum conditions (*e*) at the centre of the earth.
11. Absolute zero temperature is taken as
    (*a*) – 273°C (*b*) 273°C
    (*c*) 237°C (*d*) – 373°C.
12. Which of the following is *correct* ?
    (*a*) Absolute pressure = gauge pressure + atmospheric pressure
    (*b*) Gauge pressure = absolute pressure + atmospheric pressure
    (*c*) Atmospheric pressure = absolute pressure + gauge pressure
    (*d*) Absolute pressure = gauge pressure – atmospheric pressure.

13. The unit of energy in SI units is
(*a*) Joule (J) (*b*) Joule metre (Jm)
(*c*) Watt (W) (*d*) Joule/metre (J/m).

14. One watt is equal to
(*a*) 1 Nm/s (*b*) 1 N/min
(*c*) 10 N/s (*d*) 100 Nm/s
(*e*) 100 Nm/m.

15. One joule (J) is equal to
(*a*) 1 Nm (*b*) kNm
(*d*) 10 Nm/s (*d*) 10 kNm/s.

16. The amount of heat required to raise the temperature of 1 kg of water through 1°C is called
(*a*) specific heat at constant volume (*b*) specific heat at constant pressure
(*c*) kilo calorie (*d*) none of the above.

17. The heating and expanding of a gas is called
(*a*) thermodynamic system (*b*) thermodynamic cycle
(*c*) thermodynamic process (*d*) thermodynamic law.

18. A series of operations, which take place in a certain order and restore the initial condition is known as
(*a*) reversible cycle (*b*) irreversible cycle
(*c*) thermodynamic cycle (*d*) none of the above.

19. The condition for the reversibility of a cycle is
(*a*) the pressure and temperature of the working substance must not differ, appreciably, from those of the surroundings at any stage in the process
(*b*) all the processes, taking place in the cycle of operation, must be extremely slow
(*c*) the working parts of the engine must be friction free
(*d*) there should be no loss of energy during the cycle of operation
(*e*) all of the above (*f*) none of the above.

20. In an irreversible process, there is a
(*a*) loss of heat (*b*) no loss of heat
(*c*) gain of heat (*d*) no gain of heat.

21. The main cause of the irreversibility is
(*a*) mechanical and fluid friction (*b*) unrestricted expansion
(*c*) heat transfer with a finite temperature difference
(*d*) all of the above (*e*) none of the above.

22. According to kinetic theory of heat
(*a*) temperature should rise during boiling (*b*) temperature should fall during freezing
(*c*) at low temperature all bodies are in solid state
(*d*) at absolute zero there is absolutely no vibration of molecules
(*e*) none of the above.

23. A system comprising a single phase is called a
(*a*) closed system (*b*) open system
(*c*) isolated system (*d*) homogeneous system
(*e*) heterogeneous system.

ANSWERS

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

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THEORETICAL QUESTIONS

1. Define a thermodynamic system. Differentiate between open system, closed system and an isolated system.
2. How does a homogeneous system differ from a heterogeneous system ?
3. What do you mean by a pure substance ?
4. Explain the following terms :
   (*i*) State, (*ii*) Process, and (*iii*) Cycle.
5. Explain briefly zeroth law of thermodynamics.
6. What is a quasi-static process ?
7. What do you mean by 'reversible work' ?

UNSOLVED EXAMPLES

1. A cylindrical vessel of 60 cm diameter and 80 cm height contains 3.2 kg of a gas. The pressure measured with manometer indicates 60 cm of Hg above atmosphere when barometer reads 760 mm of Hg. Find : (*i*) The absolute pressure of the gas in the vessel in bar, and (*ii*) Specific volume and density of gas. [**Ans.** (*i*) 1.81 bar ; (*ii*) 0.09 m^3/kg ; 11.11 kg/m^3]
2. A force of 1600 N is applied uniformly on a piston of 8 cm diameter. Determine the pressure on the piston. [**Ans.** 319 kN/m^2]
3. An oil of specific gravity 0.8 is contained in a tube to a depth of 80 cm. Determine the gauge pressure at this depth in kN/m^2. [**Ans.** 6.275 kN/m^2]
4. A vacuum recorded in the condenser of a steam power plant is 745 mm of Hg. Find the absolute pressure in the condenser in Pa. The barometer reading is 760 mm of Hg. [**Ans.** 2262 Pa]
5. The pressure of gas in a pipe line is measured with a mercury manometer having one limb open to the atmosphere. If the difference in the height of mercury in the two limbs is 562 mm, calculate the gas pressure. The barometer reads 761 mm Hg, the acceleration due to gravity is 9.79 m/s^2, and the density of mercury is 13640 kg/m^3. [**Ans.** 1.77 bar]
6. A turbine is supplied with steam at a gauge pressure of 1.4 MPa. After expansion in the turbine the steam flows into a condenser which is maintained at a vacuum of 710 mm Hg. The barometric pressure is 772 mm Hg. Express the inlet and exhaust steam pressures in pascals (absolute). Take the density of mercury as 13600 kg/m^3. [**Ans.** 1.503 MPa ; 8.27 kPa]
7. Gas from a bottle of compressed helium is used to inflate an inelastic flexible balloon, originally folded completely flat to a volume of 0.5 m^3. If the barometer reads 760 mm Hg, what is the amount of work done upon the atmosphere by the balloon ? [**Ans.** 50.66 kJ]
8. A rubber balloon (flexible) is to be filled with hydrogen from a storage cylinder which contains hydrogen at 5 bar pressure until it has a volume of 1 m^3. The atmospheric pressure may be taken as 1.013×10^2 N/m^2. Determine the work done by the system comprising the hydrogen initially in the bottle. [**Ans.** 101.3 kJ]
9. When the valve of the evacuated bottle is opened, atmospheric air rushes into it. If the atmospheric pressure is 101.325 kPa, and 1.2 m^3 of air (measured at atmospheric conditions) enters into the bottle, calculate the work done by the air. [**Ans.** – 60.8 kJ]
10. A pump forces 1.2 m^3/min of water horizontally from an open well to a closed tank where the pressure is 0.9 MPa. Compute the work the pump must do upon the water in an hour just to force the water into the tank against the pressure. [**Ans.** 15972 kJ]
11. 14.5 litres of gas at 172 MN/m^2 is expanded at constant pressure until its volume becomes 130.5 litres. Determine the work done by the gas. [**Ans.** 199.5 kJ]
12. 1 kg of a fluid is compressed reversibly according to a law $pv = 0.25$ where p is in bar and v is in m^3/kg. The final volume is 1/4 of the initial volume. Calculate the work done on the fluid and sketch the process on a p-v diagram. [**Ans.** 34660 Nm]

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13. A gas having a volume of 0.05 m^3 and pressure of 6.9 bar expands reversibly in a cylinder behind a piston according to law pv = constant until the volume is 0.08 m^3. Calculate the work done by the gas. Also sketch the process on a p-v diagram. [**Ans.** 15300 Nm]

14. A certain fluid at 10 bar is contained in a cylinder behind a piston, the initial volume being 0.05 m^3. Calculate the work done by the fluid when it expands reversibly, (*a*) At constant pressure to final volume of 0.2 m^3 ; (*b*) According to linear law to final volume of 0.2 m^3 and a final pressure of 2 bar ; (*c*) According to a law pV = constant to a final volume of 0.1 m^3 ; (*d*) According to law pV^3 = constant to a final volume of 0.06 m^3. Sketch all processes on p-V diagram. [**Ans.** 150000 ; 90000 ; 34700 ; 7640 Nm]

15. A fluid undergoes the following processes :

(*i*) Heated reversibly at a constant pressure of 1.05 bar until it has a specific volume of 0.1 m^3/kg.

(*ii*) It is then compressed reversibly according to a law pv = constant to a pressure of 4.2 bar.

(*iii*) It is then allowed to expand reversibly according to a law $pv^{1.3}$ = constant.

(*iv*) Finally it is heated at constant volume back to initial conditions.

The work done in the constant pressure process is 515 Nm and the mass of fluid present is 0.2 kg. Calculate the net work done on or by the fluid in the cycle and sketch the cycle on a p-v diagram.

[**Ans.** – 422 Nm]

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2 Properties of Pure Substances

2.1. Definition of the pure substance. 2.2. Phase change of a pure substance. 2.3. p-T (pressure-temperature) diagram for a pure substance. 2.4. p-V-T (pressure-volume-temperature) surface. 2.5. Phase change terminology and definitions. 2.6. Property diagrams in common use. 2.7. Formation of steam. 2.8. Important terms relating to steam formation. 2.9. Thermodynamic properties of steam and steam tables. 2.10. External work done during evaporation. 2.11. Internal latent heat. 2.12. Internal energy of steam. 2.13. Entropy of water. 2.14. Entropy of evaporation. 2.15. Entropy of wet steam. 2.16. Entropy of superheated steam. 2.17. Enthalpy-entropy (h-s) chart or Mollier diagram. 2.18. Determination of dryness fraction of steam—Tank or bucket calorimeter—Throttling calorimeter—Separating and throttling calorimeter—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

2.1. DEFINITION OF THE PURE SUBSTANCE

A pure substance is a system which is (*i*) *homogeneous in composition*, (*ii*) *homogeneous in chemical aggregation*, and (*iii*) *invariable in chemical aggregation.*

— "*Homogeneous in composition*" means that the composition of each part of the system is the *same* as the composition of *every other part*. "*Composition* means the relative proportions of the chemical elements into which the sample can be analysed. It does not matter how these elements are combined.

For example in Fig. 2.1 system (*a*), comprising steam and water, is homogeneous in composition, since chemical analysis would reveal that hydrogen and oxygen atoms are presents in the ratio 2 : 1 whether the sample be taken from the steam or from the water. The same is true of system (*b*) containing uncombined hydrogen and oxygen gas in the atomic ratio 2 : 1 in the upper part, and water in the lower part. System (*c*) however, is not homogeneous in composition, for the hydrogen and oxygen are present in the ratio 1 : 1 in the upper part, but in the ratio 2 : 1 (as water) in the lower part.

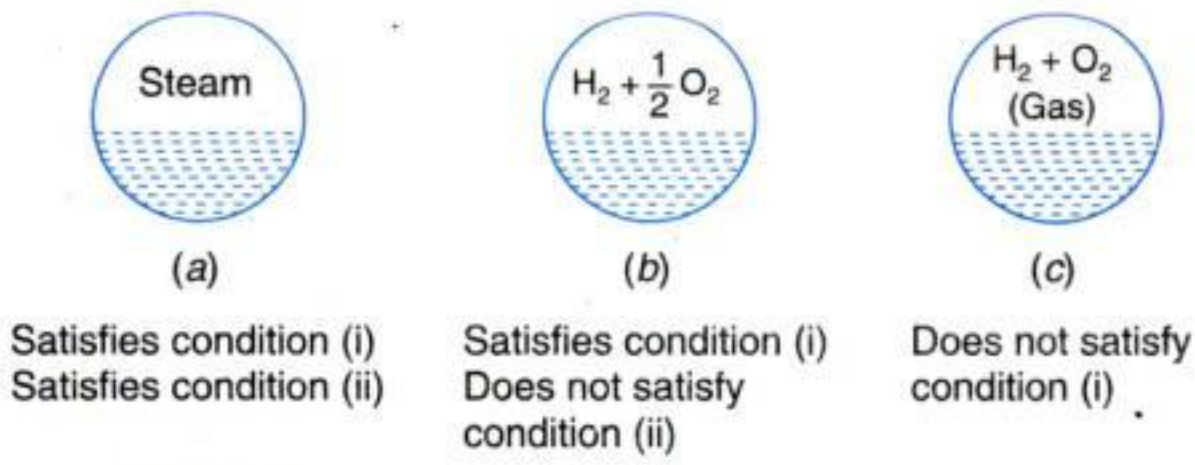

Fig. 2.1. Illustrating the definition of a pure substance.

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— "*Homogeneous in chemical aggregation*" means that the chemical elements must be combined chemically in the same way in all parts of the system. Consideration of Fig. 2.1 again shows that the system (*a*) satisfies this condition also ; for steam and water consist of identical molecules. System (*b*) on the other hand is not homogeneous in chemical aggregation since in the upper part of the system the hydrogen and oxygen are not combined chemically (individual atoms of *H* and *O* are not uniquely associated), whereas in the lower part of the system the hydrogen and oxygen are combined to form water.

Note however that a uniform mixture of steam, hydrogen gas, and oxygen gas would be regarded as homogeneous in both composition and chemical aggregation whatever the relative proportions of the components.

— "*Invariable in chemical aggregation*" means that the state of chemical combination of the system does not change with *time* (condition (*ii*) referred to variation with *position*). Thus a mixture of hydrogen and oxygen, which changed into steam during the time that the system was under consideration, would not be a pure substance.

2.2. PHASE CHANGE OF A PURE SUBSTANCE

Let us consider 1 kg of liquid water at a temperature of 20°C in a cylinder fitted with a piston, which exerts on the water a constant pressure of one atmosphere (1.0132 bar) as shown in Fig. 2.2 (*i*).

— As the water is heated slowly its temperature rises until the temperature of the liquid water becomes 100°C. During the process of heating, the *volume slightly increases* as indicated by the line 1–2 on the temperature-specific volume diagram (Fig. 2.3). The piston starts moving upwards.

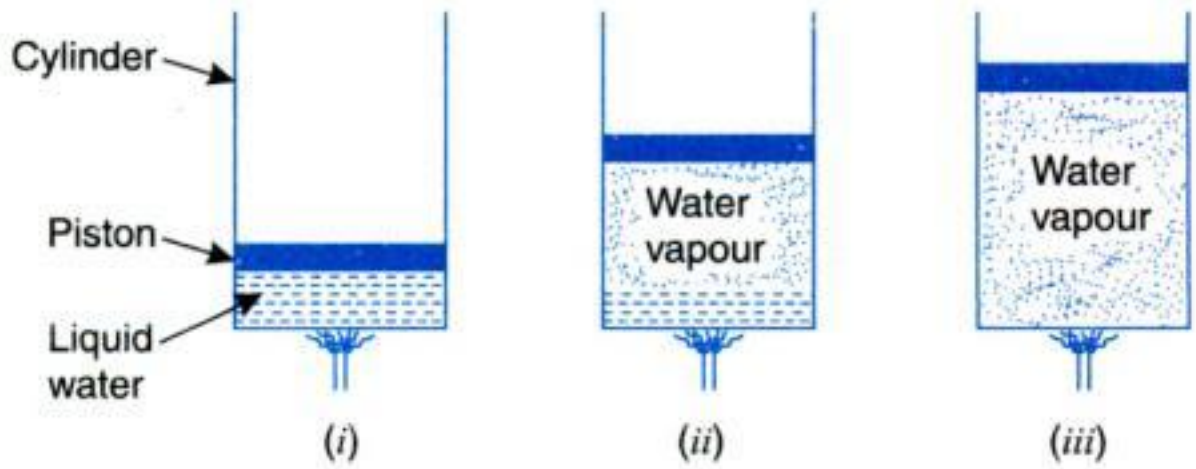

Fig. 2.2. Phase change of water at constant pressure from liquid to vapour phase.

— If the heating of the liquid, after it attains a temperature of 100°C, is continued it *undergoes a change in phase*. A portion of the liquid water changes into vapour as shown in Fig. 2.2 (*ii*). This state is described by the line 2–3 in Fig. 2.3. The amount of heat required to convert the liquid water completely into vapour under this condition is called the *heat of vapourisation*. The temperature at which vapourisation takes place at a given pressure is called the *saturation temperature* and the given pressure is called the *saturation pressure*.

During the process represented by the line 2–3 (Fig. 2.3) the volume increases rapidly and piston moves upwards Fig. 2.2 (*iii*).

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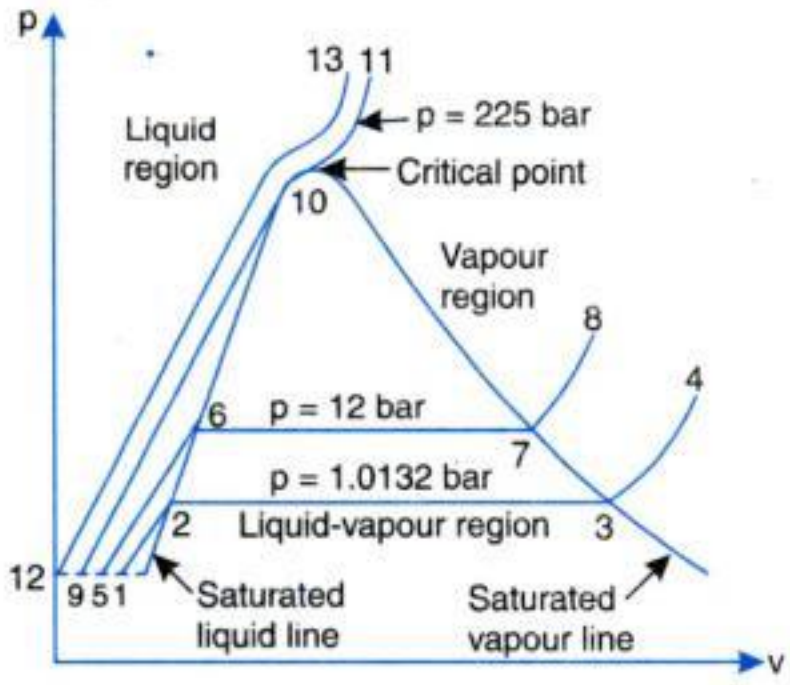

Fig. 2.3

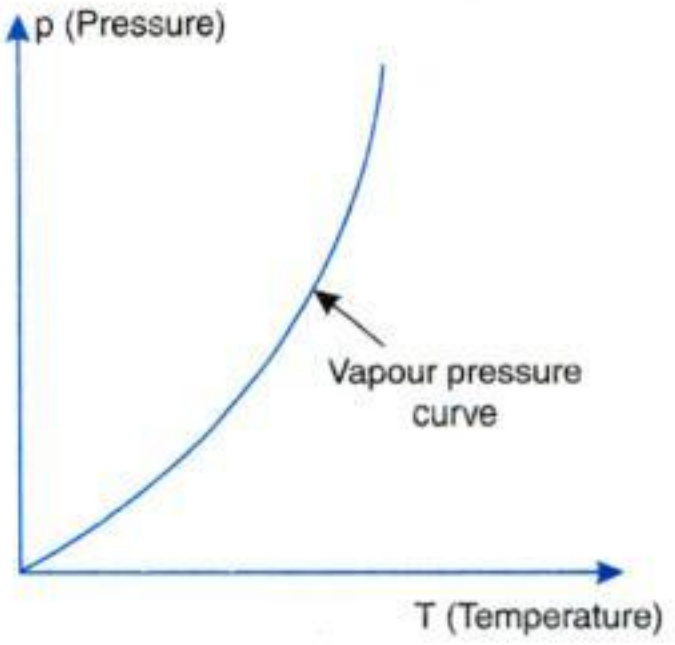

Fig. 2.4. Vapour pressure curve for water.

For a pure substance, definite relationship exists between the saturation pressure and saturation temperature as shown in Fig. 2.4, the curve so obtained is called *vapour pressure curve*.

— It may be noted that if the temperature of the liquid water on cooling becomes lower than the saturation temperature for the given pressure, the liquid water is called a *sub-cooled liquid*. The point '1' (in Fig. 2.3) illustrates this situation, when the liquid water is cooled under atmospheric pressure to a temperature of 20°C, which is below the saturation temperature (100°C).

— Further, at point '1' the temperature of liquid is 20°C and corresponding to this temperature, the saturation pressure is 0.0234 bar, which is lower than the pressure on the liquid water, which is 1 atmosphere. Thus the pressure on the liquid water is greater than the saturation pressure at a given temperature. In this condition, the liquid water is known as the *compressed liquid*.

The term *compressed liquid* or *sub-cooled liquid* is used to distinguish it from *saturated liquid. All points in the liquid region indicate the states of the compressed liquid.*

— When all the liquid has been evaporated completely and heat is further added, the *temperature of the vapour increases*. The curve 3-4 in Fig. 2.3 describes the process. When the temperature increases above the saturation temperature (in this case 100°C), the vapour is known as the *superheated vapour* and the temperature at this state is called the *superheated temperature*. There is *rapid increase in volume* and the piston moves upwards [Fig. 2.2 (*iii*)].

The difference between the superheated temperature and the saturation temperature at the given pressure is called the *degree of superheat*.

— If the above mentioned heating process is repeated at different pressures a number of curve similar to 1-2-3-4 are obtained. Thus, if the heating of the liquid water in the piston cylinder arrangement takes place under a constant pressure of 12 bar with an initial temperature of 20°C until the liquid water is converted into superheated steam, then curve 5-6-7-8 will represent the process.

— In the above heating process, it may be noted that, as the pressure increases the *length of constant temperature vapourisation gets reduced*.

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From the heating process at a constant pressure of 225 bar represented by the curve 9-10-11 in Fig. 2.3, it can be seen that there is *no constant temperature vapourisation line*. The specific volume of the saturated liquid and of the saturated vapour is the same, *i.e.*, $v_f = v_g$. Such a state of the substance is called the *critical state*. The parameters like temperature, pressure, volume, etc., at such a state are called *critical parameters*.

— The curve 12-13 (Fig. 2.3) *represents a constant pressure heating process, when the pressure is greater than the critical pressure.* At this state, *the liquid water is directly converted into superheated steam.* As there is no definite point at which the liquid water changes into superheated steam, it is generally called *liquid water when the temperature is less than the critical temperature and superheated steam when the temperature is above the critical temperature.*

2.3. p-T (Pressure-Temperature) DIAGRAM FOR A PURE SUBSTANCE

If the vapour pressure of a solid is measured at various temperatures until the *triple point* is reached and then that of the liquid is measured until the critical point is reached, the result when plotted on a *p-T* diagram appears as in Fig. 2.5.

If the substance at the triple point is compressed until there is no vapour left and the pressure on the resulting mixture of liquid and solid is increased, the temperature will have to be changed for equilibrium to exist between the solid and the liquid.

Measurements of these pressures and temperatures give rise to a third curve on the *p-T* diagram, starting at the triple point and continuing indefinitely.

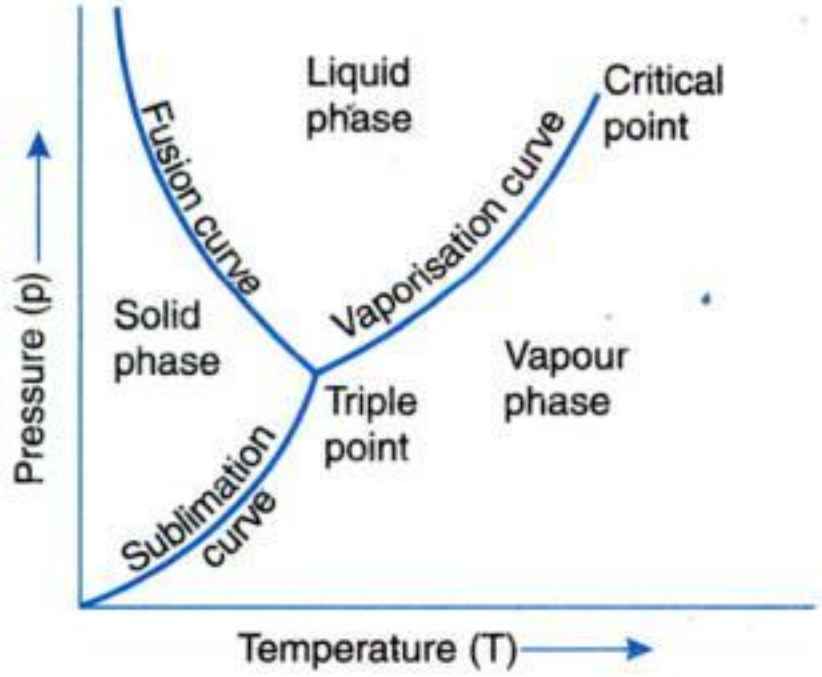

Fig. 2.5. *p*-T diagram for a substance such as water.

The points representing the coexistence of (*i*) *solid* and *vapour* lie on the '*sublimation curve*', (*ii*) *liquid* and *vapour* lie on the '*vapourisation curve*', (*iii*) *liquid* and *solid* lie on the '*fusion curve*'. In the particular case of *water*, the sublimation curve is called the *frost line*, the vapourisation curve is called the *steam line*, and the fusion curve is called the *ice line*.

The slopes of sublimation and the vapourisation curves for all substances are *positive*. The slope of the fusion curve, however may be positive or negative. The fusion curve of *most substances* have a *positive slope*. Water is one of the important exceptions.

Triple point

The triple point is merely the point of intersection of sublimation and vapourisation curves. It must be understood that only on *p-T* diagram is the triple point represented by a *point*. On *p-V* diagram it is a *line*, and on a *U-V* diagram it is a *triangle*.

— The pressure and temperature at which all three phases of a pure substance coexist may be measured with the apparatus that is used to measure vapour pressure.

— Triple-point data for some interesting substances are given in Table 2.1.

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Table 2.1. Triple-point Data

S. No.	*Substance*	*Temp., K*	*Pressure, mm Hg*
1.	Hydrogen (normal)	13.96	54.1
2.	Deuterium (normal)	18.63	128
3.	Neon	24.57	324
4.	Nitrogen	63.18	94
5.	Oxygen	54.36	1.14
6.	Ammonia	195.40	45.57
7.	Carbon dioxide	216.55	3.880
8.	Sulphur dioxide	197.68	1.256
9.	Water	273.16	4.58

2.4. p-V-T (Pressure-Volume-Temperature) SURFACE

A detailed study of the heating process reveals that the temperature of the solid rises and then during the change of phase from solid to liquid (or solid to vapour) the temperature remains constant. This phenomenon is common to all phase changes. Since the temperature is constant, pressure and temperature are not independent properties and cannot be used to specify state during a change of phase.

The combined picture of change of pressure, specific volume and temperature may be shown on a three dimensional state model. Fig. 2.6 illustrates the equilibrium states for a pure substance which expands on fusion. Water is an example of a substance that exhibits this phenomenon.

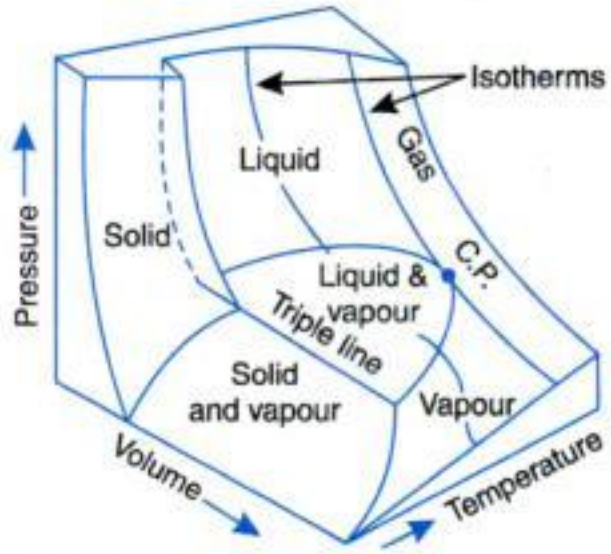

Fig. 2.6. A pressure-volume-temperature (p-V-T) surface.

All the equilibrium states lie on the surface of the model. States represented by the space above or below the surface are not possible. It may be seen that the *triple point appears as a line in this representation*. The point *C.P.* is called the *critical point* and no liquid phase exists at temperatures above the isotherms through this point. The term evaporation is meaningless in this situation.

At the critical point the temperature and pressure are called the critical temperature and the critical pressure respectively and when the temperature of a substance is above the critical value, it is called a gas. It is not possible to cause a phase change in a gas unless the temperature is lowered to a value less than the critical temperature. Oxygen and nitrogen are examples of gases that have critical temperatures below normal atmospheric temperature.

2.5. PHASE CHANGE TERMINOLOGY AND DEFINITIONS

Suffices :

Solid	i
Liquid	f
Vapour	g

Phase change	Name	Process	Process suffix
1. Solid-liquid	Fusion	Freezing, melting	*if*
2. Solid-vapour	Sublimation	Frosting, defrosting	*ig*
3. Liquid-vapour	Evaporation	Evaporating, Condensing	*fg*

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Triple point—The only state at which the solid, liquid and vapour phases *coexist in equilibrium.*

Critical point (C.P.). The *limit of distinction between a liquid and vapour.*

Critical pressure. The pressure at the critical point.

Critical temperature. The temperature at the critical point.

Gas—A vapour whose *temperature is greater than the critical temperature.*

Liquid-vapour terms : Refer Fig. 2.7.

Saturation temperature. The phase change temperature corresponding to the saturation pressure. Sometimes called the *boiling temperature.*

Saturation pressure. The phase change pressure.

Compressed liquid. Liquid whose temperature is *lower than the saturation temperature.* Sometimes called a *sub-cooled liquid.*

Saturated liquid. Liquid at the saturation temperature corresponding to the saturation pressure. That is liquid about to commence evaporating, represented by the point *f* on a diagram.

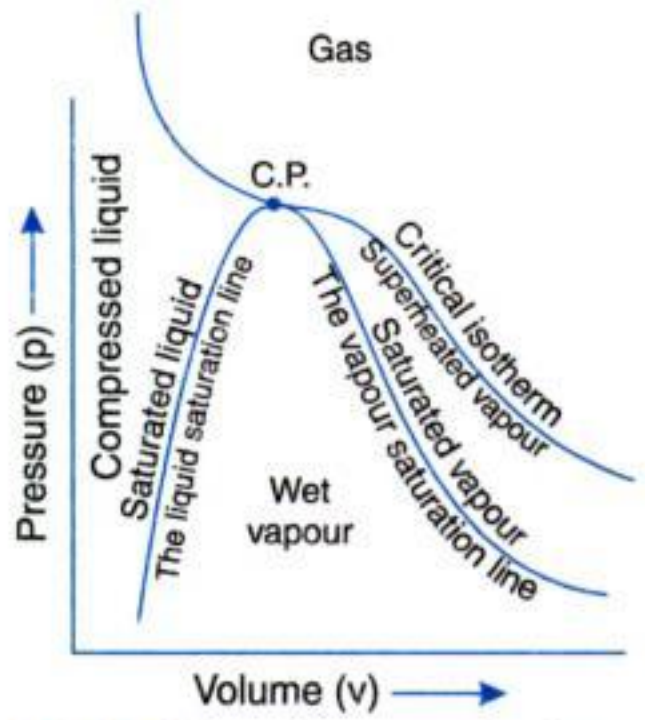

Fig. 2.7. Phase change terminology.

Saturated vapour. A term including wet and dry vapour.

Dry (saturated) vapour. Vapour which has just completed evaporation. The pressure and temperature of the vapour are the saturation values. Dry vapour is represented by a point *g* on a diagram.

Wet vapour. The mixture of saturated liquid and dry vapour during the phase change.

Superheated vapour. Vapour whose temperature is greater than the saturation temperature corresponding to the pressure of the vapour.

Degree of superheat. The term used for the numerical amount by which the temperature of a superheated vapour exceeds the saturation temperature.

2.6. PROPERTY DIAGRAMS IN COMMON USE

Besides *p-V* diagram which is useful because *pressure and volume are easily visualised* and the T-*s* chart which is used in *general thermodynamic work*, there are other charts which are of *practical use for particular applications.* The *specific enthalpy-specific entropy chart is used for steam plant work and the pressure-specific enthalpy chart is used in refrigeration work.* Sketches of these charts are shown in Fig. 2.8. These charts are drawn for H_2O (water and steam) and represent the correct shape of the curves for this substance.

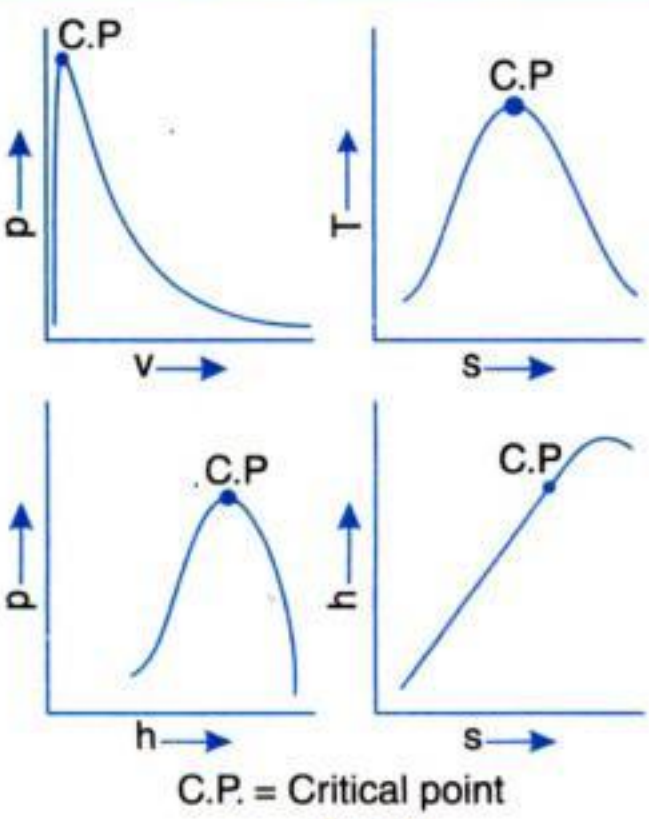

Fig. 2.8

2.7. FORMATION OF STEAM

The process of formation of steam is discussed in detail in the following few pages :

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Consider a cylinder fitted with a piston which can move freely upwards and downwards in it. Let, for the sake of simplicity, there be 1 kg of water at 0°C with volume v_f m^3 under the piston [Fig 2.9 (*i*)]. Further let the piston is loaded with load W to ensure heating at constant pressure. Now if the heat is imparted to water, a rise in temperature will be noticed and this rise will continue till boiling point is reached. The temperature at which water starts boiling depends upon the pressure and as such for *each pressure* (under which water is heated) *there is a different boiling point*. This boiling temperature is known as the temperature of formation of steam or *saturation temperature*.

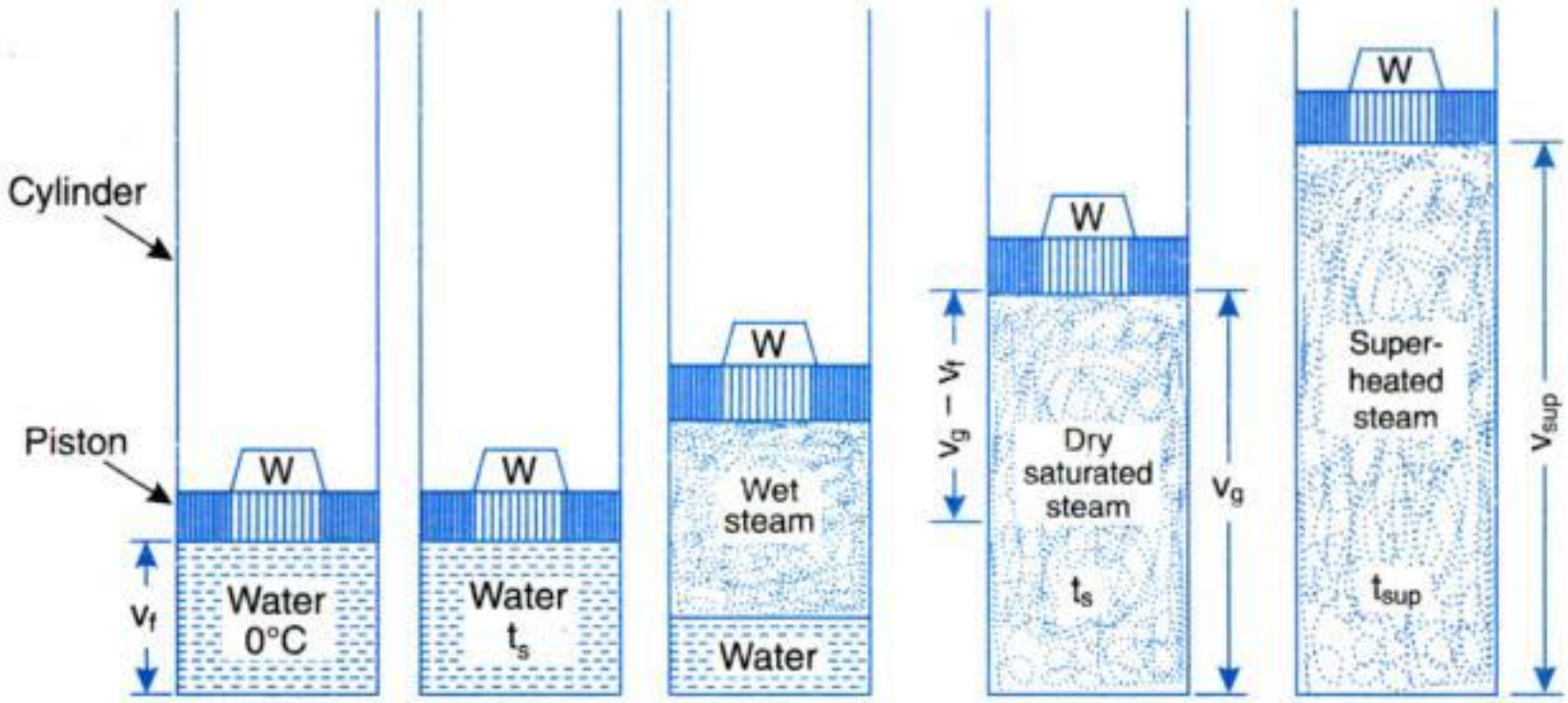

Fig. 2.9. Formation of steam.

It may be noted during heating up to boiling point that there will be slight increase in volume of water due to which piston moves up and hence work is obtained as shown in Fig. 2.9 (*ii*). This work, however, is so *small* that is can be *neglected*.

Now, if supply of heat to water is continued it will be noticed that rise of temperature after the boiling point is reached *nil* but piston starts moving upwards which indicates that there is increase is volume which is only possible if steam formation occurs. The heat being supplied does not show any rise of temperature but changes water into vapour state (steam) and is known as *latent heat* or *hidden heat*. So long as the steam is in contact with water, it is called *wet steam* [Fig. 2.9 (*iii*)] and if heating of steam is further progressed [as shown in Fig. 2.9 (*iv*)] such that all the water particles associated with steam are evaporated, the steam so obtained is called *dry and saturated steam*. If v_g m^3 is the volume of 1 kg of dry and saturated steam then work done on the piston will be

$$p(v_g - v_f) \qquad ...(2.1)$$

where p is the constant pressure (due to weight 'W' on the piston).

Again, if supply of heat to the dry and saturated steam is continued at constant pressure there will be increase in temperature and volume of steam. The steam so obtained is called *super-*

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heated steam and it *behaves like a perfect gas*. This phase of steam formation is illustrated in Fig. 2.9 (*v*).

Fig. 2.10 shows the graphical representation of formation of steam.

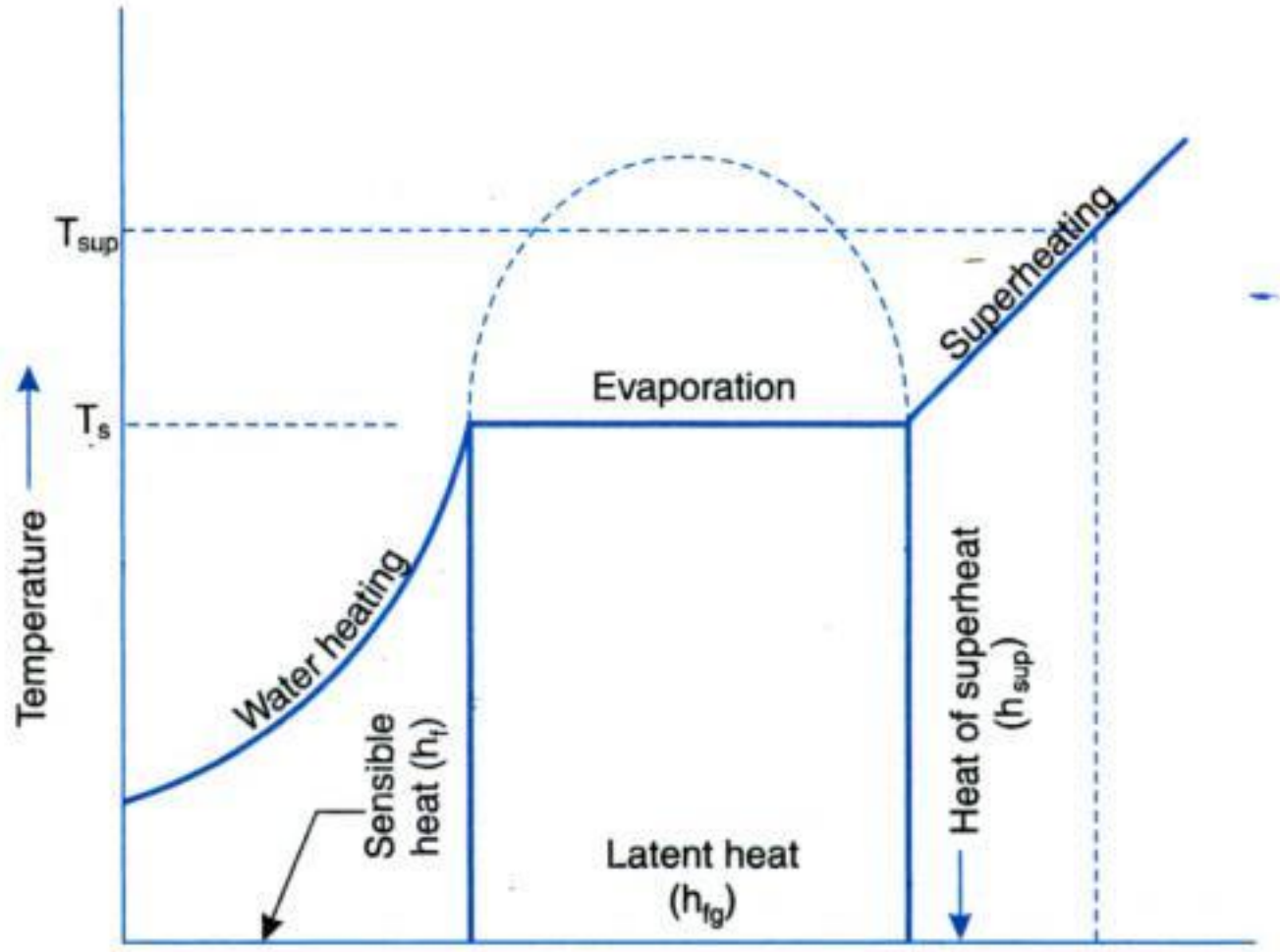

Fig. 2.10. Graphical representation of formation of steam.

2.8. IMPORTANT TERMS RELATING TO STEAM FORMATION

1. Sensible heat of water (h_f). *It is defined as the quantity of heat absorbed by 1 kg of water when it is heated from 0°C (freezing point) to boiling point.* It is also called total heat (or enthalpy) of water or liquid heat invariably. It is reckoned from 0°C where sensible heat is taken as zero. If 1 kg of water is heated from 0°C to 100°C the sensible heat added to it will be 4.18 × 100 = 418 kJ but if water is at say 20°C initially then sensible heat added will be 4.18 × (100 – 20) = 334.4 kJ. This type of heat is denoted by letter h_f and its value can be directly read from the steam tables.

Note. The value of specific heat of water may be taken as 4.18 kJ/kg K at low pressures but at high pressures it is different from this value.

2. Latent heat or hidden heat (h_{fg}). *It is the amount of heat required to convert water at a given temperature and pressure into steam at the same temperature and pressure.* It is expressed by the symbol h_{fg} and its value is available from steam tables. *The value of latent heat is not constant and varies according to pressure variation.*

3. Dryness fraction (x). The term dryness fraction is related with wet steam. *It is defined as the ratio of the mass of actual dry steam to the mass of steam containing it.* It is usually expressed by the symbol '*x*' or '*q*'.

If m_s = Mass of dry steam contained in steam considered, and

m_w = Weight of water particles in suspension in the steam considered,

Then,
$$x = \frac{m_s}{m_s + m_w} \qquad ...(2.2)$$

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Thus if in 1 kg of wet steam 0.9 kg is the dry steam and 0.1 kg water particles then $x = 0.9$.

Note. No steam can be completely dry and saturated, so long as it is in contact with the water from which it is being formed.

4. Total heat or enthalpy of wet steam (h). *It is defined as the quantity of heat required to convert 1 kg of water at 0°C into wet steam at constant pressure.* It is the sum of total heat of water and the latent heat and this sum is also called **enthalpy.**

In other words, $$h = h_f + xh_{fg} \quad ...(2.3)$$

If steam is dry and saturated, then $x = 1$ and $h_g = h_f + h_{fg}$.

5. Superheated steam. When steam is heated after it has become dry and saturated, it is called superheated steam and the process of heating is called *superheating. Superheating is always carried out at constant pressure.* The additional amount of heat supplied to the steam during superheating is called as *'Heat of superheat'* and can be calculated by using the specific heat of superheated steam at constant pressure (c_{ps}), the value of which varies from 2.0 to 2.1 kJ/kg K depending upon pressure and temperature.

If T_{sup}., T_s are the temperatures of superheated steam in K and wet or dry steam, then $(T_{sup} - T_s)$ is called *'degree of superheat'*.

The total heat of superheated steam is given by

$$h_{sup} = h_f + h_{fg} + c_{ps}\,(T_{sup} - T_s) \quad ...(2.4)$$

Superheated steam behaves like a gas and therefore it follows the gas laws. The value of n for this type of steam is 1.3 and the law for the adiabatic expansion is $pv^{1.3}$ = constant.

The ***advantages*** obtained by using *'superheated'* steam are as follows :

(*i*) By superheating steam, its heat content and hence its capacity to do work is increased without having to increase its pressure.

(*ii*) Superheating is done in a superheater which obtains its heat from waste furnace gases which would have otherwise passed uselessly up the chimney.

(*iii*) High temperature of superheated steam results in an increase in thermal efficiency.

(*iv*) Since the superheated steam is at a temperature above that corresponding to its pressure, it can be considerably cooled during expa]nsion in an engine before its temperature falls below that at which it will condense and thereby become wet. Hence, heat losses due to condensation of steam on cylinder walls etc., are avoided to a great extent.

6. Volume of wet and dry steam. If the steam has dryness fraction of x, then 1 kg of this steam will contain x kg of dry steam and $(1 - x)$ kg of water. If v_f is the volume of 1 kg of water and v_g is the volume of 1 kg of perfect dry steam (also known as specific volume), then volume of 1 kg of wet steam = volume of dry steam + volume of water.

$$= xv_g + (1 - x)v_f \quad ...(2.5)$$

Note. The volume of v_f at low pressures is very small and is generally neglected. Thus is general, the volume of 1 kg of wet steam is given by, xv_g and density $\frac{1}{xv_g}$ kg/m^3.

$$= xv_g + v_f - xv_f$$
$$= v_f + x(v_g - v_f)$$
$$= v_f + xv_{fg} \quad ...[2.5\ (a)]$$
$$= v_f + xv_{fg} + v_{fg} - v_{fg}$$
$$= (v_f + v_{fg}) - (1 - x)\,v_{fg}$$
$$= v_g - (1 - x)v_{fg} \quad ...[2.5\ (b)]$$

7. Volume of superheated steam. As superheated steam behaves like a perfect gas its volume can be found out in the same way as the gases.

If v_g = Specific volume of dry steam at pressure p,

T_s = Saturation temperature in K,
T_{sup} = Temperature of superheated steam in K, and
v_{sup} = Volume of 1 kg of superheated steam at pressure p,

Then $$\frac{p \,.\, v_g}{T_s} = \frac{p \,.\, v_{sup}}{T_{sup}}$$

or $$v_{sup} = \frac{v_g \, T_{sup}}{T_s} \qquad ...(2.6)$$

2.9. THERMODYNAMIC PROPERTIES OF STEAM AND STEAM TABLES

In engineering problem, for any fluid which is used as working fluid, the six basic thermodynamic properties required are : p (pressure), T (temperature), v (volume), u (internal energy), h (enthalpy) and s (entropy). These properties must be known at different pressure for analysing the thermodynamic cycles used for work producing devices. The values of these properties are determined theoretically or experimentally and are tabulated in the form of tables which are known as '*Steam Tables*'. The properties of wet steam are then computed from such tabulated data. Tabulated values are also available for superheated steam. It may be noted that *steam has only one saturation temperature at each pressure.*

Following are the thermodynamic properties of steam which are tabulated in the form of table :

p = Absolute pressure (bar or kPa) ;
t_s = Saturation temperature (°C) ;
h_f = Enthalpy of saturated liquid (kJ/kg) ;
h_{fg} = Enthalpy or latent heat of vapourisation (kJ/kg) ;
h_g = Enthalpy of saturated vapour (steam) (kJ/kg) ;
s_f = Entropy of saturated liquid (kJ/kg K) ;
s_{fg} = Entropy of vapourisation (kJ/kg K) ;
s_g = Entropy of saturated vapour (steam) (kJ/kg K) ;
v_f = Specific volume of saturated liquid (m^3/kg) ;
v_g = Specific volume of saturated vapour (steam) (m^3/kg).

Also, $h_{fg} = h_g - h_f$ ..... Change of enthalpy during evaporation
$s_{fg} = s_g - s_f$ ..... Change of entropy during evaporation
$v_{fg} = v_g - v_f$ ..... Change of volume during evaporation.

The above mentioned properties at different pressures are tabulated in the form of tables as under :

The internal energy of steam ($u = h - pv$) is also tabulated in some steam tables.

STEAM TABLES

Absolute pressure	*Temperature*	*Specific enthalpy kJ/kg*			*Specific entropy kJ/kg K*			*Specific volume m^3/kg*	
bar, p	t_s	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	v_f	v_g
1.0	99.6	417.5	2257.9	2675.4	1.3027	6.0571	7.3598	0.001043	1.6934
50.0	263.9	1154.9	1639.7	2794.2	2.9206	3.0529	5.9735	0.001286	0.00394
100.0	311.1	1408.0	1319.7	2727.7	3.3605	2.2593	5.6198	0.001452	0.01811

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2.10. EXTERNAL WORK DONE DURING EVAPORATION

When water is evaporated to form *saturated* steam, its volume increases from v_f to v_g at a constant pressure, and thus external work is done by steam due to increase in volume. The energy for doing the work is obtained during the absorption of latent heat. This work is called **external work of evaporation** and is given by $p(v_g - v_f)$.

i.e., External work of evaporation = $p(v_g - v_f)$ ...(2.7)

As at low pressure v_f is very small and hence neglected, work of evaporation is

$$p \cdot v_g \qquad ...(2.8)$$

In case of *wet* steam with dryness fraction x, work of evaporation will be

$$pxv_g \qquad ...(2.9)$$

2.11. INTERNAL LATENT HEAT

The latent heat consists of true latent heat and the work of evaporation. This true latent heat is called the *internal latent heat* and may also be found as follows :

$$\text{Internal latent heat} = h_{fg} - \frac{pv_g}{J} \qquad ...(2.10)$$

J = 1 in SI units.

2.12. INTERNAL ENERGY OF STEAM

It is defined as the actual energy stored in the steam. As per previous articles, the total heat of steam is sum of sensible heat, internal latent heat and the external work of evaporation. Work of evaporation is not stored in the steam as it is utilised in doing external work. Hence the internal energy of steam could be found by subtracting work of evaporation from the total heat.

In other words,

$$h = \frac{pv_g}{J} + u, \text{ where } u \text{ is internal energy of 1 kg of steam at pressure } p$$

or $$u = h - \frac{pv_g}{J}$$

In case of wet steam with dryness fraction 'x'

$$u = h - \frac{pxv_g}{J} \qquad ...(2.11)$$

and if steam is superheated to a volume of v_{sup} per kg.

$$h_{sup} = h_f + h_{fg} + c_{ps}\,(T_{sup} - T_s)$$

and $$u = h_{sup} - \frac{p \cdot v_{sup}}{J} \qquad ...(2.12)$$

2.13. ENTROPY OF WATER

(**Note.** For definition of entropy phase refer Article 4.12).

Consider 1 kg of water being heated from temperature T_1 to T_2 at constant pressure. The change in entropy will be given by,

$$ds = \frac{dQ}{T} = c_{pw} \cdot \frac{dT}{T}$$

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Integrating both sides, we get

$$\int_{s_1}^{s_2} ds = \int_{T_1}^{T_2} c_{pw} \frac{dT}{T}$$

$$s_2 - s_1 = c_{pw} \log_e \frac{T_2}{T_1} \quad ...(2.13)$$

If 0°C is taken as datum, then entropy of water per kg at any temperature T above this datum will be

$$s_f = c_{pw} \log_e \frac{T}{273} \quad ...(2.14)$$

2.14. ENTROPY OF EVAPORATION

The change of entropy (ds) is given by,

$$ds = \frac{dQ}{T}$$

or $$s_2 - s_1 = \frac{Q}{T}, \quad \text{where } Q \text{ is the heat absorbed.}$$

When water is evaporated to steam completely the heat absorbed is the latent heat and this heat goes into water without showing any rise of temperature.

Then $$Q = h_{fg}$$

and $$s_{evap.} = \frac{h_{fg}}{T_s} \quad ...(2.15)$$

However, in case of wet steam with dryness fraction x the evaporation will be partial and heat absorbed will be xh_{fg} per kg of steam. The change of entropy will be $\frac{xh_{fg}}{T_s}$.

2.15. ENTROPY OF WET STEAM

The total entropy of wet steam is the sum of entropy of water (s_f) and entropy of evaporation (s_{fg}).

In other words, $$s_{wet} = s_f + \frac{xh_{fg}}{T_s} \quad ...(2.16)$$

where s_{wet} = Total entropy of wet steam,

s_f = Entropy of water, and

$\frac{xh_{fg}}{T_s}$ = Entropy of evaporation.

If steam is dry and saturated, *i.e.*, $x = 1$, then

$$s_g = s_f + \frac{h_{fg}}{T_s} \quad ...(2.17)$$

2.16. ENTROPY OF SUPERHEATED STEAM

Let 1 kg of dry saturated steam at T_s (saturation temperature of steam) be heated to T_{sup}. If specific heat at constant pressure is c_{ps}, then change of entropy during superheating at constant pressure p

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$$= c_{ps} \log_e \left(\frac{T_{sup}}{T_s} \right).$$

Total entropy of superheated steam above the freezing point of water.

s_{sup} = Entropy of dry saturated steam + change of entropy during superheating

$$= s_f + \frac{h_{fg}}{T_s} + c_{ps} \log_e \left(\frac{T_{sup}}{T_s} \right) = s_g + c_{ps} \log_e \left(\frac{T_{sup}}{T_s} \right) \quad ...(2.18)$$

2.17. ENTHALPY-ENTROPY (h-s) CHART OR MOLLIER DIAGRAM

Dr. Mollier, in 1904, conceived the idea of plotting total heat against entropy, and his diagram is more widely used than any other entropy diagram, since the work done on vapour cycles can be scaled from this diagram directly *as a length ; whereas on T-s diagram it is represented by an area*.

A sketch of the *h-s* chart is shown in Fig. 2.11.

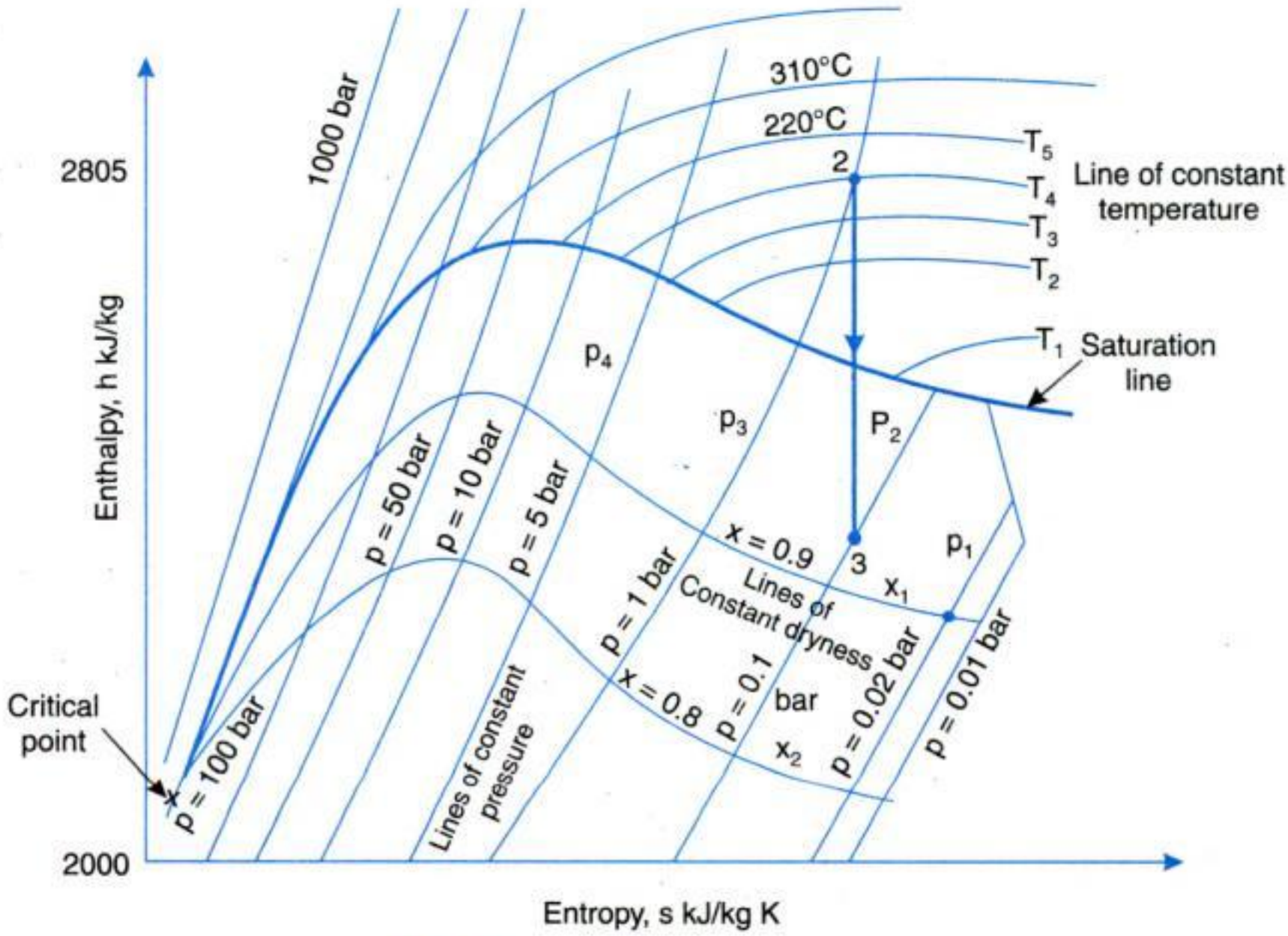

Fig. 2.11. Enthalpy-entropy (*h-s*) chart.

— Lines of constant pressure are indicated by p_1, p_2 etc., lines of constant temperature by T_1, T_2, etc.

— Any two independent properties which appear on the chart are sufficient to define the state (*e.g.*, p_1 and x_1 define state 1 and h can be read off the vertical axis).

— In the superheat region, pressure and temperature can define the state (*e.g.*, p_3 and T_4 define the state 2, and h_2 can be read off).

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— A line of constant entropy between two state points 2 and 3 defines the properties at all points during an *isentropic process* between the two states.

Example 2.1. *Calculate the dryness fraction (quality) of steam which has 1.5 kg of water in suspension with 50 kg of steam.*

Solution. Mass of dry steam, $m_s = 50$ kg

Mass of water in suspension, $m_w = 1.5$ kg

$$\therefore \textbf{Dryness fraction, } x = \frac{\text{Mass of dry steam}}{\text{Mass of dry steam + mass of water in suspension}}$$

$$= \frac{m_s}{m_s + m_w} = \frac{50}{50 + 1.5} = \textbf{0.971.} \quad \textbf{(Ans.)}$$

☞ **Example 2.2.** *A vessel having a volume of 0.6 m^3 contains 3.0 kg of liquid water and water vapour mixture in equilibrium at a pressure of 0.5 MPa. Calculate :*

(i) Mass and volume of liquid ;

(ii) Mass and volume of vapour.

Solution. Volume of the vessel, $V = 0.6$ m^3

Mass of liquid water and water vapour, $m = 3.0$ kg

Pressure, $p = 0.5$ MPa $= 5$ bar

Thus, specific volume, $v = \frac{V}{m} = \frac{0.6}{3.0} = 0.2$ m^3/kg

At 5 bar : From steam tables,

$$v_{fg} = v_g - v_f = 0.375 - 0.00109 = 0.3739 \text{ m}^3/\text{kg}$$

We know that, $v = v_g - (1 - x)\, v_{fg}$, where x = quality of the vapour.

$$0.2 = 0.375 - (1 - x) \times 0.3739$$

$$\therefore \quad (1 - x) = \frac{(0.375 - 0.2)}{0.3739} = 0.468$$

or $$x = 0.532$$

(i) **Mass and volume of liquid, $\mathbf{m_{liq.}}$ = ? $\mathbf{V_{liq.}}$ = ?**

$$\mathbf{m_{liq.}} = m(1 - x) = 3.0 \times 0.468 = \textbf{1.404 kg.} \quad \textbf{(Ans.)}$$

$$\mathbf{V_{liq.}} = m_{liq.}\, v_f = 1.404 \times 0.00109 = \textbf{0.0015 m}^3\textbf{.} \quad \textbf{(Ans.)}$$

(ii) **Mass and volume of vapour, $\mathbf{m_{vap.}}$ = ? $\mathbf{V_{vap.}}$ = ?**

$$\mathbf{m_{vap.}} = m.x = 3.0 \times 0.532 = \textbf{1.596 kg.} \quad \textbf{(Ans.)}$$

$$\mathbf{V_{vap.}} = m_{vap.}\, v_g = 1.596 \times 0.375 = \textbf{0.5985 m}^3\textbf{.} \quad \textbf{(Ans.)}$$

☞ **Example 2.3.** *A vessel having a capacity of 0.05 m^3 contains a mixture of saturated water and saturated steam at a temperature of 245°C. The mass of the liquid present is 10 kg. Find the following :*

(i) The pressure, *(ii) The mass,*

(iii) The specific volume, *(iv) The specific enthalpy,*

(v) The specific entropy, and *(vi) The specific internal energy.*

Solution. From steam tables, corresponding to 245°C :

$$p_{sat} = 36.5 \text{ bar}, \; v_f = 0.001239 \text{ m}^3/\text{kg}, \; v_g = 0.0546 \text{ m}^3/\text{kg}$$

$$h_f = 1061.4 \text{ kJ/kg}, \; h_{fg} = 1740.2 \text{ kJ/kg}, \; s_f = 2.7474 \text{ kJ/kg K}$$

$$s_{fg} = 3.3585 \text{ kJ/kg K.}$$

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(*i*) **The pressure = 36.5 bar** (or 3.65 MPa). **(Ans.)**

(*ii*) **The mass, m :**

Volume of liquid, $V_f = m_f v_f$

$= 10 \times 0.001239 = 0.01239 \text{ m}^3$

Volume of vapour, $V_g = 0.05 - 0.01239 = 0.03761 \text{ m}^3$

$\therefore$ Mass of vapour, $m_g = \dfrac{V_g}{v_g} = \dfrac{0.03761}{0.0546} = 0.688 \text{ kg}$

$\therefore$ The total mass of mixture,

$$\mathbf{m} = m_f + m_g = 10 + 0.688 = \mathbf{10.688 \text{ kg.}} \quad \textbf{(Ans.)}$$

(*iii*) **The specific volume, v :**

Quality of the mixture,

$$x = \frac{m_g}{m_g + m_f} = \frac{0.688}{0.688 + 10} = 0.064$$

$\therefore$ $\mathbf{v} = v_f + x v_{fg}$

$= 0.001239 + 0.064 \times (0.0546 - 0.001239)$. $(\because \quad v_{fg} = v_g - v_f)$

$= \mathbf{0.004654 \text{ m}^3/\text{kg.}}$ **(Ans.)**

(*iv*) **The specific enthalpy, h :**

$\mathbf{h} = h_f + x h_{fg}$

$= 1061.4 + 0.064 \times 1740.2 = \mathbf{1172.77 \text{ kJ/kg.}}$ **(Ans.)**

(*v*) **The specific entropy, s :**

$\mathbf{s} = s_f + x s_{fg}$

$= 2.7474 + 0.064 \times 3.3585 = \mathbf{2.9623 \text{ kJ/kg K.}}$ **(Ans.)**

(*vi*) **The specific internal energy, u :**

$\mathbf{u} = h - pv$

$$= 1172.77 - \frac{36.5 \times 10^5 \times 0.004654}{1000} = \mathbf{1155.78 \text{ kJ/kg.}} \quad \textbf{(Ans.)}$$

Example 2.4. *Determine the amount of heat, which should be supplied to 2 kg of water at 25°C to convert it into steam at 5 bar and 0.9 dry.*

Solution. Mass of water to be converted to steam, $m_w = 2$ kg

Temperature of water, $t_w = 25°\text{C}$

Pressure and dryness fraction of steam = 5 bar, 0.9 dry

At 5 bar : From steam tables,

$h_f = 640.1$ kJ/kg ; $h_{fg} = 2107.4$ kJ/kg

Enthalpy of 1 kg of steam (above 0°C)

$h = h_f + x h_{fg}$

$= 640.1 + 0.9 \times 2107.4 = 2536.76$ kJ/kg

Sensible heat associated with 1 kg of water

$= m_w \times c_{pw} \times (t_w - 0)$

$= 1 \times 4.18 \times (25 - 0) = 104.5$ kJ

Net quantity of heat to be supplied per kg of water

$= 2536.76 - 104.5 = 2432.26$ kJ

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Total amount of heat to be supplied

$= 2 \times 2432.26 =$ **4864.52 kJ.** **(Ans.)**

Example 2.5. *What amount of heat would be required to produce 4.4 kg of steam at a pressure of 6 bar and temperature of 250°C from water at 30°C ? Take specific heat for superheated steam as 2.2 kJ/kg K.*

Solution. Mass of steam to be produced, $m = 4.4$ kg

Pressure of steam, $p = 6$ bar

Temperature of steam, $t_{sup} = 250°C$

Temperature of water $= 30°C$

Specific heat of steam, $c_{ps} = 2.2$ kJ/kg

At 6 bar, 250°C : From steam tables,

$t_s = 158.8°C$, $h_f = 670.4$ kJ/kg, $h_{fg} = 2085$ kJ/kg

Enthalpy of 1 kg superheated steam reckoned from 0°C,

$$h_{sup} = h_f + h_{fg} + c_{ps} (T_{sup} - T_s)$$
$$= 670.4 + 2085 + 2.2(250 - 158.8)$$
$$= 2956 \text{ kJ}$$

Amount of heat already with 1 kg of water

$= 1 \times 4.18 \times (30 - 0) = 125.4$ kJ

Net amount of heat required to be supplied per kg

$= 2956 - 125.4 = 2830.6$ kJ

Total amount of heat required

$= 4.4 \times 2830.6 =$ **12454.6 kJ.** **(Ans.)**

☞ **Example 2.6.** *Determine the mass of 0.15 m^3 of wet steam at a pressure of 4 bar and dryness fraction 0.8. Also calculate the heat of 1 m^3 of steam.*

Solution. Volume of wet steam, $v = 0.15$ m^3

Pressure of wet steam, $p = 4$ bar

Dryness fraction, $x = 0.8$

At 4 bar. From steam tables,

$v_g = 0.462$ m^3/kg, $h_f = 604.7$ kJ/kg, $h_{fg} = 2133$ kJ/kg

$$\therefore \quad \text{Density} = \frac{1}{x v_g} = \frac{1}{0.8 \times 0.462} = 2.7056 \text{ kg/m}^3$$

Mass of 0.15 m^3 of steam

$= 0.15 \times 2.7056 =$ **0.4058 kg.** **(Ans.)**

Total heat of 1 m^3 of steam which has a mass of 2.7056 kg

$= 2.7056\ h$ (where h is the total heat of 1 kg of steam)

$= 2.7056\ (h_f + x h_{fg})$

$= 2.7056(604.7 + 0.8 \times 2133)$

$=$ **6252.9 kJ.** **(Ans.)**

Example 2.7. *1000 kg of steam at a pressure of 16 bar and 0.9 dry is generated by a boiler per hour. The steam passes through a superheater via boiler stop valve where its temperature is raised to 380°C. If the temperature of feed water is 30°C, determine :*

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(i) The total heat supplied to feed water per hour to produce wet steam.

(ii) The total heat absorbed per hour in the superheater.

Take specific heat for superheated steam as 2.2 kJ/kg K.

Solution. Mass of steam generated, m = 1000 kg/h

Pressure of steam, p = 16 bar

Dryness fraction, x = 0.9

Temperature of superheated steam,

$$T_{sup} = 380 + 273 = 653 \text{ K}$$

Temperature of feed water = 30°C

Specific heat of superheated steam, c_{ps} = 2.2 kJ/kg K.

At 16 bar. From steam tables,

$$t_s = 201.4°\text{C} \ (T_s = 201.4 + 273 = 474.4 \text{ K}) ;$$

$$h_f = 858.6 \text{ kJ/kg} ; \ h_{fg} = 1933.2 \text{ kJ/kg}$$

(i) Heat supplied to feed water per hour to produce wet steam is given by :

$$H = m \ [(h_f + xh_{fg}) - 1 \times 4.18 \times (30 - 0)]$$

$$= 1000 \ [(858.6 + 0.9 \times 1933.2) - 4.18 \times 30]$$

$$= 1000(858.6 + 1739.88 - 125.4)$$

$$= \mathbf{2473.08 \times 10^3 \text{ kJ.}} \quad \textbf{(Ans.)}$$

(ii) Heat absorbed by superheater per hour

$$= m[(1 - x) \ h_{fg} + c_{ps} \ (T_{sup} - T_s)]$$

$$= 1000[(1 - 0.9) \times 1933.2 + 2.2 \ (653 - 474.4)]$$

$$= 1000(193.32 + 392.92)$$

$$= \mathbf{586.24 \times 10^3 \text{ kJ.}} \quad \textbf{(Ans.)}$$

Example 2.8. *Using steam tables, determine the mean specific heat for superheated steam :*

(i) at 0.75 bar, between 100°C and 150°C ;

(ii) at 0.5 bar, between 300°C and 400°C.

Solution. *(i)* **At 0.75 bar.** From steam tables ;

At 100°C, h_{sup} = 2679.4 kJ/kg

At 150°C, h_{sup} = 2778.2 kJ/kg

$\therefore$ $2778.2 = 2679.4 + c_{ps} \ (150 - 100)$

i.e., $$c_{ps} = \frac{2778.2 - 2679.4}{50} = \mathbf{1.976.} \quad \textbf{(Ans.)}$$

(ii) **At 0.5 bar.** From steam tables ;

At 300°C, h_{sup} = 3075.5 kJ/kg

At 400°C, h_{sup} = 3278.9 kJ/kg

$\therefore$ $3278.9 = 3075.5 + c_{ps} \ (400 - 300)$

i.e., $$c_{ps} = \frac{3278.9 - 3075.5}{100} = \mathbf{2.034.} \quad \textbf{(Ans.)}$$

☞ **Example 2.9.** *A pressure cooker contains 1.5 kg of saturated steam at 5 bar. Find the quantity of heat which must be rejected so as to reduce the quality to 60% dry. Determine the pressure and temperature of the steam at the new state.*

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Solution. Mass of steam in the cooker = 1.5 kg

Pressure of steam, $p = 5$ bar

Initial dryness fraction of steam, $x_1 = 1$

Final dryness fraction of steam, $x_2 = 0.6$

Heat to be rejected :

Pressure and temperature of the steam at the new state :

At 5 bar. From steam tables,

$t_s = 151.8°C$; $h_f = 640.1$ kJ/kg ;

$h_{fg} = 2107.4$ kJ/kg ; $v_g = 0.375$ m³/kg

Thus, the volume of pressure cooker

$$= 1.5 \times 0.375 = 0.5625 \text{ m}^3$$

Internal energy of steam per kg at initial point 1,

$$u_1 = h_1 - p_1 v_1$$

$$= (h_f + h_{fg}) - p_1 v_{g_1} \qquad (\because \ v_1 = v_{g1})$$

$$= (640.1 + 2107.4) - 5 \times 10^5 \times 0.375 \times 10^{-3}$$

$$= 2747.5 - 187.5 = 2560 \text{ kJ/kg}$$

Also, $V_1 = V_2$ (V_2 = volume at final condition)

i.e., $0.5625 = 1.5[(1 - x_2)\, v_{f2} + x_2 v_{g2}]$

$$= 1.5\, x_2 v_{g2} \qquad (\because \ v_{f2} \text{ is negligible})$$

$$= 1.5 \times 0.6 \times v_{g2}$$

$$\therefore \quad v_{g2} = \frac{0.5625}{1.5 \times 0.6} = 0.625 \text{ m}^3/\text{kg}.$$

From steam tables corresponding to 0.625 m³/kg,

$p_2 \simeq$ **2.9 bar**, $t_s =$ **132.4°C**, $h_f = 556.5$ kJ/kg, $h_{fg} = 2166.6$ kJ/kg

Internal energy of steam per kg, at final point 2,

$$u_2 = h_2 - p_2 v_2$$

$$= (h_{f_2} + x_2 h_{fg_2}) - p_2 x v_{g2} \qquad (\because \ v_2 = x v_{g2})$$

$$= (556.5 + 0.6 \times 2166.6) - 2.9 \times 10^5 \times 0.6 \times 0.625 \times 10^{-3}$$

$$= 1856.46 - 108.75 = 1747.71 \text{ kJ/kg}.$$

Heat transferred at constant volume per kg

$$= u_2 - u_1 = 1747.71 - 2560 = -812.29 \text{ kJ/kg}$$

Thus, **total heat transferred**

$$= -812.29 \times 1.5 = -\mathbf{1218.43 \text{ kJ}}. \quad \textbf{(Ans.)}$$

Negative sign indicates that heat has been **rejected.**

☞ **Example 2.10.** *A spherical vessel of 0.9 m³ capacity contains steam at 8 bar and 0.9 dryness fraction. Steam is blown off until the pressure drops to 4 bar. The valve is then closed and the steam is allowed to cool until the pressure falls to 3 bar. Assuming that the enthalpy of steam in the vessel remains constant during blowing off periods, determine :*

(i) The mass of steam blown off ;

(ii) The dryness fraction of steam in the vessel after cooling ;

(iii) The heat lost by steam per kg during cooling.

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Solution. Capacity of the spherical vessel, $V = 0.9\ m^3$

Pressure of the steam, $p_1 = 8$ bar

Dryness fraction of steam, $x_1 = 0.9$

Pressure of steam after blow off, $p_2 = 4$ bar

Final pressure of steam, $p_3 = 3$ bar.

(*i*) **The mass of steam blown off :**

The mass of steam in the vessel

$$m_1 = \frac{V}{x_1 v_{g_1}} = \frac{0.9}{0.9 \times 0.24} = 4.167 \text{ kg} \qquad (\because \text{ At 8 bar} : v_g = 0.24\ m^3/kg)$$

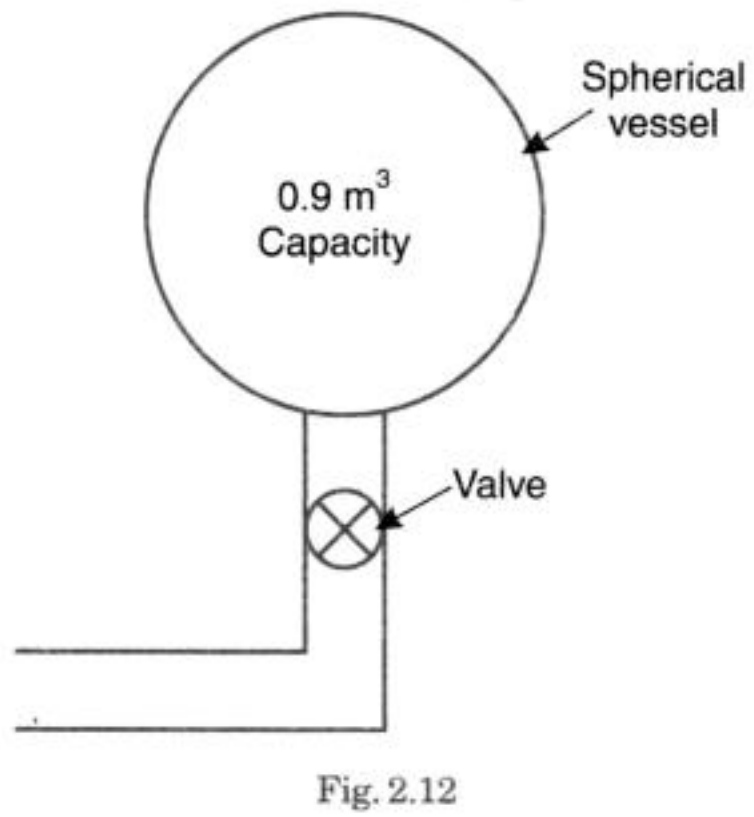

Fig. 2.12

The enthalpy of steam before blowing off (per kg)

$$= h_{f_1} + x_1 h_{fg_1} = 720.9 + 0.9 \times 2046.5 \text{ ..... at pressure } 8 \text{ bar}$$

$$= 2562.75 \text{ kJ/kg}$$

Enthalpy before blowing off = Enthalpy after blowing off

$$\therefore \quad 2562.75 = (h_{f_2} + x_2 h_{fg_2}) \text{ at pressure 4 bar}$$

$$= 604.7 + x_2 \times 2133 \text{ ..... at pressure } 4 \text{ bar}$$

$$\therefore \quad x_2 = \frac{2562.75 - 604.7}{2133} = 0.918$$

Now the mass of steam in the vessel after blowing off,

$$m_2 = \frac{0.9}{0.918 \times 0.462} = 2.122 \text{ kg} \qquad [v_{g_2} = 0.462\ m^3/kg \text{ ..... at 4 bar}]$$

Mass of steam blown off, $m = m_1 - m_2 = 4.167 - 2.122$

$$= \mathbf{2.045\ kg.} \quad \textbf{(Ans.)}$$

(*ii*) **Dryness fraction of steam in the vessel after cooling,** x_3 :

As it is constant volume cooling

$$\therefore \quad x_2 v_{g_2} \text{ (at 4 bar)} = x_3 v_{g_3} \text{ (at 3 bar)}$$

$$0.918 \times 0.462 = x_3 \times 0.606$$

$$\therefore \quad x_3 = \frac{0.918 \times 0.462}{0.606} = \mathbf{0.699.} \ \textbf{(Ans.)}$$

(*iii*) **Heat lost during cooling :**

Heat lost during cooling = $m(u_3 - u_2)$, where u_2 and u_3 are the internal energies of steam before starting cooling or after blowing and at the end of the cooling.

$$\therefore \quad u_2 = h_2 - p_2 x_2 v_{g_2} = (h_{f_2} + x_2 h_{fg2}) - p_2 x_2 v_{g2}$$

$$= (604.7 + 0.918 \times 2133) - 4 \times 10^5 \times 0.918 \times 0.462 \times 10^{-3}$$

$$= 2562.79 - 169.65 = 2393.14 \text{ kJ/kg}$$

$$u_3 = h_3 - p_3 x_3 v_{g_3} = (h_{f_3} + x_3 h_{fg3}) - p_3 x_3 v_{g3}$$

$$= (561.4 + 0.669 \times 2163.2) - 3 \times 10^5 \times 0.699 \times 0.606 \times 10^{-3}$$

$$= 2073.47 - 127.07 = 1946.4 \text{ kJ/kg}$$

$\therefore$ Heat transferred during cooling

$$= 2.045\ (1946.4 - 2393.14) = -913.6 \text{ kJ.}$$

i.e., **Heat lost during cooling = 913.6 kJ.** **(Ans.)**

Example 2.11. *If a certain amount of steam is produced at a pressure of 8 bar and dryness fraction 0.8. Calculate :*

(*i*) *External work done during evaporation.*

(*ii*) *Internal latent heat of steam.*

Solution. Pressure of steam, p = 8 bar

Dryness fraction, $x = 0.8$

At 8 bar. From steam tables,

$$v_g = 0.240 \text{ m}^3/\text{kg}, \ h_{fg} = 2046.5 \text{ kJ/kg}$$

(*i*) **External work done during evaporation**

$$= pxv_g = 8 \times 10^5 \times 0.8 \times 0.24 \text{ Nm}$$

$$= \frac{8 \times 10^5 \times 0.8 \times 0.24}{10^3} = \mathbf{153.6 \text{ kJ.}} \ \textbf{(Ans.)}$$

(*ii*) **Internal latent heat** $= xh_{fg}$ – external work done

$$= 0.8 \times 2046.5 - 153.6$$

$$= \mathbf{1483.6 \text{ kJ.}} \ \textbf{(Ans.)}$$

Example 2.12. *A quantity of steam at 10 bar and 0.85 dryness occupies 0.15 m³. Determine the heat supplied to raise the temperature of the steam to 300°C at constant pressure and percentage of this heat which appears as external work.*

Take specific heat of superheated steam as 2.2 kJ/kg K.

Solution. Pressure of steam, $p_1 = p_2 = 10$ bar

Dryness fraction, $x_1 = 0.85$

Volume of steam, $V_1 = 0.15 \text{ m}^3$

Final temperature of steam, $t_{sup_2} = 300°\text{C}$

Specific heat of superheated steam, $c_{ps} = 2.2$ kJ/kg K

$$\text{Mass of steam} = \frac{V_1}{x_1 v_{g_1}} = \frac{0.15}{0.85 \times 0.194} = 0.909 \text{ kg} \quad (\because \text{ At 10 bar} : v_g = 0.194 \text{ m}^3/\text{kg})$$

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Heat supplied per kg of steam

$$= (1 - x_1)\, h_{fg_1} + c_{ps}\,(300 - 179.9)$$

$$= (1 - 0.85)2013.6 + 2.2 \times 120.1 \qquad \left[\because \begin{array}{l} t_s = 179.9°\text{C} \ldots \text{at 10 bar,} \\ \text{and } h_{fg} = 2013.6 \text{ kJ/kg} \end{array}\right]$$

$$= 566.26 \text{ kJ/kg}$$

Total heat supplied

$$= 0.909 \times 566.26 = \mathbf{514.7\ kJ.} \quad \textbf{(Ans.)}$$

External work done during this process

$$= p\,(v_{sup_2} - x_1 v_{g_1}) \qquad [\because p_1 = p_2 = p]$$

$$= 10 \times 10^5 \left[\left(v_{g_1} \times \frac{T_{sup_2}}{T_{s_1}}\right) - x_1 v_{g_1}\right] \times 10^{-3} \quad \left[\because \frac{v_{g_1}}{T_{s_1}} = \frac{v_{sup_2}}{T_{sup_2}}\ i.e., v_{sup_2} = \frac{v_{g_1} \times T_{sup_2}}{T_{s_1}}\right]$$

$$= 10 \times 10^5 \left[0.194 \times \frac{(300 + 273)}{(179.9 + 273)} - 0.85 \times 0.194\right] \times 10^{-3}$$

$$= \frac{10 \times 10^5}{10^3}\ (0.245 - 0.165) = 80 \text{ kJ/kg}$$

∴ **Percentage of total heat supplied (per kg) which appears as external work**

$$= \frac{80}{566.26} = 0.141 = \mathbf{14.1\%.} \quad \textbf{(Ans.)}$$

Example 2.13. *Find the specific volume, enthalpy and internal energy of wet steam at 18 bar, dryness fraction 0.85.*

Solution. Pressure of steam, $p = 18$ bar

Dryness fraction, $x = 0.85$

From steam tables corresponding to 18 bar pressure :

$h_f = 884.6$ kJ/kg, $h_{fg} = 1910.3$ kJ/kg, $v_g = 0.110$ m³/kg, $u_f = 883$ kJ/kg, $u_g = 2598$ kJ/kg.

(*i*) **Specific volume of wet steam,**

$$v = x v_g = 0.85 \times 0.110 = \mathbf{0.0935\ m^3/kg.} \quad \textbf{(Ans.)}$$

(*ii*) **Specific enthalpy of wet steam,**

$$h = h_f + x h_{fg} = 884.6 + 0.85 \times 1910.3$$

$$= \mathbf{2508.35\ kJ/kg.} \quad \textbf{(Ans.)}$$

(*iii*) **Specific internal energy of wet steam,**

$$u = (1 - x)u_f + x u_g$$

$$= (1 - 0.85) \times 883 + 0.85 \times 2598$$

$$= \mathbf{2340.75\ kJ/kg.} \quad \textbf{(Ans.)}$$

Example 2.14. *Find the dryness fraction, specific volume and internal energy of steam at 7 bar and enthalpy 2550 kJ/kg.*

Solution. Pressure of steam, $p = 7$ bar

Enthalpy of steam, $h = 2550$ kJ

From steam tables corresponding to 7 bar pressure :

$h_f = 697.1$ kJ/kg, $h_{fg} = 2064.9$ kJ/kg, $v_g = 0.273$ m³/kg,

$u_f = 696$ kJ/kg, $u_g = 2573$ kJ/kg.

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(*i*) **Dryness fraction, x :**

At 7 bar, h_g = 2762 kJ/kg, hence since the actual enthalpy is given as 2550 kJ/kg, the steam must be in the wet vapour state.

Now, using the equation,

$$h = h_f + xh_{fg}$$

$$\therefore \quad 2550 = 697.1 + x \times 2064.9$$

i.e.,
$$x = \frac{2550 - 697.1}{2064.9} = 0.897$$

Hence, **dryness fraction = 0.897. (Ans.)**

(*ii*) **Specific volume of wet steam,**

$$v = xv_g = 0.897 \times 0.273 = \mathbf{0.2449\ m^3/kg.}\ \textbf{(Ans.)}$$

(*iii*) **Specific internal energy of wet steam,**

$$u = (1 - x)u_f + xu_g$$
$$= (1 - 0.897) \times 696 + 0.897 \times 2573$$
$$= \mathbf{2379.67\ kJ/kg.}\ \textbf{(Ans.)}$$

Example 2.15. *Steam at 120 bar has a specific volume of 0.01721 m^3/kg, find the temperature, enthalpy and the internal energy.*

Solution. Pressure of steam, p = 120 bar

Specific volume, v = 0.01721 m^3/kg

(*i*) **Temperature :**

First it is necessary to decide whether the steam is wet, dry saturated or superheated.

At 120 bar, v_g = 0.0143 m^3/kg, which is less than the actual specific volume of 0.01721 m^3/kg, and hence the steam is **superheated.**

From the superheat tables at 120 bar, the specific volume is 0.01721 m^3/kg at a temperature of **350°C. (Ans.)**

(*ii*) **Enthalpy :**

From the steam tables the specific enthalpy at 120 bar, 350°C,

$$h = \mathbf{2847.7\ kJ/kg.}\ \textbf{(Ans.)}$$

(*iii*) **Internal energy :**

To find internal energy, using the equation,

$$u = h - pv$$

$$= 2847.7 - \frac{120 \times 10^5 \times 0.01721}{10^3}$$

$$= \mathbf{2641.18\ kJ/kg.}\ \textbf{(Ans.)}$$

Example 2.16. *Steam at 140 bar has an enthalpy of 3001.9 kJ/kg, find the temperature, the specific volume and the internal energy.*

Solution. Pressure of steam, p = 140 bar

Enthalpy of steam, h = 3001.9 kJ/kg

(*i*) **Temperature :**

At 140 bar, h_g = 2642.4 kJ, which is less than the actual enthalpy of 3001.9 kJ/kg, and hence the steam is **superheated.**

From superheat tables at 140 bar, h = 3001.9 kJ/kg at a temperature of **400°C. (Ans.)**

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(*ii*) The **specific volume,** v = **0.01722 m³/kg. (Ans.)**

∴ The **internal energy** (specific),

$$u = h - pv = 3001.9 - \frac{140 \times 10^5 \times 0.01722}{10^3}$$

= **2760.82 kJ/kg. (Ans.)**

☞ **Example 2.17.** *Calculate the internal energy per kg of superheated steam at a pressure of 10 bar and a temperature of 300°C. Also find the change of internal energy if this steam is expanded to 1.4 bar and dryness fraction 0.8.*

Solution. At 10 bar, 300°C. From steam tables for superheated steam.

$$h_{sup} = 3051.2 \text{ kJ/kg } (T_{sup} = 300 + 273 = 573 \text{ K})$$

and corresponding to 10 bar (from tables of dry saturated steam)

$$T_s = 179.9 + 273 = 452.9 \text{ K} ; v_g = 0.194 \text{ m}^3/\text{kg}$$

To find $v_{sup.}$, using the relation,

$$\frac{v_g}{T_s} = \frac{v_{sup}}{T_{sup}}$$

$$\therefore \qquad v_{sup} = \frac{v_g \times T_{sup}}{T_s} = \frac{0.194 \times 573}{452.9} = 0.245 \text{ m}^3/\text{kg.}$$

Internal energy of superheated steam at 10 bar,

$$u_1 = h_{sup} - pv_{sup}$$
$$= 3051.2 - 10 \times 10^5 \times 0.245 \times 10^{-3}$$
$$= \mathbf{2806.2 \text{ kJ/kg.}} \textbf{ (Ans.)}$$

At 1.4 bar. From steam tables ;

$$h_f = 458.4 \text{ kJ/kg}, h_{fg} = 2231.9 \text{ kJ/kg} ; v_g = 1.236 \text{ m}^3/\text{kg}$$

Enthalpy of wet steam (after expansion)

$$h = h_f + xh_{fg}$$
$$= 458.4 + 0.8 \times 2231.9 = 2243.92 \text{ kJ.}$$

Internal energy of this steam,

$$u_2 = h - pxv_g$$
$$= 2243.92 - 1.4 \times 10^5 \times 0.8 \times 1.236 \times 10^{-3}$$
$$= 2105.49 \text{ kJ}$$

Hence change of internal energy per kg

$$u_2 - u_1 = 2105.49 - 2806.2$$
$$= \mathbf{-700.7 \text{ kJ.}} \textbf{ (Ans.)}$$

Negative sign indicates **decrease** in internal energy.

Example 2.18. *Find the internal energy of 1 kg of steam at 20 bar when*

(i) it is superheated, its temperature being 400°C ;

(ii) it is wet, its dryness being 0.9.

Assume superheated steam to behave as a perfect gas from the commencement of superheating and thus obeys Charle's law. Specific heat for steam = 2.3 kJ/kg K.

Solution. Mass of steam = 1 kg

Pressure of steam, p = 20 bar

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Temperature of superheated steam = 400°C (T_{sup} = 400 + 273 = 673 K)

Dryness fraction, $x = 0.9$

Specific heat of superheated steam, c_{ps} = 2.3 kJ/kg K

(*i*) **Internal energy of 1 kg of superheated steam :**

At 20 bar. From steam tables,

t_s = 212.4°C ; h_f = 908.6 kJ/kg ; h_{fg} = 1888.6 kJ/kg, v_g = 0.0995 m³/kg

Now, $h_{sup} = h_f + h_{fg} + c_{ps}\,(T_{sup} - T_s)$

$= 908.6 + 1888.6 + 2.3(400 - 212.4)$

$= 3228.68$ kJ/kg

Also, $h_{sup} = u + p \,.\, v_{sup}$

or $u = h_{sup} - p \,.\, v_{sup}$

The value of v_{sup} can be found out by Charle's law

$$\frac{v_g}{T_g} = \frac{v_{sup}}{T_{sup}}$$

$$\therefore \quad v_{sup} = \frac{v_g \times T_{sup}}{T_s} = \frac{0.0995 \times 673}{(212.4 + 273)} = 0.1379 \text{ m}^3/\text{kg}$$

Hence internal energy, $u = 3228.68 - 20 \times 10^5 \times 0.1379 \times 10^{-3}$

= 2952.88 kJ/kg. **(Ans.)**

(*ii*) **Internal energy of 1 kg of wet steam :**

$h = h_f + xh_{fg} = 908.6 + 0.9 \times 1888.6 = 2608.34$ kJ/kg

Again $h = u + p \,.\, x \,.\, v_g$

$\therefore \quad u = h - p \,.\, x \,.\, v_g = 2608.34 - 20 \times 10^5 \times 0.9 \times 0.0995 \times 10^{-3}$

$= 2429.24$ kJ/kg

Hence internal energy = 2429.24 kJ/kg. **(Ans.)**

Example 2.19. *Two boilers one with superheater and other without superheater are delivering equal quantities of steam into a common main. The pressure in the boilers and main is 20 bar. The temperature of steam from a boiler with a superheater is 350°C and temperature of the steam in the main is 250°C.*

Determine the quality of steam supplied by the other boiler. Take c_{ps} = 2.25 *kJ/kg.*

Solution. Boiler B_1. 20 bar, 350°C :

Enthalpy, $h_1 = h_{g_1} + c_{ps}\,(T_{sup} - T_s)$

$= 2797.2 + 2.25(350 - 212.4)$

$= 3106.8$ kJ/kg ...(*i*)

Boiler B_2. 20 bar (temperature not known) :

$h_2 = h_{f_2} + x_2 h_{fg_2}$

$= (908.6 + x_2 \times 1888.6)$ kJ/kg ...(*ii*)

Main. 20 bar, 250°C.

Total heat of 2 kg of steam in the steam main

$= 2[h_g + c_{ps}\,(T_{sup} - T_s)]$

$= 2[2797.2 + 2.25\,(250 - 212.4)] = 5763.6$ kJ ...(*iii*)

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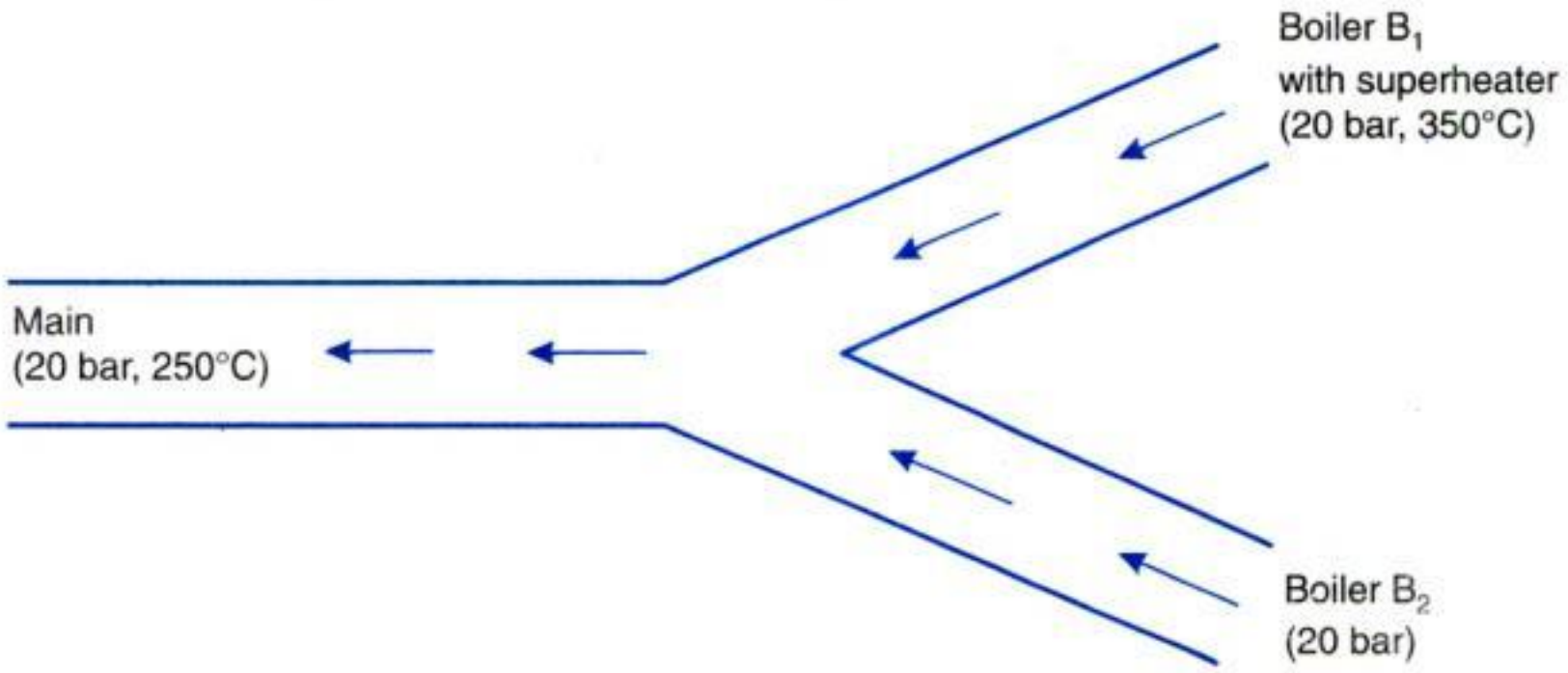

Fig. 2.13

Equating (*i*) and (*ii*) with (*iii*), we get

$$3106.8 + 908.6 + x_2 \times 1888.6 = 5763.6$$

$$4015.4 + 1888.6x_2 = 5763.6$$

$$\therefore \quad x_2 = \frac{5763.6 - 4015.4}{1888.6} = 0.925$$

Hence, **quality of steam supplied by the other boiler = 0.925.** **(Ans.)**

Example 2.20. *Determine the entropy of 1 kg of wet steam at a pressure of 6 bar and 0.8 dry, reckoned from freezing point (0°C).*

Solution. Mass of wet steam, $m = 1$ kg

Pressure of steam, $p = 6$ bar

Dryness fraction, $x = 0.8$

At 6 bar. From steam tables,

$$t_s = 158.8°C, \; h_{fg} = 2085 \text{ kJ/kg}$$

Entropy of wet steam is given by

$$s_{wet} = c_{pw} \log_e \frac{T_s}{273} + \frac{xh_{fg}}{T_s} \qquad \text{(where } c_{pw} = \text{specific heat of water)}$$

$$= 4.18 \log_e \left(\frac{158.8 + 273}{273}\right) + \frac{0.8 \times 2085}{(158.8 + 273)}$$

$$= 1.9165 + 3.8700 = 5.7865 \text{ kJ/kg K}$$

Hence, **entropy of wet steam = 5.7865 kJ/kg K.** **(Ans.)**

Example 2.21. *Steam enters an engine at a pressure 10 bar absolute and 400°C. It is exhausted at 0.2 bar. The steam at exhaust is 0.9 dry. Find :*

(i) Drop in enthalpy ;

(ii) Change in entropy.

Solution. Initial pressure of steam, $p_1 = 10$ bar

Initial temperature of steam, $t_{sup} = 400°C$

Final pressure of steam, $p_2 = 0.2$ bar

Final condition of steam, $x_2 = 0.9$

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At 10 bar, 400°C. From steam tables,

$$h_{sup} = 3263.9 \text{ kJ/kg} \; ; \; s_{sup} = 7.465 \text{ kJ/kg K}$$

i.e., $h_1 = h_{sup} = 3263.9$ kJ/kg and $s_1 = s_{sup} = 7.465$ kJ/kg K

At 0.2 bar. From steam tables,

$$h_f = 251.5 \text{ kJ/kg} \; ; \; h_{fg} = 2358.4 \text{ kJ/kg} \; ;$$
$$s_f = 0.8321 \text{ kJ/kg K} \; ; \; s_g = 7.9094 \text{ kJ/kg K}$$

Also, $h_2 = h_{f_2} + x_2 h_{fg_2} = 251.5 + 0.9 \times 2358.4$
$= 2374$ kJ/kg.

Also, $s_2 = s_{f_2} + x_2 s_{fg_2}$

$= s_{f_2} + x_2(s_{g_2} - s_{f_2})$
$= 0.8321 + 0.9(7.9094 - 0.8321)$
$= 7.2016$ kJ/kg K

Hence, (*i*) **Drop in enthalpy,**

$= h_1 - h_2 = 3263.9 - 2374 =$ **889.9 kJ/kg.** **(Ans.)**

(*ii*) **Change in entropy**

$= s_1 - s_2 = 7.465 - 7.2016$
$=$ **0.2634 kJ/kg K (decrease).** **(Ans.)**

Example 2.22. *Find the entropy of 1 kg of superheated steam at a pressure of 12 bar and a temperature of 250°C. Take specific heat of superheated steam as 2.1 kJ/kg K.*

Solution. Mass of steam, $m = 1$ kg

Pressure of steam, $p = 12$ bar

Temperature of steam, $T_{sup} = 250 + 273 = 523$ K

Specific heat of superheated steam, $c_{ps} = 2.1$ kJ/kg K

At 12 bar. From steam tables,

$$T_s = 188 + 273 = 461 \text{ K}, \; h_{fg} = 1984.3 \text{ kJ/kg}$$

∴ Entropy of 1 kg of superheated steam,

$$s_{sup} = c_{pw} \log_e \frac{T_s}{273} + \frac{h_{fg}}{T_s} + c_{ps} \log_e \frac{T_{sup}}{T_s}$$

$$= 4.18 \log_e \left(\frac{461}{273}\right) + \frac{1984.3}{461} + 2.1 \times \log_e \left(\frac{523}{461}\right)$$

$$= 2.190 + 4.304 + 0.265$$

$=$ **6.759 kJ/kg.** **(Ans.)**

Example 2.23. *A piston-cylinder contains 3 kg of wet steam at 1.4 bar. The initial volume is 2.25 m^3. The steam is heated until its temperature reaches 400°C . The piston is free to move up or down unless it reaches the stops at the top. When the piston is up against the stops the cylinder volume is 4.65 m^3. Determine the amount of work and heat transfer to or from steam.*

(U.P.S.C. 1998)

Solution. Initial volume per kg of steam $= \frac{2.25}{3} = 0.75$ m³/kg

Specific volume of steam at 1.4 bar $= 1.2363$ m³/kg

Dryness fraction of initial steam $= \frac{0.75}{12363} = 0.607$

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At 1.4 bar, the enthalpy of 3 kg of steam

$$= 3\left[h_f + xh_{fg}\right] = 3\ [\ 458.4 + 0.607 \times 2231.9] = 5439.5 \text{ kJ}$$

At 400°C, volume of steam per kg = $\frac{4.65}{3}$ = 1.55 m³/kg

At 400°C, when v_{sup} = 1.55 m³/kg, from steam tables,

Pressure of steam = 2.0 bar

Saturation temperature

= 120.2°C, h = 3276.6 kJ/kg

Degree of superheat

$$= t_{sup} - t_s = 400 - 120.2 = 279.8°C$$

Enthalpy of superheated steam at 2.0 bar,

400°C = 3 × 3276.6 = 9829.8 kg

Heat *added during the process*

= 9829.8 – 5439.5 = **4390.3 kJ.** **(Ans.)**

Internal energy of 0.607 dry steam at 1.4 bar

= 3 × 1708 = 5124 kJ.

Internal energy of superheated steam at 2 bar, 400°C

$$= 3(h_{sup} - pv) = 3(3276.6 - 2 \times 10^2 \times 1.55) = 8899.8 \text{ kJ}$$

(∵ 1 bar = 10^2 kPa)

Change in internal energy = 8899.8 – 5124 = 3775.8 kJ

Hence, **work done** = 4390.3 – 3775.8 = **614.5 kJ.** **(Ans.)** (∵ W = Q – ΔU)

2.18. DETERMINATION OF DRYNESS FRACTION OF STEAM

The dryness fraction of steam can be measured by using the following *calorimeters* :

1. Tank or bucket calorimeter
2. Throttling calorimeter
3. Separating and throttling calorimeter.

2.18.1. Tank or Bucket Calorimeter

The dryness fraction of steam can be found with the help of tank calorimeter as follows :

A known mass of steam is passed through a known mass of water and steam is completely condensed. The heat lost by steam is equated to heat gained by the water.

Fig. 2.14 shows the arrangement of this calorimeter.

The steam is passed through the sampling tube into the bucket calorimeter containing *a known* mass of water.

The weights of calorimeter with water before mixing with steam and after mixing the steam are obtained by weighing.

The temperature of water before and after mixing the steam are measured by *mercury thermometer*.

The pressure of steam passed through the sampling tube is measured with the help of *pressure gauge*.

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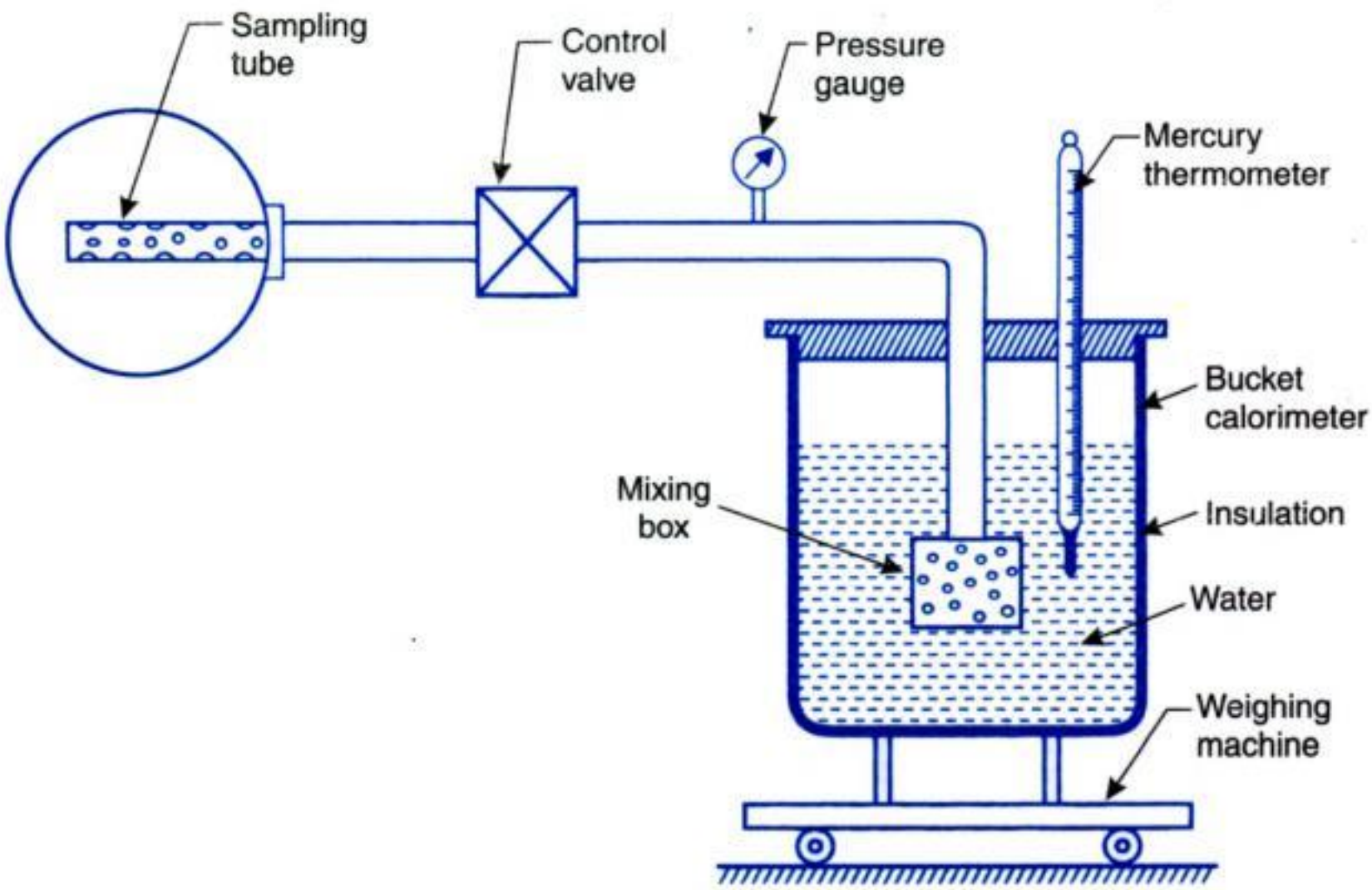

Fig. 2.14. Tank or bucket calorimeter.

Let, p_s = Gauge pressure of steam (bar),
p_a = Atmospheric pressure (bar),
t_s = Saturation temperature of steam known from steam table at pressure $(p_s + p_a)$,
h_{fg} = Latent heat of steam,
x = Dryness fraction of steam,
c_{pw} = Specific heat of water,
c_{pc} = Specific heat of calorimeter,
m_c = Mass of calorimeter, kg,
m_{cw} = Mass of calorimeter and water, kg,
$m_w = (m_{cw} - m_c)$ = Mass of water in calorimeter, kg,
m_{cws} = Mass of calorimeter, water and condensed steam, kg,
$m_s = (m_{cws} - m_{cw})$ = Mass of steam condensed in calorimeter, kg,
t_{cw} = Temperature of water and calorimeter before mixing the steam, °C, and
t_{cws} = Temperature of water and calorimeter after mixing the steam, °C.

Neglecting the losses and *assuming* that the *heat lost by steam is gained by water and calorimeter*, we have

$$(m_{cws} - m_{cw})\,[xh_{fg} + c_{pw}\,(t_s - t_{cws})] = (m_{cw} - m_c)c_{pw}\,(t_{cws} - t_{cw}) + m_c\,c_{pc}\,(t_{cws} - t_{cw})$$

$$\therefore \quad m_s[xh_{fg} + c_{pw}\,(t_s - t_{cws})] = (t_{cws} - t_{cw})\,[(m_{cw} - m_c)(c_{pw} + m_c c_{pc})] \qquad \text{...(2.19)}$$

or $$m_s[xh_{fg} + c_{pw}\,(t_s - t_{cws})] = (t_{cws} - t_{cw})(m_w c_{pw} + m_c c_{pc})$$

The $m_c c_{pc}$ is known as *water equivalent of calorimeter*.

The value of dryness fraction 'x' can be found by solving the above equation.

The value of dryness fraction found by this method involves some *inaccuracy* since losses due to convection and radiation are *not* taken into account.

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The calculated value of dryness fraction neglecting losses is *always less* than the actual value of the dryness.

Example 2.24. *Steam at a pressure of 5 bar passes into a tank containing water where it gets condensed. The mass and temperature in the tank before the admission of steam are 50 kg and 20°C respectively. Calculate the dryness fraction of steam as it enters the tank if 3 kg of steam gets condensed and resulting temperature of the mixture becomes 40°C. Take water equivalent of tank as 1.5 kg.*

Solution. Pressure of steam, $p = 5$ bar
Mass of water in the tank = 50 kg
Initial temperature of water = 20°C
Amount of steam condensed, $m_s = 3$ kg
Final temperature after condensation of steam = 40°C
Water equivalent of tank = 1.5 kg

Dryness fraction of steam, x :

At 5 bar. From steam tables,

$$h_f = 640.1 \text{ kJ/kg} \; ; \; h_{fg} = 2107.4 \text{ kJ/kg}$$

Total mass of water, m_w = mass of water in the tank + water equivalent of tank
= 50 + 1.5 = 51.5 kg

Also, heat lost by steam = heat gained by water

$$m_s [(h_f + xh_{fg}) - 1 \times 4.18 (40 - 0)] = m_w[1 \times 4.18 (40 - 20)]$$

or $$3[(640.1 + x \times 2107.4) - 4.18 \times 40] = 51.5 \times 4.18 \times 20$$

or $$3(472.9 + 2107.4x) = 4305.4$$

or $$472.9 + 2107.4x = 1435.13$$

$$\therefore \quad x = \frac{1435.13 - 472.9}{2107.4} = 0.456.$$

Hence, **dryness fraction of steam = 0.456.** **(Ans.)**

Example 2.25. *Steam at a pressure of 1.1 bar and 0.95 dry is passed into a tank containing 90 kg of water at 25°C. The mass of tank is 12.5 kg and specific heat of metal is 0.42 kJ/kg K. If the temperature of water rises to 40°C after the passage of the steam, determine the mass of steam condensed. Neglect radiation and other losses.*

Solution. Pressure of steam, $p = 1.1$ bar
Dryness fraction of steam, $x = 0.95$
Mass of water in the tank = 90 kg
Initial temperature of water in the tank = 25°C
Mass of tank = 12.5 kg
Specific heat of metal = 0.42 kJ/kg K
Final temperature of water = 40°C.

Mass of steam condensed, m_s :

Since the radiation losses are neglected,

$\therefore$ Heat lost by steam = Heat gained by water

or $$m_s[(h_f + xh_{fg}) - 1 \times 4.18 (40 - 0)] = m[1 \times 4.18(40 - 25)]$$

But $$m = m_1 + m_2$$

where, m_1 = Mass of cold water in the vessel before steam supply, and
m_2 = Water equivalent of vessel = 0.42 × 12.5 = 5.25 kg

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At 1.1 bar. From steam tables,

$$h_f = 428.8 \text{ kJ/kg} \; ; \; h_{fg} = 2250.8 \text{ kJ/kg}$$

$$\therefore \quad m_s \, [(428.8 + 0.95 \times 2250.8) - 1 \times 4.18 \times 40]$$
$$= (90 + 5.25) \, [1 \times 4.18 \times (40 - 25)]$$

$$m_s \, [2567.06 - 167.20] = 95.25 \times 62.7$$

i.e., $$2399.86 m_s = 5972.17$$

$$\therefore \quad m_s = 2.488 \text{ kg}$$

Hence, **mass of steam condensed = 2.488 kg.** **(Ans.)**

2.18.2. Throttling Calorimeter

The dryness fraction of wet steam can be determined by using a throttling calorimeter which is illustrated diagrammatically in Fig. 2.15.

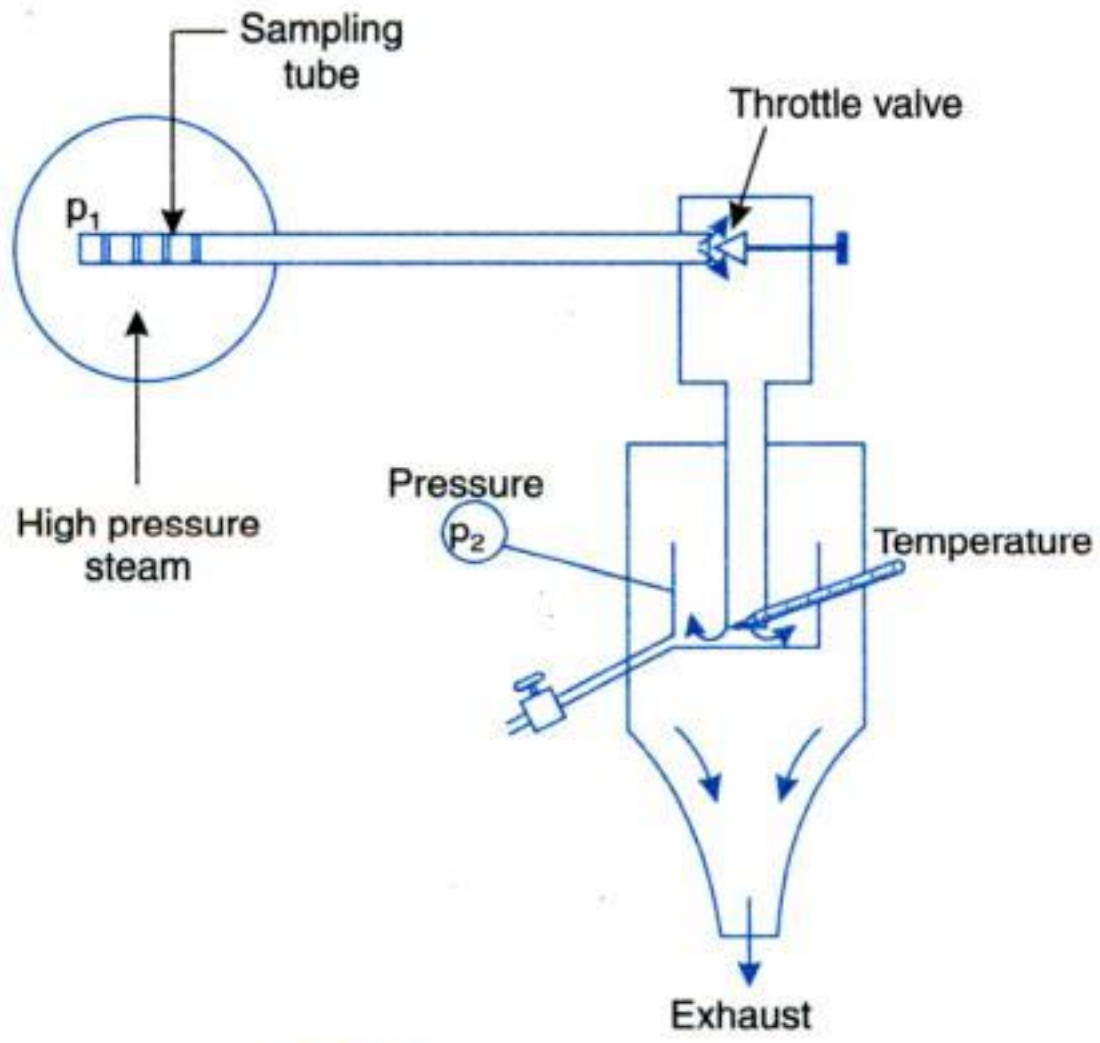

Fig. 2.15. Throttling calorimeter.

The steam to be sampled is taken from the pipe by means of suitable positioned and dimensioned sampling tube. It passes into an insulated container and is throttled through an orifice to atmospheric pressure. Here the temperature is taken and the steam ideally should have about 5.5 K of superheat.

The throttling process is shown on h-s diagram in Fig. 2.16 by the line 1-2. If steam initially wet is throttled through a sufficiently large pressure drop, then the steam at state 2 will become superheated. State 2 can then be defined by the *measured pressure and temperature*. The enthalpy, h_2 can then be found and hence

$$h_2 = h_1 = (h_{f_1} + x_1 h_{fg_1}) \text{ at } p_1$$

[where $h_2 = h_{f_2} + h_{fg_2} + c_{ps} \, (T_{sup_2} - T_{s_2})$]

$$\therefore \quad x_1 = \frac{h_2 - h_{f_1}}{h_{fg_1}} \qquad \ldots(2.20)$$

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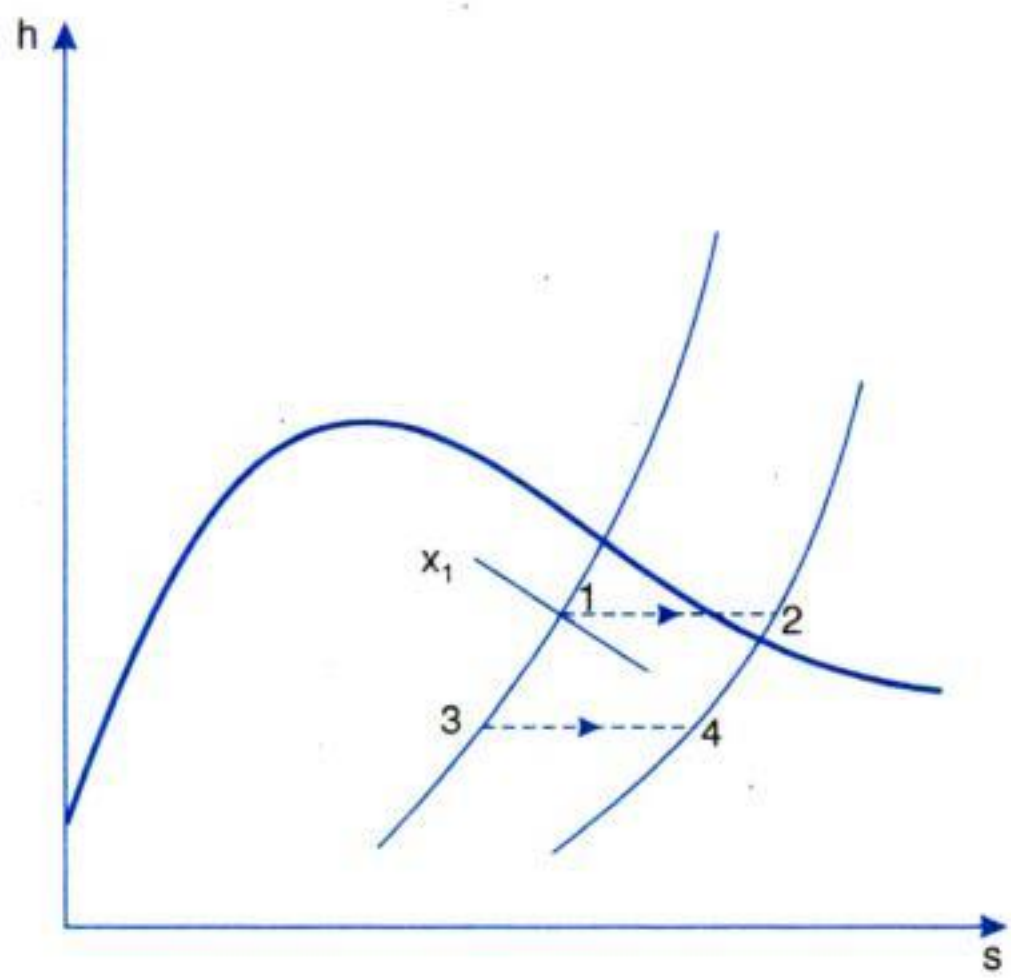

Fig. 2.16. Throttling process.

Hence, the dryness fraction is determined and state 1 is defined.

Example 2.26. *A throttling calorimeter is used to measure the dryness fraction of the steam in the steam main which has steam flowing at a pressure of 8 bar. The steam after passing through the calorimeter is at 1 bar pressure and 115°C.*

Calculate the dryness fraction of the steam in the main. Take $c_{ps} = 2.1$ *kJ/kg K.*

Solution. Condition of steam *before throttling* :

$$p_1 = 8 \text{ bar}, x_1 = ?$$

Condition of steam *after throttling* :

$$p_2 = 1 \text{ bar}, \; t_2 = t_{sup_2} = 115°C$$

As throttling is a *constant enthalpy* process

$$\therefore \quad h_1 = h_2$$

i.e.,
$$h_{f_1} + x_1 h_{gf_1} = h_{f_2} + h_{fg_2} + c_{p_s} (T_{\sup_2} - T_{s_2}) \quad [\because \; T_{sup_2} = 115 + 273 = 388 \text{ K}$$
$$T_{s_2} = 99.6 + 273 = 372.6 \text{ K (at 1 bar)}]$$

$$720.9 + x_1 \times 2046.5 = 417.5 + 2257.9 + 2.1(388 - 372.6)$$

$$720.9 + 2046.5 \, x_1 = 2707.7$$

$$\therefore \quad x_1 = \frac{2707.7 - 720.9}{2046.5} = 0.97$$

Hence, **dryness fraction of steam in the main = 0.97.** **(Ans.)**

2.18.3. Separating and Throttling Calorimeter

If the steam whose dryness fraction is to be determined is *very wet* then throttling to atmospheric pressure *may not be sufficient to ensure superheated steam at exit*. In this case it is *necessary to dry the steam partially, before throttling*. This is done by *passing the steam sample from the main through a separating calorimeter* as shown in Fig. 2.17. The steam is made to change direction suddenly, and the water, being denser than the dry steam is separated out. The quantity of water which is separated out (m_w) is measured at the separator, the steam remaining,

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which *now has a higher dryness fraction*, is passed through the *throttling calorimeter*. With the combined separating and throttling calorimeter it is *necessary* to condense the steam after throttling and measure the amount of condensate (m_s). If a throttling calorimeter only is sufficient, there is no need to measure condensate, the pressure and temperature measurements at exit being sufficient.

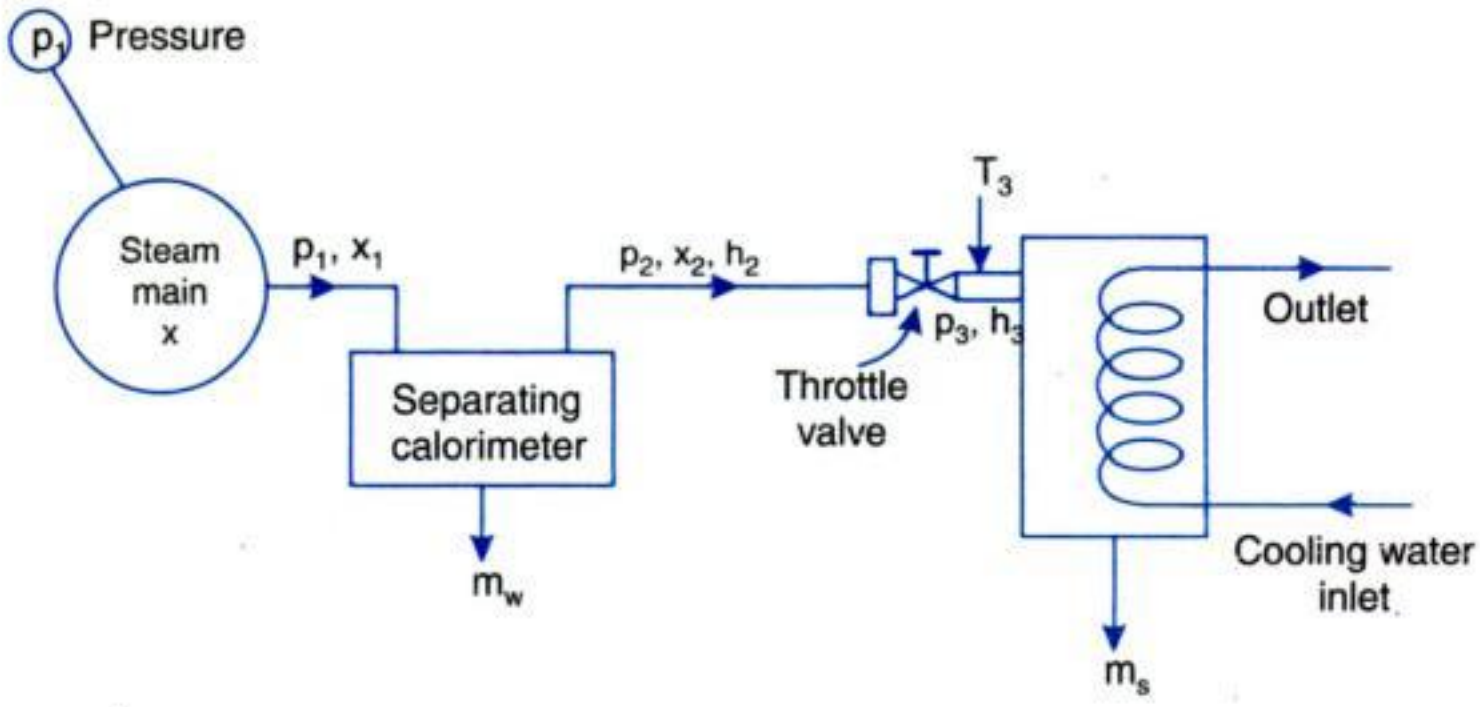

Fig. 2.17. Separating and throttling calorimeter.

Dryness fraction at 2 is x_2, therefore, the mass of dry steam leaving the separating calorimeter is equal to $x_2 m_s$ and this must be the mass of dry vapour in the sample drawn from the main at state 1.

$$\text{Hence fraction in main, } x_1 = \frac{\text{Mass of dry vapour}}{\text{Total mass}} = \frac{x_2\, m_s}{m_w + m_s}.$$

The dryness fraction, x_2, can be determined as follows :

$$^*h_3 = h_2 = h_{f_2} + x_2 h_{fg_2} \text{ at } p_2 \; [^*h_3 = h_{f_3} + h_{fg_3} + c_{ps}\,(T_{sup_3} - T_{s_3}) \text{ at pressure } p_3]$$

or,
$$x_2 = \frac{h_3 - h_{f_2}}{h_{fg_2}}$$

The values of h_{f_2} and h_{fg_2} are read from steam tables at pressure p_2. The pressure in the separator is small so that p_1 is approximately equal to p_2.

Example 2.27. *The following observations were taken with a separating and a throttling calorimeter arranged in series :*

Water separated = 2 kg, steam discharged from the throttling calorimeter = 20.5 kg, temperature of steam after throttling = 110°C, initial pressure = 12 bar abs., barometer = 760 mm of Hg, final pressure = 5 mm of Hg.

Estimate the quality of steam supplied.

Solution. Quantity of water separated out, m_w = 2 kg

Steam (condensate) discharged from the throttling calorimeter, m_s = 20.5 kg

Temperature of steam after throttling, t_{sup} = 110°C

Initial pressure of steam, p_1 = 12 bar abs.

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Final pressure of steam, $p_3 = 760 + 5 = 765$ mm

$$= \frac{765}{1000} \times 1.3366 \qquad (\because \quad 1 \text{ m Hg} = 1.3366 \text{ bar})$$

$$\simeq 1 \text{ bar}$$

From steam tables :

At $p_1 = p_2 =$ **12 bar :** $h_f = 798.4$ kJ/kg, $h_{fg} = 1984.3$ kJ/kg

At $p_3 =$ **1 bar :** $t_s = 99.6°C$, $h_f = 417.5$ kJ/kg, $h_{fg} = 2257.9$ kJ/kg

$t_{sup} = 110°C$ (given)

Also, $h_3 = h_2$

$$(h_{f_3} + h_{fg_3}) + c_{ps}(T_{\sup_3} - T_{s_3}) = hf_2 + x_2 h_{fg_2}$$

Taking $c_{ps} = 2$ kJ/kg K, we get

$$417.5 + 2257.9 + 2[(110 + 273) - (99.6 + 273)] = 798.4 + x_2 \times 1984.3$$

$$2696.2 = 798.4 + 1984.3\, x_2$$

$$\therefore \quad x_2 = \frac{2696.2 - 798.4}{1984.3} = 0.956$$

Now, **quality of steam supplied,**

$$x_1 = \frac{x_2 m_s}{m_w + m_s} = \frac{0.956 \times 20.5}{2 + 20.5}$$

$$= \mathbf{0.87.} \quad \text{(Ans.)}$$

☞ **Example 2.28.** *The following data were obtained in a test on a combined separating and throttling calorimeter :*

Pressure of steam sample = 15 bar, pressure of steam at exit = 1 bar, temperature of steam at the exit = 150°C, discharge from separating calorimeter = 0.5 kg/min, discharge from throttling calorimeter = 10 kg/min.

Determine the dryness fraction of the sample steam.

Solution. Pressure of steam sample, $p_1 = p_2 = 15$ bar

Pressure of steam at the exit, $p_3 = 1$ bar

Temperature of steam at the exit, $t_{sup_3} = 150°C$

Discharge from separating calorimeter, $m_w = 0.5$ kg/min

Discharge from throttling calorimeter, $m_s = 10$ kg/min

From steam tables :

At $p_1 = p_2 =$ **15 bar** : $h_{f_2} = 844.7$ kJ/kg, $h_{fg_2} = 1945.2$ kJ/kg

At $p_3 =$ **1 bar and 150°C** : $h_{sup_3} = 2776.4$ kJ/kg

Also, $h_2 = h_3$

$$h_{f_2} + x_2 h_{fg_2} = h_{sup_3}$$

$$844.7 + x_2 \times 1945.2 = 2776.4$$

$$\therefore \quad x_2 = \frac{2776.4 - 844.7}{1945.2} = 0.993$$

Now, **quality of steam supplied,**

$$x_1 = \frac{x_2 m_s}{m_s + m_w} = \frac{0.993 \times 10}{10 + 0.5} = \mathbf{0.946.} \quad \text{(Ans.)}$$

HIGHLIGHTS

1. A pure substance is a system which is (*i*) homogeneous in composition, (*ii*) homogeneous in chemical aggregation, (*iii*) invariable in chemical aggregation.
2. The triple point is merely the point of intersection of sublimation and vapourisation curves. It must be understood that only on *p-T* diagram is the triple point represented by a point. On *p-V* diagram it is a line, and on a *U-V* diagram it is a triangle.
3. Steam as a vapour does not obey laws of perfect gases unless and until it is highly in super dry condition.
4. *Dryness fraction* is the ratio of the mass of actual dry steam to the mass of steam containing it.

   i.e., $$\text{Dryness fraction} = \frac{m_s}{m_s + m_w}$$

   where, m_s = Mass of dry steam contained in steam considered

   m_w = Mass of water particles in suspension in the steam considered.
5. Superheated steam behaves like a gas and therefore, it follows gas laws. The law for adiabatic expansion is $pv^{1.3} = C$.
6. External work of evaporation $= p(v_g - v_f)$

   Internal latent heat $= h_{fg} - \frac{pv_g}{J}$

   Internal energy of steam, u :

   (*i*) For wet steam : $$u = h - \frac{p.x.v_g}{J}$$

   (*ii*) For superheated steam : $u = h_{sup} - \frac{p \cdot v_{sup}}{J}$ (J = 1, in SI units)
7. Entropy of water when heated from temperature T_1 to T_2 ;

   $$s_2 - s_1 = c_{pw} \log_e \frac{T_2}{T_1}$$

   If 0°C is taken as the datum then entropy of water at any temperature T, above this datum will be

   $$s_f = c_{pw} \log_e \frac{T}{273}$$

   Entropy of evaporation :

   $$s_{evap.} = \frac{h_{fg}}{T_s}$$ ...(when water is evaporated to steam completely)

   $$= \frac{xh_{fg}}{T_s}$$ ...(when water is evaporated partially and dryness fraction of steam is x)

   Entropy of steam :

   $$s_{wet} = s_f + \frac{xh_{fg}}{T_s}$$ ...[wet steam ($x < 1$)]

   $$s_g = s_f + \frac{h_{fg}}{T_s}$$ ...[dry and saturated steam ($x = 1$)]

   $$s_{sup} = s_f + \frac{h_{fg}}{T_s} + c_{ps} \log_e \frac{T_{sup}}{T_s}$$ ...(superheated steam)

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8. Mollier chart/diagram is more widely used than any other entropy diagram, since the work done on vapour cycles can be scaled from this diagram directly as a length, whereas on T-s diagram it is represented by an area.

9. *Different processes :*

(*i*) Constant volume heating or cooling

$$x_1 v_{g_1} = x_2 v_{g_2}$$

$$x_1 v_{g_1} = v_{sup_2} = v_{g_2} \cdot \frac{T_{sup_2}}{T_{s_2}}$$

(*ii*) Constant pressure heating or cooling

$$Q = h_2 - h_1$$

(*iii*) Isentropic expansion (non-flow process)

$$W = (u_1 - u_2) \text{ and } s_1 = s_2$$

(*iv*) Throttling

$$h_{f_1} + x_1 h_{fg_1} = h_{f_2} + x_2 h_{fg_2} \quad \text{...(For wet condition)}$$

$$= h_{f_2} + h_{fg_2} + c_{ps}(T_{sup} - T_{s_2}) \quad \text{...(For superheated condition)}$$

10. Dryness fraction of steam can determined by the following methods :

(*i*) Bucket calorimeter

(*ii*) Throttling calorimeter

(*iii*) Separating and throttling calorimeter.

OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. (*a*) Specific volume of water decreases on freezing

(*b*) Boiling point of water decreases with increasing pressure

(*c*) Specific volume of CO_2 increases on freezing

(*d*) Freezing temperature of water decreases with increasing pressure.

2. (*a*) The slope of vapourisation curve is always negative

(*b*) The slope of vapourisation curve is always positive

(*c*) The slope of sublimation curve in negative for all pure substances

(*d*) The slope of fusion curve is positive for all pure substances.

3. (*a*) The process of passing from liquid to vapour is condensation

(*b*) An isothermal line is also a constant pressure line during wet region

(*c*) Pressure and temperature are independent during phase change

(*d*) The term dryness fraction is used to describe the fraction by mass of liquid in the mixture of liquid water and water vapour.

4. The latent heat of vapourisation at critical point is

(*a*) less than zero (*b*) greater than zero

(*c*) equal to zero (*d*) none of the above.

5. (*a*) Critical point involves equilibrium of solid and vapour phases

(*b*) Critical point involves equilibrium of solid and liquid phases

(*c*) Critical point involves equilibrium of solid, liquid and vapour phases

(*d*) Triple point involves equilibrium of solid, liquid and vapour phases.

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6. With the increase in pressure
   (*a*) boiling point of water increases and enthalpy of evaporation increases
   (*b*) boiling point of water increases and enthalpy of evaporation decreases
   (*c*) boiling point of water decreases and enthalpy of evaporation increases.
7. With increase in pressure
   (*a*) enthalpy of dry saturated steam increases
   (*b*) enthalpy of dry saturated steam decreases
   (*c*) enthalpy of dry saturated steam remains same
   (*d*) enthalpy of dry saturated steam first increases and then decreases.
8. Dryness fraction of steam is defined as
   (*a*) mass of water vapour in suspension/(mass of water vapour in suspension + mass of dry steam)
   (*b*) mass of dry steam/mass of water vapour in suspension
   (*c*) mass of dry steam/(mass of dry steam + mass of water vapour in suspension)
   (*d*) mass of water vapour in suspension/mass of dry steam.
9. The specific volume of water when heated at 0°C
   (*a*) first increases and then decreases
   (*b*) first decreases and then increases
   (*c*) increases steadily
   (*d*) decreases steadily.
10. Only throttling calorimeter is used for measuring
   (*a*) very low dryness fraction upto 0.7
   (*b*) very high dryness fraction upto 0.98
   (*c*) dryness fraction of only low pressure steam
   (*d*) dryness fraction of only high pressure steam.
11. Heat of superheated steam is given by
   (*a*) $h_{sup} = h_f + h_{fg} + c_{ps} \log_e \frac{T_{sup}}{T_s}$
   (*b*) $h_{sup} = h_f + xh_{fg}$
   (*c*) $h_{sup} = h_f + h_{fg}$
   (*d*) $h_{sup} = h_f + xh_{fg} + c_{ps} \log_e \frac{T_s}{273}$.
12. Volume of wet steam (per kg) with dryness fraction x is given by
   (*a*) $x^3 v_g$
   (*b*) xv_f
   (*c*) $x^2(v_g - v_f)$
   (*d*) $x^2 v_g$
   (*e*) none of the above.
13. Internal latent heat is given by
   (*a*) $h_{fg} - \frac{pv_g}{J}$
   (*b*) $h_g - \frac{pv_g}{J}$
   (*c*) $h_{sup} - \frac{pv_f}{J}$
   (*d*) $h_{fg} + \frac{pv_g}{J}$
   (*e*) none of the above.
14. Entropy of 1 kg of water at T K is given by
   (*a*) $c_{pw} \log_e \frac{T}{273}$
   (*b*) $c_{pw} \log_e \frac{T_2}{T_1}$
   (*c*) $c_{pw} \log_{10} \frac{T}{273}$
   (*d*) $c_{pw} \log_e \frac{T_2}{273}$
   (*e*) none of the above.
15. Entropy of wet steam (1 kg) is given by
   (*a*) $s_f + \frac{xh_{fg}}{T_s}$
   (*b*) $s_g + \frac{xh_{fg}}{T_s}$
   (*c*) $s_f + \frac{h_{fg}}{T_s}$
   (*d*) $s_f + c_{ps} \log_e \frac{T_{sup}}{T_s}$
   (*e*) none of the above.

16. In throttling process

(a) $h_1^2 = h_2$ (b) $h_1 = h_2$

(c) $h_1 = h_2 + \frac{h_{fg}}{T_s}$ (d) $h_2 = h_1 + \frac{h_{fg}}{T_s}$

(e) none of the above.

17. In isentropic process

(a) $W = 2(u_2 - u_1)$ (b) $W = (u_2 - u_1)^2$

(c) $W = u_2 - u_1$ (d) $W = (u_2 - u_1)^{1/2}$

(e) none of the above.

ANSWERS

1.	(d)	2.	(a)	3.	(b)	4.	(c)	5.	(d)	6.	(b)	7.	(b)
8.	(c)	9.	(b)	10.	(b)	11.	(a)	12.	(e)	13.	(a)	14.	(a)
15.	(a)	16.	(b)	17.	(c).								

THEORETICAL QUESTIONS

1. What is a pure substance ?
2. Draw and explain a p-T (pressure-temperature) diagram for a pure substance.
3. What is a triple point ?
4. Explain with a neat diagram p-V-T surface.
5. Does wet steam obey laws of perfect gases ?
6. Describe the process of formation of steam and give its graphical representation also.
7. Explain the following terms relating to steam formation :

(i) Sensible heat of water, (ii) Latent heat of steam,

(iii) Dryness fraction of steam, (iv) Enthalpy of wet steam, and

(v) Superheated steam.

8. What advantages are obtained if superheated steam is used in steam prime movers ?
9. What do you mean by the following :

(i) Internal latent heat (ii) Internal energy of steam

(iii) External work of evaporation (iv) Entropy of evaporation

(v) Entropy of wet steam (vi) Entropy of superheated steam.

10. Write a short note on Mollier chart.
11. Draw a neat sketch of throttling calorimeter and explain how dryness fraction of steam is determined ; clearly explain its limitations.
12. Describe with a neat sketch a separating-throttling calorimeter for measuring the dryness fraction of steam.

UNSOLVED EXAMPLES

1. Find the specific volume, enthalpy and internal energy of wet steam at 18 bar, dryness fraction 0.9.

[**Ans.** 0.0994 m^3/kg ; 2605.8 kJ/kg ; 2426.5 kJ/kg]

2. Find the dryness fraction, specific volume and internal energy of steam at 7 bar and enthalpy 2600 kJ/kg.

[**Ans.** 0.921 ; 0.2515 m^3/kg, 2420 kJ/kg]

3. Steam at 110 bar has a specific volume of 0.0196 m^3/kg, find the temperature, the enthalpy and the internal energy.

[**Ans.** 350°C ; 2889 kJ/kg ; 2673.4 kJ/kg]

4. Steam at 150 bar has an enthalpy of 3309 kJ/kg, find the temperature, the specific volume and the internal energy. [**Ans.** 500°C ; 0.02078 m^3/kg ; 2997.3 kJ/kg]

5. Steam at 19 bar is throttled to 1 bar and the temperature after throttling is found to be 150°C. Calculate the initial dryness fraction of the steam. [**Ans.** 0.989]

6. Find the internal energy of one kg of steam at 14 bar under the following conditions :
   (*i*) When the steam is 0.85 dry ;
   (*ii*) When steam is dry and saturated ; and
   (*iii*) When the temperature of steam is 300°C. Take c_{ps} = 2.25 kJ/kg K.
   [**Ans.** (*i*) 2327.5 kJ/kg ; (*ii*) 2592.5 kJ/kg ; (*iii*) 2784 kJ/kg]

7. Calculate the internal energy of 0.3 m^3 of steam at 4 bar and 0.95 dryness. If this steam is superheated at constant pressure through 30°C, determine the heat added and change in internal energy.
   [**Ans.** 2451 kJ/kg ; 119 kJ ; 107.5 kJ/kg]

8. Water is supplied to the boiler at 15 bar and 80°C and steam is generated at the same pressure at 0.9 dryness. Determine the heat supplied to the steam in passing through the boiler and change in entropy.
   [**Ans.** 2260.5 kJ/kg ; 4.92 kJ/kg K]

9. A cylindrical vessel of 5 m^3 capacity contains wet steam at 1 bar. The volume of vapour and liquid in the vessel are 4.95 m^3 and 0.05 m^3 respectively. Heat is transferred to the vessel until the vessel is filled with saturated vapour. Determine the heat transfer during the process. [**Ans.** 104.93 MJ]

10. A pressure cooker contains 1.5 kg of steam at 5 bar and 0.9 dryness when the gas was switched off. Determine the quantity of heat rejected by the pressure cooker when the pressure in the cooker falls to 1 bar. [**Ans.** – 2355 kJ]

11. A vessel of spherical shape having a capacity of 0.8 m^3 contains steam at 10 bar and 0.95 dryness. Steam is blown off until the pressure drops to 5 bar. The valve is then closed and the steam is allowed to cool until the pressure falls to 4 bar. Assuming that the enthalpy of steam in the vessel remains constant during blowing off periods, determine :
   (*i*) The mass of steam blown-off,
   (*ii*) The dryness fraction of steam in the vessel after cooling, and
   (*iii*) The heat lost by steam per kg during cooling. [**Ans.** (*i*) 2.12 kg ; (*ii*) 0.78 ; (*iii*) – 820 kJ]

12. Two boilers one with superheater and other without superheater are delivering equal quantities of steam into a common main. The pressure in the boilers and the main is 15 bar. The temperature of the steam from a boiler with a superheater is 300°C and temperature of the steam in the main is 200°C. Determine the quality of steam supplied by the other boiler. [**Ans.** 0.89]

13. A tank of capacity 0.5 m^3 is connected to a steam pipe through a valve which carries steam at 14 bar and 300°C. The tank initially contains steam at 3.5 bar and saturated condition. The valve in the line connecting the tank is opened and the steam is allowed to pass into the tank until the pressure in the tank becomes 14 bar.
   Determine the mass of steam that entered into the tank. [**Ans.** 1.565 kg]

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3 First Law of Thermodynamics

3.1. Internal energy. 3.2. Law of conservation of energy. 3.3. First law of thermodynamics. 3.4. Application of first law to a process. 3.5. Energy—a property of system. 3.6. Perpetual motion machine of the first kind—PMM 1. 3.7. Energy of an isolated system. 3.8. The perfect gas—The characteristic equation of state—Specific heats—Joule's law—Relationship between two specific heats—Enthalpy—Ratio of specific heats. 3.9. Application of first law of thermodynamics to non-flow or closed system. 3.10. Application of first law to steady flow process. 3.11. Energy relations for flow process. 3.12. Engineering applications of steady flow energy equation (S.F.E.E.)—Water turbine—Steam or gas turbine—Centrifugal water pump—Centrifugal compressor—Reciprocating compressor—Boiler—Condenser—Evaporator—Steam nozzle. 3.13. Throttling process and Joule-Thompson porous plug experiment. 3.14. Heating-cooling and expansion of vapours. 3.15. Unsteady flow processes. Additional/Typical Worked Examples—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

3.1. INTERNAL ENERGY

It is the heat energy stored in a gas. If a certain amount of heat is supplied to a gas the result is that temperature of gas may increase or volume of gas may increase thereby doing some external work or both temperature and volume may increase ; but it will be decided by the conditions under which the gas is supplied heat. *If during heating of the gas the temperature increases its internal energy will also increase.*

Joule's law of internal energy states that internal energy of a perfect gas is a *function of temperature only.* In other words, internal energy of a gas is dependent on the temperature change only and is not affected by the change in pressure and volume.

We do not know how to find the absolute quantity of internal energy in any substance ; however, what is needed in engineering is the change of internal energy (ΔU).

3.2. LAW OF CONSERVATION OF ENERGY

In the early part of nineteenth century the scientists developed the concept of energy and hypothesis that it can be neither created nor destroyed ; this came to be known as the *law of the conservation of energy*. The first law of thermodynamics is merely one statement of this general law/principle with particular reference to heat energy and mechanical energy *i.e.*, work.

3.3. FIRST LAW OF THERMODYNAMICS

It is observed that when a system is made to undergo a complete cycle then network is done *on* or *by* the system. Consider a cycle in which network is done by the system. Since energy cannot be created, this mechanical energy must have been supplied from some source of energy. Now the system has been returned to its initial state : Therefore, its *intrinsic* energy is unchanged, and hence the mechanical energy has not been provided by the system itself. The only other energy

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involved in the cycle is the heat which was supplied and rejected in various processes. Hence, by the law of conservation of energy, the network done by the system is equal to the net heat supplied to the system. The First Law of Thermodynamics can, therefore, be stated as follows :

"When a system undergoes a thermodynamic cycle then the net heat supplied to the system from the surroundings is equal to network done by the system on its surroundings".

or $$\oint dQ = \oint dW$$

where $\oint$ represents the sum for a complete cycle.

The first law of Thermodynamics *cannot be proved analytically, but experimental evidence has repeatedly confirmed its validity*, and since no phenomenon has been shown to contradict it, the first law is accepted as a *law of nature.* It may be remarked that no restriction was imposed which limited the application of first law to reversible energy transformation. Hence, the first law applies to reversible as well as irreversible transformations : For non-cyclic process, a more general formulation of first law of thermodynamics is required. A new concept which involves a term called *internal energy* fulfills this need.

— The First Law of Thermodynamics may also be stated as follows :

"Heat and work are mutually convertible but since energy can neither be created nor destroyed, the total energy associated with an energy conversion remains constant".

Or

— **"No machine can produce energy without corresponding expenditure of energy, *i.e.*, it is impossible to construct a perpetual motion machine of first kind".**

Fig. 3.1 shows the experiment for checking first law of thermodynamics.

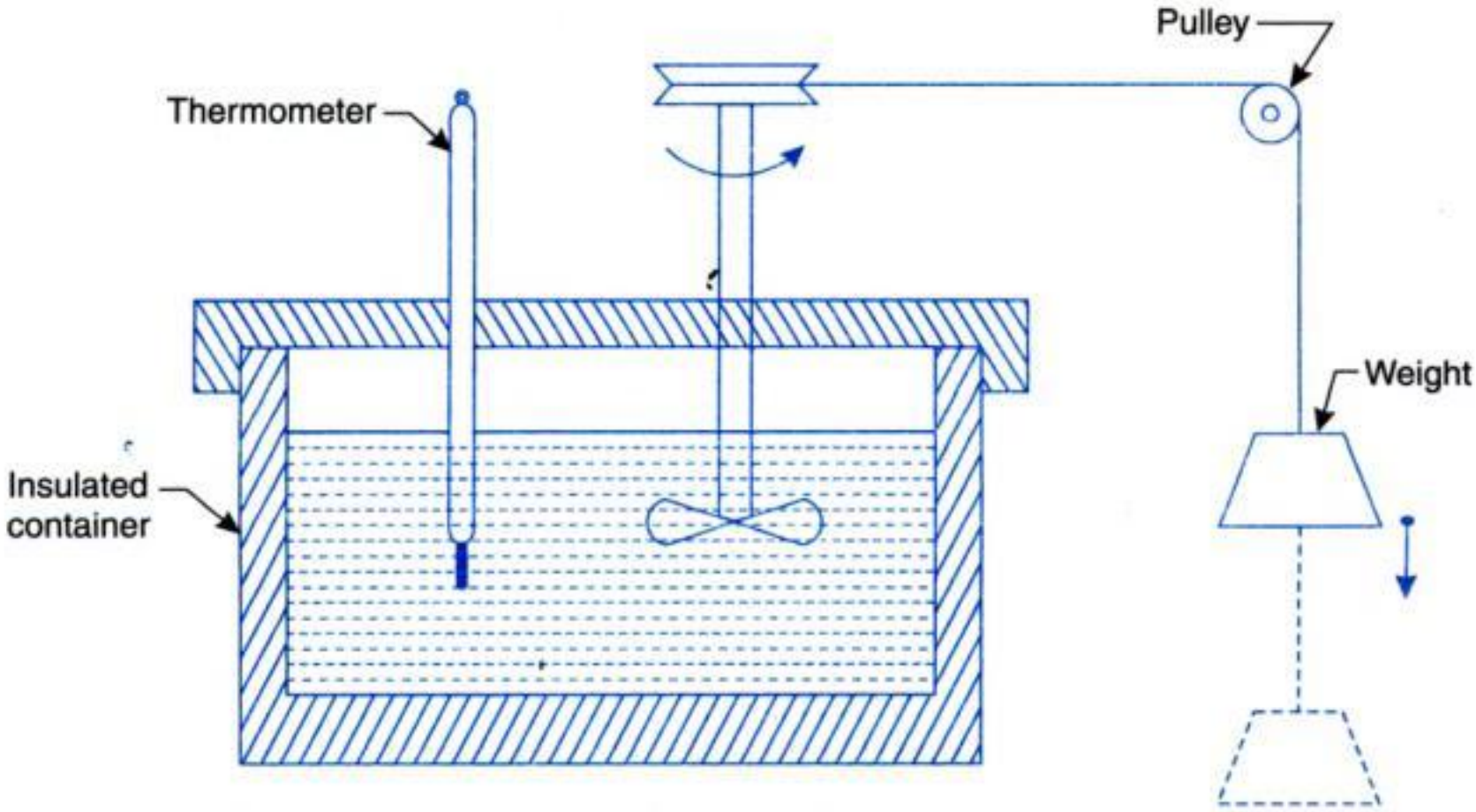

Fig. 3.1. Heat and work.

The work input to the paddle wheel is measured by the fall of weight, while the corresponding temperature rise of liquid in the insulated container is measured by the thermometer. It is already known to us from experiments on heat transfer that temperature rise can also be produced by heat transfer. The experiments show : (*i*) A definite quantity of work is always required to

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accomplish the same temperature rise obtained with a unit amount of heat. (*ii*) Regardless of whether the temperature of liquid is raised by work transfer or heat transfer, the liquid can be returned by heat transfer in opposite direction to the identical state from which it started. The above results lead to the inference that *work and heat* are different forms of something more general, which is called *energy*.

— It can be stated as an invariable experience that whenever a physical system passes through a complete cycle the algebraic sum of the work transfers during the cycle $\oint dW$ bears a definite ratio to the algebraic sum of the heat transfers during the cycle, $\oint dQ$. This may be expressed by the equation,

$$\oint dW = J \oint dQ \qquad \ldots(3.1)$$

where J is the proportionality constant and is known as *Mechanical Equivalent of heat*. In S.I. units its value is unity, *i.e.*, 1 Nm/J.

3.4. APPLICATION OF FIRST LAW TO A PROCESS

When a process is executed by a system, the *change in stored energy of the system is numerically equal to the net heat interactions minus the network interaction during the process.*

$$\therefore \quad E_2 - E_1 = Q - W$$

$$\therefore \quad \Delta E = Q - W \qquad [\text{or } Q = \Delta E + W]$$

or
$$\int_1^2 d(Q - W) = \Delta E = E_2 - E_1 \qquad \ldots(3.2)$$

where E represents the *total internal energy.*

If the electric, magnetic and chemical energies are *absent* and changes in potential and kinetic energy for a *closed system* are neglected, the above equation can be written as

$$\int_1^2 d(Q - W) = \Delta U = U_2 - U_1 \qquad \ldots(3.3)$$

$$\therefore \quad Q - W = \Delta U = U_2 - U_1 \qquad \ldots(3.4)$$

Generally, when heat is added to a system its temperature rises and external work is performed due to increase in volume of the system. *The rise in temperature is an indication of increase of internal energy.*

Heat *added* to the system will be considered as *positive* and the heat *removed* or *rejected*, from the system, as *negative.*

3.5. ENERGY—A PROPERTY OF SYSTEM

Consider a system which changes its state from state 1 to state 2 by following the path L, and returns from state 2 to state 1 by following the path M (Fig. 3.2). So the system undergoes a cycle. Writing the first law for path L

$$Q_L = \Delta E_L + W_L \qquad \ldots(3.5)$$

and for path M

$$Q_M = \Delta E_M + W_M \qquad \ldots(3.6)$$

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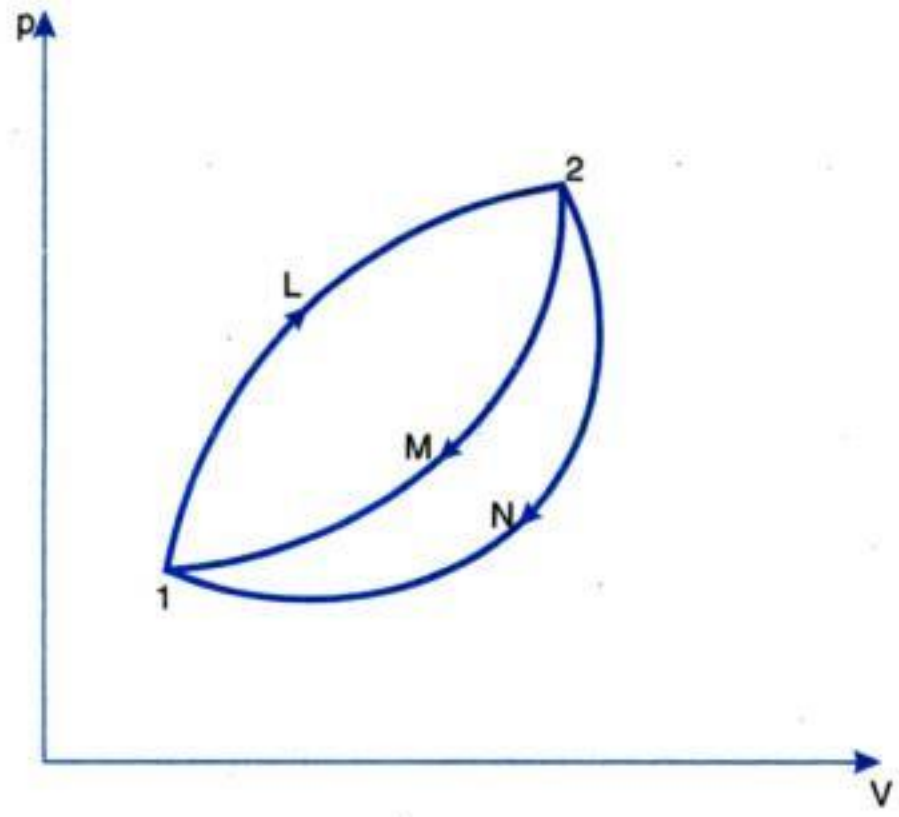

Fig. 3.2. Energy—a property of system.

The processes L and M together constitute a cycle, for which

$$\oint dW = \oint dQ$$

$$W_L + W_M = Q_L + Q_M$$

or
$$Q_L - W_L = W_M - Q_M \qquad ...(3.7)$$

From equations (3.5), (3.6) and (3.7), it yields

$$\Delta E_L = - \Delta E_M \qquad ...(3.8)$$

Similarly, had the system returned from state 2 to state 1 by following the path N instead of path M

$$\Delta E_L = - \Delta E_N \qquad ...(3.9)$$

From equations (3.8) and (3.9),

$$\Delta E_M = \Delta E_N \qquad ...(3.10)$$

Thus, it is seen that the *change in energy between two states of a system is the same, whatever path the system may follow in undergoing that change of state.* If some arbitrary value of energy is assigned to state 2, the value of energy at state 1 is fixed independent of the path the system follows. Therefore, *energy has a definite value for every state of the system.* Hence, it is *a point function and a property of the system.*

3.6. PERPETUAL MOTION MACHINE OF THE FIRST KIND—PMM 1

The first law of thermodynamics states the general principle of the conservation of energy. Energy is neither created nor destroyed, but only gets transformed from one form to another. *There can be no machine which would continuously supply mechanical work without some form of energy disappearing simultaneously* (Fig. 3.3). Such a *fictitious machine* is called a ***perpetual motion machine of the first kind,*** or in brief, PMM 1. A PMM 1 is thus ***impossible.***

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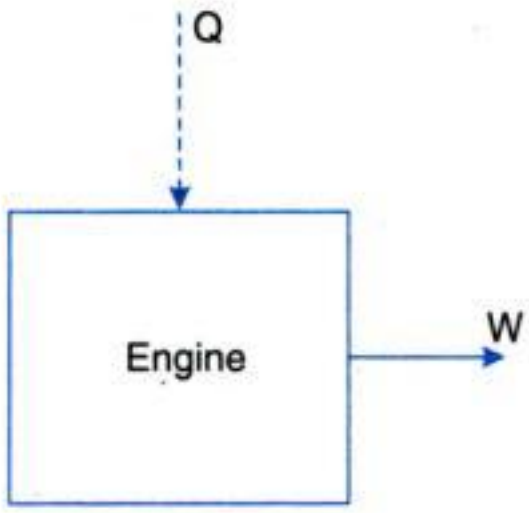

Fig. 3.3. A PPM 1.

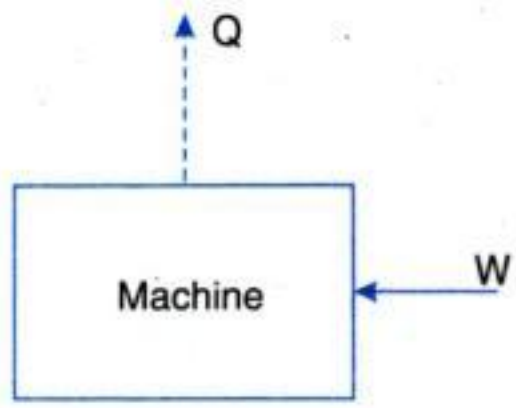

Fig. 3.4. The converse of PMM 1.

— The converse of the above statement is also true, *i.e.*, there can be no machine which would continuously consume work without some other form of energy appearing simultaneously (Fig. 3.4).

3.7. ENERGY OF AN ISOLATED SYSTEM

An isolated system is one in which there is no interaction of the system with the surroundings.

For an isolated system,

$$dQ = 0,\ dW = 0$$

The first law of thermodynamics gives

$$dE = 0$$

or

$$E = \text{constant}$$

The energy of an isolated system is always constant.

3.8. THE PERFECT GAS

3.8.1. The Characteristic Equation of State

— At temperatures that are considerably in excess of critical temperature of a fluid, and also at very low pressure, the vapour of fluid tends to obey the equation

$$\frac{pv}{T} = \text{constant} = R$$

In practice, no gas obeys this law rigidly, but many gases tend towards it.

An imaginary ideal gas which obeys this law is called a *perfect gas*, and the equation $\frac{pv}{T} = R$, is called the *characteristic equation of a state of a perfect gas.* The constant R is called the *gas constant*. Each perfect gas has a different gas constant.

Units of R are Nm/kg K or kJ/kg K.

Usually, the characteristic equation is written as

$$pv = RT \qquad ...(3.11)$$

or for m kg, occupying V m^3

$$pV = mRT \qquad ...(3.12)$$

— The characteristic equation in *another form*, can be derived by using kilogram-mole as a unit.

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The *kilogram-mole* is defined as a quantity of a gas equivalent to M kg of the gas, where M is the molecular weight of the gas (*e.g.*, since the molecular weight of oxygen is 32, then 1 kg mole of oxygen is equivalent to 32 kg of oxygen).

As per definition of the kilogram-mole, for m kg of a gas, we have

$$m = nM \qquad ...(3.13)$$

where n = number of moles.

Note. Since the standard of mass is the kg, kilogram-mole will be written simply as mole.

Substituting for m from eqn. (3.13) in eqn. (3.12) gives

$$pV = nMRT$$

or
$$MR = \frac{pV}{nT}$$

According to *Avogadro's hypothesis* the volume of 1 mole of any gas is the same as the volume of 1 mole of any other gas, when the gases are at the same temperature and pressure. Therefore, $\frac{V}{n}$ is the same for all gases at the same value of p and T. That is the quantity $\frac{pV}{nT}$ is a *constant* for all gases. This constant is called *universal gas constant*, and is given the symbol, R_0.

i.e.,
$$MR = R_0 = \frac{pV}{nT}$$

or
$$pV = nR_0T \qquad ...(3.14)$$

Since $MR = R_0$, then

$$R = \frac{R_0}{M} \qquad ...(3.15)$$

It has been found experimentally that the volume of 1 mole of any perfect gas at 1 bar and 0°C is approximately 22.71 m^3.

Therefore, from eqn. (3.14),

$$R_0 = \frac{pV}{nT} = \frac{1 \times 10^5 \times 22.71}{1 \times 273.15}$$

$$= 8314.3 \text{ Nm/mole K}$$

Using eqn. (3.15), the gas constant for any gas can be found when the molecular weight is known.

Example. For oxygen which has a molecular weight of 32, the gas constant

$$R = \frac{R_0}{M} = \frac{8314}{32} = 259.8 \text{ Nm/kg K.}$$

3.8.2. Specific Heats

— The specific heat of a solid or liquid is usually defined as the *heat required to raise unit mass through one degree temperature rise.*

— For small quantities, we have

$$dQ = mcdT$$

where m = mass,

c = specific heat, and

dT = temperature rise.

For a gas there are an infinite number of ways in which heat may be added between any two temperatures, and hence a *gas could have an infinite number of specific heats.* However, only two specific heats for gases are defined.

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Specific heat at constant volume, c_v

and *Specific heat at constant pressure,* c_p.

We have

$dQ = m\, c_p\, dT$ For a reversible non-flow process at *constant pressure* ...(3.16)

and $dQ = m\, c_v\, dT$ For a reversible non-flow process at *constant volume* ...(3.17)

The values of c_p and c_v, for a perfect gas, are constant for anyone gas at all pressures and temperatures. Hence, integrating eqns. (3.16) and (3.17), we have

Flow of heat in a reversible constant pressure process

$$= mc_p\,(T_2 - T_1) \qquad ...(3.18)$$

Flow of heat in a reversible constant volume process

$$= mc_v\,(T_2 - T_1) \qquad ...(3.19)$$

In case of *real gases,* c_p and c_v *vary with temperature,* but a suitable *average value* may be used for most practical purposes.

3.8.3. Joule's Law

Joule's law states as follows :

"The internal energy of a perfect gas is a function of the absolute temperature only."

i.e., $u = f(T)$

To evaluate this function let 1 kg of a perfect gas be heated at constant volume.

According to non-flow energy equation,

$$dQ = du + dW$$

$dW = 0$, since volume remains constant

$$\therefore \quad dQ = du$$

At constant volume for a perfect gas, from eqn. (3.17), for 1 kg

$$dQ = c_v dT$$

$$\therefore \quad dQ = du = c_v dT$$

and integrating $u = c_v\, T + K$, K being constant.

According to Joule's law $u = f(T)$, which means that internal energy varies linearly with absolute temperature. Internal energy can be made zero at any arbitrary reference temperature. For a perfect gas it can be assumed that $u = 0$ when $T = 0$, hence constant K is zero.

i.e., Internal energy, $u = c_v\, T$ for a perfect gas ...(3.20)

or For mass m, of a perfect gas

Internal energy, $U = mc_v\, T$ ...(3.21)

For a perfect gas, in any process between states 1 and 2, we have from eqn. (3.21)

Gain in internal energy,

$$U_2 - U_1 = mc_v\,(T_2 - T_1) \qquad ...(3.22)$$

Eqn. (3.22) gives the gains of internal energy for a perfect gas between two states *for any process, reversible* or *irreversible.*

3.8.4. Relationship between Two Specific Heats

Consider a perfect gas being heated at constant pressure from T_1 to T_2.

According to non-flow equation,

$$Q = (U_2 - U_1) + W$$

Also for a perfect gas,

$$U_2 - U_1 = mc_v\,(T_2 - T_1)$$

$$Q = mc_v\,(T_2 - T_1) + W$$

In a constant pressure process, the work done by the fluid,

$$W = p(V_2 - V_1)$$

$$= mR(T_2 - T_1) \qquad \left[\because \begin{array}{l} p_1V_1 = mRT_1 \\ p_2V_2 = mRT_2 \\ p_1 = p_2 = p \text{ in this case} \end{array}\right]$$

On substituting

$$Q = mc_v\,(T_2 - T_1) + mR\,(T_2 - T_1) = m(c_v + R)\,(T_2 - T_1)$$

But for a constant pressure process,

$$Q = mc_p\,(T_2 - T_1)$$

By equating the two expressions, we have

$$m(c_v + R)(T_2 - T_1) = mc_p(T_2 - T_1)$$

$$\therefore \quad c_v + R = c_p$$

or
$$c_p - c_v = R \qquad \text{...(3.23)}$$

Dividing both sides by c_v, we get

$$\frac{c_p}{c_v} - 1 = \frac{R}{c_v}$$

$$\therefore \quad c_v = \frac{R}{\gamma - 1} \qquad \text{...[3.23 (a)]}$$

(where $\gamma = c_p/c_v$)

Similarly, dividing both sides by c_p, we get

$$c_p = \frac{\gamma R}{\gamma - 1} \qquad \text{...[3.23 (b)]}$$

$$\left[\begin{array}{l} \text{In M.K.S. units : } c_p - c_v = \dfrac{R}{J};\ c_v = \dfrac{R}{J(\gamma - 1)},\ c_p = \dfrac{\gamma R}{(\gamma - 1)J} \\ \text{In S.I. units the value of } J \text{ is unity.} \end{array}\right]$$

3.8.5. Enthalpy

— One of the fundamental quantities which occur invariably in thermodynamics is the sum of internal energy (u) and pressure volume product (pv). This sum is called **Enthalpy** (h).

i.e., $$h = u + pv \qquad \text{...(3.24)}$$

— The enthalpy of a fluid is the property of the fluid, since it consists of the sum of a property and the product of the two properties. Since enthalpy is a property like internal energy, pressure, specific volume and temperature, it can be introduced into any problem whether the process is a flow or a non-flow process.

The total enthalpy of mass, m, of a fluid can be

$$H = U + pV, \text{ where } H = mh.$$

For a **perfect gas,**

Referring equation (3.24),

$$h = u + pv$$

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$$= c_v T + RT \qquad [\because \quad pv = RT]$$
$$= (c_v + R)T$$
$$= c_p T \qquad [\because \quad c_p = c_v + R]$$

i.e., $h = c_p T$

and $H = mc_p T$.

(Note that, since it has been assumed that $u = 0$ at $T = 0$, then $h = 0$ at $T = 0$).

3.8.6. Ratio of Specific Heats

The ratio of specific heat at constant pressure to the specific heat at constant volume is given the symbol γ (gamma).

i.e., $$\gamma = \frac{c_p}{c_v} \qquad ...(3.25)$$

Since $c_p = c_v + R$, it is clear that c_p must be *greater* than c_v for any perfect gas. It follows, therefore, that the ratio, $\frac{c_p}{c_v} = \gamma$ is *always greater than unity.*

In general, the approximate values of γ are as follows :

For *monoatomic* gases such as *argon, helium* = 1.6.

For *diatomic* gases such as *carbon monoxide, hydrogen, nitrogen* and *oxygen* = 1.4.

For triatomic gases such as *carbon dioxide* and *sulphur dioxide* = 1.3.

For some hydrocarbons the value of γ is quite low.

[*e.g.*, for ethane γ = 1.22, and for isobutane γ = 1.11]

3.9. APPLICATION OF FIRST LAW OF THERMODYNAMICS TO NON-FLOW OR CLOSED SYSTEM

1. Reversible Constant Volume Process (v = constant)

In a constant volume process the working substance is contained in a rigid vessel, hence the boundaries of the system are immovable and no work can be done on or by the system, other than paddle-wheel work input. It will be assumed that '*constant volume*' implies zero work unless stated otherwise.

Fig. 3.5 shows the system and states before and after the heat addition at constant volume.

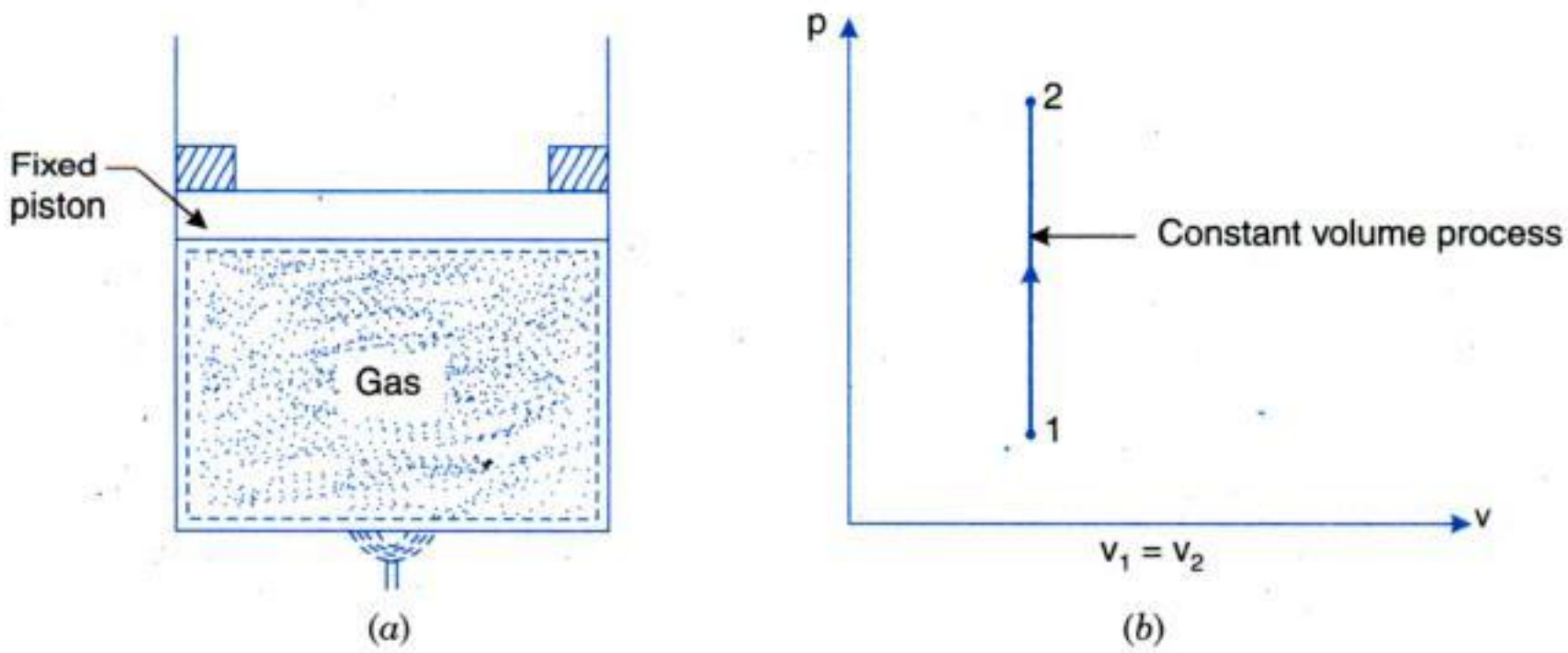

Fig. 3.5. Reversible constant volume process.

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Considering mass of the working substance *unity* and applying first law of thermodynamics to the process

$$Q = (u_2 - u_1) + W \qquad ...(3.26)$$

The work done $W = \int_1^2 p dv = 0$ as $dv = 0$.

$$\therefore \qquad Q = (u_2 - u_1) = c_v(T_2 - T_1) \qquad ...[3.27\ (a)]$$

where c_v = Specific heat at constant volume.

For mass, m, of working substance

$$Q = U_2 - U_1 = mc_v(T_2 - T_1) \qquad ...[3.27\ (b)]$$

$[\because \quad mu = U]$

2. Reversible Constant Pressure Process (p = constant)

It can be seen from Fig. 3.5 (b) that when the boundary of the system is *inflexible* as in a constant volume process, then the pressure rises when heat is supplied. Hence, for a constant pressure process, the boundary must move against an external resistance as heat is supplied ; for instance a gas [Fig. 3.6 (a)] in a cylinder behind a piston can be made to undergo a constant pressure process. Since the *piston is pushed through a certain distance* by the force exerted by the gas, then the work is done by the gas on its surroundings.

Fig. 3.6 shows the system and states before and after the heat addition at constant pressure.

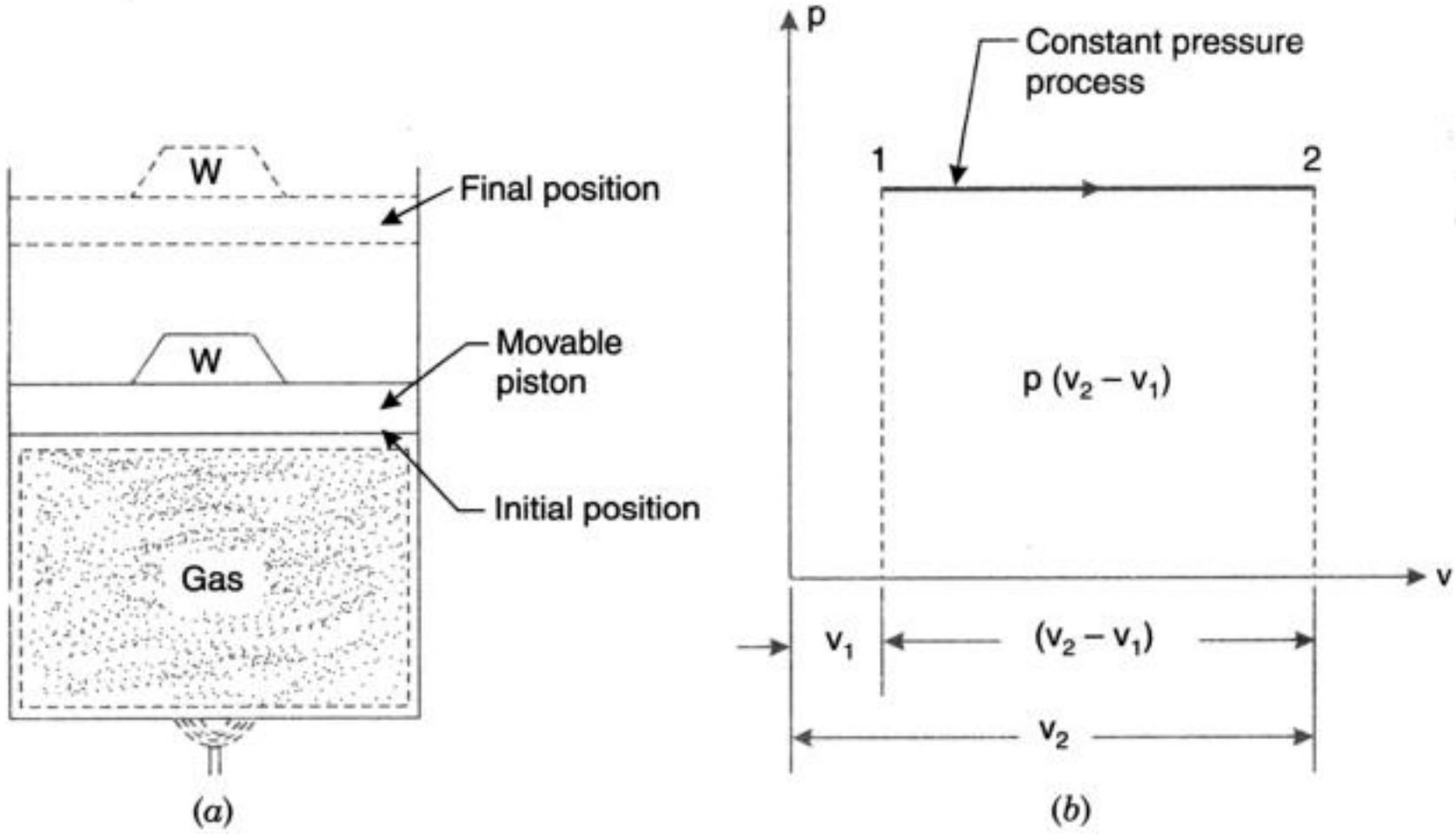

Fig. 3.6. Reversible constant pressure process.

Considering unit mass of working substance and applying first law of thermodynamics to the process

$$Q = (u_2 - u_1) + W$$

The work done, $W = \int_1^2 p dv = p(v_2 - v_1)$

$$\therefore \qquad Q = (u_2 - u_1) + p(v_2 - v_1) = u_2 - u_1 + pv_2 - pv_1$$

$$= (u_2 + pv_2) - (u_1 + pv_1) = h_2 - h_1 \qquad [\because \quad h = u + pv]$$

or $$Q = h_2 - h_1 = c_p (T_2 - T_1) \qquad ...(3.28)$$

where h = Enthalpy (specific), and

c_p = Specific heat at constant pressure.

For mass, m, of working substance

$$Q = H_2 - H_1 = mc_p (T_2 - T_1) \qquad ...[3.28\ (a)]$$

$$[\because \quad mh = H]$$

3. Reversible Temperature or Isothermal Process (pv = constant, T = constant)

A *process at a constant temperature is called an isothermal process.* When a working substance in a cylinder behind a piston expands from a high pressure to a low pressure there is a tendency for the temperature to fall. In an isothermal expansion heat must be added continuously in order to keep the temperature at the initial value. Similarly in an isothermal compression heat must be removed from the working substance continuously during the process.

Fig. 3.7 shows the system and states before and after the heat addition at constant temperature.

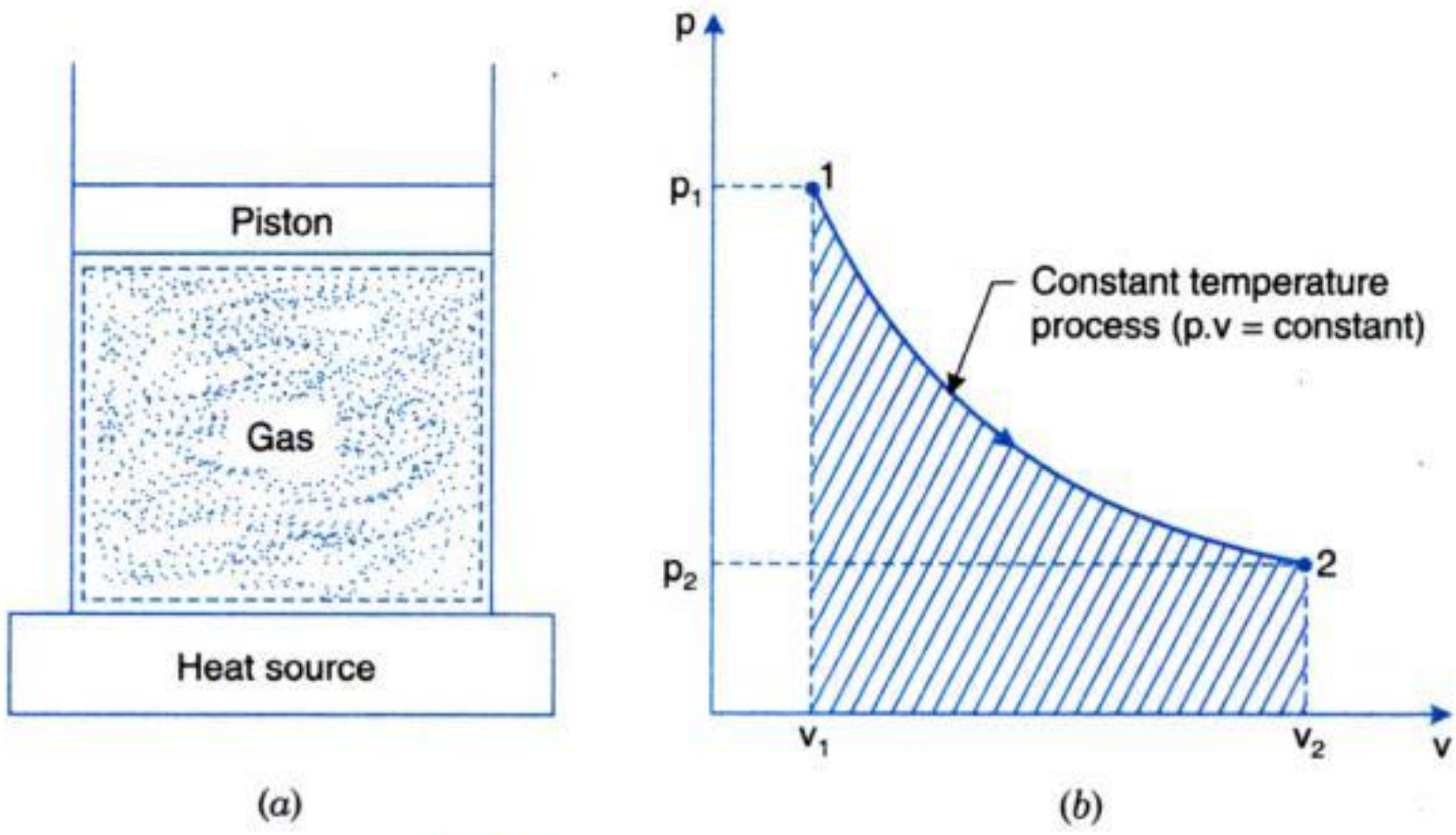

Fig. 3.7. Reversible isothermal process.

Considering unit mass of working substance and applying first law to the process

$$Q = (u_2 - u_1) + W$$

$$= c_v (T_2 - T_1) + W$$

$$= 0 + W \qquad [\because \quad T_2 = T_1]$$

The work done, $W = \int_1^2 p dv$

In this case pv = constant or $p = \dfrac{C}{v}$ (where C = constant)

$$\therefore \qquad W = \int_{v_1}^{v_2} C \frac{dv}{v} = C\left[\log_e v\right]_{v_1}^{v_2} = C \log_e \frac{v_2}{v_1}$$

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The constant C can either be written as p_1v_1 or as p_2v_2, since

$$p_1v_1 = p_2v_2 = \text{constant, } C$$

i.e., $$W = p_1v_1 \log_e \frac{v_2}{v_1} \text{ per unit mass of working substance}$$

or $$W = p_2v_2 \log_e \frac{v_2}{v_1} \text{ per unit mass of working substance}$$

$$\therefore \quad Q = W = p_1v_1 \log_e \frac{v_2}{v_1} \qquad \text{...(3.29)}$$

For mass, m, of the working substance

$$Q = p_1V_1 \log_e \frac{V_2}{V_1} \qquad \text{...[3.29 (a)]}$$

or $$Q = p_1V_1 \log_e \frac{p_1}{p_2} \quad \left[\because \frac{V_2}{V_1} = \frac{p_1}{p_2}\right] \qquad \text{...[3.29 (b)]}$$

4. Reversible Adiabatic Process (pv^γ = constant)

An **adiabatic process** *is one in which no heat is transferred to or from the fluid during the process.* Such a process can be reversible or irreversible. The reversible adiabatic non-flow process will be considered in this section.

Considering unit mass of working substance and applying first law to the process

$$Q = (u_2 - u_1) + W$$

$$O = (u_2 - u_1) + W$$

or $$W = (u_1 - u_2) \text{ for any adiabatic process} \qquad \text{...(3.30)}$$

Eqn. (3.30) is true for an adiabatic process whether the process is reversible or not. In an adiabatic expansion, the work done W by the fluid is at the expense of a reduction in the internal energy of the fluid. Similarly in an adiabatic compression process all the work done on the fluid goes to increase the internal energy of the fluid.

For an adiabatic process to take place, perfect thermal insulation for the system must be available.

To derive the law pv^γ = constant :

To obtain a law relating p and v for a reversible adiabatic process let us consider the non-flow energy equation in differential form,

$$dQ = du + dW$$

For a *reversible process*

$$dW = pdv$$

$$\therefore \quad dQ = du + pdv = 0$$

(Since for an adiabatic process $Q = 0$)

Also for a perfect gas

$$pv = RT \text{ or } p = \frac{RT}{v}$$

Hence substituting,

$$du + \frac{RTdv}{v} = 0$$

Also $$u = c_vT \text{ or } du = c_vdT$$

$$\therefore \quad c_vdT + \frac{RTdv}{v} = 0$$

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Dividing both sides by T, we get

$$c_v \frac{dT}{T} + \frac{R dv}{v} = 0$$

Integrating

$$c_v \log_e T + R \log_e v = \text{constant}$$

Substituting $T = \frac{pv}{R}$

$$c_v \log_e \frac{pv}{R} + R \log_e v = \text{constant}$$

Dividing throughout both sides by c_v

$$\log_e \frac{pv}{R} + \frac{R}{c_v} . \log_e v = \text{constant}$$

Again $$c_v = \frac{R}{(\gamma - 1)} \quad \text{or} \quad \frac{R}{c_v} = \gamma - 1$$

Hence substituting

$$\log_e \frac{pv}{R} + (\gamma - 1) \log_e v = \text{constant}$$

$$\therefore \quad \log_e \frac{pv}{R} + \log_e v^{\gamma - 1} = \text{constant}$$

$$\log_e \frac{pv \times v^{\gamma - 1}}{R} = \text{constant}$$

i.e., $$\log_e \frac{pv^\gamma}{R} = \text{constant}$$

i.e., $$\frac{pv^\gamma}{R} = e^{\text{constant}} = \text{constant}$$

or $$pv^\gamma = \text{constant} \qquad \text{...(3.31)}$$

Expression for work W :

A reversible adiabatic process for a perfect gas is shown on a p-v diagram in Fig. 3.8 (b).

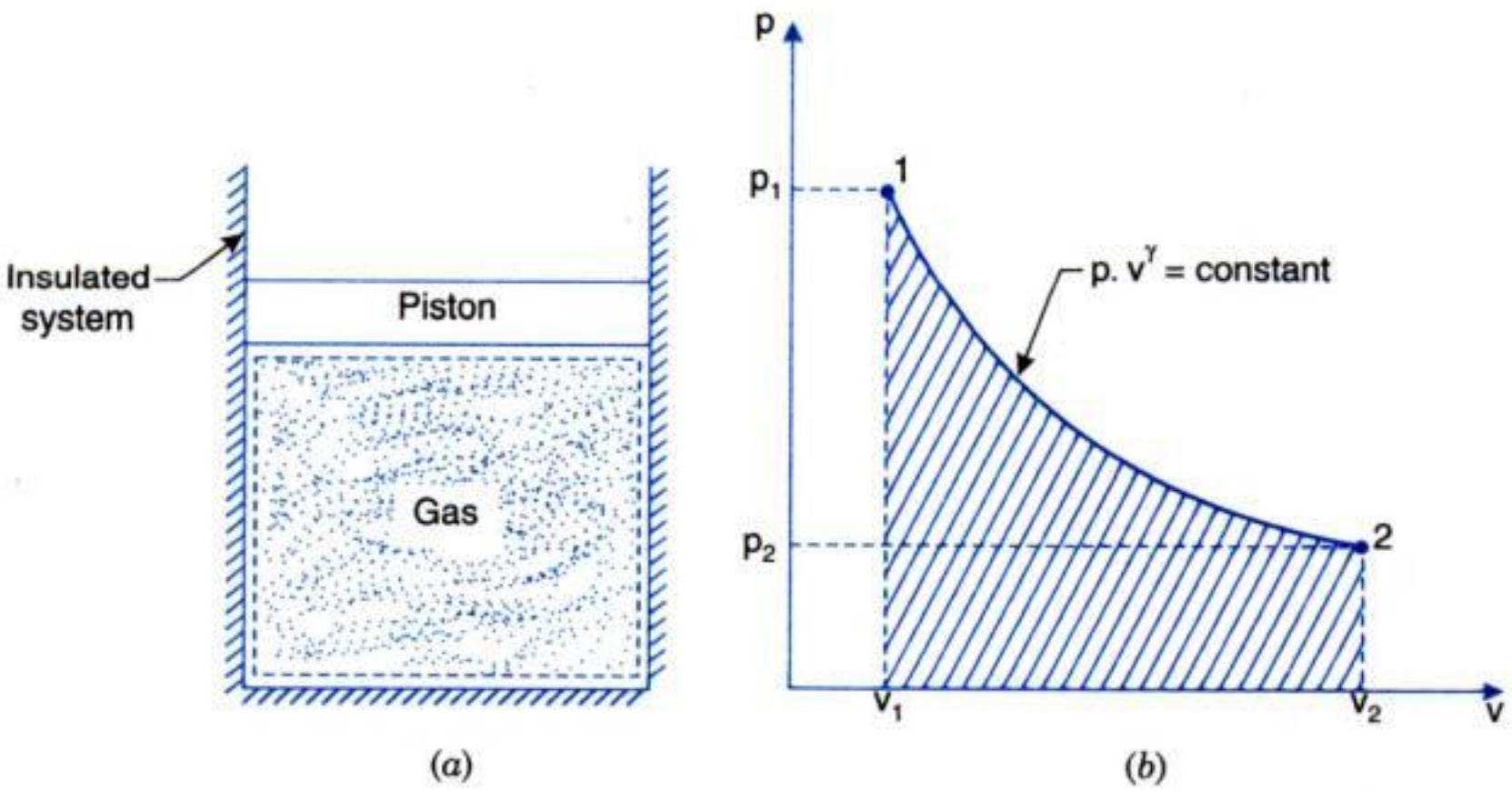

Fig. 3.8. Reversible adiabatic process.

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The work done is given by the shaded area, and this area can be evaluated by integration.

i.e.,
$$W = \int_{v_1}^{v_2} p\,dv$$

Therefore, since pv^γ = constant, C, then

$$W = \int_{v_1}^{v_2} C\,\frac{dv}{v^\gamma} \qquad \left[\because\ p = \frac{C}{v^\gamma}\right]$$

i.e.,
$$W = C\int_{v_1}^{v_2} \frac{dv}{v^\gamma} = C\left|\frac{v^{-\gamma+1}}{-\gamma+1}\right|_{v_1}^{v_2}$$

$$= C\left(\frac{{v_2}^{-\gamma+1} - {v_1}^{-\gamma+1}}{1-\gamma}\right) = C\left(\frac{{v_1}^{-\gamma+1} - {v_2}^{-\gamma+1}}{\gamma-1}\right)$$

The constant in this equation can be written as $p_1{v_1}^\gamma$ or as $p_2{v_2}^\gamma$. Hence,

$$W = \frac{p_1\,{v_1}^\gamma {v_1}^{-\gamma+1} - p_2\,{v_2}^\gamma {v_2}^{-\gamma+1}}{\gamma-1} = \frac{p_1v_1 - p_2v_2}{\gamma-1}$$

i.e.,
$$W = \frac{p_1v_1 - p_2v_2}{\gamma-1} \qquad \text{...(3.32)}$$

or
$$W = \frac{R(T_1 - T_2)}{\gamma-1} \qquad \text{...(3.33)}$$

Relationship between T and v, and T and p :

By using equation $pv = RT$, the relationship between T and v, and T and p, may by derived as follows :

i.e.,
$$pv = RT$$

$\therefore$
$$p = \frac{RT}{v}$$

Putting this value in the equation pv^γ = constant

$$\frac{RT}{v}.v^\gamma = \text{constant}$$

i.e.,
$$Tv^{\gamma-1} = \text{constant} \qquad \text{...(3.34)}$$

Also $v = \frac{RT}{p}$; hence substituting in equation pv^γ = constant

$$p\left(\frac{RT}{p}\right)^\gamma = \text{constant}$$

$\therefore$
$$\frac{T^\gamma}{p^{\gamma-1}} = \text{constant}$$

or
$$\frac{T}{(p)^{\frac{\gamma-1}{\gamma}}} = \text{constant} \qquad \text{...(3.35)}$$

Therefore, for a reversible adiabatic process for a perfect gas between states 1 and 2, we can write :

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From eqn. (3.31),

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma} \quad \text{or} \quad \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} \qquad ...(3.36)$$

From eqn. (3.34),

$$T_1 v_1^{\gamma - 1} = T_2 v_2^{\gamma - 1} \text{ or } \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} \qquad ...(3.37)$$

From eqn. (3.35),

$$\frac{T_1}{(p_1)^{\frac{\gamma-1}{\gamma}}} = \frac{T_2}{(p_2)^{\frac{\gamma-1}{\gamma}}} \quad \text{or} \quad \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad ...(3.38)$$

From eqn. (3.30), the work done in an adiabatic process per kg of gas is given by $W = (u_1 - u_2)$. The gain in internal energy of a perfect gas is given by equation :

$$u_2 - u_1 = c_v (T_2 - T_1) \qquad \text{(for 1 kg)}$$

$$\therefore \quad W = c_v (T_1 - T_2)$$

Also, we know that

$$c_v = \frac{R}{\gamma - 1}$$

Hence substituting, we get

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

Using equation, $pv = RT$

$$W = \frac{p_1 v_1 - p_2 v_2}{\gamma - 1}$$

This is the same expression obtained before as eqn. (3.32).

5. Polytropic Reversible Process (pv^n = constant)

It is found that many processes in practice approximate to a reversible law of form pv^n = constant, where n is a constant. Both vapours and perfect gases obey this type of law closely in many non-flow processes. Such processes are *internally reversible*.

We know that for any reversible process,

$$W = \int p \, dv$$

For a process in pv^n = constant, we have

$$p = \frac{C}{v^n} \text{ , where } C \text{ is a constant}$$

$$\therefore \quad W = C \int_{v_1}^{v_2} \frac{dv}{v^n} = C \left| \frac{v^{-n+1}}{-n+1} \right| = C \left(\frac{v_2^{-n+1} - v_1^{-n+1}}{-n+1} \right)$$

i.e.,

$$W = C \left(\frac{v_1^{-n+1} - v_2^{-n+1}}{n-1} \right) = \frac{p_1 v_1^{n} v_1^{-n+1} - p_2 v_2^{n} v_2^{-n+1}}{n-1}$$

(since the constant C, can be written as $p_1 v_1^{n}$ or as $p_2 v_2^{n}$)

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i.e., Work done, $$W = \frac{p_1v_1 - p_2v_2}{n-1} \quad ...(3.39)$$

or $$W = \frac{R(T_1 - T_2)}{n-1} \quad ...(3.40)$$

Eqn. (3.39) is true for any working substance undergoing a reversible polytropic process. It follows also that for any polytropic process, we can write

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n \quad ...(3.41)$$

The following relations can be derived (following the same procedure as was done under reversible adiabatic process)

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1} \quad ...(3.42)$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \quad ...(3.43)$$

Heat transfer during polytropic process (for perfect gas pv = RT) :

Using non-flow energy equation, the heat flow/transfer during the process can be found,

i.e., $$Q = (u_2 - u_1) + W$$

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

i.e., $$Q = \frac{R(T_1 - T_2)}{n-1} - c_v(T_1 - T_2)$$

Also $$c_v = \frac{R}{(\gamma - 1)}$$

On substituting,

$$Q = \frac{R}{n-1}(T_1 - T_2) - \frac{R}{(\gamma-1)}(T_1 - T_2)$$

i.e., $$Q = R(T_1 - T_2)\left(\frac{1}{n-1} - \frac{1}{\gamma-1}\right)$$

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

$\therefore$ $$Q = \frac{(\gamma - n)}{(\gamma-1)}\frac{R(T_1 - T_2)}{(n-1)}$$

or $$Q = \left(\frac{\gamma - n}{\gamma - 1}\right)W \qquad \left[\because W = \frac{R(T_1 - T_2)}{(n-1)}\right] \quad ...(3.44)$$

In a polytropic process, the *index n depends only on the heat and work quantities* during the process. The various processes considered earlier are special cases of polytropic process for a perfect gas. For example,

When $n = 0$ pv^0 = constant *i.e.*, p = constant

When $n = \infty$ pv^∞ = constant

or $p^{1/\infty}\, v$ = constant, *i.e.*, v = constant

When $n = 1$ pv = constant, *i.e.*, T = constant

[since $(pv)/T$ = constant for a perfect gas]

When $n = \gamma$ pv^γ = constant, *i.e.*, reversible adiabatic

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This is illustrated on a p-v diagram in Fig. 3.9.

(*i*) State 1 to state A is *constant pressure cooling* ($n = 0$).

(*ii*) State 1 to state B is *isothermal compression* ($n = 1$).

(*iii*) State 1 to state C is *reversible adiabatic compression* ($n = \gamma$).

(*iv*) State 1 to state D is *constant volume heating* ($n = \infty$).

Similarly,

(*i*) State 1 to state A' is *constant pressure heating* ($n = 0$).

(*ii*) State 1 to state B' is *isothermal expansion* ($n = 1$).

(*iii*) State 1 to state C' is *reversible adiabatic expansion* ($n = \gamma$).

(*iv*) State 1 to state D' is *constant volume cooling* ($n = \infty$).

It may be noted that, since γ is always greater than unity, than process 1 to C must lie between processes 1 to B and 1 to D ; similarly, process 1 to C' must lie between processes 1 to B' and 1 to D'.

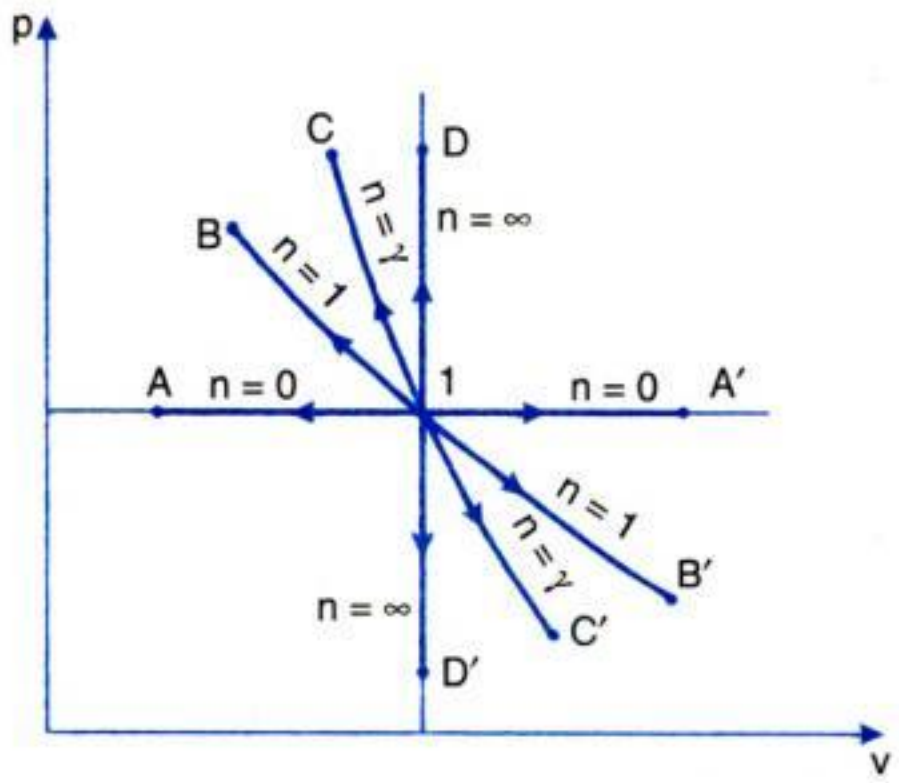

Fig. 3.9

6. Free Expansion

Consider two vessels 1 and 2 interconnected by a short pipe with a valve A, and perfectly thermally insulated [Fig. 3.10]. Initially let the vessel 1 be filled with a fluid at a certain pressure, and let 2 be completely evacuated. When the valve A is opened the fluid in 1 will expand rapidly to fill both vessels 1 and 2. The pressure finally will be lower than the initial pressure in vessel 1. This is known as *free or unresisted expansion*. The process is *highly irreversible* ; since the fluid is eddying continuously during the process. Now applying first law of thermodynamics (or non-flow energy equation) between the initial and final states,

$$Q = (u_2 - u_1) + W$$

In this process, no work is done on or by the fluid, since the boundary of the system does not move. No heat flows to or from the fluid since the system is well lagged. The process is therefore, *adiabatic but irreversible*.

i.e., $$u_2 - u_1 = 0 \quad \text{or} \quad u_2 = u_1$$

In a free expansion, therefore, the internal energy initially equals the initial energy finally.

For a perfect gas,

$$u = c_v T$$

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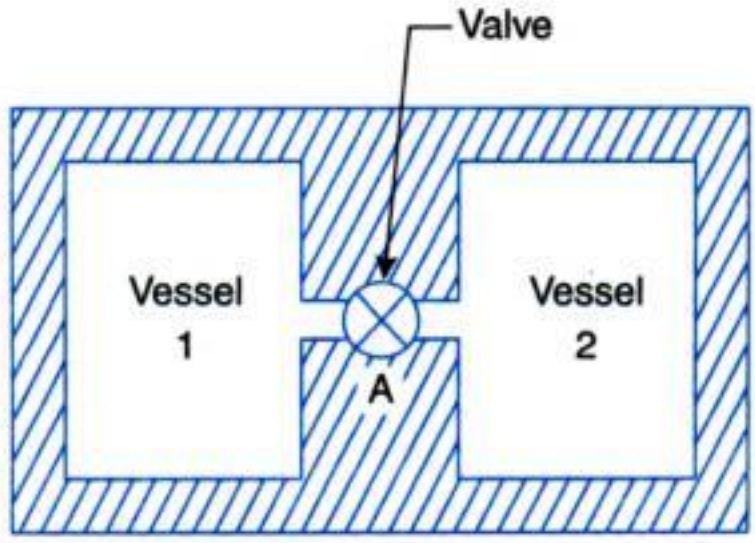

Fig. 3.10. Free expansion.

∴ For a free expansion of a perfect gas,

$$c_v T_1 = c_v T_2 \quad i.e., \qquad T_1 = T_2$$

That is, for a perfect gas undergoing a free expansion, the initial temperature is equal to the final temperature.

Table 3.1. Summary of Processes for Perfect Gas (Unit mass)

Process	*Index* n	*Heat added*	$\int_1^2 pdv$	p, v, T *relations*	*Specific heat, c*
Constant pressure	$n = 0$	$c_p(T_2 - T_1)$	$p(v_2 - v_1)$	$\frac{T_2}{T_1} = \frac{v_2}{v_1}$	c_p
Constant volume	$n = \infty$	$c_v(T_2 - T_1)$	0	$\frac{T_1}{T_2} = \frac{p_1}{p_2}$	c_v
Constant temperature	$n = 1$	$p_1 v_1 \log_e \frac{v_2}{v_1}$	$p_1 v_1 \log_e \frac{v_2}{v_1}$	$p_1 v_1 = p_2 v_2$	∞
Reversible adiabatic	$n = \gamma$	0	$\frac{p_1 v_1 - p_2 v_2}{\gamma - 1}$	$p_1 v_1^{\gamma} = p_2 v_2^{\gamma}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$	0
Polytropic	$n = n$	$c_n(T_2 - T_1) = c_v\left(\frac{\gamma - n}{1 - n}\right) \times (T_2 - T_1)$ $= \frac{\gamma - n}{\gamma - 1} \times$ work done (non-flow)	$\frac{p_1 v_1 - p_2 v_2}{n - 1}$	$p_1 v_1^{n} = p_2 v_2^{n}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n - 1} = \left(\frac{p_2}{p_1}\right)^{\frac{n - 1}{n}}$	$c_n = c_v\left(\frac{\gamma - n}{1 - n}\right)$

Note. Equations must be used keeping dimensional consistence.

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Example 3.1. *In an internal combustion engine, during the compression stroke the heat rejected to the cooling water is 50 kJ/kg and the work input is 100 kJ/kg.*

Calculate the change in internal energy of the working fluid stating whether it is a gain or loss.

Solution. Heat rejected to the cooling water, $Q = -50$ kJ/kg

(–ve sign since heat is rejected)

Work input, $W = -100$ kJ/kg

(–ve sign since work is supplied to the system)

Using the relation, $Q = (u_2 - u_1) + W$

$$-50 = (u_2 - u_1) - 100$$

or $$u_2 - u_1 = -50 + 100 = 50 \text{ kJ/kg}$$

Hence, **gain in internal energy = 50 kJ/kg.** **(Ans.)**

Example 3.2. *In an air motor cylinder the compressed air has an internal energy of 450 kJ/kg at the beginning of the expansion and an internal energy of 220 kJ/kg after expansion. If the work done by the air during the expansion is 120 kJ/kg, calculate the heat flow to and from the cylinder.*

Solution. Internal energy at beginning of the expansion,

$$u_1 = 450 \text{ kJ/kg}$$

Internal energy after expansion,

$$u_2 = 220 \text{ kJ/kg}$$

Work done by the air during expansion,

$$W = 120 \text{ kJ/kg}$$

Heat flow, Q :

Using the relation, $Q = (u_2 - u_1) + W$

$\therefore$ $Q = (220 - 450) + 120$

$= -230 + 120 = -110$ kJ/kg

Hence, **heat rejected by air = 110 kJ/kg.** **(Ans.)**

Example 3.3. *0.3 kg of nitrogen gas at 100 kPa and 40°C is contained in a cylinder. The piston is moved compressing nitrogen until the pressure becomes 1 MPa and temperature becomes 160°C. The work done during the process is 30 kJ.*

Calculate the heat transferred from the nitrogen to the surroundings.

c_v for nitrogen = 0.75 kJ/kg K.

Solution. Mass of nitrogen, $m = 0.3$ kg

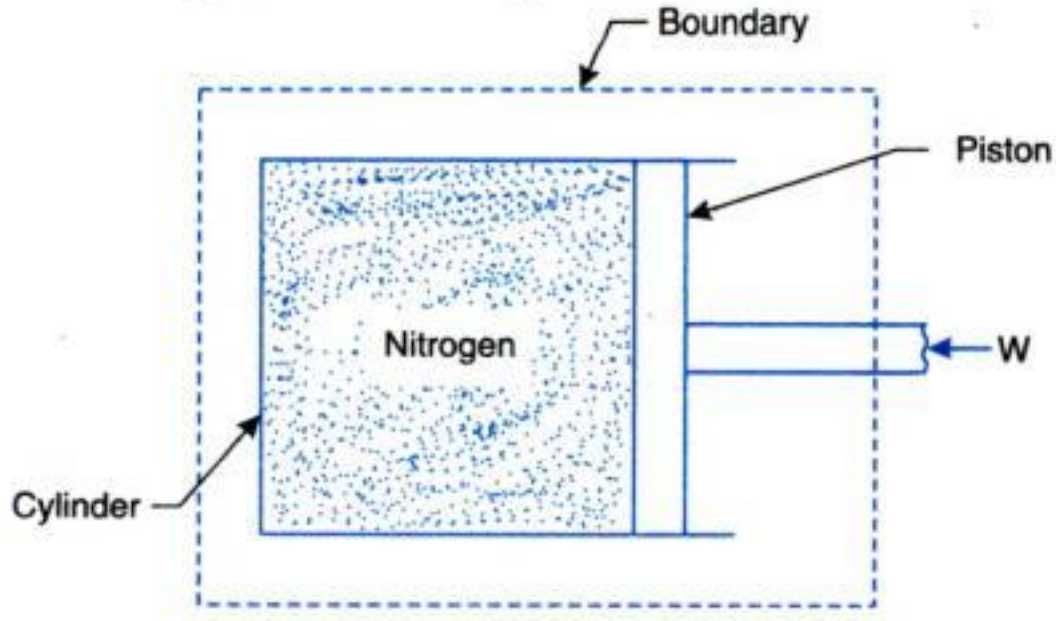

Fig. 3.11

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Temperature before compression = 40°C or 313 K

Temperature after compression = 160°C or 433 K

The work done during the compression process, $W = -30$ kJ

According to first law of thermodynamics,

$$Q = \Delta U + W = (U_2 - U_1) + W$$
$$= mc_v (T_2 - T_1) + W$$
$$= 0.3 \times 0.75(433 - 313) - 30 = -3 \text{ kJ}$$

Hence, **heat 'rejected' during the process = 3 kJ.** **(Ans.)**

Note. Work, W has been taken –ve because it has been supplied from outside.

Example 3.4. *When a stationary mass of gas was compressed without friction at constant pressure its initial state of 0.4 m^3 and 0.105 MPa was found to change to final state of 0.20 m^3 and 0.105 MPa. There was a transfer of 42.5 kJ of heat from the gas during the process.*

How much did the internal energy of the gas change ?

Solution.

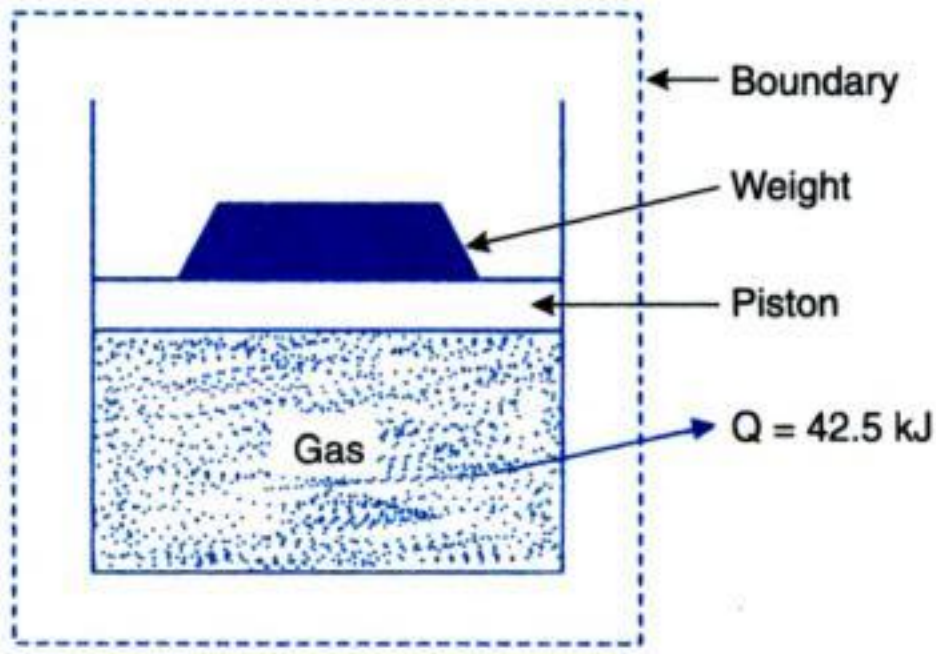

Fig. 3.12

Initial state

Pressure of gas, $p_1 = 0.105$ MPa

Volume of gas, $V_1 = 0.4$ m^3

Final state

Pressure of gas, $p_2 = 0.105$ MPa

Volume of gas, $V_2 = 0.20$ m^3

Process used : *Constant pressure*

Heat transferred, $Q = -42.5$ kJ

(–ve sign indicates that heat is rejected)

Change in internal energy, $\Delta U = U_2 - U_1$:

First law for a stationary system in a process gives

$$Q = \Delta U + W$$

or
$$Q_{1-2} = (U_2 - U_1) + W_{1-2} \quad ...(i)$$

Here
$$W_{1-2} = \int_{V_1}^{V_2} p dV = p(V_2 - V_1)$$

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$= 0.105(0.20 - 0.40)$ MJ $= -21$ kJ [$\because$ 1 MJ $= 10^3$ kJ]

Substituting this value of $W_{1\text{-}2}$ in equation (i), we get

$$-42.5 = (U_2 - U_1) - 21$$

$\therefore$ $$U_2 - U_1 = -42.5 + 21 = -21.5 \text{ kJ}$$

Hence **'decrease' in internal energy = 21.5 kJ.** **(Ans.)**

Example 3.5. *A container is divided into compartments by a partition. The container is completely insulated so that there is no heat transfer. One portion contains gas at temperature T_1 and pressure p_1 while the other portion also has the same gas but at temperature T_2 and pressure p_2.*

How will the First Law of Thermodynamics conclude the result if partition is removed ?

Solution. Refer Fig. 3.13.

According to First Law of Thermodynamics,

$$\delta Q = \delta U + \delta W$$

When partition removed, $\delta Q = 0$

$$\delta W = 0$$

$\therefore$ $$\delta U = 0.$$

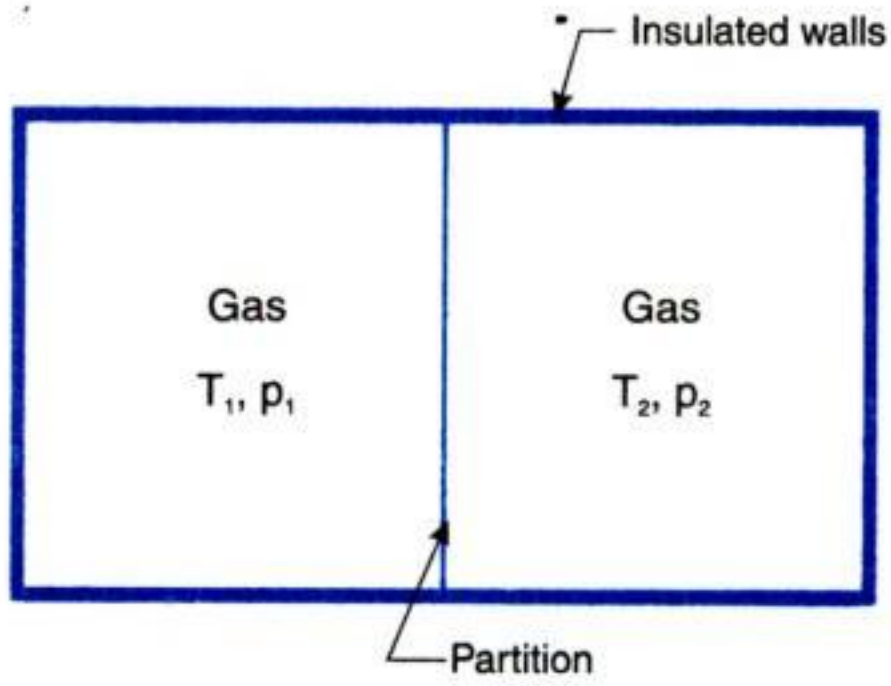

Fig. 3.13

Conclusion. *There is conservation of internal energy.*

Example 3.6. *Air enters a compressor at 10^5 Pa and 25°C having volume of 1.8 m^3/kg and is compressed to 5×10^5 Pa isothermally.*

Determine : (i) Work done ;

(ii) Change in internal energy ; and

(iii) Heat transferred.

Solution. Initial pressure of air, $p_1 = 10^5$ Pa

Initial temperature of air, $T_1 = 25 + 273 = 298$ K

Final pressure of air, $p_2 = 5 \times 10^5$ Pa

Final temperature of air, $T_2 = T_1 = 298$ K (isothermal process)

Since, it is a closed steady state process, we can write down the first law of thermodynamics as,

$$Q = (u_2 - u_1) + W \text{ .....per kg}$$

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(*i*) *For isothermal process* :

$$W_{1-2} = \int_1^2 p \, . \, dv = p_1 v_1 \log_e \left(\frac{p_1}{p_2} \right)$$

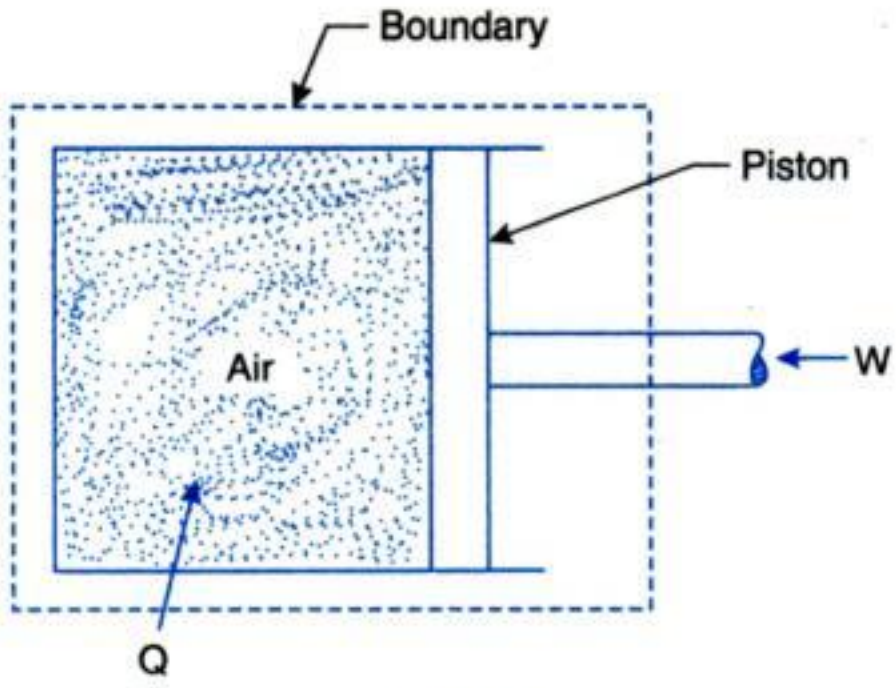

Fig. 3.14

as $p_1 v_1 = p_2 v_2$ for isothermal process

$\therefore$ $$W_{1-2} = - 10^5 \times 1.8 \log_e \left(\frac{1 \times 10^5}{5 \times 10^5} \right)$$

$$= - 2.897 \times 10^5 = - 289.7 \text{ kJ/kg.}$$

(– ve sign indicates that the work is supplied to the air)

$\therefore$ **Work done on the air = 289.7 kJ/kg.** **(Ans.)**

(*ii*) Since temperature is constant,

$\therefore$ $$u_2 - u_1 = 0$$

$\therefore$ **Change in internal energy = zero.** **(Ans.)**

(*iii*) Again, $$Q_{1-2} = (u_2 - u_1) + W$$

$$= 0 + (- 289.7) = - 289.7 \text{ kJ}$$

(– ve sign indicates that heat is lost from the system to the surroundings)

$\therefore$ **Heat rejected = 289.7 kJ/kg.** **(Ans.)**

Example 3.7. *A cylinder containing the air comprises the system. Cycle is completed as follows :*

(*i*) *82000 Nm of work is done by the piston on the air during compression stroke and 45 kJ of heat are rejected to the surroundings.*

(*ii*) *During expansion stroke 100000 Nm of work is done by the air on the piston. Calculate the quantity of heat added to the system.*

Solution. Refer Fig. 3.15.

Compression stroke. Process 1–2 :

Work done by the piston *on the air*, $W_{1-2} = - 82000$ Nm (= – 82 kJ)

Heat rejected to the system, $Q_{1-2} = - 45$ kJ

Now, $$Q_{1-2} = (U_2 - U_1) + W$$

$$- 45 = (U_2 - U_1) + (- 82)$$

$\therefore$ $$(U_2 - U_1) = 37 \text{ kJ} \qquad ...(i)$$

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Expansion stroke. Process 2–1 :

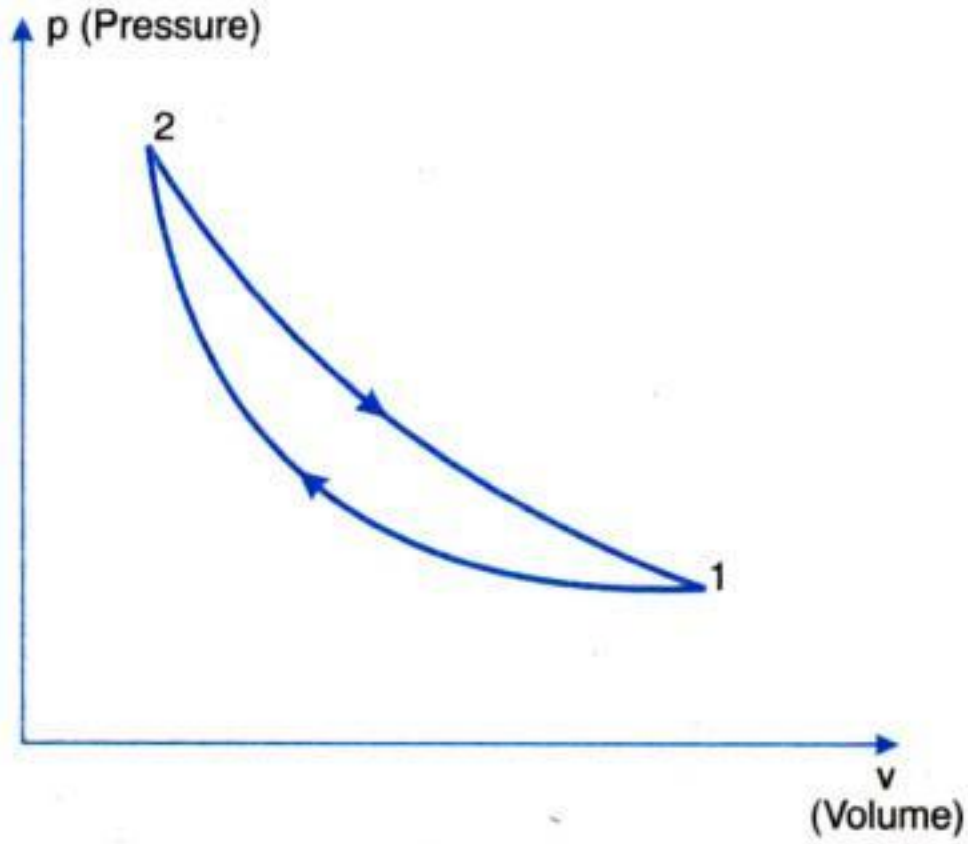

Fig. 3.15

Work done by air on the piston, W_{2-1} = 100000 N-m (= 100 kJ)

Now, $$Q_{2-1} = (U_1 - U_2) + W$$
$$= -37 + 100 \text{ kJ} = 63 \text{ kJ}$$

Hence, **quantity of heat added to the system = 63 kJ.** **(Ans.)**

☞ **Example 3.8.** *A tank containing air is stirred by a paddle wheel. The work input to the paddle wheel is 9000 kJ and the heat transferred to the surroundings from the tank is 3000 kJ.*

Determine : (i) Work done ;

(ii) Change in internal energy of the system.

Solution. Refer to Fig. 3.16.

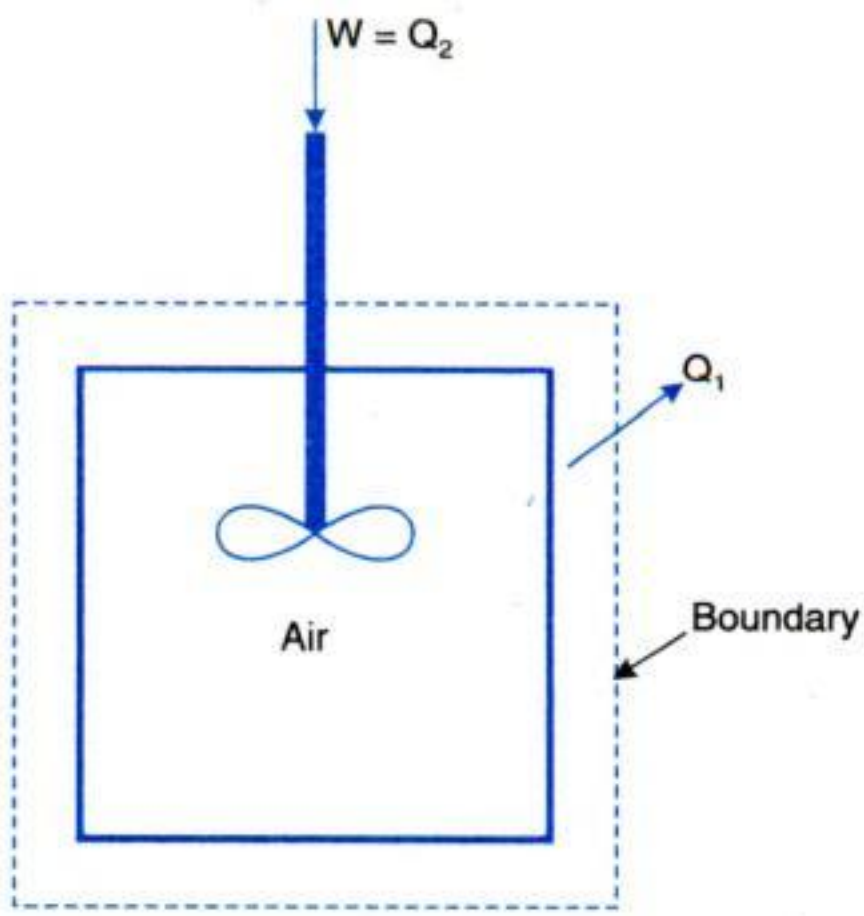

Fig. 3.16

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Work input to the paddle wheel = 9000 kJ

Heat transferred to the surroundings from the tank = 3000 kJ

As it is a closed system, the first law of thermodynamics can be written as

$$U_1 - Q + W = U_2 \qquad ...(i)$$

The *work* enters into the tank in the form of energy only so this should be considered as *heat input*.

$$\therefore \quad Q = Q_1 - Q_2$$
$$= 3000 - 9000 = -6000 \text{ kJ}.$$

(*i*) Since volume does not change (being constant volume process)

$\therefore$ Work done, $W = 0$

Putting the value of $W = 0$ in equation (*i*), we get

(*ii*) $$U_1 - (-6000) + 0 = U_2$$

$$\therefore \quad U_2 - U_1 = 6000 \text{ kJ}$$

Hence, **change in internal energy (increase) = 6000 kJ.** **(Ans.)**

Example 3.9. *A stone of 20 kg mass and a tank containing 200 kg water comprise a system. The stone is 15 m above the water level initially. The stone and water are at the same temperature initially. If the stone falls into water, then determine ΔU, ΔPE, ΔKE, Q and W, when*

(i) the stone is about to enter the water,

(ii) the stone has come to rest in the tank, and

(iii) the heat is transferred to the surroundings in such an amount that the stone and water come to their initial temperature.

Solution. Refer Fig. 3.17.

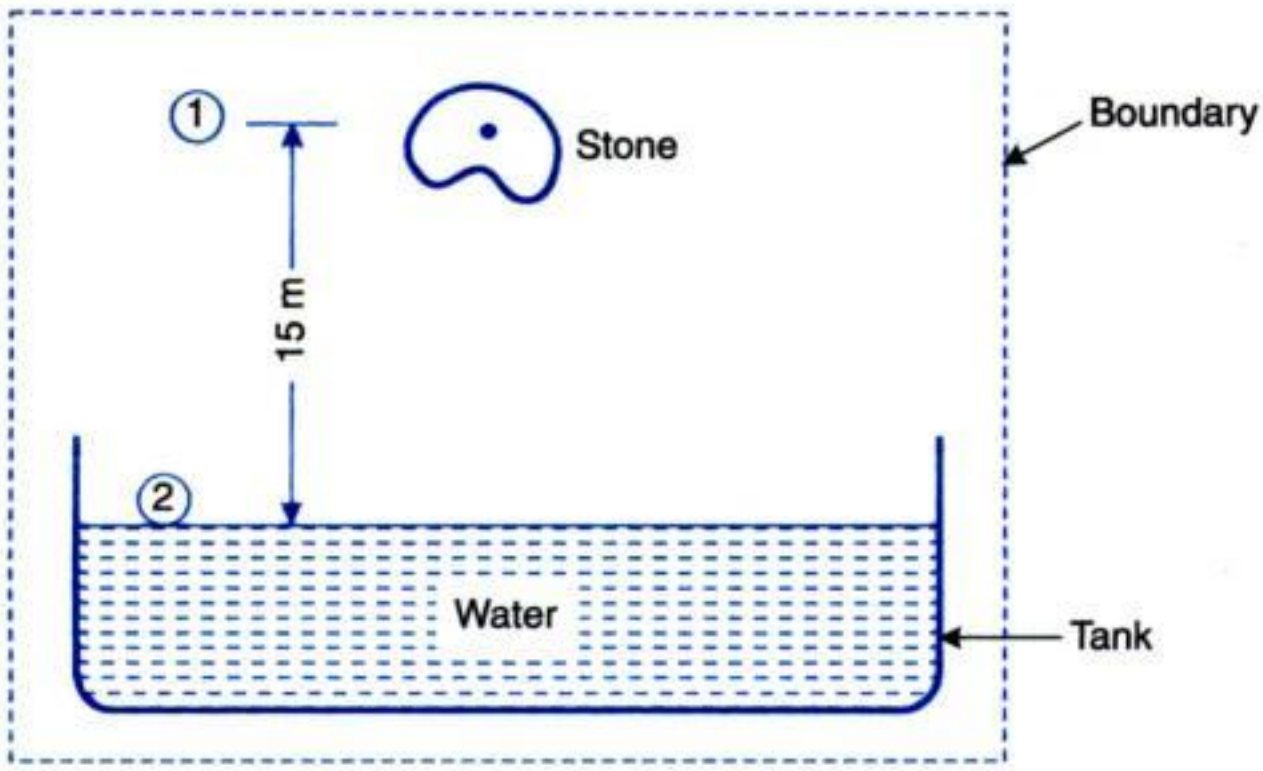

Fig. 3.17

Mass of stone = 20 kg

Mass of water in the tank = 200 kg

Height of stone above water level = 15 m

Applying the first law of thermodynamics,

$$Q = (U_2 - U_1) + m\left[\frac{C_2^2 - C_1^2}{2}\right] + mg\,(Z_2 - Z_1) + W$$
$$= \Delta U + \Delta KE + \Delta PE + W \qquad ...(1)$$

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Here Q = Heat leaving the boundary.

(*i*) When the stone is about to enter the water,

$$Q = 0,\ W = 0,\ \Delta U = 0$$

$\because$ $$-\Delta KE = \Delta PE = mg\,(Z_2 - Z_1)$$

$$= 20 \times 9.81\,(0 - 15) = -2943 \text{ J}$$

$\therefore$ $$\Delta KE = 2943 \text{ J}$$

and $$\Delta PE = \mathbf{-2943\ J.}\quad \textbf{(Ans.)}$$

(*ii*) When the stone dips into the tank and comes to rest

$$Q = 0,\ W = 0,\ \Delta KE = 0$$

Substituting these values in eqn. (1), we get

$$0 = \Delta U + 0 + \Delta PE + 0$$

$\therefore$ $$\Delta U = -\Delta PE = -(-2943) = \mathbf{2943\ J.}\quad \textbf{(Ans.)}$$

This shows that the internal energy (temperature) of the system increases.

(*iii*) When the water and stone come to their initial temperature,

$$W = 0,\ \Delta KE = 0$$

Substituting these values in eqn. (1), we get

$\therefore$ $$Q = -\Delta U = \mathbf{-2943\ J.}\quad \textbf{(Ans.)}$$

The *negative* sign shows that the *heat is lost from the system to the surroundings.*

☞**Example 3.10.** *When a system is taken from state l to state m, in Fig. 3.18, along path lqm, 168 kJ of heat flows into the system, and the system does 64 kJ of work :*

(i) How much will be the heat that flows into the system along path lnm if the work done is 21 kJ ?

(ii) When the system is returned from m to l along the curved path, the work done on the system is 42 kJ. Does the system absorb or liberate heat, and how much of the heat is absorbed or liberated ?

(iii) If $U_l = 0$ and $U_n = 84$ kJ, find the heat absorbed in the processes ln and nm.

Solution. Refer to Fig. 3.18.

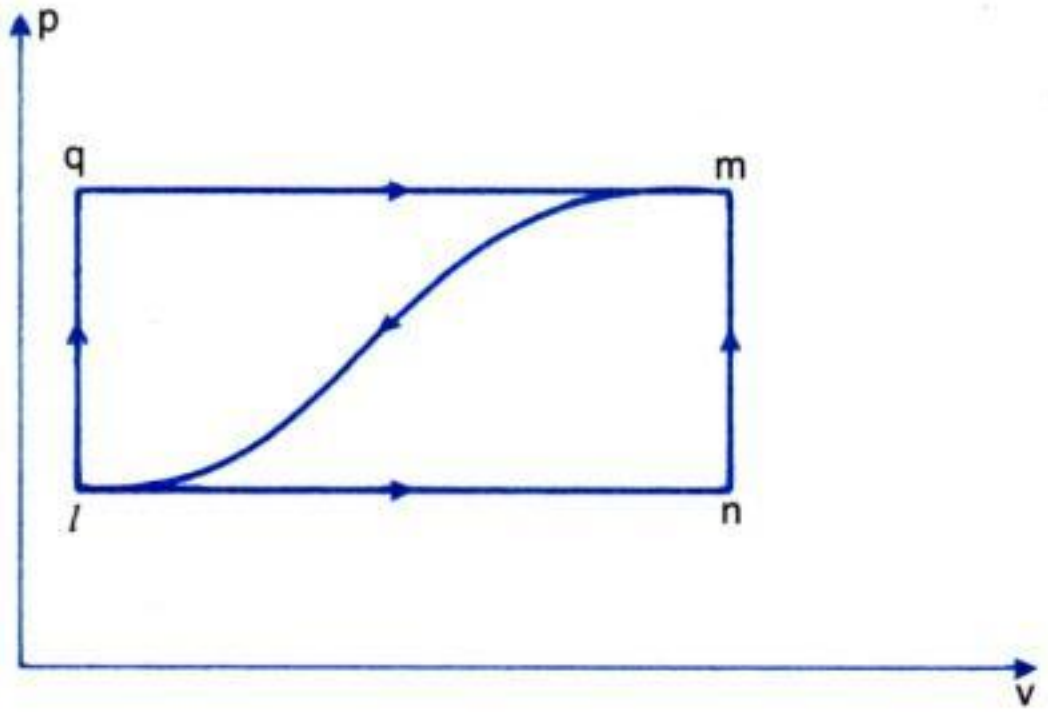

Fig. 3.18

$$Q_{l\text{-}q\text{-}m} = 168 \text{ kJ}$$

$$W_{l\text{-}q\text{-}m} = 64 \text{ kJ}$$

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We have, $Q_{l-q-m} = (U_m - U_l) + W_{l-q-m}$

$168 = (U_m - U_l) + 64$

$\therefore \quad U_m - U_l = \mathbf{104\ kJ.}$ **(Ans.)**

(i) $Q_{l-n-m} = (U_m - U_l) + W_{l-n-m}$

$= 104 + 21 = \mathbf{125\ kJ.}$ **(Ans.)**

(ii) $Q_{m-l} = (U_l - U_m) + W_{m-l}$

$= -104 + (-42) = \mathbf{-146\ kJ.}$ **(Ans.)**

The system liberates 146 kJ.

(iii) $W_{l-n-m} = W_{l-n} + W_{n-m} = W_{l-m} = 21$ kJ

[$\because$ $W_{n-m} = 0$, since volume does not change.]

$\therefore \quad Q_{l-n} = (U_n - U_l) + W_{l-n}$

$= (84 - 0) + 21 = \mathbf{105\ kJ.}$ **(Ans.)**

Now $Q_{l-m-n} = 125$ kJ $= Q_{l-n} + Q_{n-m}$

$\therefore \quad Q_{n-m} = 125 - Q_{l-n}$

$= 125 - 105 = \mathbf{20\ kJ.}$ **(Ans.)**

Example 3.11. *In a system, executing a non-flow process, the work and heat per degree change of temperature are given by*

$$\frac{dW}{dT} = 200\ \text{W-s/°C and } \frac{dQ}{dT} = 160\ \text{J/°C}$$

What will be the change of internal energy of the system when its temperature changes from

$$T_1 = 55°C \text{ to } T_2 = 95°C\ ?$$

Solution. Initial temperature, $T_1 = 55°C$; Final temperature, $T_2 = 95°C$

$$\frac{dW}{dT} = 200\ \text{W-s/°C} ; \frac{dQ}{dT} = 160\ \text{J/°C}.$$

Change of internal energy :

Now, $\frac{dW}{dT} = 200$ W-s/°C

$\therefore \quad W = \int_{T_1}^{T_2} 200\, dT = \Big| 200\, T \Big|_{T_1}^{T_2} = \Big| 200\, T \Big|_{55}^{95}$

$= 200\ (95 - 55) = 8000$ W-s $= 8000$ J [$\because$ 1 W-s = 1 J]

Also, $\frac{dQ}{dT} = 160$ J/°C

$\therefore \quad Q = \int_{T_1}^{T_2} 160\, dT = \Big| 160\, T \Big|_{T_1}^{T_2}$

$= \Big| 160\, T \Big|_{55}^{95} = 160\ (95 - 55) = 6400$ J

Applying the first law of thermodynamics to the given non-flow system,

$Q = \Delta U + W$

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$$\therefore \quad 6400 = \Delta U + 8000$$

or $$\Delta U = -1600 \text{ J} = \mathbf{-1.6 \text{ kJ}}. \quad \textbf{(Ans.)}$$

The –ve sign indicates that there is *decrease* in internal energy.

Example 3.12. *A fluid system, contained in a piston and cylinder machine, passes through a complete cycle of four processes. The sum of all heat transferred during a cycle is – 340 kJ. The system completes 200 cycles per min.*

Complete the following table showing the method for each item, and compute the net rate of work output in kW.

Process	*Q (kJ/min)*	*W (kJ/min)*	*ΔE (kJ/min)*
1–2	*0*	*4340*	—
2–3	*42000*	*0*	—
3–4	*–4200*	—	*–73200*
4–1	—	—	—

Solution. Sum of all heat transferred during the cycle = – 340 kJ.

Number of cycles completed by the system = 200 cycles/min.

Process 1–2 :

$$Q = \Delta E + W$$

$$0 = \Delta E + 4340$$

$$\therefore \quad \Delta E = -4340 \text{ kJ/min}.$$

Process 2–3 :

$$Q = \Delta E + W$$

$$42000 = \Delta E + 0$$

$$\Delta E = 42000 \text{ kJ/min}.$$

Process 3–4 :

$$Q = \Delta E + W$$

$$-4200 = -73200 + W$$

$$\therefore \quad W = 69000 \text{ kJ/min}.$$

Process 4–1 :

$$\underset{\text{cycle}}{\Sigma Q} = -340 \text{ kJ}$$

The system completes 200 cycles/min

$$\because \quad Q_{1-2} = Q_{2-3} + Q_{3-4} + Q_{4-1} = -340 \times 200 = -68000 \text{ kJ/min}$$

or $$0 + 42000 + (-4200) + Q_{4-1} = -68000$$

$$Q_{4-1} = -105800 \text{ kJ/min}.$$

Now, $\int dE = 0$, since cyclic integral of any property is zero.

$$\Delta E_{1-2} + \Delta E_{2-3} + \Delta E_{3-4} + \Delta E_{4-1} = 0$$

$$-4340 + 42000 + (-73200) + \Delta E_{4-1} = 0$$

$$\therefore \quad \Delta E_{4-1} = 35540 \text{ kJ/min}.$$

$$\therefore \quad W_{4-1} = Q_{4-1} - \Delta E_{4-1}$$

$$= -105800 - 35540 = -141340 \text{ kJ/min}$$

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The completed table is given below :

Process	*Q(kJ/min)*	*W(kJ/min)*	*ΔE(kJ/min)*
1–2	0	4340	–4340
2–3	42000	0	42000
3–4	–4200	69000	–73200
4–1	–105800	–141340	35540

Since $\underset{\text{cycle}}{\Sigma Q} = \underset{\text{cycle}}{\Sigma W}$

Rate of work output = – 68000 kJ/min $= -\frac{68000}{60}$ kJ/s or kW

$$= \mathbf{1133.33\ kW.}\ \textbf{(Ans.)}$$

Example 3.13. *The power developed by a turbine in a certain steam plant is 1200 kW. The heat supplied to the steam in the boiler is 3360 kJ/kg, the heat rejected by the system to cooling water in the condenser is 2520 kJ/kg and the feed pump work required to pump the condensate back into the boiler is 6 kW.*

Calculate the steam flow round the cycle in kg/s.

Solution. The power developed by the turbine = 1200 kW

The heat supplied to the steam in the boiler = 3360 kJ/kg

The heat rejected by the system to cooling water = 2520 kJ/kg

Feed pump work = 6 kW

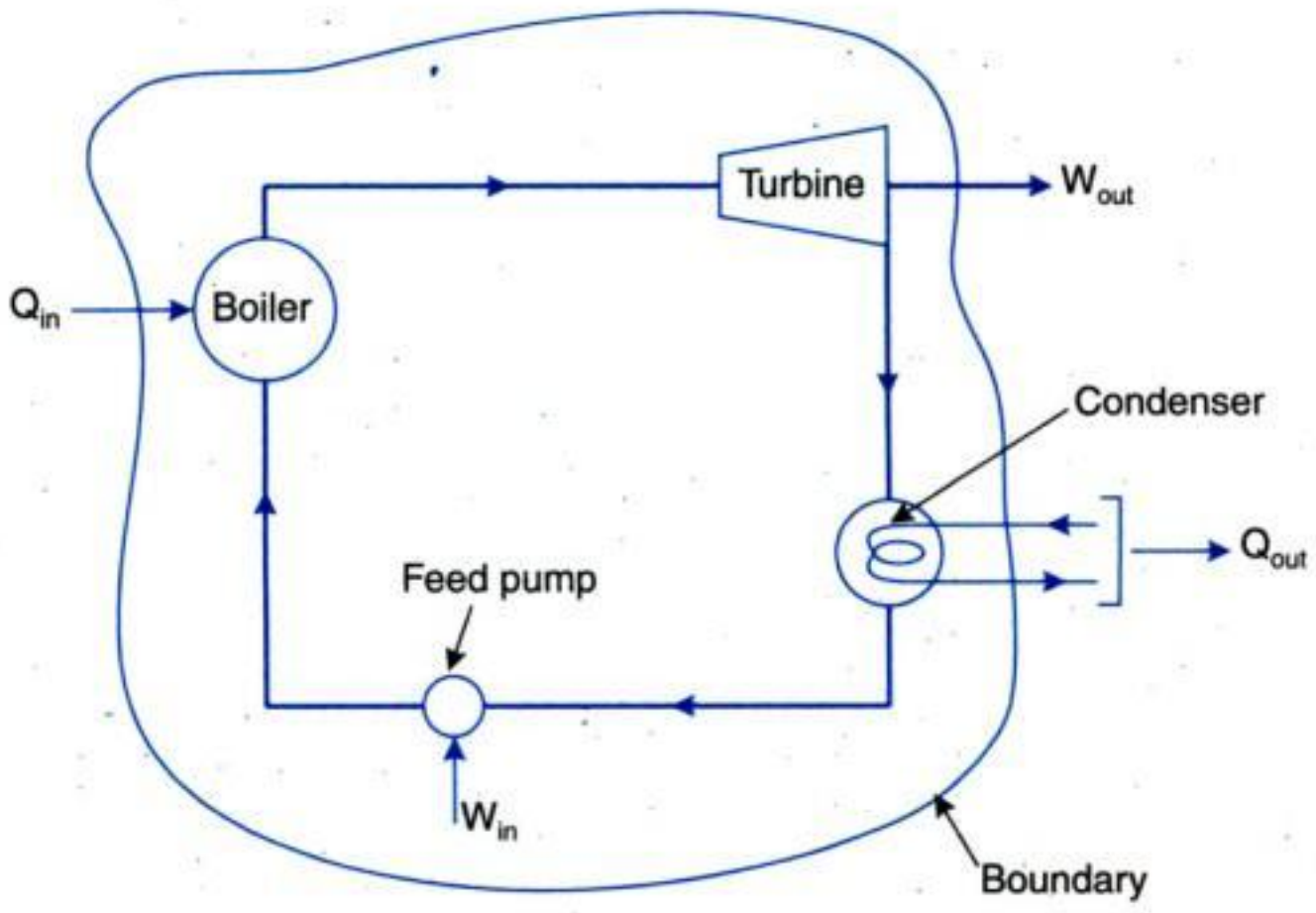

Fig. 3.19

Fig. 3.19 shows the cycle. A boundary is shown which encompasses the entire plant. Strictly, this boundary should be thought of as encompassing the working fluid only.

$$\oint dQ = 3360 - 2520 = 840 \text{ kJ/kg}$$

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Let the system flow be in kg/s.

$$\therefore \qquad \oint dQ = 840\ \dot{m}\ \text{kJ/s}$$

$$\oint dW = 1200 - 6 = 1194\ \text{kJ/s}$$

But $$\oint dQ = \oint dW$$

i.e., $$840 \cdot \dot{m} = 1194$$

$$\therefore \qquad \dot{m} = \frac{1194}{840} = 1.421\ \text{kg/s}$$

$\because$ **Steam flow round the cycle = 1.421 kg/s.** **(Ans.)**

Example 3.14. *A closed system of constant volume experiences a temperature rise of 25°C when a certain process occurs. The heat transferred in the process is 30 kJ. The specific heat at constant volume for the pure substance comprising the system is 1.2 kJ/kg°C, and the system contains 2.5 kg of this substance. Determine :*

(i) The change in internal energy ;

(ii) The work done.

Solution. Temperature rise, $(T_2 - T_1) = 25°C$

The heat transferred in the process, $Q = 30$ kJ

Specific heat at constant volume, $c_v = 1.2$ kJ/kg°C

Mass of the substance, $m = 2.5$ kg

Now, $$\Delta U = m \int_{T_1}^{T_2} c_v\, dT$$

$$= 2.5 \int_{T_1}^{T_2} 1.2\, dT = 3.0 \times (T_2 - T_1)$$

$$= 3.0 \times 25 = \mathbf{75\ kJ}$$

Hence, **the change in internal energy is 75 kJ.** **(Ans.)**

According to the first law of thermodynamics,

$$Q = \Delta U + W$$

$$\therefore \qquad 30 = 75 + W$$

$$\therefore \qquad W = 30 - 75 = -45\ \text{kJ}$$

Hence, **the work done** $= \mathbf{-45\ kJ.}$ **(Ans.)**

It may be observed that *even though the volume is constant the work is not zero.* Clearly, the process is **irreversible.**

Example 3.15. *A system receives 50 kJ of heat while expanding with volume change of 0.14 m^3 against an atmosphere of 1.2 × 10^5 N/m^2. A mass of 90 kg in the surroundings is also lifted through a distance of 5.5 metres.*

(i) Find the change in energy of the system.

(ii) The system is returned to its initial volume by an adiabatic process which requires 110 kJ of work. Find the change in energy of the system.

(iii) For the combined processes of (i) and (ii) determine the change in energy of the system.

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Solution. Heat received by the system,

$$Q = 50 \text{ kJ}$$

Change in volume $\Delta V = 0.14 \text{ m}^3$

Pressure $= 1.2 \times 10^5 \text{ N/m}^2$

Mass lifted in the surroundings $= 90 \text{ kg}$

Distance through which lifted $= 5.5 \text{ m}$

Work done during adiabatic process = – 110 kJ.

(*i*) $Q = \Delta E + W$ ...(*i*)

Now, $W = p.\Delta V + W_{net}$

$$= \left(\frac{1.2 \times 10^5 \times 0.14}{1000}\right) + \left(\frac{90 \times 5.5 \times 9.8}{1000}\right) \text{kJ}$$

$= 16.8 + 4.85 = 21.65 \text{ kJ}$

But [from (*i*)], $\Delta E = Q - W$

$= 50 - 21.65 = \mathbf{28.35 \text{ kJ}}$. **(Ans.)**

(*ii*) Since the process is adiabatic,

$$Q = 0$$

and $\Delta E = - W$

$= - (- 110) = \mathbf{110 \text{ kJ}}$. **(Ans.)**

(*iii*) Change in internal energy,

$$\Delta E = Q - W$$

$= 50 - [(- 110) + 21.65] = \mathbf{138.35 \text{ kJ}}$. **(Ans.)**

☞ **Example 3.16.** *A fluid system undergoes a non-flow frictionless process following the pressure-volume relation as* $p = \frac{5}{V} + 1.5$ *where p is in bar and V is in* m^3. *During the process the volume changes from 0.15* m^3 *to 0.05* m^3 *and the system rejects 45 kJ of heat. Determine :*

(i) Change in internal energy ;

(ii) Change in enthalpy.

Solution. Pressure-volume relation : $p = \frac{5}{V} + 1.5$

Initial volume, $V_1 = 0.15 \text{ m}^3$

Final volume, $V_2 = 0.05 \text{ m}^3$

Heat *rejected* by the system, $Q = - 45 \text{ kJ}$

Work done is given by,

$$W = \int_1^2 p \, . \, dV = \int_{V_1}^{V_2} \left(\frac{5}{V} + 1.5\right) dV$$

$$= \int_{0.15}^{0.05} \left(\frac{5}{V} + 1.5\right) dV = \left[5 \log_e \frac{V_2}{V_1} + 1.5 (V_2 - V_1)\right]_{0.15}^{0.05} \times 10^5 \text{ Nm}$$

$$= 10^5 \left[5 \log_e \frac{0.05}{0.15} + 1.5 (0.05 - 0.15)\right] = 10^5 (- 5.49 - 0.15) \text{ Nm}$$

$= - 5.64 \times 10^5 \text{ Nm} = - 5.64 \times 10^5 \text{ J}$ [$\because$ 1 Nm = 1 J]

$= - 564 \text{ kJ}$.

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(*i*) Applying the first law energy equation,

$$Q = \Delta U + W$$

$$-45 = \Delta U + (-564)$$

$$\therefore \quad \Delta U = 519 \text{ kJ.} \quad \textbf{(Ans.)}$$

This shows that the internal energy is **increased.**

(*ii*) Change in enthalpy,

$$\Delta H = \Delta U + \Delta (pV)$$

$$= 519 \times 10^3 + (p_2V_2 - p_1V_1)$$

$$p_1 = \frac{5}{V_1} + 1.5 = \frac{5}{0.15} + 1.5 = 34.83 \text{ bar}$$

$$= 34.83 \times 10^5 \text{ N/m}^2$$

$$p_2 = \frac{5}{V_2} + 1.5 = \frac{5}{0.05} + 1.5$$

$$= 101.5 \text{ bar} = 101.5 \times 10^5 \text{ N/m}^2$$

$$\therefore \quad \Delta H = 519 \times 10^3 + (101.5 \times 10^5 \times 0.05 - 34.83 \times 10^5 \times 0.15)$$

$$= 519 \times 10^3 + 10^3(507.5 - 522.45)$$

$$= 10^3(519 + 507.5 - 522.45) = 504 \text{ kJ}$$

$\therefore$ **Change in enthalpy = 504 kJ.** **(Ans.)**

☞ **Example 3.17.** *The following equation gives the internal energy of a certain substance*

$$u = 3.64\, pv + 90$$

where u is kJ/kg, p is in kPa and v is in m^3/kg.

A system composed of 3.5 kg of this substance expands from an initial pressure of 500 kPa and a volume of 0.25 m^3 to a final pressure 100 kPa in a process in which pressure and volume are related by $pv^{1.25}$ = constant.

(i) If the expansion is quasi-static, find Q, ΔU and W for the process.

(ii) In another process, the same system expands according to the same pressure-volume relationship as in part (i), and from the same initial state to the same final state as in part (i), but the heat transfer in this case is 32 kJ. Find the work transfer for this process.

(iii) Explain the difference in work transfer in parts (i) and (ii).

Solution. Internal energy equation : $u = 3.64\, pv + 90$

Initial volume, $V_1 = 0.25 \text{ m}^3$

Initial pressure, $p_1 = 500 \text{ kPa}$

Final pressure, $p_2 = 100 \text{ kPa}$

Process : $pv^{1.25} = \text{constant.}$

(*i*) Now, $u = 3.64\, pv + 90$

$$\Delta u = u_2 - u_1$$

$$= 3.64\,(p_2v_2 - p_1v_1) \quad \text{...per kg}$$

$$\therefore \quad \Delta U = 3.64\,(p_2V_2 - p_1V_1) \quad \text{...for 3.5 kg}$$

Now, $p_1V_1^{1.25} = p_2V_2^{1.25}$

$$V_2 = V_1 \left(\frac{p_1}{p_2}\right)^{1/1.25} = 0.25 \left(\frac{500}{100}\right)^{1/1.25}$$

$$= 0.906 \text{ m}^3$$

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$\therefore$ $\Delta U = 3.64\ (100 \times 10^3 \times 0.906 - 500 \times 10^3 \times 0.25)$ J [$\because$ 1 Pa = 1 N/m^2]

$= 3.64 \times 10^5\ (0.906 - 5 \times 0.25)$ J

$= -\ 3.64 \times 10^5 \times 0.344$ J $= -\ 125.2$ kJ

i.e., **$\Delta U = -\ 125.2$ kJ.** **(Ans.)**

For a **quasi-static process**

$$W = \int p dV = \frac{p_1V_1 - p_2V_2}{n-1}$$

$$= \frac{(500 \times 10^3 \times 0.25 - 100 \times 10^3 \times 0.906)}{(1.25 - 1)} = \frac{125 - 90.6}{0.25}\ \text{kJ} = 137.6\ \text{kJ}$$

$\therefore$ $Q = \Delta U + W$

$= -\ 125.2 + 137.6 = 12.4$ kJ

i.e., **$Q = 12.4$ kJ.** **(Ans.)**

(*ii*) Here $Q = 32$ kJ

Since the end states are the same, ΔU would remain the same as in (*i*)

$\therefore$ $W = Q - \Delta U = 32 - (-\ 125.2)$

$= 157.2$ kJ. **(Ans.)**

(*iii*) The work in (*ii*) is **not equal** to $\int p\ dV$ since the process is **not quasi-static.**

Example 3.18. *The properties of a system, during a reversible constant pressure non-flow process at $p = 1.6$ bar, changed from $v_1 = 0.3$ m^3/kg, $T_1 = 20°C$ to $v_2 = 0.55$ m^3/kg, $T_2 = 260°C$. The specific heat of the fluid is given by*

$$c_p = \left(1.5 + \frac{75}{T + 45}\right)\ \text{kJ/kg°C, where } T \text{ is in °C.}$$

Determine : (i) Heat added/kg ; (ii) Work done/kg ;

(iii) Change in internal energy/kg ; (iv) Change in enthalpy/kg.

Solution. Initial volume, $v_1 = 0.3$ m^3/kg

Initial temperature, $T_1 = 20°C$

Final volume, $v_2 = 0.55$ m^3/kg

Final temperature, $T_2 = 260°C$

Constant pressure, $p = 1.6$ bar

Specific heat at constant pressure, $c_p = \left(1.5 + \frac{75}{T + 45}\right)$ kJ/kg°C

(*i*) The **heat added** per kg of fluid is given by

$$Q = \int_{T_1}^{T_2} c_p\ dT = \int_{20}^{260} \left(1.5 + \frac{75}{T + 45}\right) dT$$

$$= \Big| 1.5\,T + 75 \log_e (T + 45) \Big|_{20}^{260}$$

$$= 1.5\ (260 - 20) + 75 \times \log_e \left(\frac{260 + 45}{20 + 45}\right) = 475.94\ \text{kJ}$$

$\therefore$ **Heat added** **$= 475.94$ kJ/kg.** **(Ans.)**

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(*ii*) The *work done* per kg of fluid is given by

$$W = \int_{v_1}^{v_2} p dv = p(v_2 - v_1) = 1.6 \times 10^5 (0.55 - 0.3) \text{ Nm}$$

$$= 40 \times 10^3 \text{ J} = 40 \text{ kJ}$$

$\therefore$ **Work done = 40 kJ/kg.** **(Ans.)**

(*iii*) **Change in internal energy,**

$$\Delta u = Q - W = 475.94 - 40 = \mathbf{435.94 \text{ kJ/kg.}} \quad \textbf{(Ans.)}$$

(*iv*) **Change in enthalpy,** (for **non-flow process**)

$$\Delta h = Q = \mathbf{475.94 \text{ kJ/kg.}} \quad \textbf{(Ans.)}$$

Example 3.19. *1 kg of gaseous CO_2 contained in a closed system undergoes a reversible process at constant pressure. During this process 42 kJ of internal energy is decreased. Determine the work done during the process.*

Take c_p = 840 J/kg°C and c_v = 600 J/kg°C.

Solution. Mass CO_2, m = 1 kg

Decrease in internal energy, $\Delta u = -42 \text{ kJ} = -42 \times 10^3 \text{ J}$

Specific heat at constant pressure, $c_p = 840$ J/kg°C

Specific heat at constant volume, $c_v = 600$ J/kg°C

Let, initial temperature of CO_2 $= T_1$

Final temperature of CO_2 $= T_2$

Now change in internal energy,

$$\Delta U = m \times c_v (T_2 - T_1)$$

$$-42 \times 10^3 = 1 \times 600 (T_2 - T_1)$$

$$\therefore \quad T_2 - T_1 = -\frac{42 \times 10^3}{600} = -70°\text{C}$$

The heat supplied or rejected,

$$Q = mc_p (T_2 - T_1)$$

$$= 1 \times 840 \times (-70) = -58800 \text{ J or } -58.8 \text{ kJ}$$

Applying first law to the process,

$$Q = \Delta U + W$$

$$-58.8 = -42 + W \text{ or } W = -16.8 \text{ kJ}$$

$\therefore$ **Work done during the process = – 16.8 kJ.** **(Ans.)**

Example 3.20. *A fluid is contained in a cylinder by a spring-loaded, frictionless piston so that the pressure in the fluid is a linear function of the volume ($p = a + bV$). The internal energy of the fluid is given by the following equation*

$$U = 42 + 3.6\, pV$$

where U is in kJ, p in kPa, and V in cubic metre. If the fluid changes from an initial state of 190 kPa, 0.035 m^3 to a final state of 420 kPa, 0.07 m^3, with no work other than that done on the piston, find the direction and magnitude of the work and heat transfer.

Solution. Relation between pressure and volume, $p = a + bV$.

Equation of internal energy : $U = 42 + 3.6pV$

Initial pressure, $p_1 = 190$ kPa

Initial volume, $V_1 = 0.035 \text{ m}^3$

Final pressure, $p_2 = 420$ kPa

Final volume, $V_2 = 0.07 \text{ m}^3$

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The change in internal energy of the fluid during the process

$$U_2 - U_1 = (42 + 3.6p_2V_2) - (42 + 3.6p_1V_1)$$
$$= 3.6(p_2V_2 - p_1V_1)$$
$$= 3.6(4.2 \times 10^5 \times 0.07 - 1.9 \times 10^5 \times 0.035)\ \text{J}$$
$$= 360(4.2 \times 0.07 - 1.9 \times 0.035)\ \text{kJ}$$
$$= 81.9\ \text{kJ}$$

Now, $p = a + bV$

$$190 = a + b \times 0.035 \quad ...(i)$$
$$420 = a + b \times 0.07 \quad ...(ii)$$

Subtracting (i) from (ii), we get

$$230 = 0.035\ b \text{ or } b = \frac{230}{0.035} = 6571\ \text{kN/m}^5$$

and $a = -40\ \text{kN/m}^2$

Work transfer involved during the process

$$W_{1-2} = \int_{V_1}^{V_2} pdV = \int_{V_1}^{V_2} (a + bV)\,dV = a(V_2 - V_1) + b\left(\frac{V_2^2 - V_1^2}{2}\right)$$

$$= (V_2 - V_1)\left[a + \frac{b}{2}(V_1 + V_2)\right]$$

$$= (0.07 - 0.035)\left[-40\ \text{kN/m}^2 + \frac{6571}{2}\ \text{kN/m}^5\ (0.035 + 0.07)\right] = 10.67\ \text{kJ}$$

∴ Work done by the system = 10.67 kJ. (Ans.)

Heat transfer involved,

$$Q_{1-2} = (U_2 - U_1) + W_{1-2} = 81.9 + 10.67 = 92.57\ \text{kJ}.$$

92.57 kJ of heat flow into the system during the process. (Ans.)

Example 3.21. *90 kJ of heat are supplied to a system at a constant volume. The system rejects 95 kJ of heat at constant pressure and 18 kJ of work is done on it. The system is brought to original state by adiabatic process. Determine :*

(i) The adiabatic work ;

(ii) The values of internal energy at all end states if initial value is 105 kJ.

Solution. Refer Fig. 3.20.

Heat supplied at constant volume = 90 kJ

Heat rejected at constant pressure = – 95 kJ

Work done *on* the system = – 18 kJ

Initial value of internal energy, $U_l = 105$ kJ

Process *l–m* (*constant volume*) :

$$W_{l-m} = 0$$
$$Q_{l-m} = 90 = U_m - U_l$$

∴ $U_m = U_l + 90 = 105 + 90$

$$= 195\ \text{kJ}$$

Process *m–n* (*constant pressure*) :

$$Q_{m-n} = (U_n - U_m) + W_{m-n}$$
$$-95 = (U_n - U_m) - 18$$

∴ $U_n - U_m = -77\ \text{kJ}$

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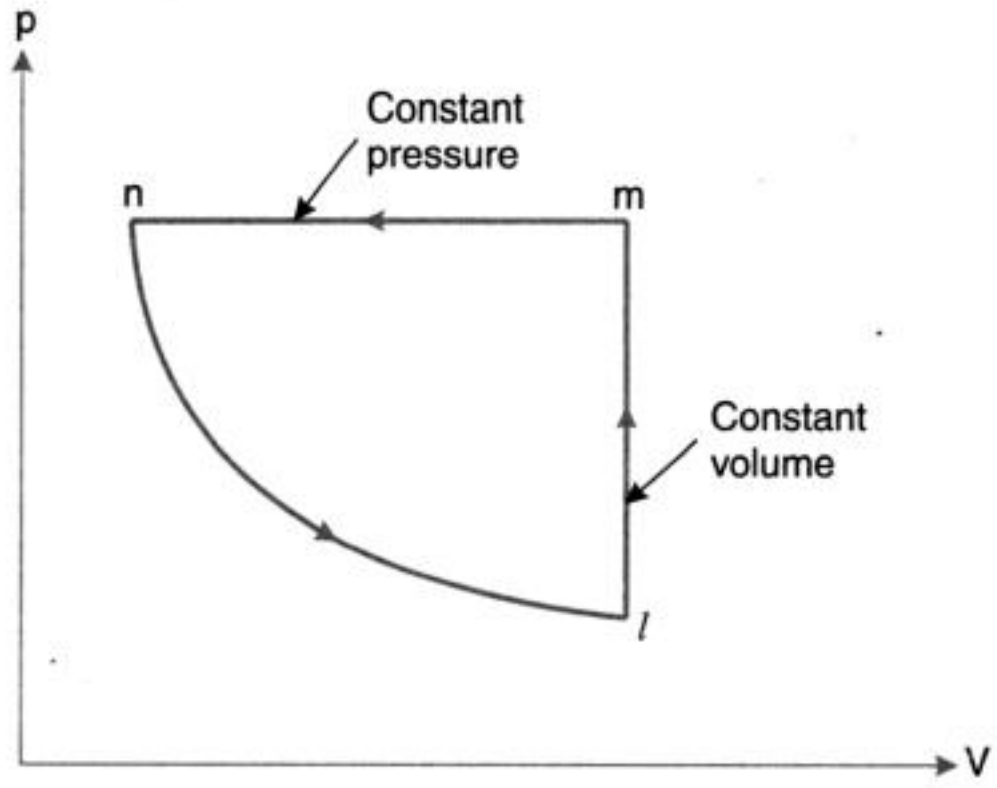

Fig. 3.20

$\therefore$ $U_n = 195 - 77 = 118$ kJ

$Q_{n-l} = 0$ being adiabatic process

$\therefore$ $\oint \delta Q = 90 - 95 = -5$ kJ

and $\oint \delta W = -18 + W_{n-l} = -5$

$\therefore$ $W_{n-l} = -5 + 18 = 13$ kJ

Hence, $\mathbf{W_{n-l} = 13\ kJ\ ;\ U_l = 105\ kJ\ ;\ U_m = 195\ kJ\ ;\ U_n = 118\ kJ.}$ **(Ans.)**

Example 3.22. *A movable frictionless piston closes a fully insulated cylinder on one side and offers a constant resistance during its motion. A paddle work is drawn into the cylinder and does work on the system.*

Prove that the paddle work is equal to change in enthalpy.

Solution. Refer to Fig. 3.21.

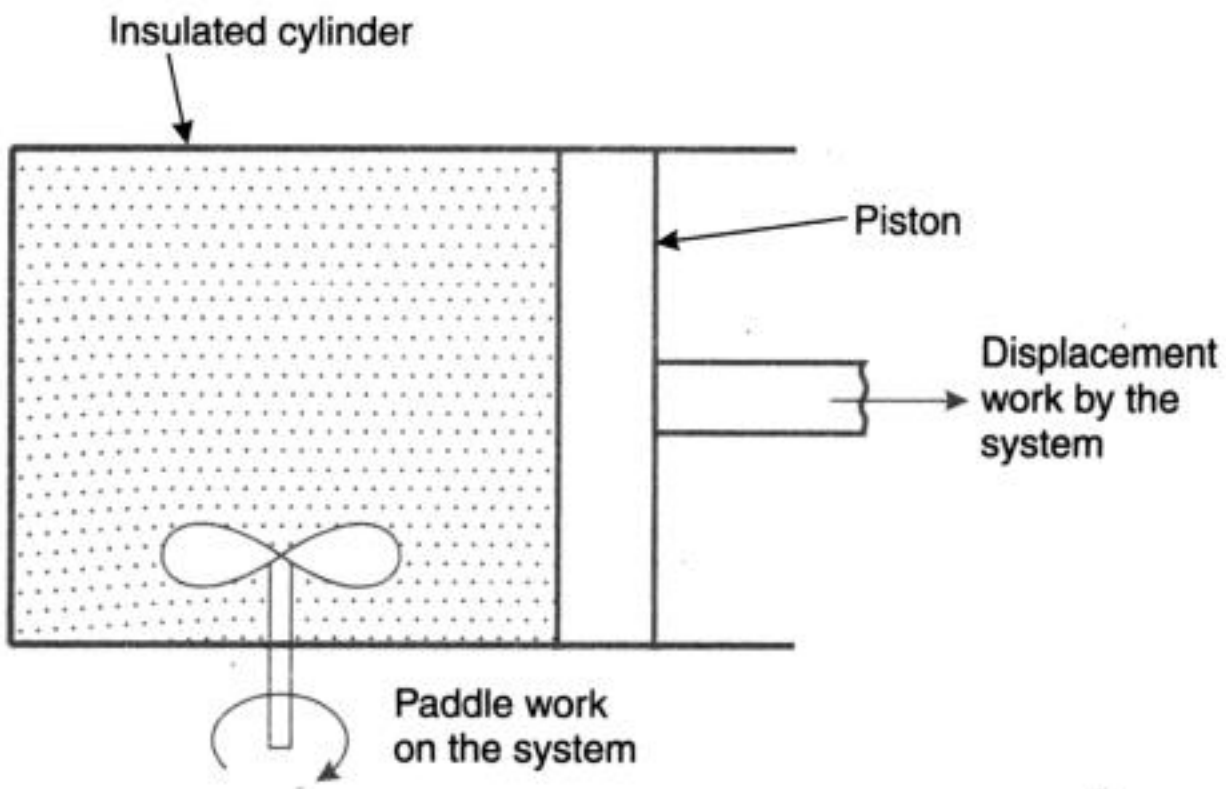

Fig. 3.21

$$Q = W_{paddle} = \Delta U + p\Delta V = \Delta U + \Delta(pV) = \Delta(U + pV) = \Delta H$$

Hence, **paddle work is equal to change in enthalpy. (Ans.)**

☞**Example 3.23.** *0.2 m^3 of air at 4 bar and 130°C is contained in a system. A reversible adiabatic expansion takes place till the pressure falls to 1.02 bar. The gas is then heated at constant pressure till enthalpy increases by 72.5 kJ. Calculate :*

(i) The work done ;

(ii) The index of expansion, if the above processes are replaced by a single reversible polytropic process giving the same work between the same initial and final states.

Take c_p = 1 kJ/kg K, c_v = 0.714 kJ/kg K.

Solution. Refer to Fig. 3.22.

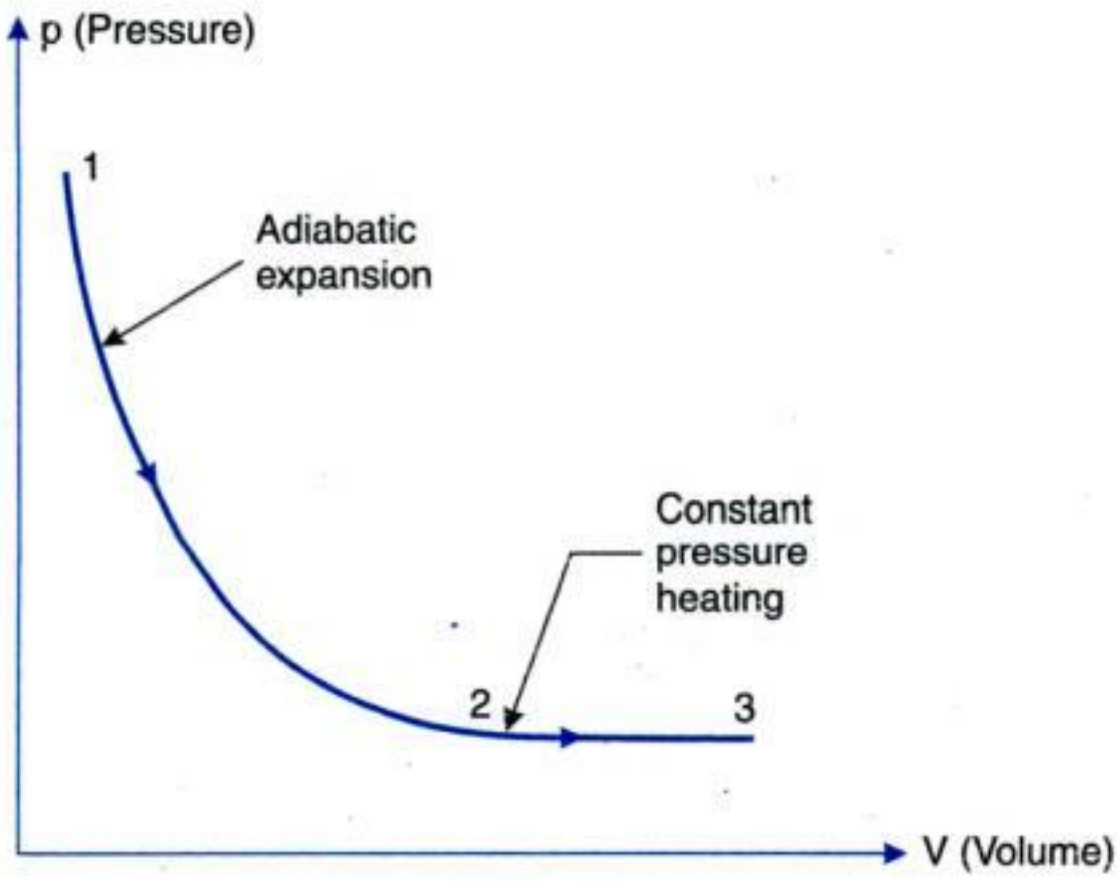

Fig. 3.22

Initial volume, $V_1 = 0.2\ m^3$

Initial pressure, $p_1 = 4\ bar = 4 \times 10^5\ N/m^2$

Initial temperature, $T_1 = 130 + 273 = 403\ K$

Final pressure after adiabatic expansion,

$$p_2 = 1.02\ bar = 1.02 \times 10^5\ N/m^2$$

Increase in enthalpy during constant pressure process

$$= 72.5\ kJ.$$

(i) **Work done :**

Process 1–2 : *Reversibe adiabatic process* :

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

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

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Also, $$\gamma = \frac{c_p}{c_v} = \frac{1}{0.714} = 1.4$$

$\therefore$ $$V_2 = 0.2 \times \left(\frac{4 \times 10^5}{1.02 \times 10^5}\right)^{\frac{1}{1.4}} = 0.53 \text{ m}^3$$

Also, $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$$

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

$$= 403 \left(\frac{1.02 \times 10^5}{4 \times 10^5}\right)^{\frac{1.4 - 1}{1.4}} = 272.7 \text{ K}$$

Mass of the gas,

$$m = \frac{p_1 V_1}{R T_1} \qquad [\because \quad pV = mRT]$$

where, $R = (c_p - c_v) = (1 - 0.714)$ kJ/kg K

$= 0.286$ kJ/kg K $= 286$ J/kg K or 286 Nm/kg K

$\therefore$ $$m = \frac{4 \times 10^5 \times 0.2}{286 \times 403} = 0.694 \text{ kg.}$$

Process 2–3. *Constant pressure* :

$$Q_{2\text{-}3} = mc_p (T_3 - T_2)$$

$$72.5 = 0.694 \times 1 \times (T_3 - 272.7)$$

$\therefore$ $$T_3 = \frac{72.5}{0.694} + 272.7 = 377 \text{ K}$$

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

or $$\frac{0.53}{272.7} = \frac{V_3}{377}$$

$\therefore$ $$V_3 = \frac{0.53 \times 377}{272.7} = 0.732 \text{ m}^3$$

Work done by the path 1-2-3 is given by

$$W_{1\text{-}2\text{-}3} = W_{1\text{-}2} + W_{2\text{-}3}$$

$$= \frac{p_1 V_1 - p_2 V_2}{\gamma - 1} + p_2 (V_3 - V_2)$$

$$= \frac{4 \times 10^5 \times 0.2 - 1.02 \times 10^5 \times 0.53}{1.4 - 1} + 1.02 \times 10^5 (0.732 - 0.53)$$

$$= \frac{10^5 (4 \times 0.2 - 1.02 \times 0.53)}{0.4} + 1.02 \times 10^5 (0.732 - 0.53)$$

$$= 64850 + 20604 = 85454 \text{ Nm or J}$$

Hence, **total work done = 85454 Nm or J. (Ans.)**

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(*ii*) **Index of expansion, n :**

If the work done by the polytropic process is the same,

$$W_{1\text{-}2\text{-}3} = W_{1\text{-}3} = \frac{p_1V_1 - p_3V_3}{n-1}$$

$$85454 = \frac{4 \times 10^5 \times 0.2 - 1.02 \times 10^5 \times 0.732}{(n-1)} = \frac{5336}{n-1}$$

$$\therefore \qquad n = \frac{5336}{85454} + 1$$

i.e., $n = 1.062$

Hence, **value of index = 1.062. (Ans.)**

Example 3.24. *The following is the equation which connects u, p and v for several gases*

$$u = a + bpv$$

where a and b are constants. Prove that for a reversible adiabatic process,

$$pv^\gamma = \text{constant, where } \gamma = \frac{b+1}{b}.$$

Solution. Consider a unit mass.

For a reversible adiabatic process, first law gives

$$0 = du + pdv$$

$$\therefore \qquad \frac{du}{dv} = -p \qquad \ldots(i)$$

Also, $u = a + bpv$

$$\therefore \qquad \frac{du}{dv} = \frac{d(a+bpv)}{dv} = bv\,\frac{dp}{dv} + bp$$

$$= b\left(p + v \cdot \frac{dp}{dv}\right) \qquad \ldots(ii)$$

Equating (*i*) and (*ii*), we get

$$b\left(p + v \cdot \frac{dp}{dv}\right) = -p$$

$$bp + b \cdot v \cdot \frac{dp}{dv} = -p$$

$$bp + p + bv \cdot \frac{dp}{dv} = 0$$

$$p(b+1) + bv \cdot \frac{dp}{dv} = 0$$

Multiplying both sides by $\frac{dv}{bpv}$, we get

$$\left(\frac{b+1}{b}\right)\frac{dv}{v} + \frac{dp}{p} = 0$$

or $$\frac{dp}{p} + \left(\frac{b+1}{b}\right)\frac{dv}{v} = 0$$

$$d(\log_e p) + \left(\frac{b+1}{b}\right) d(\log_e v) = 0$$

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Also, $\frac{b+1}{b} = \gamma$ ...(Given)

$\therefore$ $d(\log_e p) + \gamma d(\log_e v) = 0$

Integrating, we get $pv^\gamma = \text{constant}.$

Example 3.25. *A 15 cm diameter vertical cylinder, closed by a piston contains a combustible mixture at a temperature of 30°C. The piston is free to move and its weight is such that the mixture pressure is 3 bar. Upper surface of the piston is exposed to the atmosphere. The mixture is ignited. As the reaction proceeds, the piston moves slowly upwards and heat transfer to the surroundings takes place. When the reaction is complete and the contents have been reduced to the initial temperature of 30°C, it is found that the piston has moved upwards a distance of 8.5 cm and the magnitude of heat transfer is 4 kJ. Evaluate :*

(i) The work ;

(ii) Decrease in internal energy of the system.

Solution. Diameter of vertical cylinder, $d = 15$ cm (or 0.15 m)

Temperature of combustible mixture = 30°C (or 303 K)

Pressure of the mixture = 3 bar = 3×10^5 N/m^2

Upward displacement of the system = 8.5 cm (or 0.085 m)

Magnitude of heat transfer, $Q = -4$ kJ ...(*i*)

(*i*) Work done by the system, $W = \int p dv$

$= 3 \times 10^5 \int dv$ [$\because$ p = constant = 3×10^5 N/m^2]

$= 3 \times 10^5 \left[\frac{\pi}{4} \times (0.15)^2 \times 0.085\right]$ Nm

$= 450.62$ Nm or J $= 0.4506$ kJ

$\therefore$ $W = 0.4506$ kJ.

(*ii*) By first law of thermodynamics,

$$Q = \Delta U + W$$

$$-4 = \Delta U + 0.4506$$

$\therefore$ $\Delta U = -4.4506$ kJ

$\therefore$ **Decrease in internal energy = 4.4506 kJ.** **(Ans.)**

Example 3.26. *A housewife, on a warm summer day, decides to beat the heat by closing the windows and doors in the kitchen and opening the refrigerator door. At first she feels cool and refreshed, but after a while the effect begins to wear off.*

Evaluate the situation as it relates to First Law of Thermodynamics, considering the room including the refrigerator as the system.

Solution. Initially, the temperature of air in the room falls when it communicates with the cool refrigerator with its door open. This makes the housewife feel cool.

Considering the room and its contents as the system, and assuming the walls, windows and doors non-conducting, we find, $Q = 0$.

To operate the refrigerator, *electricity* is supplied from outside and hence *external work W is done on the system*.

Applying the first law to the system,

$$Q = \Delta U + W$$

$$0 = \Delta U + (-W)$$

$\therefore$ $\Delta U = W$

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The right hand side is a positive figure indicating the increase in energy of the system with time. As the energy is increasing the temperature of air increases and hence the effect of coolness gradually begins to wear off.

It may be pointed out here that in this case the energy rise manifests itself in a rise in temperature.

☞**Example 3.27.** *A cylinder contains 0.45 m^3 of a gas at 1×10^5 N/m^2 and 80°C. The gas is compressed to a volume of 0.13 m^3, the final pressure being 5×10^5 N/m^2. Determine :*

(i) The mass of gas ;

(ii) The value of index 'n' for compression ;

(iii) The increase in internal energy of the gas ;

(iv) The heat received or rejected by the gas during compression.

Take $\gamma = 1.4$, $R = 294.2$ J/kg°C.

Solution. Initial volume of gas, $V_1 = 0.45$ m^3

Initial pressure of gas, $p_1 = 1 \times 10^5$ N/m^2

Initial temperature, $T_1 = 80 + 273 = 353$ K

Final volume after compression, $V_2 = 0.13$ m^3

The final pressure, $p_2 = 5 \times 10^5$ N/m^2.

(*i*) To find mass '*m*' using the relation

$$m = \frac{p_1 V_1}{R T_1} = \frac{1 \times 10^5 \times 0.45}{294.2 \times 353} = \mathbf{0.433\ kg.}\quad \textbf{(Ans.)}$$

(*ii*) To find index '*n*' using the relation

$$p_1 V_1^{\,n} = p_2 V_2^{\,n}$$

or
$$\left(\frac{V_1}{V_2}\right)^n = \frac{p_2}{p_1}$$

$$\left(\frac{0.45}{0.13}\right)^n = \left(\frac{5 \times 10^5}{1 \times 10^5}\right) = 5$$

or
$$(3.46)^n = 5$$

Taking log on both sides, we get

$$n \log_e 3.46 = \log_e 5$$

$$n = \log_e 5 / \log_e 3.46 = \mathbf{1.296.}\quad \textbf{(Ans.)}$$

(*iii*) In a polytropic process,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1} = \left(\frac{0.45}{0.13}\right)^{1.296-1} = 1.444$$

$\therefore$
$$T_2 = 353 \times 1.444 = 509.7\ \text{K}$$

Now, increase in internal energy,

$$\Delta U = m c_v (T_2 - T_1)$$

$$= 0.433 \times \frac{R}{(\gamma - 1)} (T_2 - T_1) \qquad \left[\because c_v = \frac{R}{(\gamma - 1)}\right]$$

$$= 0.433 \times \frac{294.2}{(1.4 - 1)\,1000} (509.7 - 353)$$

$$= \mathbf{49.9\ kJ.}\quad \textbf{(Ans.)}$$

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(*iv*) $Q = \Delta U + W$

Now, $$W = \frac{p_1V_1 - p_2V_2}{n-1} = \frac{mR(T_1 - T_2)}{n-1}$$

$$= \frac{0.433 \times 294.2(353 - 509.7)}{1.296 - 1}$$

$$= -67438 \text{ Nm or } -67438 \text{ J} = -67.44 \text{ kJ}$$

$\therefore$ $Q = 49.9 + (-67.44) = -17.54$ kJ

$\therefore$ **Heat rejected = 17.54 kJ.** **(Ans.)**

Example 3.28. *Air at 1.02 bar, 22°C, initially occupying a cylinder volume of 0.015 m^3, is compressed reversibly and adiabatically by a piston to a pressure of 6.8 bar. Calculate :*

(*i*) *The final temperature ;*

(*ii*) *The final volume ;*

(*iii*) *The work done.*

Solution. Initial pressure, $p_1 = 1.02$ bar

Initial temperature, $T_1 = 22 + 273 = 295$ K

Initial volume, $V_1 = 0.015$ m^3

Final pressure, $p_2 = 6.8$ bar

Law of compression : $pv^\gamma = C$

(*i*) **Final temperature :**

Using the relation,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_2}{295} = \left(\frac{6.8}{1.02}\right)^{\frac{1.4-1}{1.4}} \qquad [\because \quad \gamma \text{ for air} = 1.4]$$

$\therefore$ $$T_2 = 295\left(\frac{6.8}{1.02}\right)^{\frac{1.4-1}{1.4}} = 507.24 \text{ K}$$

i.e., **Final temperature** = 507.24 − 273 = **234.24°C.** **(Ans.)**

(*ii*) **Final volume :**

Using the relation,

$$p_1V_1^\gamma = p_2V_2^\gamma$$

$$\frac{p_1}{p_2} = \left(\frac{V_2}{V_1}\right)^\gamma \quad \text{or} \quad \frac{V_2}{V_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}$$

$\therefore$ $$V_2 = V_1 \times \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} = 0.015 \times \left(\frac{1.02}{6.8}\right)^{\frac{1}{1.4}} = 0.00387 \text{ m}^3$$

i.e., **Final volume = 0.00387 m^3.** **(Ans.)**

Now, work done *on the air*,

$$W = \frac{mR\,(T_1 - T_2)}{(\gamma - 1)} \qquad ...(i)$$

where m is the mass of air and is found by the following relation,

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$$pV = mRT$$

$$\therefore \quad m = \frac{p_1 V_1}{RT_1} = \frac{1.02 \times 10^5 \times 0.015}{0.287 \times 10^3 \times 295} \qquad [\because \ R \text{ for air} = 0.287 \times 10^3]$$

$$= 0.01807 \text{ kg}$$

$$\therefore \quad W = \frac{0.01807 \times 0.287 \times 10^3 (295 - 507.24)}{(1.4 - 1)} = -2751 \text{ J or } -2.751 \text{ kJ}$$

i.e., **Work done = 2.751 kJ.** (Ans.)

(– ve sign indicates that work is done on the air).

Example 3.29. *0.44 kg of air at 180°C expands adiabatically to three times its original volume and during the process, there is a fall in temperature to 15°C. The work done during the process is 52.5 kJ. Calculate c_p and c_v.*

Solution. Refer to Fig. 3.23.

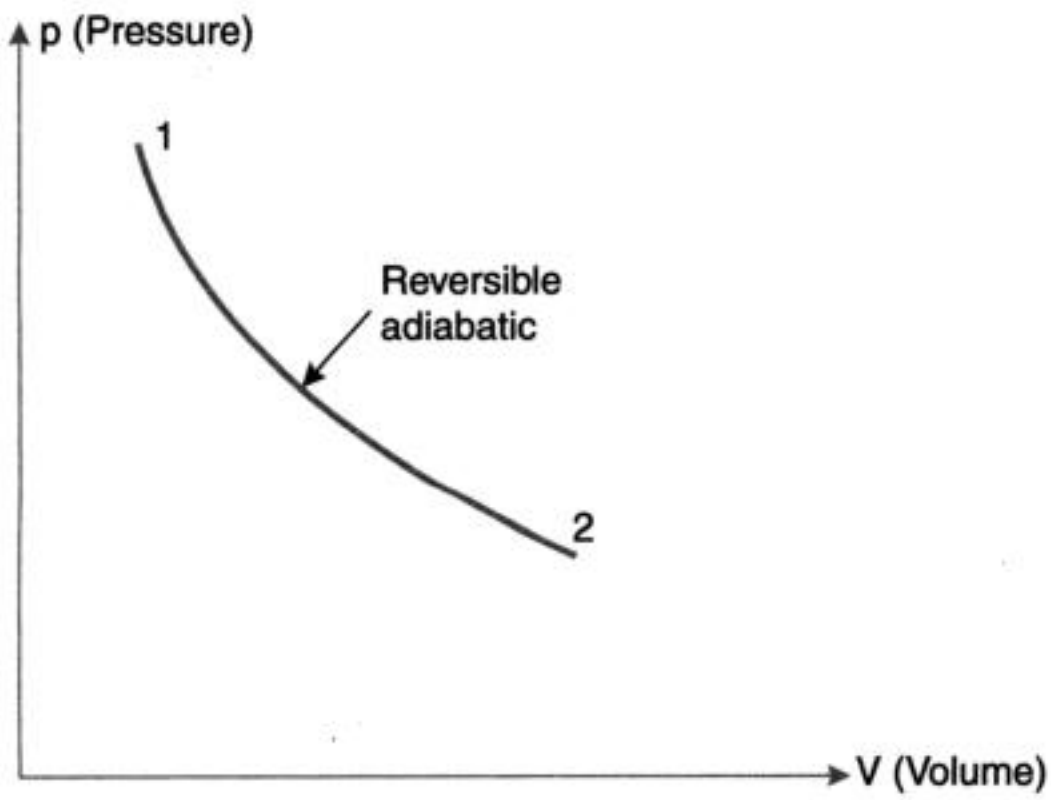

Fig. 3.23

Mass of air, $m = 0.44$ kg

Initial temperature, $T_1 = 180 + 273 = 453$ K

$$\text{Ratio} = \frac{V_2}{V_1} = 3$$

Final temperature, $T_2 = 15 + 273 = 288$ K

Work done during the process, $W_{1\text{-}2} = 52.5$ kJ

$$c_p = ?, \ c_v = ?$$

For adiabatic process, we have

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

$$\frac{288}{453} = \left(\frac{1}{3}\right)^{\gamma - 1} \qquad \text{or} \qquad 0.6357 = (0.333)^{\gamma - 1}$$

or Taking log on both sides, we get

$$\log_e (0.6357) = (\gamma - 1) \log_e (0.333)$$

$$-0.453 = (\gamma - 1) \times (-1.0996)$$

$$\therefore \quad \gamma = \frac{0.453}{1.0996} + 1 = 1.41$$

Also, $$\frac{c_p}{c_v} = \gamma = 1.41$$

Work done during adiabatic process,

$$W_{1\text{-}2} = \frac{mR(T_1 - T_2)}{\gamma - 1}$$

$$\therefore \quad 52.5 = \frac{0.44\, R(453 - 288)}{(1.41 - 1)}$$

$$\therefore \quad R = \frac{52.5\,(1.41 - 1)}{0.44\,(453 - 288)} = 0.296$$

$$\therefore \quad c_p - c_v = 0.296 \qquad [\because \quad R = c_p - c_v]$$

Also $$\frac{c_p}{c_v} = 1.41 \qquad \text{or} \qquad c_p = 1.41\, c_v$$

$$\therefore \quad 1.41\, c_v - c_p = 0.296$$

or $$c_v = \mathbf{0.722\ kJ/kg\ K.}\ \text{(Ans.)}$$

and $$c_p = \mathbf{1.018\ kJ/kg\ K.}\ \text{(Ans.)}$$

☞ **Example 3.30.** *1 kg of ethane (perfect) gas is compressed from 1.1 bar, 27°C according to a law $pV^{1.3}$ = constant, until the pressure is 6.6 bar. Calculate the heat flow to or from the cylinder walls.*

Given : Molecular weight of ethane = 30, c_p = 1.75 kJ/kg K.

Solution. Mass of ethane gas, $m = 1$ kg

Initial pressure, $p_1 = 1.1$ bar

Initial temperature, $T_1 = 27 + 273 = 300$ K

Final pressure, $p_2 = 6.6$ bar

Law of compression, $pV^{1.3} = C$

Quantity of heat transferred, Q :

Now, characteristic gas constant,

$$R = \frac{\text{Universal gas constant } (R_0)}{\text{Molecular weight } (M)}$$

$$= \frac{8314}{30} = 277.13 \text{ Nm/kg K} = 277.31 \text{ J/kg K}$$

$$= 0.277 \text{ kJ/kg K}$$

Also $$c_p - c_v = R$$

$$\therefore \quad c_v = c_p - R = 1.75 - 0.277 = 1.473 \text{ kJ/kg K}$$

$$\gamma = \frac{c_p}{c_v} = \frac{1.75}{1.473} = 1.188$$

In case of a polytropic process,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\frac{6.6}{1.1}\right)^{\frac{1.3-1}{1.3}} = 1.5119$$

$$\therefore \quad T_2 = 300 \times 1.5119 = 453.6 \text{ K}$$

Now, work done, $W = \dfrac{R(T_1 - T_2)}{n-1} = \dfrac{0.277(300 - 453.6)}{1.3 - 1} = -141.8$ kJ/kg

To find heat flow, using the relation,

$$Q = \left(\frac{\gamma - n}{\gamma - 1}\right) W = \left(\frac{1.188 - 1.3}{1.188 - 1}\right) \times -141.8 = +84.5 \text{ kJ/kg}$$

i.e., **Heat supplied = 84.5 kJ/kg.** **(Ans.)**

Example 3.31. *Fig 3.24 shows a cylinder of 8 cm inside diameter having a piston loaded with a spring (stiffness = 150 N/cm of compression). The initial pressure, volume and temperature of air in the cylinder are 3×10^5 N/m^2, 0.000045 m^3 and 20°C respectively. Determine the amount of heat added to the system so that piston moves by 3.5 cm.*

Assume c_v = 0.71 kJ/kg K and R = 0.287 kJ/kg K.

Solution. Insider diameter of the cylinder

= 8 cm

Stiffness of the spring, S = 150 N/cm

Initial pressure of air,

$p_1 = 3 \times 10^5$ N/m^2 or 30 N/cm^2

Initial volume of air,

$V_1 = 0.000045$ m^3 = 45 cm^3

Initial temperature of air,

$T_1 = 20 + 273 = 293$ K

Specific heat at constant volume,

$c_v = 0.71$ kJ/kg K

Characteristic constant for air,

$R = 0.287$ kJ/kg K

Refer Fig. 3.25.

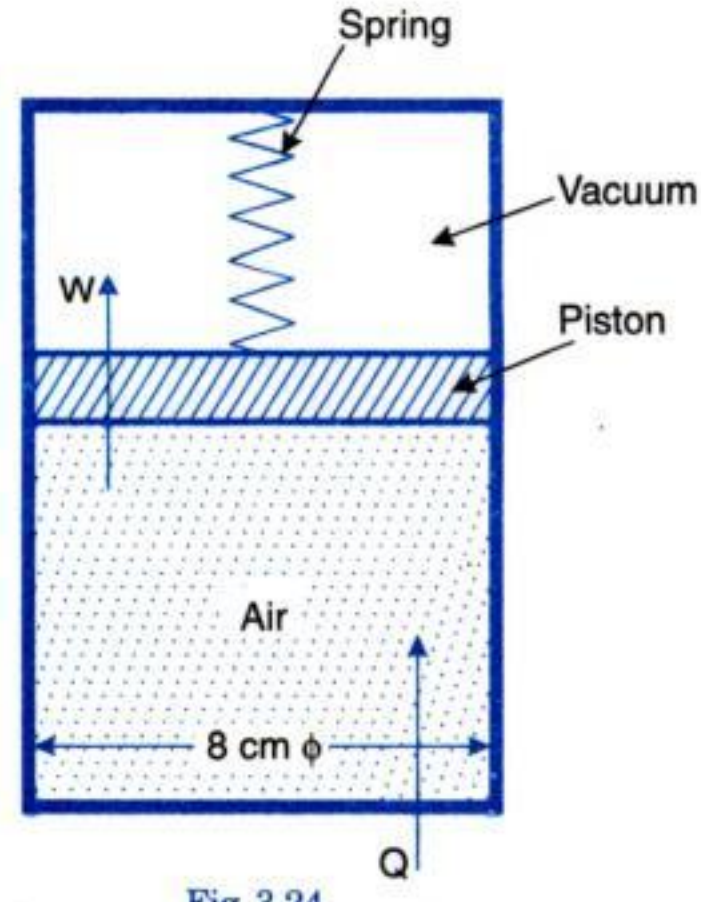

Fig. 3.24

Let, *oo* = An arbitrary datum from which the position of the lower face of the piston is to be measured,

y = Distance of the lower face of the piston,

$y = y_0$, when spring length is its free length, and

p = Pressure of air within the cylinder when $y = y_0$.

Now, force balance for the piston is given by

$$Ap = S(y - y_0) \qquad ...(i)$$

where, A = The area of the piston, and

S = Stiffness of the spring.

With heat transfer to the air, let the pressure inside the cylinder increase by dp forcing the piston to move upward by distance dy. Now the force balance for the piston is

$$A(p + dp) = S(y + dy - y_0) \qquad ...(ii)$$

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From eqns. (*i*) and (*ii*), we have

$$Adp = Sdy \qquad ...(iii)$$

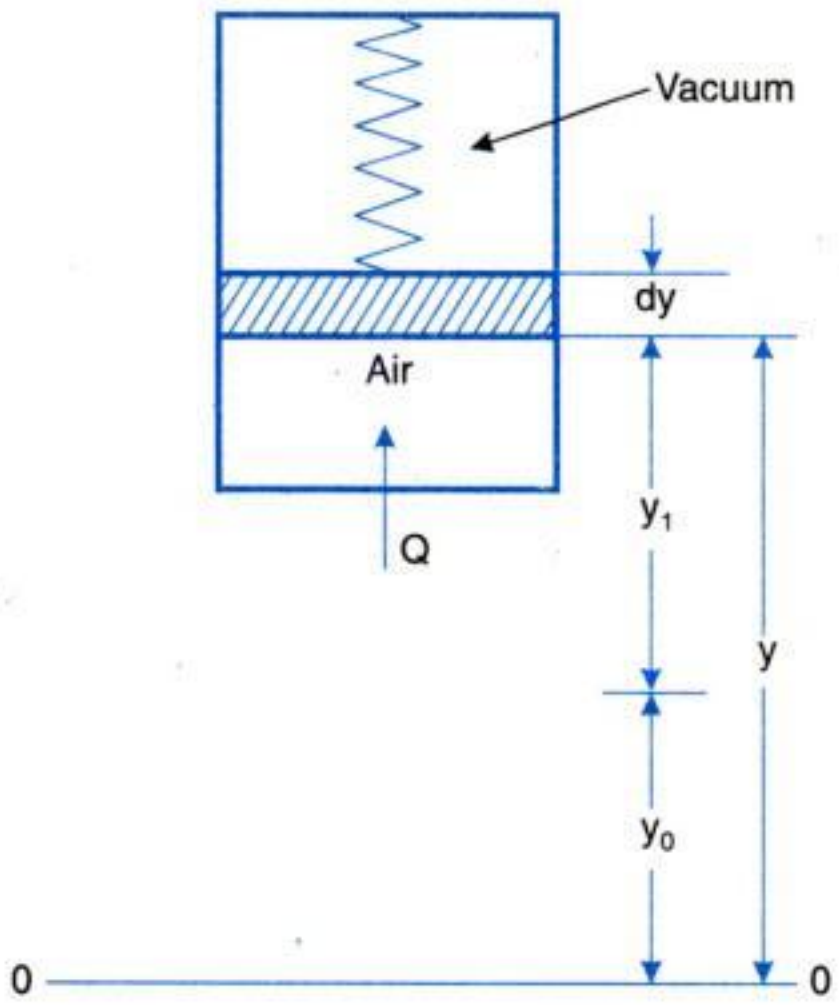

Fig. 3.25

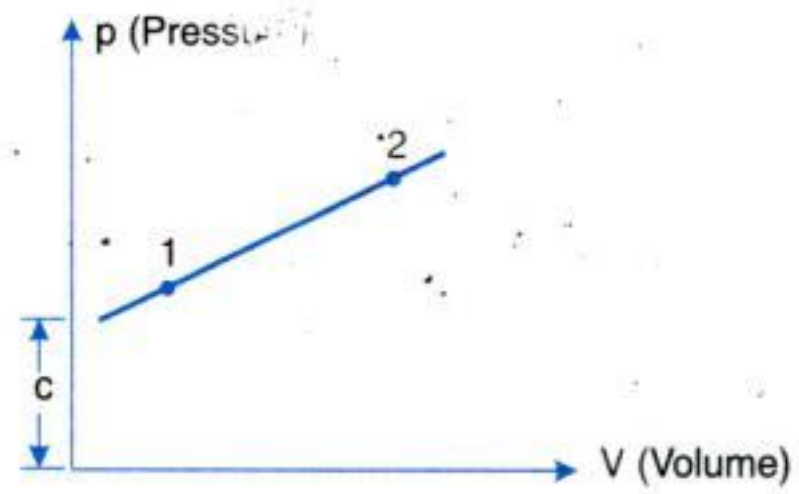

Fig. 3.26

The increase in volume dV of the gas for the piston displacement is given by

$$dV = Ady \qquad ...(iv)$$

$$\therefore \quad dp = \frac{S}{A^2}\,dy \qquad ...(v)$$

$$\therefore \quad p = \frac{S}{A^2}\,V + C \qquad ...(vi)$$

The p-V relationship for the process is a straight line (Fig. 3.26) having a slope of $\frac{S}{A^2}$ and pressure axis intercept of C. The value of C can be found out from the knowledge of pressure and volume at any state point.

Now, substituting the values of p_1, V_1, A in eqn. (*vi*), we get

$$p = \frac{150}{\left(\frac{\pi}{4} \times 8^2\right)^2} V + C$$

or $$p = 0.0594\,V + C \qquad ...(vii)$$

where p is in N/cm^2 and V is in cm^3.

$$\therefore \quad p_1 = 0.0594\,V_1 + C$$

$$30 = 0.0594 \times 45 + C$$

$$\therefore \quad C = 27.33$$

Hence, p-V relationship for the process is,

$$p = 0.0594\,V + 27.33 \qquad ...(viii)$$

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During the process the piston is moved by a distance of 3.5 cm.
This increases the volume of gas by

$$3.5 \times A^2 = 3.5 \times \left(\frac{\pi}{4} \times 8^2\right) = 175.9 \text{ cm}^3$$

Hence, the final volume of air,

$$V_2 = 45 + 175.9 = 220.9 \text{ cm}^3$$

Substituting this value in equation (*viii*), we get

$$p(= p_2) = 0.0594 \times 220.9 + 27.33 = 40.45 \text{ N/cm}^2$$

The work done W during the process is given by

$$W = \int p dV = \int_{p_1}^{p_2} \frac{A^2}{S} p dp$$

$$= \frac{A^2}{S}\left(\frac{p_2^{\,2} - p_1^{\,2}}{2}\right) = \frac{A^2}{S}\left(\frac{p_2 + p_1}{2}\right)\left(\frac{p_2 - p_1}{2}\right)$$

$$= \frac{A^2}{S}\left(\frac{p_2 + p_1}{2}\right)\frac{S}{A^2}(V_2 - V_1)$$

or
$$W = \left(\frac{p_2 + p_1}{2}\right)(V_2 - V_1) \qquad ...(ix)$$

= (Mean pressure) × (Change in volume)

$$W = \left(\frac{40.45 + 30}{2}\right) \times (220.9 - 45)$$

= 6196 N-cm or 61.96 Nm

It may be noted that work done does not cross the system boundary when spring and cylinder are considered system.

Now, to find T_2, using the relation,

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

$\therefore$ $$T_2 = \frac{p_2 V_2 T_1}{p_1 V_1} = \frac{40.45 \times 220.9 \times 293}{30 \times 45} = 1939.3 \text{ K}$$

Also, $$m = \frac{p_1 v_1}{R_1 T_1} = \frac{30 \times 45}{(0.287 \times 10^3) \times 293} = 0.0001605 \text{ kg}$$

Now, change in internal energy,

$$\Delta U = m \times c_v \times (T_2 - T_1)$$

$$= 0.0001605 \times 0.71 \times (1939.3 - 293) = 0.1876 \text{ kJ}$$

According to first law,

$$Q_{1-2} = \Delta U + W$$

$$= 0.1876 + 61.96 \times 10^{-3} = 0.2495 \text{ kJ}$$

$\therefore$ **Amount of heat added to the system = 0.2495 kJ.** **(Ans.)**

3.10. APPLICATION OF FIRST LAW TO STEADY FLOW PROCESS

Steady Flow Energy Equation (S.F.E.E.)

In many practical problems, the rate at which the fluid flows through a machine or piece of apparatus is constant. This type of flow is called *steady flow.*

Assumptions :

The following *assumptions* are made in the system analysis :

(*i*) The mass flow through the system remains constant.

(*ii*) Fluid is uniform in composition.

(*iii*) The only interaction between the system and surroundings are work and heat.

(*iv*) The state of fluid at any point remains constant with time.

(*v*) In the analysis only potential, kinetic and flow energies are considered.

Fig. 3.27 shows a schematic flow process for an open system. An open system is one in which both mass and energy may cross the boundaries. A wide interchange of energy may take place within an open system. Let the system be an automatic engine with the inlet manifold at the first state point and exhaust pipe as the second point. There would be an interchange of chemical energy in the fuel, kinetic energy of moving particles, internal energy of gas and heat transferred and shaft work within the system. From Fig. 3.27 it is obvious that if there is no variation of flow of mass or energy with time across the boundaries of the system the steady flow will prevail. The conditions may pass through the cyclic or non-cyclic changes within the system. As a result the mass entering the system equals the mass leaving, also energy entering the system equals energy leaving.

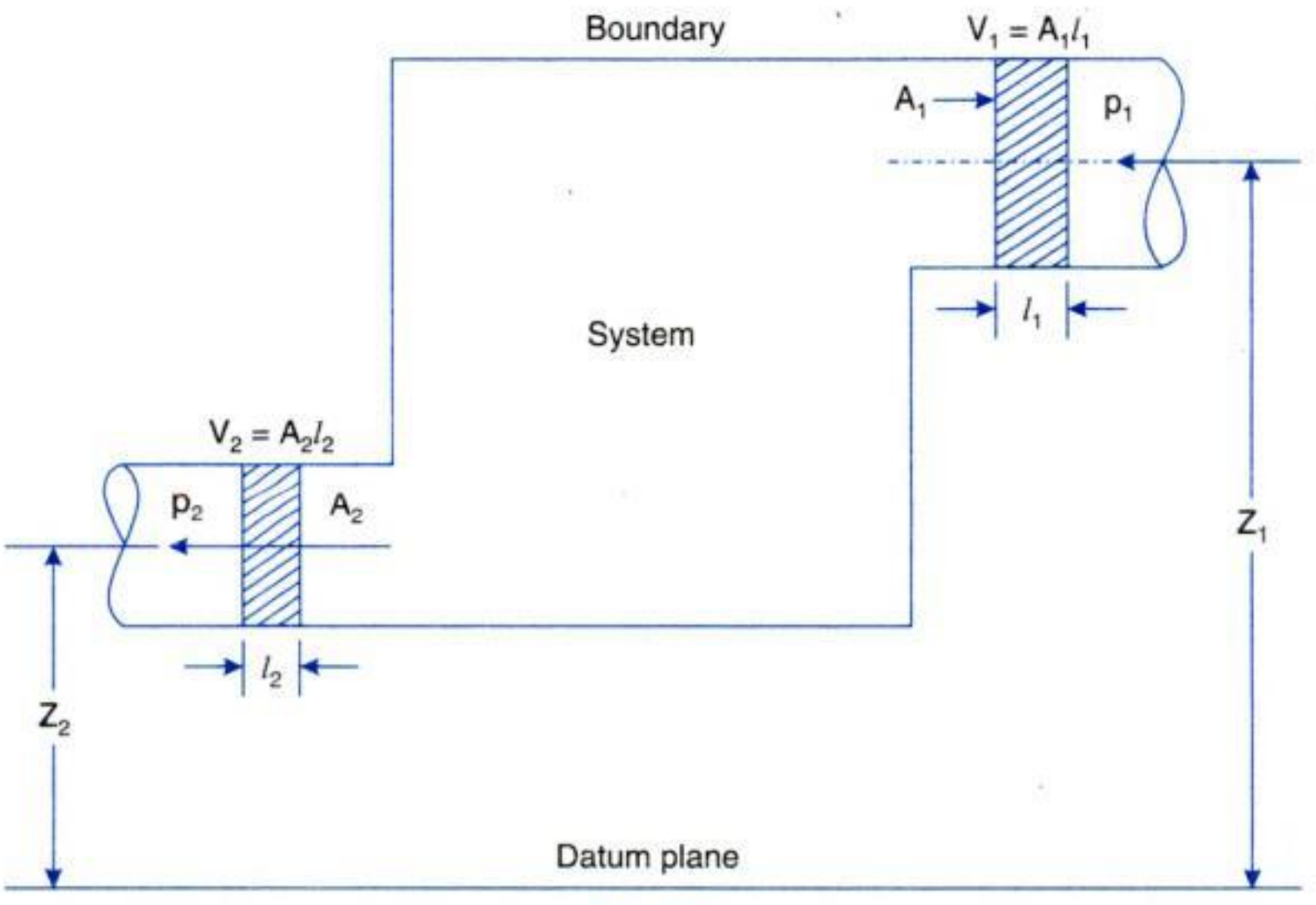

Fig. 3.27

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$$\therefore \qquad \int_1^2 pdv = -\int_1^2 vdp$$

The above equation indicates that the *area under both curves is same for an isothermal process.*

Note. In all the above equations 'v' represents volume per unit mass as mass flow is considered unity.

Now let us find out expressions for work done for different flow processes as follows :

(*i*) **Steady flow constant pressure process :**

$$W = -\int_1^2 v \,.\, dp = 0 \qquad [\because \quad dp = 0] \qquad ...(3.51)$$

(*ii*) **Steady flow constant volume process :**

$$W = -\int_1^2 Vdp = -V(p_2 - p_1) = V(p_1 - p_2)$$

i.e.,
$$W = V(p_1 - p_2) \qquad ...(3.52)$$

(*iii*) **Steady flow constant temperature process :**

The constant temperature process is represented by

$$pV = p_1V_1 = p_2V_2 = C \text{ (constant)}$$

$$\therefore \qquad W = -\int_1^2 Vdp$$

$$= -\int_1^2 \frac{C}{p} dp \qquad \left[\because V = \frac{C}{p}\right]$$

$$= -C\int_1^2 \frac{dp}{p} = -C\left|\log_e p\right|_1^2$$

$$= -C \log_e \frac{p_2}{p_1} = C \log_e \frac{p_1}{p_2}$$

i.e.,
$$W = p_1V_1 \log_e \left(\frac{p_1}{p_2}\right) \qquad ...(3.53)$$

Now substituting the values of W in the equation (3.49), considering unit mass flow :

(*a*) The energy equation for *constant pressure flow process*

$$dQ = \Delta PE + \Delta KE + \Delta h$$

$$= \Delta h \text{ (if } \Delta PE = 0 \text{ and } \Delta KE = 0).$$

(*b*) The energy equation for constant volume flow process

$$dQ = -\int_1^2 vdp + \Delta PE + \Delta KE + \Delta u + pdv + vdp$$

$$= \Delta PE + \Delta KE + \Delta u \qquad \left[\because pdv = 0 \text{ and } v \,.\, dp = \int_1^2 vdp\right]$$

$$\therefore \qquad dQ = \Delta u \text{ (if } \Delta PE = 0 \text{ and } \Delta KE = 0)$$

3.12 ENGINEERING APPLICATIONS OF STEADY FLOW ENERGY EQUATION (S.F.E.E.)

3.12.1. Water Turbine

Refer Fig. 3.29. In a water turbine, water is supplied from a height. The potential energy of water is converted into kinetic energy when it enters into the turbine and part of it is converted into useful work which is used to generate electricity.

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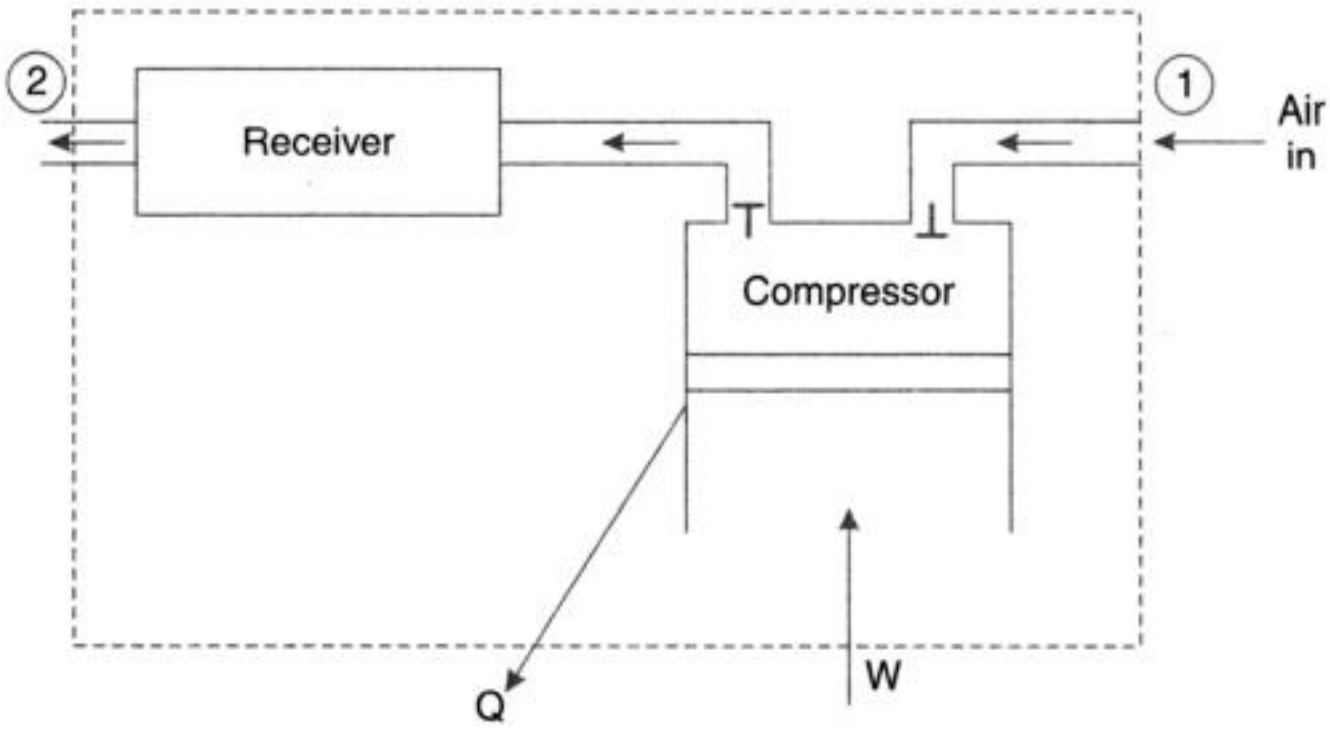

Fig. 3.33. Reciprocating compressor.

Applying energy equation to the system, we have :

$\Delta PE = 0$ and $\Delta KE = 0$ since these changes are negligible compared with other energies.

$$\therefore \quad h_1 - Q = h_2 - W \qquad ...(3.58)$$

3.12.6. Boiler

A boiler transfers heat to the incoming water and generates the steam. The system is shown in Fig. 3.34.

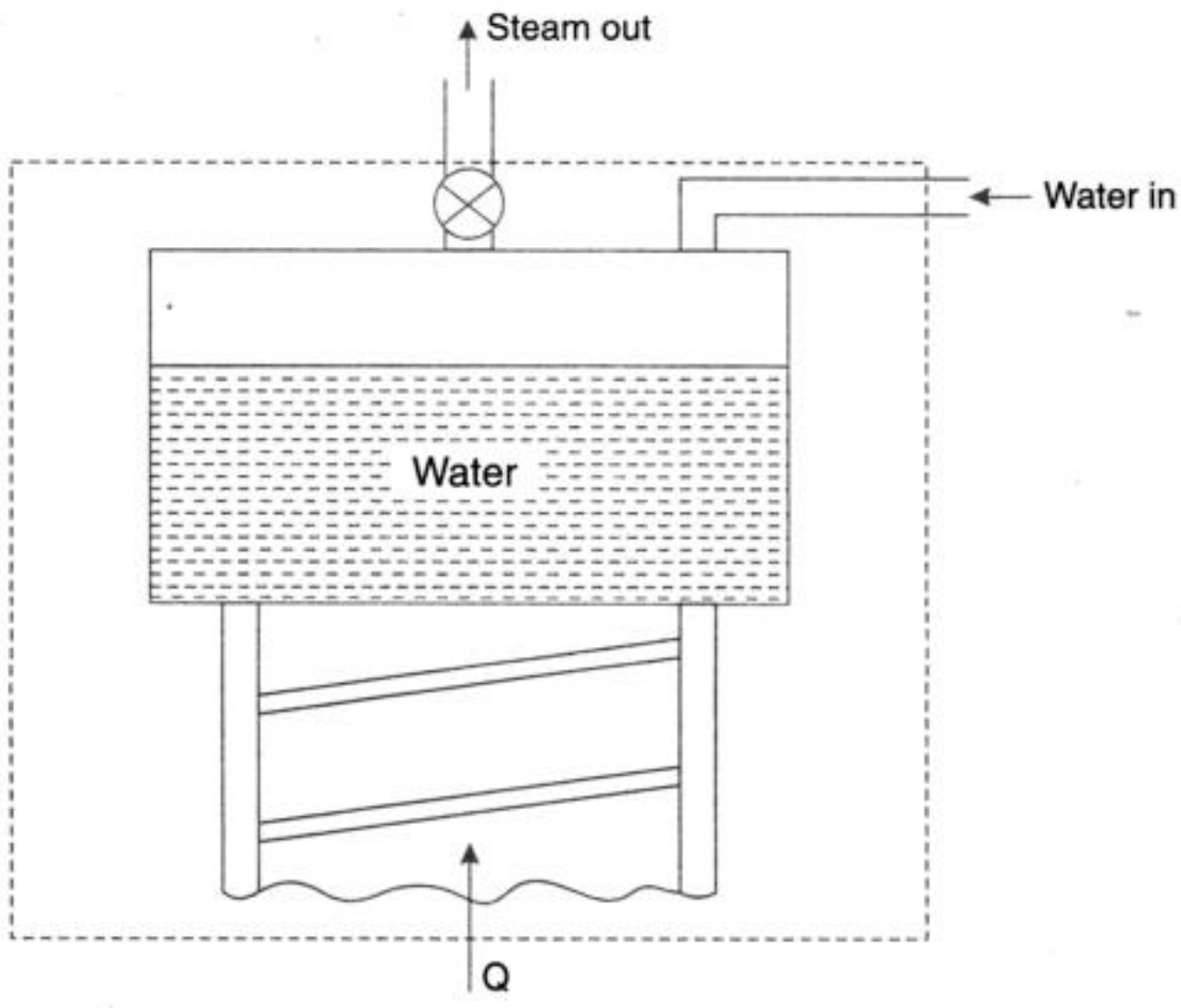

Fig. 3.34. Boiler.

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Fig. 3.37 shows a commonly used convergent-divergent nozzle.

For this system,

$$\Delta PE = 0$$
$$W = 0$$
$$Q = 0$$

Applying the energy equation to the system,

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

or $$\frac{C_2^2}{2} - \frac{C_1^2}{2} = h_1 - h_2 \quad \text{or} \quad C_2^2 - C_1^2 = 2(h_1 - h_2)$$

or $$C_2^2 = C_1^2 + 2(h_1 - h_2)$$

$\therefore$ $$C_2 = \sqrt{C_1^2 + 2(h_1 - h_2)} \qquad ...(3.62)$$

where velocity C is in m/s and enthalpy h in joules.

If $C_1 << C_2$, then

$$C_2 = \sqrt{2(h_1 - h_2)} \qquad ...[3.63\ (a)]$$

$\therefore$ $$C_2 = \sqrt{2\Delta h}\ . \qquad ...[3.63\ (b)]$$

3.13. THROTTLING PROCESS AND JOULE-THOMSON POROUS PLUG EXPERIMENT

Throttling process involves the *passage of a higher pressure fluid through a narrow constriction. The effect is the reduction in pressure and increase in volume.* This process is *adiabatic* as no heat flows from and to the system, but it is *not reversible.* It is *not an isentropic process.* The *entropy* of the fluid actually *increases.*

Such a process occurs in a flow through a *porous plug,* a *partially closed valve* and a *very narrow orifice.* The porous plug is shown in Fig. 3.38.

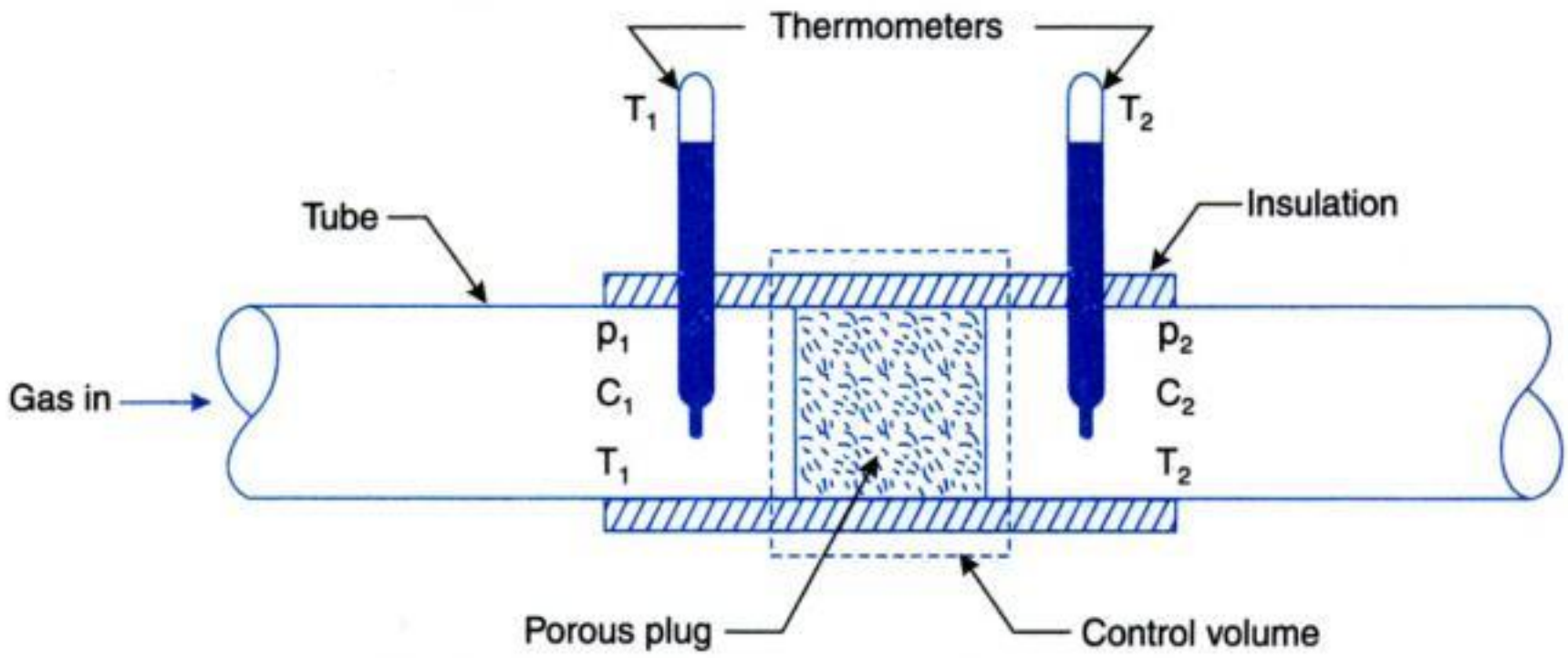

Fig. 3.38. The Joule-Thomson porous plug experiment.

In this system,

$Q = 0$ ($\because$ System is isolated)

$W = 0$ ($\because$ There is no work interaction)

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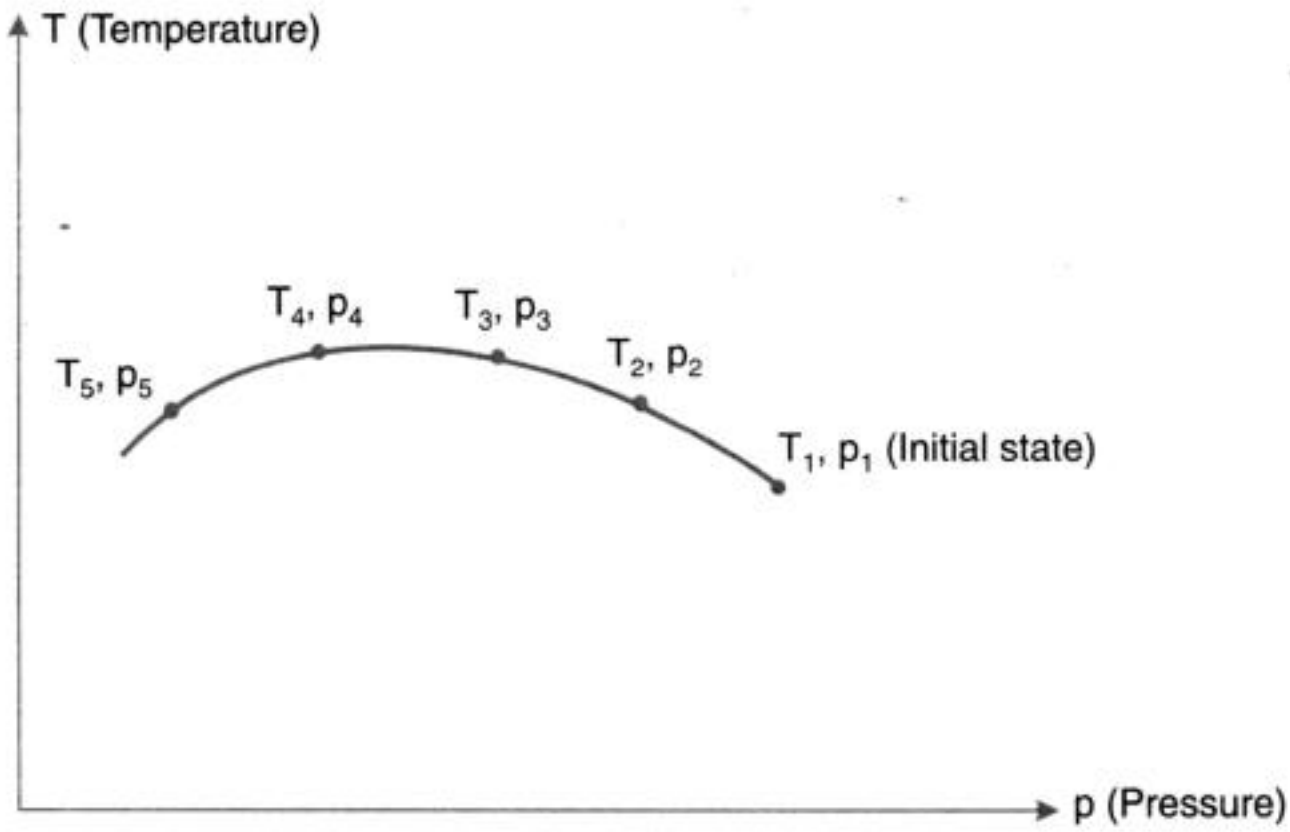

Fig. 3.39. Constant enthalpy curve.

$\Delta PE = 0$ ($\because$ Inlet and outlet are at the same level)

$\Delta KE = 0$ ($\because$ Kinetic energy does not change significantly)

Applying the energy equation to the system

$$h_1 = h_2$$

This shows that *enthalpy remains constant during adiabatic throttling process.*

The throttling process is commonly used for the following purposes :

(*i*) For determining the condition of steam (dryness fraction).

(*ii*) For controlling the speed of the turbine.

(*iii*) Used in refrigeration plant for reducing the pressure of the refrigerant before entry into the evaporator.

Throttling process frequently encountered in practice was investigated by Joule and Thomson (Lord Kelvin) in their famous *porous plug experiment* (Fig. 3.38). A stream of gas at pressure p_1 and temperature T_1 is forced continuously through a *porous plug* in a tube from which it emerges at a lower pressure p_2 and temperature T_2. The whole apparatus is *thermally insulated.*

In this process (as earlier stated)

$$h_1 = h_2$$

Whether the temperature and internal energy change in a throttling process depends on whether the fluid behaves as an ideal gas or not. Since the enthalpy of an ideal gas is a function of temperature alone, it follows that

$$T_1 = T_2 \text{ for (throttling process)}_{ideal\ gas} \qquad ...(3.64)$$

and, therefore, $u_1 = u_2$

For an *ideal gas,* therefore, the *throttling* process takes place at

(*i*) constant enthalpy,

(*ii*) constant temperature, and

(*iii*) constant internal energy.

The *enthalpy of a real gas is not a function of temperature alone.* In this case

$$T_1 \neq T_2 \qquad ...(3.65)$$

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Enthalpy of gases at the inlet, $h_1 = 1260$ kJ/kg
Enthalpy of gases at the oulet, $h_2 = 400$ kJ/kg
Velocity of gases at the inlet, $C_1 = 50$ m/s
Velocity of gases at the outlet, $C_2 = 110$ m/s.

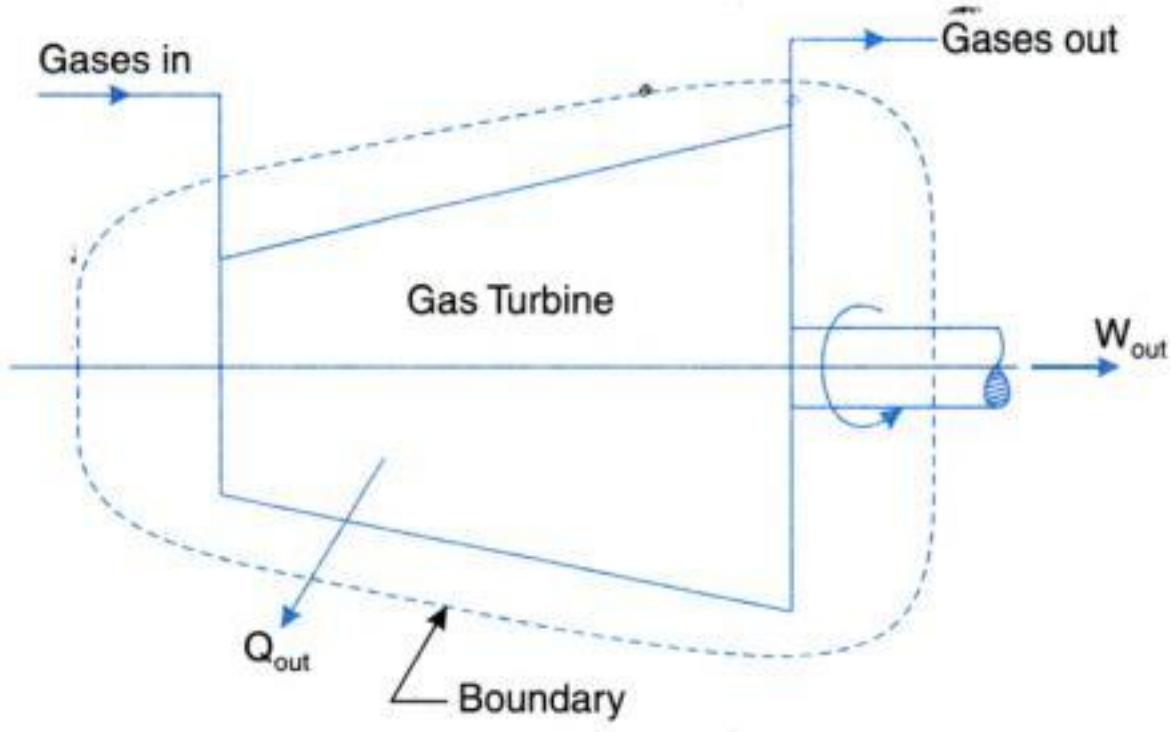

Fig. 3.42

(*i*) **Heat rejected, Q :**
Using the flow equation,

$$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W \qquad \ldots(i)\ [\because \quad Z_1 = Z_2]$$

Kinetic energy at inlet $= \frac{C_1^2}{2} = \frac{50^2}{2}$ m^2/s^2 $= \frac{50^2 \text{ kg m}^3}{2 \text{ s}^2\text{kg}} = 1250$ Nm/kg $= 1.25$ kJ/kg

Kinetic energy at outlet $= \frac{C_1^2}{2} = \frac{110^2}{2 \times 1000} = 6.05$ kJ/kg

Substituting these values in eqn. (*i*), we get

$$1260 + 1.25 + Q = 400 + 6.05 + 800$$

$\therefore \quad Q = -55.2$ kJ/kg

i.e., **Heat rejected** $= +55.2$ kJ/kg $= 55.2 \times 15$ kJ/s $=$ **828 kW.** (Ans.)

(*ii*) **Inlet area, A :**
Using the relation,

$$\dot{m} = \frac{CA}{v}$$

$$\therefore \quad A = \frac{v\dot{m}}{C} = \frac{0.45 \times 15}{50} = \mathbf{0.135\ m^2.} \quad \text{(Ans.)}$$

☞ **Example 3.34.** *In an air compressor air flows steadily at the rate of 0.5 kg/s through an air compressor. It enters the compressor at 6 m/s with a pressure of 1 bar and a specific volume of 0.85 m^3/kg and leaves at 5 m/s with a pressure of 7 bar and a specific volume of 0.16 m^3/kg. The internal energy of the air leaving is 90 kJ/kg greater than that of the air entering. Cooling water in a jacket surrounding the cylinder absorbs heat from the air at the rate of 60 kJ/s. Calculate :*

(*i*) *The power required to drive the compressor ;*

(*ii*) *The inlet and output pipe cross-sectional areas.*

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The steady flow energy equation is given by

$$h_1 + \frac{C_1^2}{2} + Z_1 g + Q = h_2 + \frac{C_2^2}{2} + Z_2 g + W \qquad ...(i)$$

Considering the nozzle as an open system, it is evident that :

— there is *no work transfer* across the boundary of the system (*i.e.*, $W = 0$)

— there is *no heat transfer* because the nozzle is insulated (*i.e.*, $Q = 0$).

— the change in potential energy is negligible since there is no significant difference in elevation between the entrance and exit of the nozzle [*i.e.*, $(Z_2 - Z_1)\, g = 0$].

Thus eqn. (*i*) reduces to

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

$$\therefore \qquad (h_2 - h_1) + \frac{C_2^2 - C_1^2}{2} = 0$$

From steam table corresponding to 6.87 bar, $h_1 = 2850$ kJ/kg

$$\therefore \qquad (h_2 - 2850) + \frac{(500)^2 - (50)^2}{2 \times 1000} = 0$$

or $$h_2 - 2850 + 123.75 = 0 \text{ or } h_2 = 2726.25 \text{ kJ}$$

Hence final enthalpy of steam = 2726.25 kJ. (Ans.)

Example 3.38. *The working fluid, in a steady flow process flows at a rate of 220 kg/min. The fluid rejects 100 kJ/s passing through the system. The conditions of the fluid at inlet and outlet are given as : $C_1 = 320$ m/s, $p_1 = 6.0$ bar, $u_1 = 2000$ kJ/kg, $v_1 = 0.36$ m^3/kg and $C_2 = 140$ m/s, $p_2 = 1.2$ bar, $u_2 = 1400$ kJ/kg, $v_2 = 1.3$ m^3/kg. The suffix 1 indicates the condition at inlet and 2 indicates at outlet of the system.*

Determine the power capacity of the system in MW.

The change in potential energy may be neglected.

Solution. Refer Fig. 3.45.

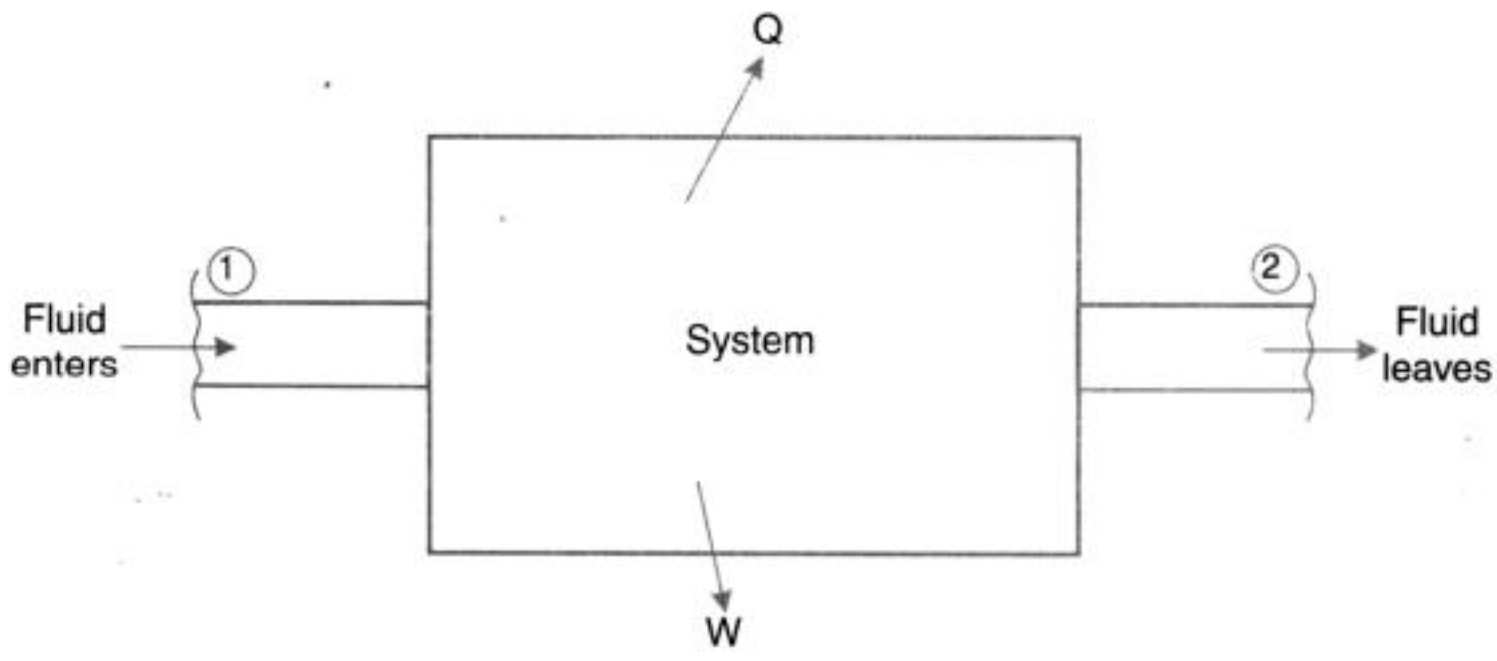

Fig. 3.45

Conditions of the fluid at point 1 :

Velocity, $C_1 = 320$ m/s

Pressure, $p_1 = 6.0 \text{ bar} = 6 \times 10^5 \text{ N/m}^2$

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Increase in enthalpy of air passing through the compressor,

$$(h_2 - h_1) = 150 \text{ kJ/kg}$$

Heat lost to the surroundings,

$$Q = -700 \text{ kJ/min} = -11.67 \text{ kJ/s.}$$

(*i*) Motor power required to drive the compressor :

Applying energy equation to the system,

$$m\left(h_1 + \frac{C_1^2}{2} + Z_1 g\right) + Q = m\left(h_2 + \frac{C_2^2}{2} + Z_2 g\right) + W$$

Now $Z_1 = Z_2$ (given)

$\therefore$ $$m\left(h_1 + \frac{C_1^2}{2}\right) + Q = m\left(h_2 + \frac{C_2^2}{2}\right) + W$$

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

$$= 0.2\left[-150 + \frac{12^2 - 90^2}{2 \times 1000}\right] + (-11.67)$$

$$= -42.46 \text{ kJ/s} = -42.46 \text{ kW}$$

$\therefore$ **Motor power required (or work done on the air) = 42.46 kW.** (Ans.)

(*ii*) Ratio of inlet to outlet pipe diameter, $\frac{d_1}{d_2}$:

The mass of air passing through the compressor is given by

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

$\therefore$ $$\frac{A_1}{A_2} = \frac{C_2}{C_1} \times \frac{v_1}{v_2} = \frac{90}{12} \times \frac{0.5}{0.14} = 26.78$$

$\therefore$ $$\left(\frac{d_1}{d_2}\right)^2 = 26.78 \quad \text{or} \quad \frac{d_1}{d_2} = 5.175$$

Hence ratio of inlet to outlet pipe diameter = 5.175. (Ans.)

Example 3.41. *In a test of water cooled air compressor, it is found that the shaft work required to drive the compressor is 175 kJ/kg of air delivered and the enthalpy of air leaving is 70 kJ/kg greater than that entering and that the increase in enthalpy of circulating water is 92 kJ/kg.*

Compute the amount of heat transfer to the atmosphere from the compressor per kg of air.

Solution. Refer Fig. 3.48.

Shaft work required to drive the compressor, $W = -175$ kJ/kg

Increase in enthalpy of air passing through the compressor, $(h_2 - h_1) = 70$ kJ/kg

Increase in enthalpy of circulating water, $Q_{water} = -92$ kJ/kg

Amount of heat transferred to atmosphere, $Q_{atm.} = ?$

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from the gas is negligible. Using the following data evaluate the relative velocity of gas leaving the jet pipe. For the gas at t = 820°C; h = 800 kJ/kg and at 910°C, 915 kJ/kg.

Solution. Pressure at entry to the engine jet pipe, $p_1 = 2$ bar

Velocity relative to the pipe, $C_1 = 300$ m/s

Heat transfer from gas, $Q = 0$

At temperature, $t_1 = 910°C$, $h_1 = 915$ kJ/kg

At temperature, $t_2 = 820°C$, $h_2 = 800$ kJ/kg

Relative velocity of gas leaving the jet pipe, C_2 :

Steady flow energy equation is given by :

$$h_1 + \frac{C_1^2}{2} + Z_1 g + Q = h_2 + \frac{C_2^2}{2} + Z_2 g + W$$

$$Q = 0$$

$$W = 0$$

$$Z_1 = Z_2 \text{ (assumed)}$$

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

$$\frac{C_2^2}{2} = (h_1 - h_2) + \frac{C_1^2}{2} = (915 - 800) \times 1000 + \frac{300^2}{2}$$

$$\therefore \quad C_2^2 = 320000 \quad \text{or} \quad C_2 = 565.7 \text{ m/s.}$$

Hence relative velocity of gas leaving the jet pipe = 565.7 m/s. (Ans.)

Example 3.45. *A centrifugal pump delivers 50 kg of water per second. The inlet and outlet pressures are 1 bar and 4.2 bar respectively. The suction is 2.2 m below the centre of the pump and delivery is 8.5 m above the centre of the pump. The suction and delivery pipe diameters are 20 cm and 10 cm respectively.*

Determine the capacity of the electric motor to run the pump.

Solution. Refer Fig. 3.51.

Quantity of water delivered by the pump, $m_w = 50$ kg/s

Inlet pressure, $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2

Outlet pressure, $p_2 = 4.2$ bar $= 4.2 \times 10^5$ N/m^2

Suction-below the centre of the pump = 2.2 m

Delivery-above the centre of the pump = 8.5 m

Diameter of suction pipe, $d_1 = 20$ cm $= 0.2$ m

Diameter of delivery pipe, $d_2 = 10$ cm $= 0.1$ m

Capacity of electric motor :

Steady flow energy equation is given by

$$m_w \left(u_1 + p_1 v_1 + \frac{C_1^2}{2} + Z_1 g \right) + Q = m_w \left(u_2 + p_2 v_2 + \frac{C_2^2}{2} + Z_2 g \right) + W \qquad ...(i)$$

Considering the datum from suction 1, as shown

$$Z_1 = 0, Z_2 = 8.5 + 2.2 = 10.7 \text{ m}$$

$$u_2 - u_1 = 0 \; ; Q = 0$$

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(*i*) **Heat exchanger :**

Rate of heat transfer :

Energy equation is given as,

$$m\left(h_1 + \frac{C_1^2}{2} + Z_1 g\right) + Q_{1\text{-}2} = m\left(h_2 + \frac{C_2^2}{2} + Z_2 g\right) + W_{1\text{-}2}$$

Here, $Z_1 = Z_2,\ C_1, C_2 = 0,\ W_{1\text{-}2} = 0$

$\therefore$ $mh_1 + Q_{1\text{-}2} = mh_2$

or $Q_{1\text{-}2} = m(h_2 - h_1)$

$= mc_p\,(t_2 - t_1) = 2.5 \times 1.005\,(820 - 20) = 2010$ kJ/s.

Hence, **rate of heat transfer = 2010 kJ/s.** **(Ans.)**

(*ii*) **Turbine :**

Power output of turbine :

Energy equation for turbine gives

$$m\left(h_2 + \frac{C_2^2}{2}\right) = m\left(h_3 + \frac{C_3^2}{2}\right) + W_{2\text{-}3} \qquad [\because\ Q_{2\text{-}3} = 0,\ Z_1 = Z_2]$$

$\therefore$ $$W_{2-3} = m\left(h_2 + \frac{C_2^2}{2}\right) - m\left(h_3 + \frac{C_3^2}{2}\right)$$

$$= m\left[(h_2 - h_3) + \left(\frac{C_2^2 - C_3^2}{2}\right)\right]$$

$$= m\left[c_p\,(t_2 - t_3) + \frac{C_2^2 - C_3^2}{2}\right]$$

$$= 2.5\left[1.005\,(820 - 620) + \frac{(40)^2 - (55)^2}{2 \times 1000}\right]$$

$= 2.5\ [201 + 0.7125] = 504.3$ kJ/s or 504.3 kW

Hence, **power output of turbine = 504.3 kW.** **(Ans.)**

(*iii*) **Nozzle :**

Velocity at exit from the nozzle :

Energy equation for nozzle gives,

$$h_3 + \frac{C_3^2}{2} = h_4 + \frac{C_4^2}{2} \qquad [\because\ W_{3\text{-}4} = 0,\ Q_{3-4} = 0,\ Z_1 = Z_2]$$

$$\frac{C_4^2}{2} = (h_3 - h_4) + \frac{C_3^2}{2} = c_p(t_3 - t_4) + \frac{C_3^2}{2}$$

$$= 1.005(620 - 510) + \frac{55^2}{2 \times 1000} = 112.062 \times 10^3\ \text{J}$$

$\therefore$ $C_4 = 473.4$ m/s.

Hence, **velocity at exit from the nozzle = 473.4 m/s.** **(Ans.)**

3.14. HEATING-COOLING AND EXPANSION OF VAPOURS

The basic energy equations for non-flow and flow processes are *also valid for vapours.*

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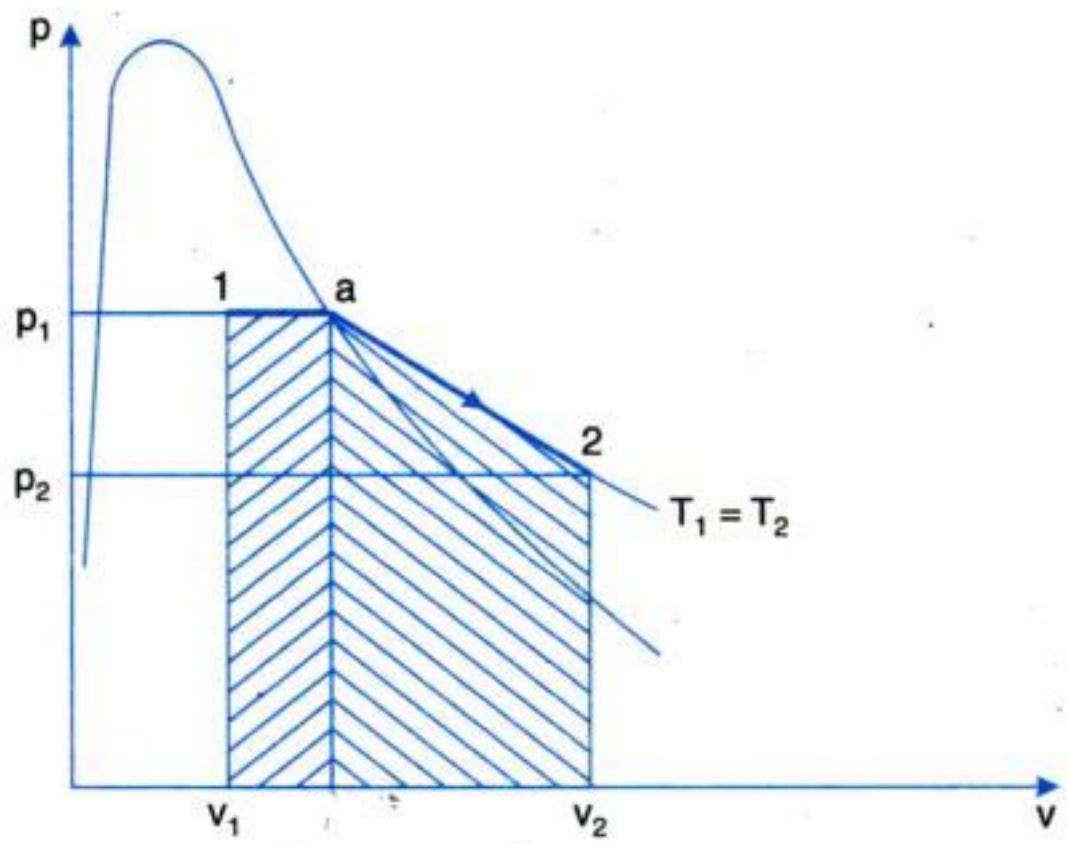
From superheated tables, $h_2 = 2839.3$ kJ/kg

Now, $Q = h_2 - h_1 = 2839.3 - 2718.28 = 121.02$ kJ/kg

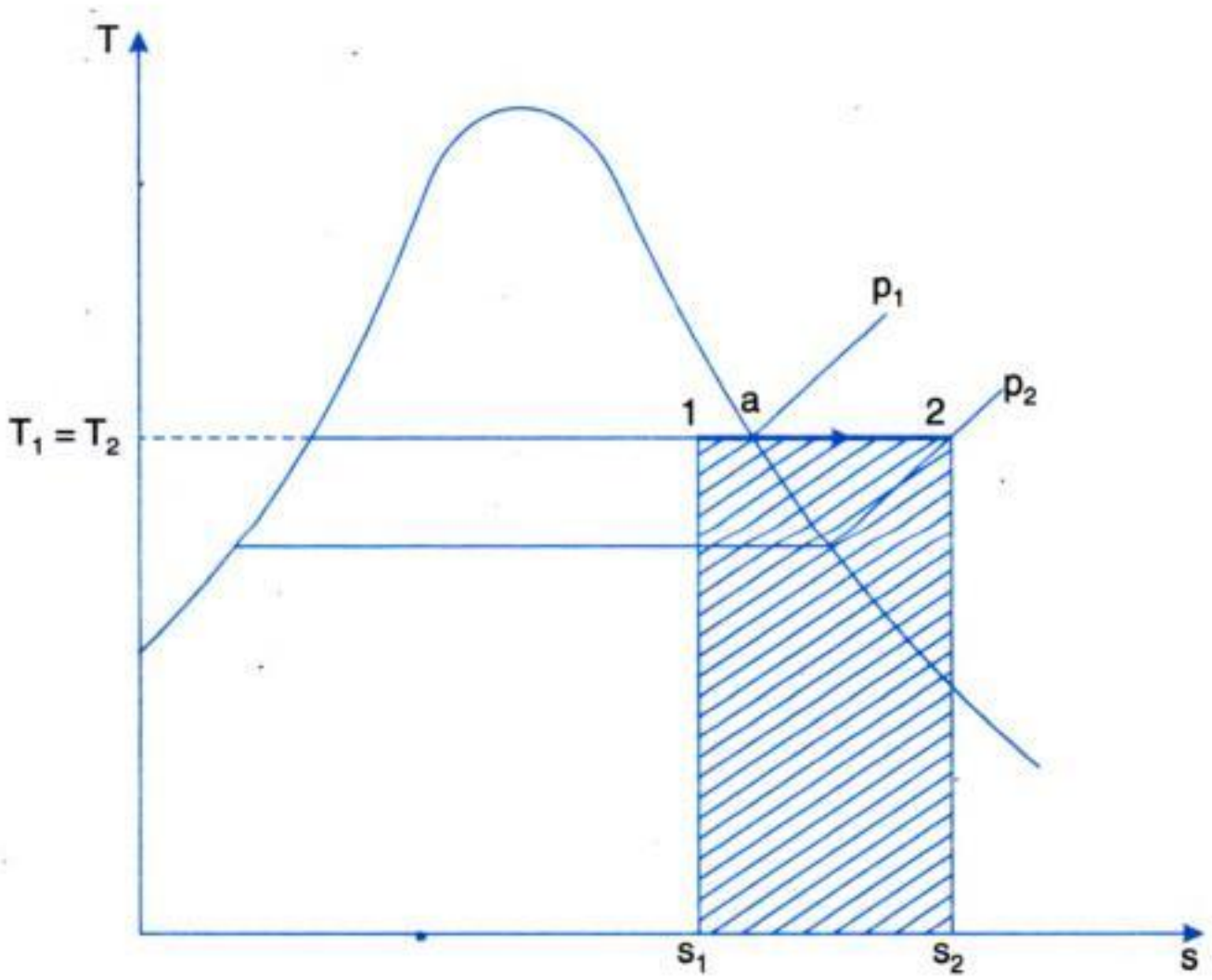
i.e., **Heat supplied = 121.02 kJ/kg.** **(Ans.)**

The *T-s* diagram showing the process is given in Fig. 3.58, the shaded area representing the heat flow.

3. Constant Temperature or Isothermal Expansion. Fig. 3.59 (*a*), (*b*) and (*c*) shows the constant temperature or isothermal expansion on *p-v*, *T-s* and *h-s* diagrams respectively.



(*a*)



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Hence, gain in internal energy

$$= u_2 - u_1 = 2567.25 - 2565 = 2.25 \text{ kJ/kg}$$

The process is shown on a p-v diagram in Fig. 3.61, the shaded area representing the work done.

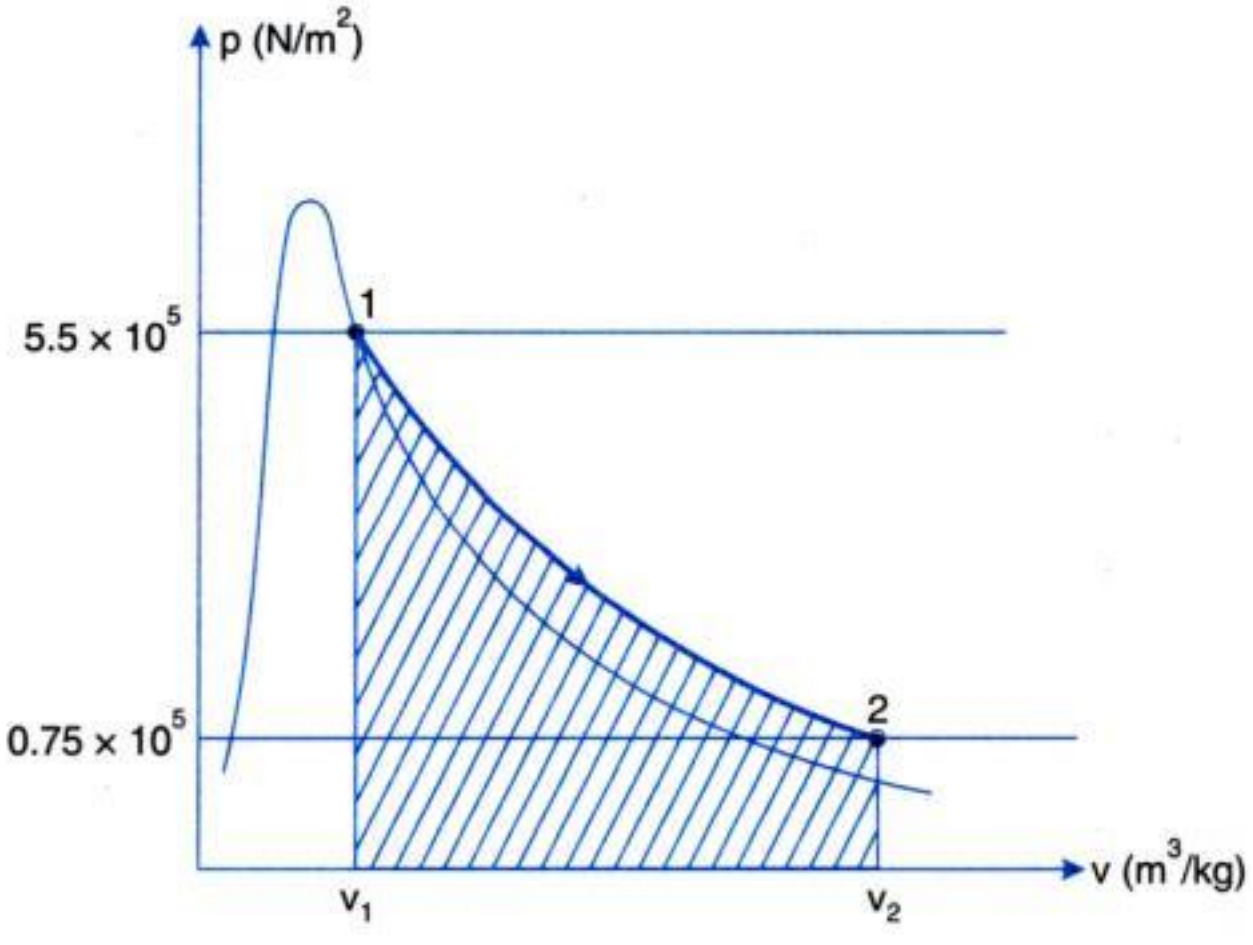

Fig. 3.61

Now,
$$W = \int_{v_1}^{v_2} p\,dv$$

$$= \int_{v_1}^{v_2} \left(\frac{\text{constant}}{v}\right) dv \qquad \left[\because \quad pv = \text{constant, and } p = \frac{\text{constant}}{v}\right]$$

$$= \text{constant} \left[\log_e v\right]_{v_1}^{v_2}$$

The constant is either p_1v_1 or p_2v_2

i.e.,
$$W = 5.5 \times 10^5 \times 0.3427 \times \log_e \frac{p_1}{p_2} \qquad \left[\because \quad p_1v_1 = p_2v_2 \quad \text{or} \quad \frac{v_2}{v_1} = \frac{p_1}{p_2}\right]$$

$$= 5.5 \times 10^5 \times 0.3427 \times \log_e \left(\frac{5.5}{0.75}\right) = 375543 \text{ Nm/kg.}$$

Using non-flow energy equation, we get

$$Q = (u_2 - u_1) + W$$

$$= 2.25 + \frac{375543}{10^3} = 378 \text{ kJ/kg}$$

i.e., **Heat supplied = 378 kJ/kg.** **(Ans.)**

Example 3.53. *Dry saturated steam at 100 bar expands isothermally and reversibly to a pressure of 10 bar. Calculate per kg of steam :*

(i) The heat supplied ;

(ii) The work done.

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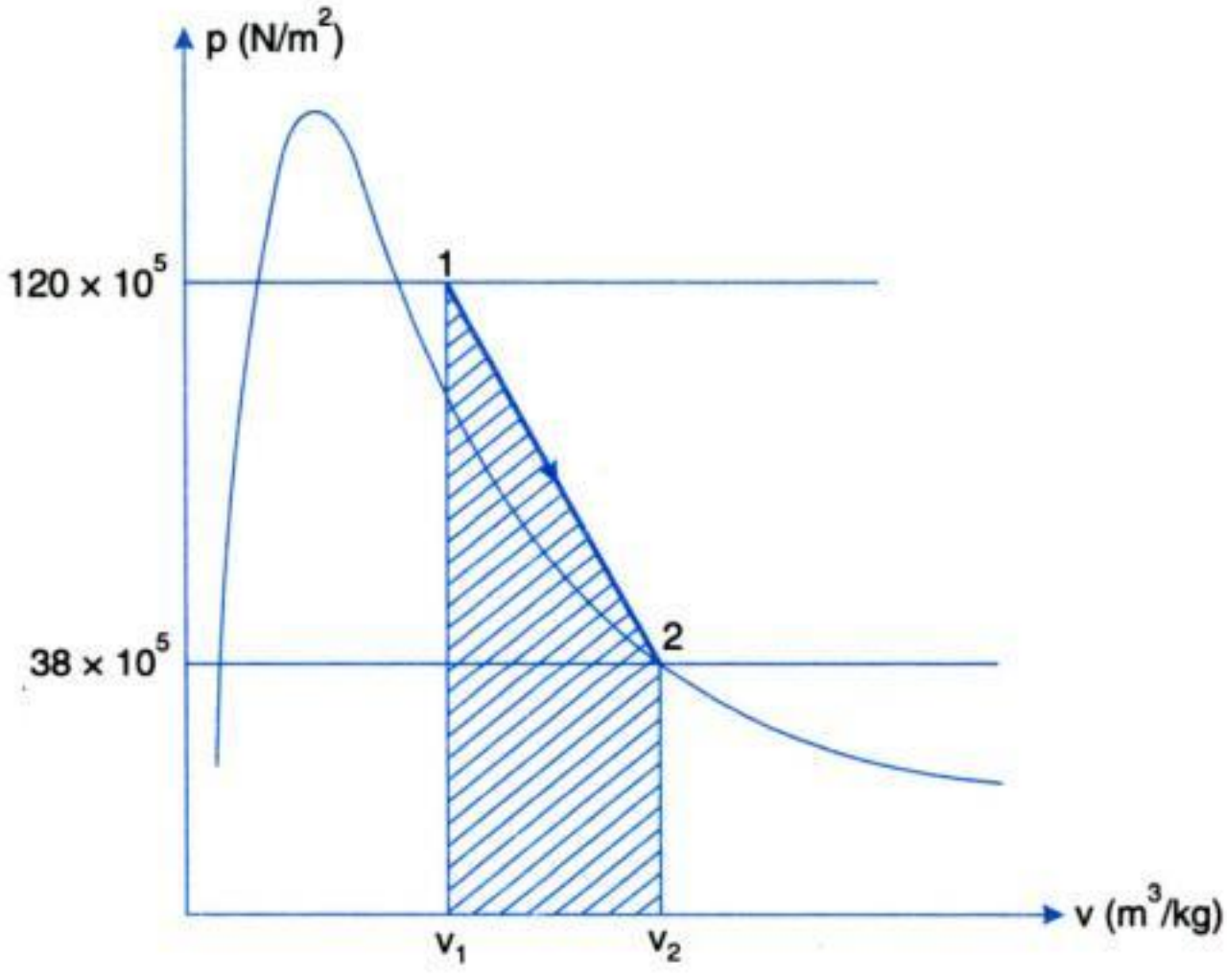


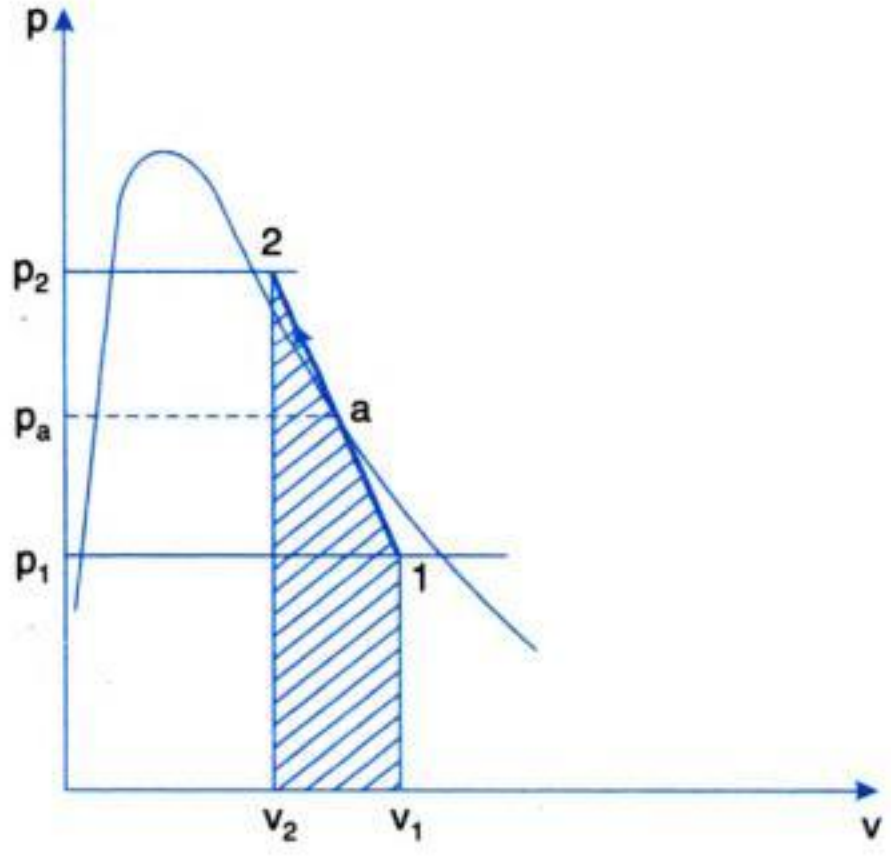
Fig. 3.64

The process is shown on p-v diagram in Fig. 3.64, the shaded area representing the work done.

5. Polytropic process. In this process, the steam follows the law pv^n = constant. This process on p-v, T-s and h-s diagrams is shown in Fig. 3.65 (a), (b) and (c).

The work done during this process is given by

$$W = \left(\frac{p_1v_1 - p_2v_2}{n-1}\right) \text{ Nm/kg}$$



(a)

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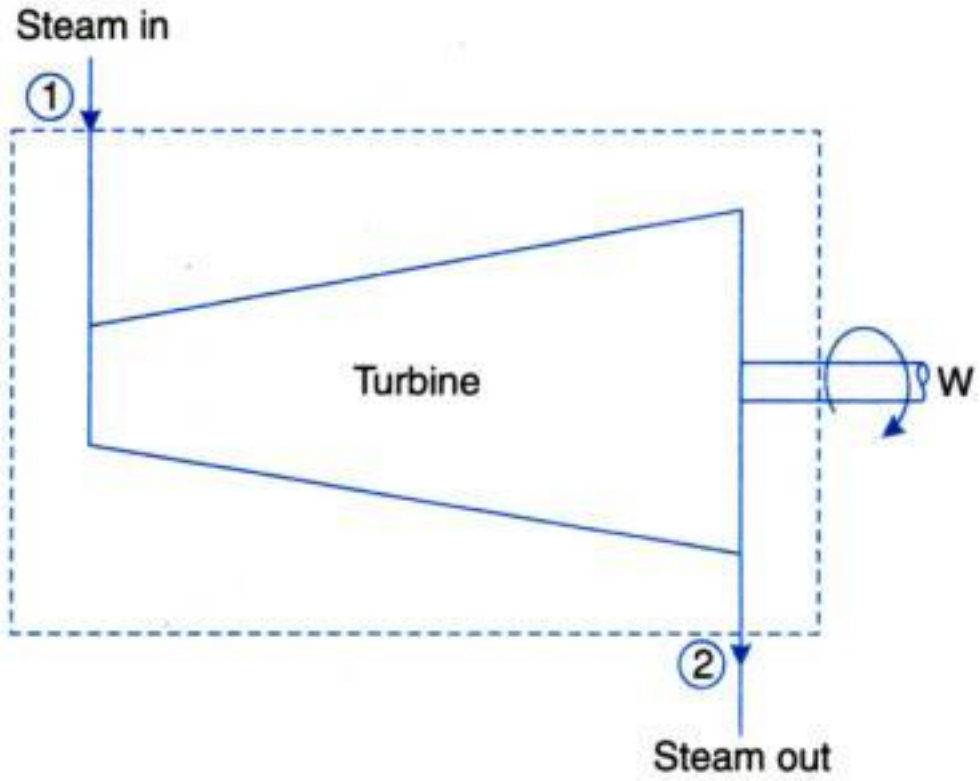
Using the non-flow energy equation,

$$Q = (u_2 - u_1) + W$$
$$= (2248.49 - 2535.46) + 450 = 163.03 \text{ kJ/kg}$$

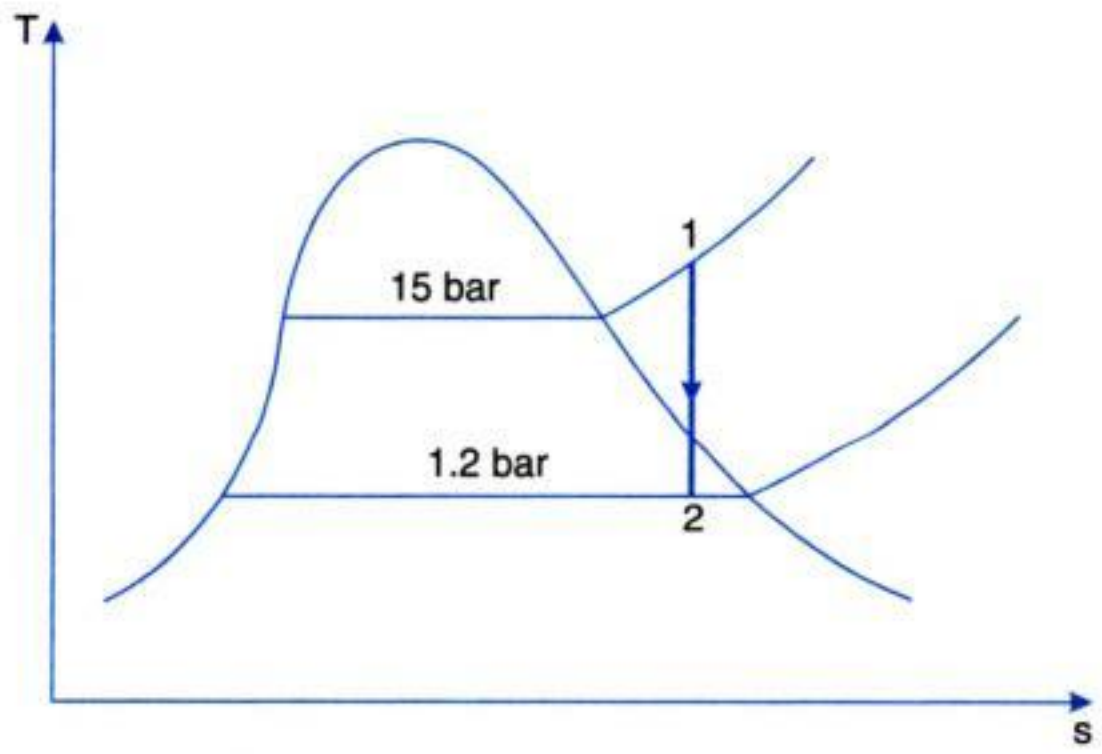
i.e., **Heat supplied = 163.03 kJ/kg.** **(Ans.)**

Example 3.56. *Steam enters a steam turbine at a pressure of 15 bar and 350°C with a velocity of 60 m/s. The steam leaves the turbine at 1.2 bar and with a velocity of 180 m/s. Assuming the process to be reversible adiabatic, determine the work done per kg of steam flow through the turbine.*

Neglect the change in potential energy.



(*a*)



(*b*)

Fig. 3.67

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3.15. UNSTEADY FLOW PROCESSES

In engineering practice, the variable flow process applications are as common as the steady flow process. The *rate of energy and mass transfer into and out of the control volume are not same in the case of unstable* (or variable or transient) *flow process.*

Following two cases only will be discussed :

1. Filling a tank.
2. Emptying a tank or tank discharge.

1. Filling a tank :

Let m_1 = Initial mass of fluid,
p_1 = Initial pressure,
v_1 = Initial specific volume,
T_1 = Initial temperature,
u_1 = Initial specific internal energy,

and m_2 = Final mass of fluid,
p_2 = Final pressure,

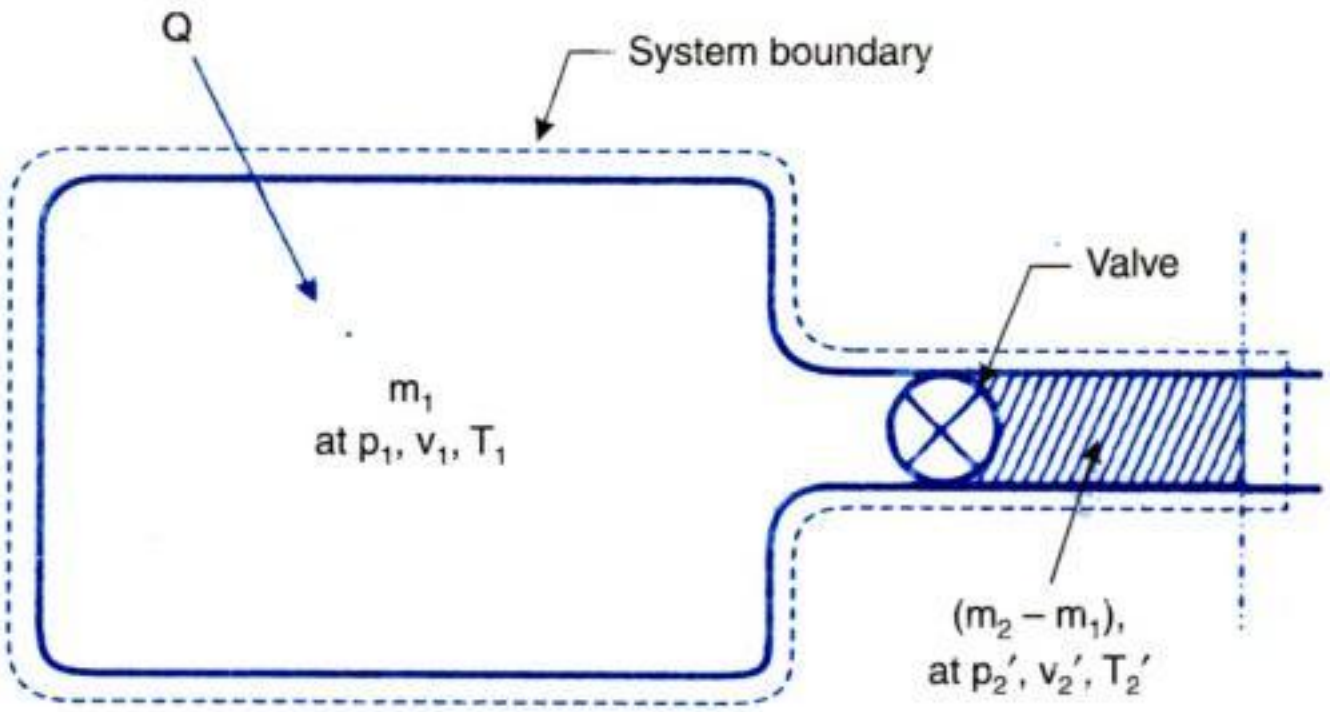

Fig. 3.75

v_2 = Final specific volume,
T_2 = Final temperature,
u_2 = Final specific internal energy,

Also, let p' = Entering fluid pressure,
v' = Entering fluid specific volume,
T' = Entering fluid temperature,
C' = Entering fluid velocity,
u' = Entering specific internal energy of fluid, and
h' = Entering specific enthalpy of fluid.

The quantity of fluid entering

$$= m_2 - m_1$$

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

or $$T_2 = \frac{p_2V_2T_1}{p_1V_1} = \frac{11.47 \times 0.035 \times 261}{4.5 \times 0.07} = \mathbf{332.6\ K.}$$ **(Ans.)**

$$\therefore \quad \text{Work done} = \frac{p_1V_1 - p_2V_2}{\gamma - 1} = \frac{4.5 \times 10^5 \times 0.07 - 11.47 \times 10^5 \times 0.035}{(1.35 - 1)}\ \text{J}$$

$$= \frac{10^5\,(4.5 \times 0.07 - 11.47 \times 0.035)}{0.35 \times 1000}\ \text{kJ}$$

$$= -24.7\ \text{kJ}\ (\textit{done on the methane})$$

(*ii*) **The piston will be in virtual equilibrium and hence zero work is effected by the piston.** **(Ans.)**

(*iii*) Work done by oxygen = work done on methane and expansion of oxygen is effected in the system

$$\therefore \quad W_{oxygen} = +24.7\ \text{kJ}$$

and $$Q = (U_2 - U_1) + W$$

Amount of oxygen present $$= \frac{p_1V_1}{RT_1} = \frac{4.5 \times 10^5 \times 0.035}{0.24 \times 1000 \times 333} = 0.197\ \text{kg}$$

and $$T_2 = \frac{p_2V_2}{p_1V_1} \times T_1 = \frac{11.47 \times 0.07 \times 333}{4.5 \times 0.035} = \mathbf{1697.5\ K.}$$ **(Ans.)**

(As the piston is free, the final pressure of oxygen and methane will be **same**).

$$\therefore \quad Q = (U_2 - U_1) + W = mc_v\,(T_2 - T_1) + W$$

$$= 0.197 \times 0.64\,(1697.5 - 333) + 24.7 = \mathbf{196.7\ kJ.}$$ **(Ans.)**

ADDITIONAL/TYPICAL WORKED EXAMPLES

Example 3.64. *Derive the equation of state for a perfect gas.*

Solution. To derive the equation of state for a perfect gas let us consider a *unit mass* of a perfect gas to change its state in the following two successive process (Fig. 3.76) :

(*i*) Process 1–2′ at constant pressure, and

(*ii*) Process 2′–2 at constant temperature,

For the *process* 1–2′, applying Charle's law, we get

$$\frac{v_1}{T_1} = \frac{v_2'}{T_2'}$$

and, since $T_2' = T_2$, we may write

$$\frac{v_1}{T_1} = \frac{v_2'}{T_2'} \qquad ...(i)$$

For the *process* 2′–2, using Boyle's law, we have

$$p_2'v_2' = p_2v_2$$

and, since $p_2' = p_1$

$$p_1v_2' = p_2v_2$$

i.e., $$v_2' = \frac{p_2v_2}{p_1} \qquad ...(ii)$$

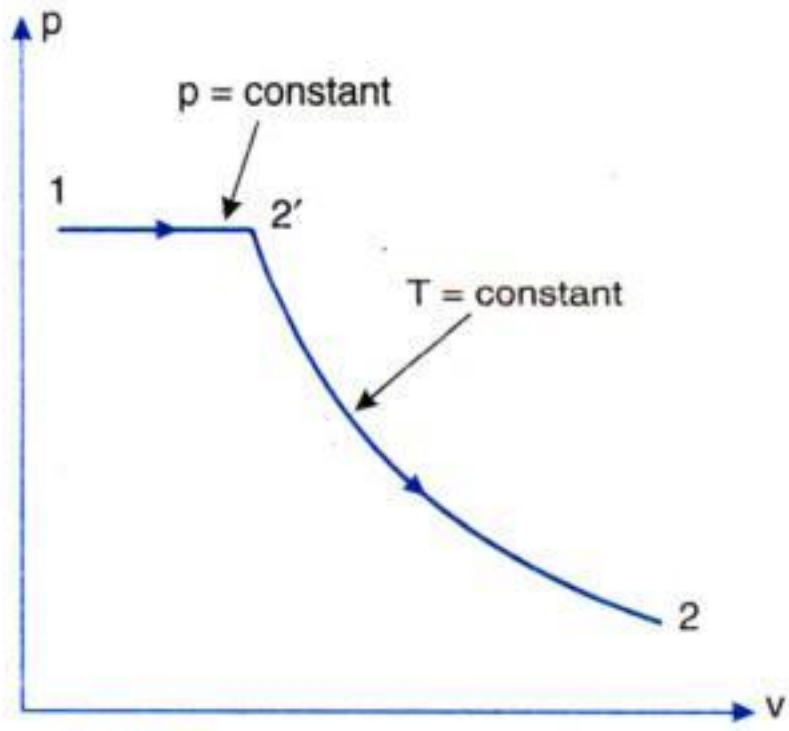

Fig. 3.76. Formulation of equation of state of a perfect gas.

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$\therefore$ *Network done* $= W_{1-2} + W_{2-3} + W_{3-1}$

$= 311.9 + 0 + (-230.87) =$ **81.03 kJ.** **(Ans.)**

For a cyclic process, $\oint \delta Q = \oint \delta W$

$\therefore$ Heat *transferred during the cycle* = **81.03 kJ.** **(Ans.)**

Example 3.67. *A system consisting of 1 kg of an ideal gas at 5 bar pressure and 0.02 m^3 volume executes a cyclic process comprising the following three distinct operations : (i) Reversible expansion to 0.08 m^3 volume, 1.5 bar pressure, presuming pressure to be a linear function of volume ($p = a + bV$), (ii) Reversible cooling at constant pressure and (iii) Reversible hyperbolic compression according to law pV = constant. This brings the gas back to initial conditions.*

(i) Sketch the cycle on p-V diagram.

(ii) Calculate the work done in each process starting whether it is done on or by the system and evaluate the net cyclic work and heat transfer.

Solution. *Given :* $m = 1$ kg ; $p_1 = 5$ bar ; $V_1 = 0.02$ m^3 ; $V_2 = 0.08$ m^3 ; $p_2 = 1.5$ bar.

(i) **p-V diagram :** *p-V* diagram of the cycle is shown in Fig. 3.79.

(ii) **Work done and heat transfer :**

- *Process 1–2 (Linear law) :*

$$p = a + bV \qquad ...(Given)$$

The values of constants a and b can be determined from the values of pressure and volume at the state points 1 and 2.

$$5 = a + 0.02b \qquad ...(i)$$

$$1.5 = a + 0.08b \qquad ...(ii)$$

From (i) and (ii) we get, $b = -58.33$ and $a = 6.167$

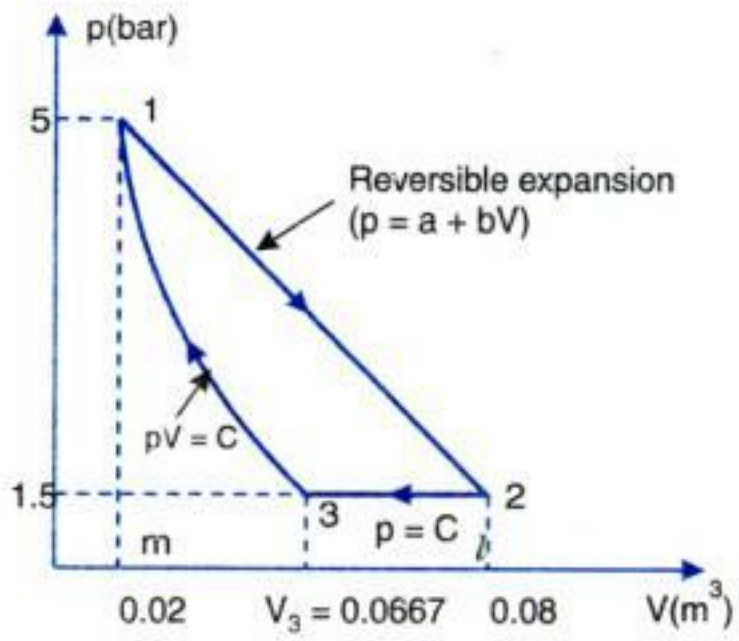

Fig. 3.79. *p-V* diagram.

$$W_{1-2} = \int_1^2 p\,dV = \int_1^2 (a + bV)\,dV$$

$$= \int_1^2 (6.167 - 58.33V)\,dV$$

$$= 10^5 \left| 6.167\,V - 58.33 \times \frac{V^2}{2} \right|_{0.02}^{0.08}$$

$$= 10^5 \left| 6.167\,(0.08 - 0.02) - 58.33 \times \frac{(0.08^2 - 0.02^2)}{2} \right| \times 10^{-3}\ \text{kJ} = \mathbf{19.5\ kJ}$$

This is work done **by the system.** **(Ans.)**

[*Alternatively :* W_{1-2} = Area under the process line 1–2

= Area of trapezium 1–2–*l*–*m*

$= \left[\frac{5 + 1.5}{2} \times 10^5\right] \times (0.08 - 0.02) = 19.5$ kJ]

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(*iii*) Explain the difference in work transfer in parts (*i*) and (*ii*).

[**Ans.** (*i*) $\Delta U = -91$ kJ, $W = 127.5$ kJ, $Q = 36.5$ kJ
(*ii*) $W = 121$ kJ, (*iii*) The work in (*ii*) in not equal to $\int pdV$ since the process is not quasi-static.]

8. A fluid is contained in a cylinder by a spring-loaded, frictionless piston so that the pressure in the fluid is linear function of the volume ($p = a + bV$). The internal energy of the fluid is given by the following equation

$$U = 34 + 3.15\,pV$$

where U is in kJ, p in kPa and V in cubic metre. If the fluid changes from an initial state of 170 kPa, 0.03 m^3 to a final state of 400 kPa, 0.06 m^3, with no work other than that done on the piston, find the direction and magnitude of the work and heat transfer.

[**Ans.** $W_{1-2} = 10.35$ kJ ;
$Q_{1-2} = 69.85$ kJ (heat flows into the system during the process)]

9. A piston cylinder arrangement has a gas in the cylinder space. During a constant pressure expansion to a larger volume the work effect for the gas are 1.6 kJ, the heat added to the gas and cylinder arrangement is 3.2 kJ and the friction between the piston and cylinder wall amounts to 0.24 kJ. Determine the change in internal energy of the entire apparatus (gas, cylinder, piston). [**Ans.** 1.84 kJ]

10. A system receives 42 kJ of heat while expanding with volume change of 0.123 m^3 against an atmosphere of 12 N/cm^2. A mass of 80 kg in the surroundings is also lifted through a distance of 6 metres.

(*i*) Find the change in energy of the system.

(*ii*) The system is returned to its initial volume by an adiabatic process which requires 100 kJ of work. Find the change in energy of system.

(*iii*) Determine the total change in energy of the system. [**Ans.** (*i*) 22.54 kJ, (*ii*) 100 kJ, (*iii*) 122.54 kJ]

11. A thermally insulated battery is being discharged at atmospheric pressure and constant volume. During a 1 hour test it is found that a current of 50 A and 2 V flows while the temperature increases from 20°C to 32.5°C. Find the change in internal energy of the cell during the period of operation. [**Ans.** -36×10^4 J]

12. In a certain steam plant the turbine develops 1000 kW. The heat supplied to the steam in the boiler is 2800 kJ/kg, the heat received by the system from cooling water in the condenser is 2100 kJ/kg and the feed pump work required to pump the condensate back into the boiler is 5 kW. Calculate the steam flow round the cycle in kg/s. [**Ans.** 1.421 kg/s]

13. In the compression stroke of an internal-combustion engine the heat rejected to the cooling water is 45 kJ/kg and the work input is 90 kJ/kg. Calculate the change in internal energy of the working fluid stating whether it is a gain or a loss. [**Ans.** 45 kJ/kg (gain)]

14. 85 kJ of heat are supplied to a system at constant volume. The system rejects 90 kJ of heat at constant pressure and 20 kJ of work is done on it. The system is brought to its original state by adiabatic process. Determine the adiabatic work. Determine also the value of internal energy at all end states if initial value is 100 kJ. [**Ans.** $W = 15$ kJ ; $U_1 = 100$ kJ, $U_2 = 185$ kJ ; $U_3 = 115$ kJ]

15. A closed system undergoes a reversible process at a constant pressure process of 3.5 bar and its volume changes from 0.15 m^3 to 0.06 m^3. 25 kJ of heat is rejected by the system during the process. Determine the change in internal energy of the system. [**Ans.** 6.5 kJ (increase)]

16. An air compressor takes in air at 10^5 Pa and 27°C having volume of 1.5 m^3/kg and compresses it to 4.5×10^5 Pa. Find the work done, heat transfer and change in internal energy if the compression is isothermal.

[**Ans.** -225 kJ ; -225 kJ ; $\Delta U = 0$]

17. A cylinder fitted with piston contains 0.2 kg of N_2 at 100 kPa and 30°C. The piston is moved compressing N_2 until the pressure becomes 1 MPa and temperature becomes 150°C. The work done during the process is 20 kJ. Determine the heat transferred from N_2 to the surroundings. Take $c_v = 0.75$ kJ/kg K for N_2.

[**Ans.** -2 kJ]

18. A closed system consisting of 1 kg of gaseous CO_2 undergoes a reversible process at constant pressure causing a decrease of 30 kJ in internal energy. Determine the work done during the process. Take $c_p = 840$ J/kg°C and $c_v = 600$ J/kg°C. [**Ans.** -12 kJ]

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49. Air at a temperature of 15°C passes through a heat exchanger at a velocity of 30 m/s, where its temperature is raised to 800°C. It then enters a turbine with the same velocity of 30 m/s and expands until the temperature falls to 650°C. On leaving the turbine, the air is taken at a velocity of 60 m/s to a nozzle where it expands until the temperature has fallen to 500°C. If the air flow rate is 2 kg/s, calculate (*i*) the rate of heat transfer to the air, (*ii*) the power output from the turbine assuming no heat loss, and (*iii*) the velocity at exit from nozzle, assuming no heat loss.

Take the enthalpy of air as $h = c_p t$, where c_p is the specific heat equal to 1.005 kJ/kg°C and t the temperature.

[**Ans.** 1580 kJ/s ; 298.8 kW ; 554 m/s]

Vapour (Steam)

50. 0.05 kg of steam is heated at a constant pressure of 2 bar until the volume occupied is 0.0658 m^3. Calculate the heat supplied and work done. [**Ans.** 18.25 kJ ; 4.304 kJ]

51. Steam at 7 bar and dryness fraction 0.9 expands in a cylinder behind a piston isothermally and reversibly to a pressure of 1.5 bar. Calculate the change of internal energy and the change of enthalpy per kg of steam. The heat supplied during the process is found to be 400 kJ/kg. Calculate the work done per kg of steam.

[**Ans.** 217.5 kJ/kg (gain) ; 245.7 kJ/kg ; 182.5 kJ/kg]

52. 1 kg of steam at 100 bar and 375°C expands reversibly in a perfectly thermally insulated cylinder behind a piston until pressure is 38 bar and the steam is then saturated.

Calculate the work done by the steam. [**Ans.** 169.7 kJ/kg]

53. In a steam engine the steam at the beginning of the expansion process is at 7 bar, dryness fraction 0.95, and the expansion follows the law $pv^{1.1}$ = constant, down to a pressure of 0.34 bar. Calculate the work done per kg of steam during the expansion, and the heat flow per kg of steam to or from the cylinder walls during the expansion. [**Ans.** 436 kJ/kg ; 155.6 kJ/kg (heat supplied)]

54. Steam at 19 bar is throttled to 1 bar and the temperature after throttling is found to be 150°C. Calculate the initial dryness fraction of the steam. [**Ans.** 0.989]

55. 1 kg of steam at 7 bar, entropy 6.5 kJ/kg K, is heated reversibly at constant pressure until the temperature is 250°C. Calculate the heat supplied, and show on a *T-s* diagram the area which represents the heat flow.

[**Ans.** 283 kJ/kg]

56. 1 kg of steam at 20 bar, dryness fraction 0.9, is heated reversibly at constant pressure to a temperature of 300°C.

Calculate the heat supplied and change of entropy and show the process on a *T-s* diagram, indicating the area which represents the heat flow. [**Ans.** 415 kJ/kg ; 0.8173 kJ/kg K]

57. Steam at 0.05 bar, 100°C is to be condensed completely by a reversible constant pressure process.

Calculate the heat to be removed per kg of steam and the change of entropy. Sketch the process on a *T-s* diagram and shade in the area which represents the heat flow. [**Ans.** 2550 kJ/kg ; 8.292 kJ/kg K]

58. 0.05 kg of steam at 10 bar, dryness fraction 0.84, is heated reversibly in a rigid vessel until the pressure is 20 bar.

Calculate the change of entropy and the heat supplied. Show the area which represents the heat supplied on a *T-s* diagram. [**Ans.** 0.0704 kJ/kg K ; 36.85 kJ]

59. 1 kg of steam undergoes a reversible isothermal process from 20 bar and 250°C to a pressure of 30 bar. Calculate the heat flow, stating whether it is supplied or rejected and sketch the process on a *T-s* diagram.

[**Ans.** – 135 kJ/kg]

60. Steam at 5 bar, 250°C, expands isentropically to a pressure of 0.7 bar. Calculate the final condition of steam.

[**Ans.** 0.967]

61. Steam expands reversibly in a cylinder behind a piston from 6 bar dry saturated, to a pressure of 0.65 bar. Assuming that the cylinder is perfectly thermally insulated, calculate the work done during the expansion per kg of steam. Sketch the process on a *T-s* diagram. [**Ans.** 323.8 kJ/kg]

62. A steam engine receives steam at 4 bar, dryness fraction 0.8, and expands it according to a law $pv^{1.05}$ = constant to a condenser pressure of 1 bar. Calculate the change of entropy per kg of steam during the expansion, and sketch the process on a *T-s* diagram. [**Ans.** 0.381 kJ/kg K]

63. Steam at 15 bar is throttled to 1 bar and a temperature of 150°C. Calculate the initial dryness fraction and the change of entropy. Sketch the process on a *T-s* diagram and state the assumptions made in the throttling process. [**Ans.** 0.992, 1.202 kJ/kg K]

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$$\text{Co-efficient of performance, } (\text{C.O.P.})_{ref.} = \frac{Q_2}{W} \qquad \ldots(4.2)$$

where, Q_2 = Heat transfer *from cold reservoir*, and
W = The net work transfer to the refrigerator.

For a **reversed heat engine** [Fig. 4.1 (*b*)] acting as a *heat pump*, the measure of success is again called the *co-efficient of performance*. It is defined by the ratio :

$$\text{Co-efficient of performance, } (\text{C.O.P.})_{heat\ pump} = \frac{Q_1}{W} \qquad \ldots(4.3)$$

where, Q_1 = Heat transfer *to hot reservoir*, and
W = Net work transfer to the heat pump.

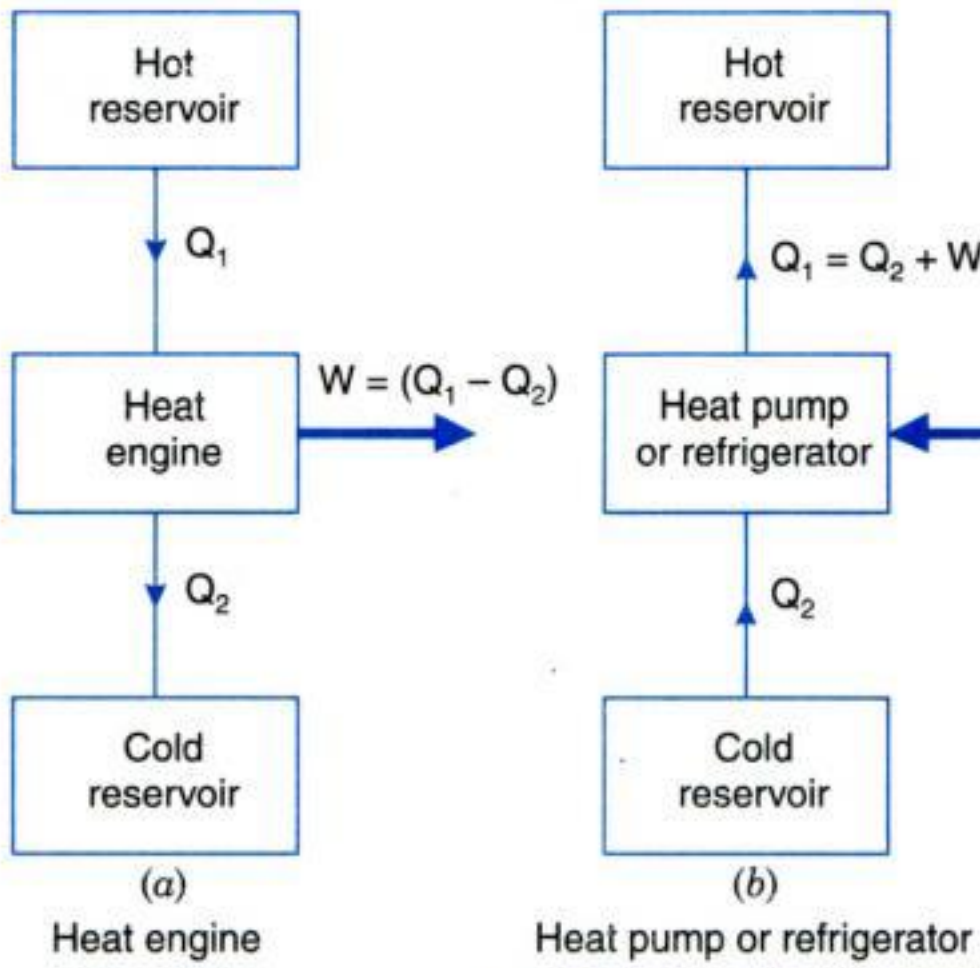

Fig. 4.1

In all the above three cases application of the first law gives the relation $Q_1 - Q_2 = W$, and this can be used to rewrite the expressions for thermal efficiency and co-efficient of performance solely in terms of the heat transfers.

$$\eta_{th} = \frac{Q_1 - Q_2}{Q_1} \qquad \ldots(4.4)$$

$$(\text{C.O.P.})_{ref} = \frac{Q_2}{Q_1 - Q_2} \qquad \ldots(4.5)$$

$$(\text{C.O.P.})_{heat\ pump} = \frac{Q_1}{Q_1 - Q_2} \qquad \ldots(4.6)$$

It may be seen that η_{th} is *always less than unity* and $(C.O.P.)_{heat\ pump}$ is *always greater than unity*.

4.3. REVERSIBLE PROCESSES

A reversible process should fulfill the following *conditions* :

1. The process should not involve friction of any kind.
2. Heat transfer should not take place with finite temperature difference.

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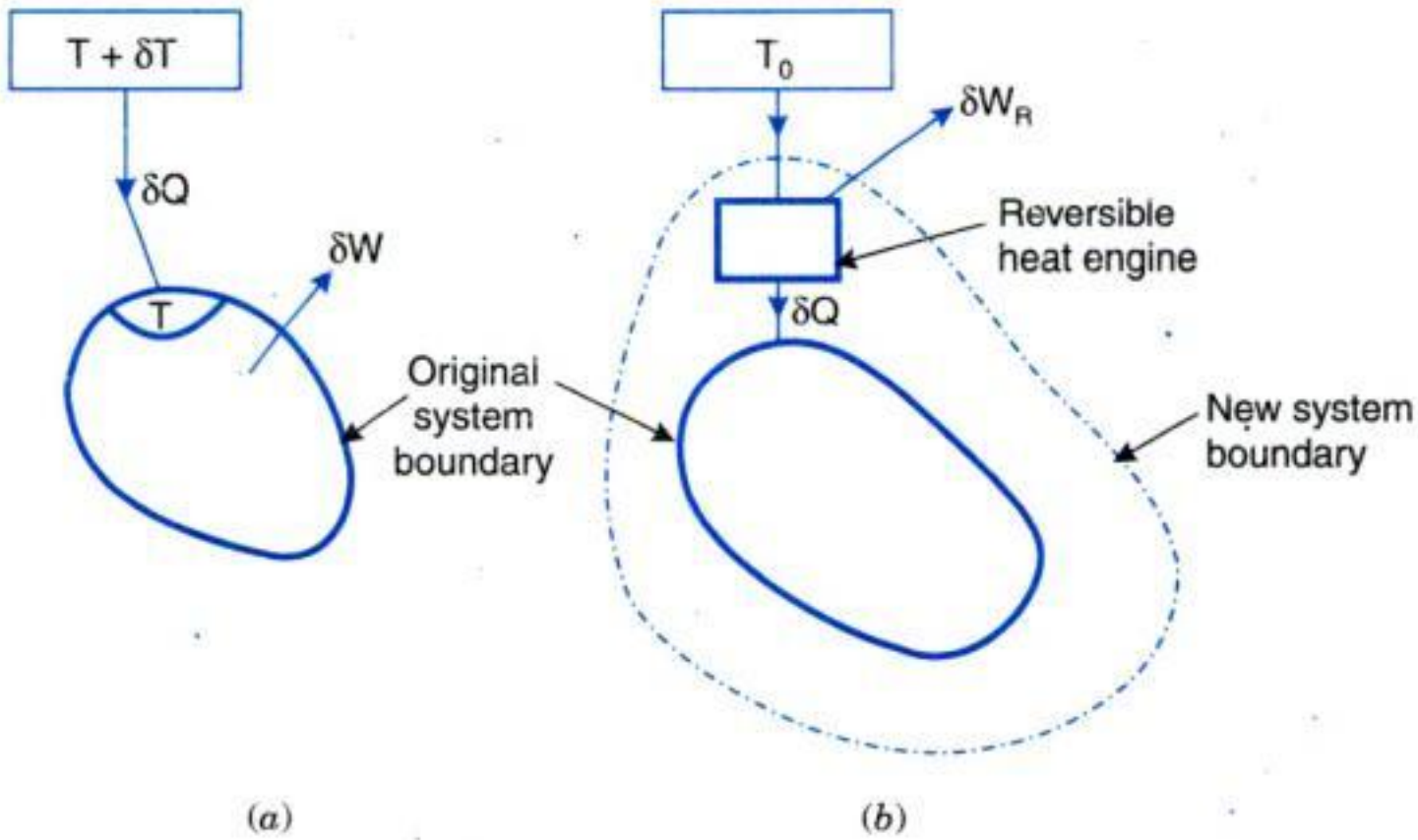

Fig. 4.4. The clausius inequality.

The system to which the heat transfer is effected is neither concerned with the source of energy it receives nor with the method of transfer, save that it must be reversible. Associated with the small heat transfer dQ to the original system is a small work transfer dW and for this system the first law gives

$$\sum_{\text{cycle}} (\delta Q - \delta W) = 0 \qquad \ldots(4.9)$$

Now consider the engine replacing the reservoirs and apply the second law to the new system in Fig. 4.4 (b). If the new system is not a perpetual motion machine of second kind, no positive work transfer is possible with a single reservoir.

Therefore,
$$\sum_{\text{cycle}} (\delta W - \delta W_R) \leq 0 \qquad \ldots(4.10)$$

But by the definition of thermodynamic temperature in equation (4.8)

$$\frac{\delta W_R}{\delta Q} = \frac{\delta Q_0 - \delta Q}{\delta Q} = \frac{T_0 - T}{T} \qquad \ldots(4.11)$$

and by combination of eqns. (4.9), (4.10) and (4.11)

$$T_0 \sum_{\text{cycle}} \left(\frac{\delta Q}{T}\right) \leq 0 \text{ but } T_0 \neq 0 \text{ and therefore ;}$$

$$\sum_{\text{cycle}} \left(\frac{\delta Q}{T}\right) \leq 0 \qquad \ldots(4.12)$$

This is known as **Clausius inequality**.

Let us now consider the case of a reversible engine for which

$$\sum_{\text{cycle}} \left(\frac{\delta Q}{T}\right) \leq 0,$$

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Let HE_A be *any* heat engine and HE_B be *any reversible* heat engine. We have to prove that efficiency of HE_B is more than that of HE_A. Let us assume that $\eta_A > \eta_B$. Let the rates of working of the engines be such that

$$Q_{1A} = Q_{1B} = Q_1$$

Since $$\eta_A > \eta_B$$

$$\frac{W_A}{Q_{1A}} > \frac{W_B}{Q_{1B}}$$

$\therefore$ $$W_A > W_B$$

Now, let HE_B be reversed. Since HE_B is a reversible heat engine, the magnitudes of heat and work transfer quantities will remain the same, but their directions will be reversed, as shown in Fig. 4.7. Since $W_A > W_B$, some part of W_A (equal to W_B) may be fed to drive the reversed heat engine $\exists H_B$. Since $Q_{1A} = Q_{1B} = Q_1$, the heat discharged by $\exists H_B$ may be supplied to HE_A. The source may, therefore, be eliminated (Fig. 4.8). The net result is that HE_A and $\exists H_B$ together constitute a heat engine which, operating in a cycle produces net work $W_A - W_B$ while exchanging heat with a single reservoir at T_2. This violates the Kelvin-Planck statement of the second law. Hence the assumption that $\eta_A > \eta_B$ is wrong.

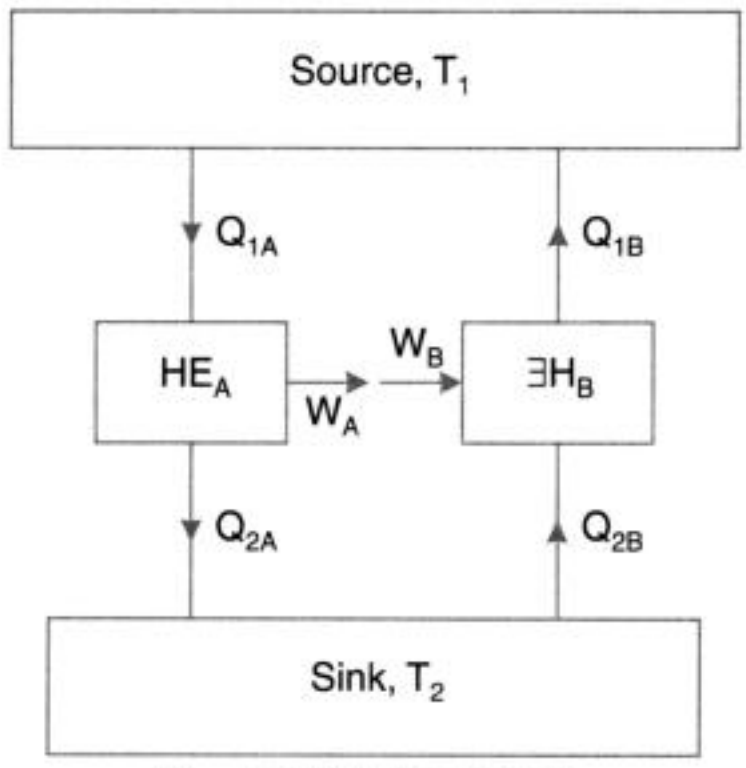

Fig. 4.7. HE_B is reversed.

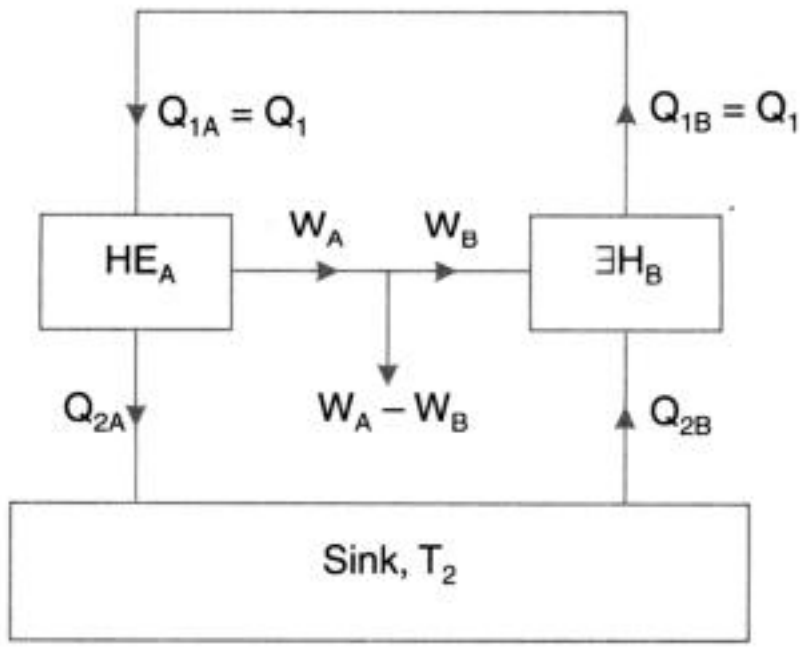

Fig. 4.8. HE_A and $\exists H_B$ together violate the Kelvin-Planck statement.

$\therefore$ $$\eta_B \geq \eta_A.$$

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Solution. (*i*) Heat requirement of the house, Q_1 (or heat rejected)

$$= 2 \times 10^5 \text{ kJ/h}$$

Work required to operate the heat pump,

$$W = 3 \times 10^4 \text{ kJ/h}$$

Now, $$Q_1 = W + Q_2$$

where Q_2 is the heat abstracted from outside.

$\therefore$ $$2 \times 10^5 = 3 \times 10^4 + Q_2$$

Thus $$Q_2 = 2 \times 10^5 - 3 \times 10^4$$

$$= 200000 - 30000 = 170000 \text{ kJ/h}$$

Hence, **heat abstracted from outside = 170000 kJ/h.** (Ans.)

(*ii*) $$(\text{C.O.P.})_{heat\ pump} = \frac{Q_1}{Q_1 - Q_2}$$

$$= \frac{2 \times 10^5}{2 \times 10^5 - 170000} = 6.66$$

Hence, **co-efficient of performance = 6.66.** (Ans.)

Note. If the heat requirements of the house were the same but this amount of heat had to be abstracted from the house and rejected out, *i.e., cooling of the house in summer*, we have

$$(\text{C.O.P.})_{refrigerator} = \frac{Q_2}{Q_1 - Q_2} = \frac{Q_2}{W}$$

$$= \frac{170000}{3 \times 10^4} = \mathbf{5.66}$$

Thus the same device has two values of C.O.P. depending upon the objective.

Example 4.6. *What is the highest possible theoretical efficiency of a heat engine operating with a hot reservoir of furnace gases at 2100°C when the cooling water available is at 15°C ?*

Solution. Temperature of furnace gases, $T_1 = 2100 + 273 = 2373$ K

Temperature of cooling water, $T_2 = 15 + 273 = 288$ K

Now, $$\eta_{max} (= \eta_{carnot}) = 1 - \frac{T_2}{T_1} = 1 - \frac{288}{2373} = \mathbf{0.878 \text{ or } 87.8\%.}$$ (Ans.)

Note. It should be noted that a system in practice operating between similar temperatures (*e.g.*, a steam generating plant) would have a thermal efficiency of about 30%. The discrepency is *due to irreversibility in the actual plant*, and *also because of deviations from the ideal Carnot cycle made for various practical reasons.*

Example 4.7. *A Carnot cycle operates between source and sink temperatures of 250°C and – 15°C. If the system receives 90 kJ from the source, find :*

(*i*) *Efficiency of the system ;* (*ii*) *The net work transfer ;*

(*iii*) *Heat rejected to sink.*

Solution. Temperature of source, $T_1 = 250 + 273 = 523$ K

Temperature of sink, $T_2 = -15 + 273 = 258$ K

Heat received by the system, $Q_1 = 90$ kJ

(*i*) **Efficiency of the system,** $$\eta_{carnot} = 1 - \frac{T_2}{T_1} = 1 - \frac{258}{523} = 0.506 = \mathbf{50.6\%.}$$ (Ans.)

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Since, $W_1 - W_2 = W = 400$ kJ

$W_2 = W_1 - W = 1670 - 400 = 1270$ kJ

$\therefore$ $Q_4 = 3.306 \times 1270 = 4198.6$ kJ

$Q_3 = Q_4 + W_2 = 4198.6 + 1270 = 5468.6$ kJ

$Q_2 = Q_1 - W_1 = 2500 - 1670 = 830$ kJ.

Heat rejection to the 50°C reservoir

$= Q_2 + Q_3 = 830 + 5468.6 =$ **6298.6 kJ.** (Ans.)

(*ii*) Efficiency of actual heat engine cycle,

$\eta = 0.45\ \eta_{max} = 0.45 \times 0.668 = 0.3$

$\therefore$ $W_1 = \eta \times Q_1 = 0.3 \times 2500 = 750$ kJ

$\therefore$ $W_2 = 750 - 400 = 350$ kJ

C.O.P. of the actual refrigerator cycle,

$$\text{C.O.P.} = \frac{Q_4}{W_2} = 0.45 \times 3.306 = 1.48$$

$\therefore$ $Q_4 = 350 \times 1.48 =$ **518 kJ.** (Ans.)

$Q_3 = 518 + 350 = 868$ kJ

$Q_2 = 2500 - 750 = 1750$ kJ

Heat rejected to 50°C reservoir

$= Q_2 + Q_3 = 1750 + 868 =$ **2618 kJ.** (Ans.)

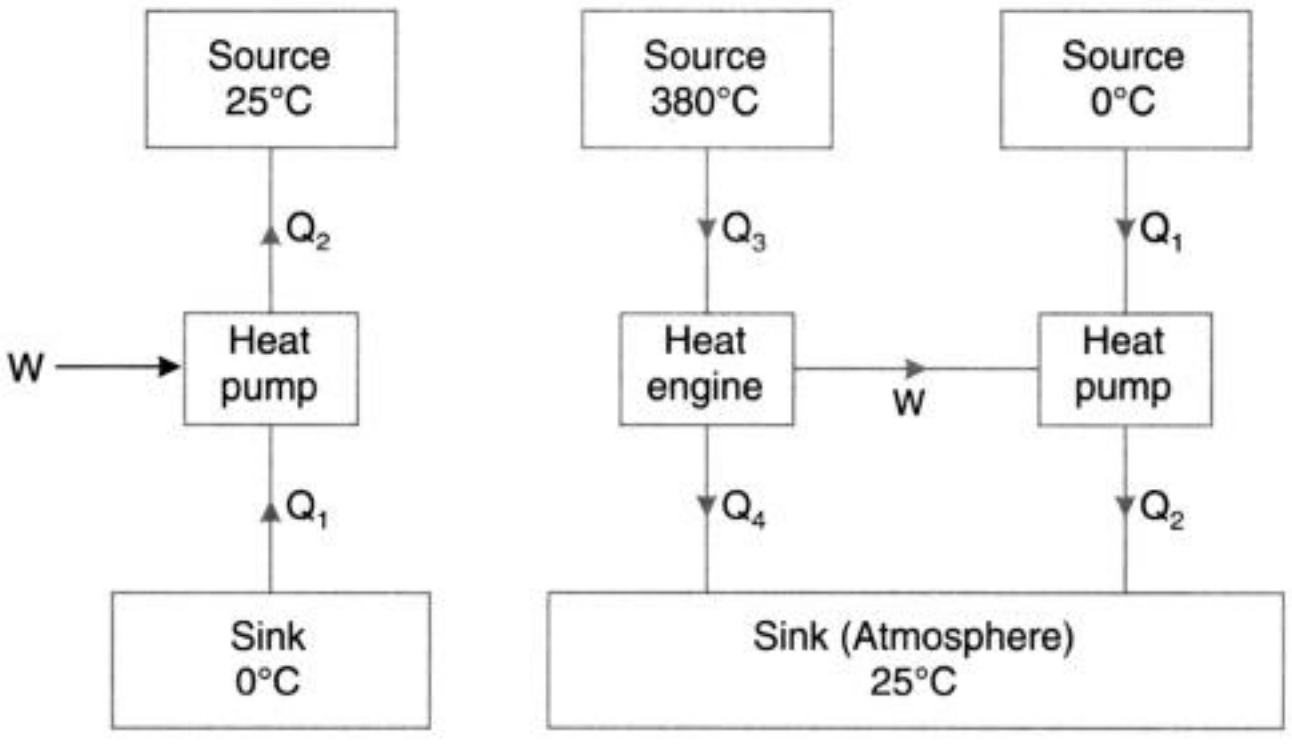
☞ **Example 4.13.** (*i*) *A reversible heat pump is used to maintain a temperature of 0°C in a refrigerator when it rejects the heat to the surroundings at 25°C. If the heat removal rate from the refrigerator is 1440 kJ/min, determine the C.O.P. of the machine and work input required.*

(*ii*) *If the required input to run the pump is developed by a reversible engine which receives heat at 380°C and rejects heat to atmosphere, then determine the overall C.O.P. of the system.*

Solution. Refer Fig. 4.15 (*a*).

(*i*) Temperature, $T_1 = 25 + 273 = 298$ K

Temperature, $T_2 = 0 + 273 = 273$ K



(*a*) Single system (*b*) Combined system

Fig. 4.15

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Solution. Boiler temperature, $T_1 = 160 + 273 = 433$ K
Condenser temperature, $T_2 = 50 + 273 = 323$ K
From steam tables :

Enthalpy of water entering boiler, $h_{f_1} = 687$ kJ/kg
Enthalpy of steam leaving boiler, $h_2 = 2760$ kJ/kg
Condenser pressure = 0.124×10^5 N/m^2
Boiler pressure = 6.18×10^5 N/m^2 .....(corresponding to 160°C)
Enthalpy of vapour leaving the turbine, $h_3 = 2160$ kJ/kg
(assuming isentropic expansion)

Enthalpy of water leaving the condenser, $h_{f4} = 209$ kJ/kg

Now $Q_{boiler}, Q_1 = h_2 - h_{f1} = 2760 - 687 = 2073$ kJ/kg

and $Q_{condenser}, Q_2 = h_{f4} - h_3 = 209 - 2160 = -1951$ kJ/kg

$$\therefore \quad \sum_{\text{cycle}} \frac{\delta Q}{T} = \frac{Q_1}{T_1} + \frac{Q_2}{T_2} = \frac{2073}{433} + \left(\frac{-1951}{323}\right)$$

$$= -1.25 \text{ kJ/kg K}$$

$$< 0. \quad \text{..... Proved.}$$

☞**Example 4.20.** *In a power plant cycle, the temperature range is 164°C to 51°C, the upper temperature being maintained in the boiler where heat is received and the lower temperature being maintained in the condenser where heat is rejected. All other processes in the steady flow cycle are adiabatic. The specific enthalpies at various points are given in Fig. 4.20. Verify the Clausius Inequality.*

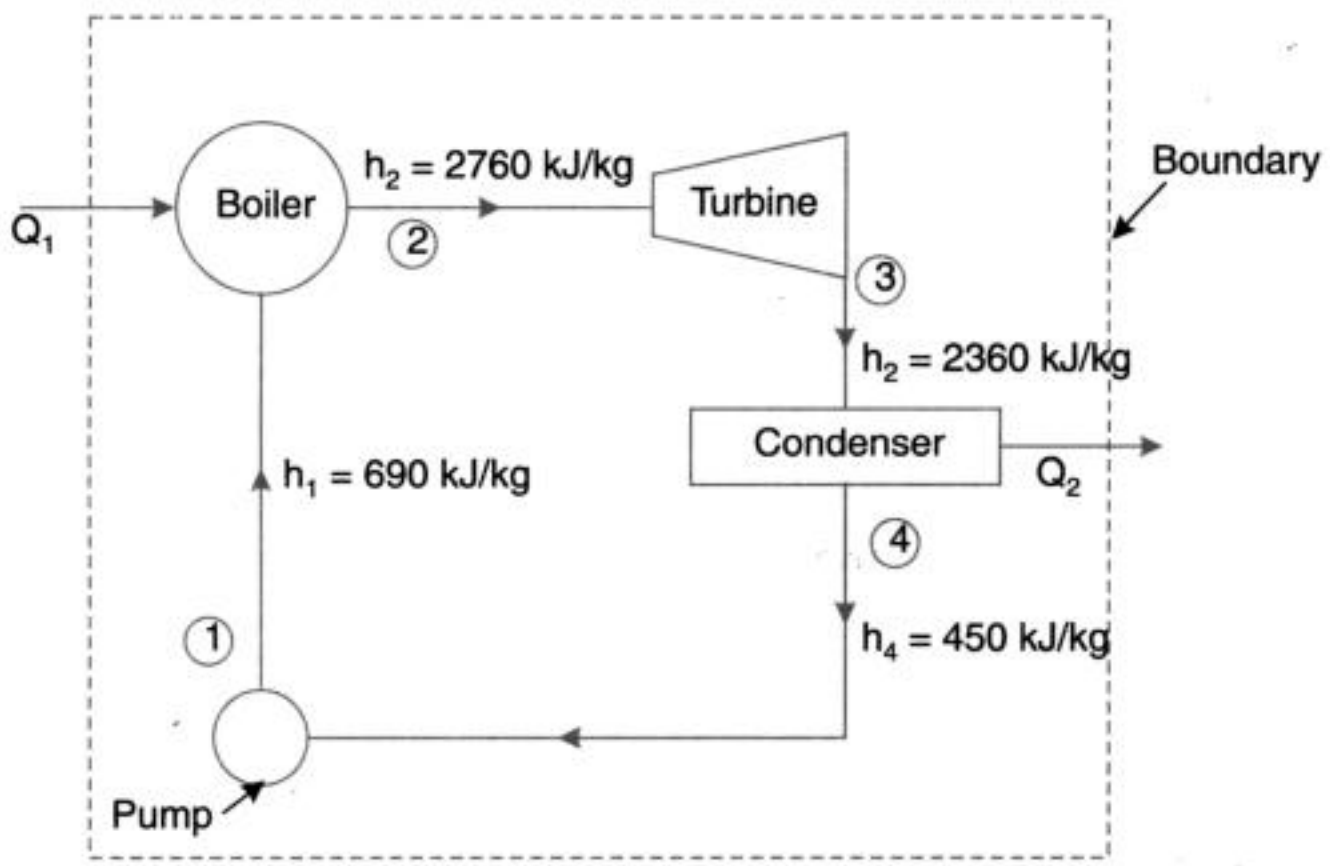

Fig. 4.20

Solution. Temperature maintained in boiler, $T_1 = 164 + 273 = 437$ K
Temperature maintained in condenser, $T_2 = 51 + 273 = 324$ K

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$$= c_v \log_e \frac{p_2}{p_1} + (c_v + R) \log_e \frac{v_2}{v_1}$$

$$= c_v \log_e \frac{p_2}{p_1} + c_p \log_e \frac{v_2}{v_1}$$

$$\therefore \quad s_2 - s_1 = c_v \log_e \frac{p_2}{p_1} + c_p \log_e \frac{v_2}{v_1} \qquad ...(4.29)$$

Again, from gas equation,

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \quad \text{or} \quad \frac{v_2}{v_1} = \frac{p_1}{p_2} \times \frac{T_2}{T_1}$$

Putting the value of $\frac{v_2}{v_1}$ in eqn. (4.28), we get

$$(s_2 - s_1) = c_v \log_e \frac{T_2}{T_1} + R \log_e \frac{p_1}{p_2} \times \frac{T_2}{T_1}$$

$$= c_v \log_e \frac{T_2}{T_1} + R \log_e \frac{p_1}{p_2} + R \log_e \frac{T_2}{T_1}$$

$$= (c_v + R) \log_e \frac{T_2}{T_1} - R \log_e \frac{p_2}{p_1}$$

$$= c_p \log_e \frac{T_2}{T_1} - R \log_e \frac{p_2}{p_1}$$

$$\therefore \quad s_2 - s_1 = c_p \log_e \frac{T_2}{T_1} - R \log_e \frac{p_2}{p_1}. \qquad ...(4.30)$$

4.17.2. Heating a Gas at Constant Volume

Refer Fig. 4.25. Let 1 kg of gas be heated at constant volume and let the change in entropy and absolute temperature be from s_1 to s_2 and T_1 to T_2 respectively.

Then $Q = c_v(T_2 - T_1)$

Differentiating to find small increment of heat dQ corresponding to small rise in temperature dT.

$$dQ = c_v dT$$

Dividing both sides by T, we get

$$\frac{dQ}{T} = c_v \cdot \frac{dT}{T}$$

or $$ds = c_v \cdot \frac{dT}{T}$$

Integrating both sides, we get

$$\int_{s_1}^{s_2} ds = c_v \int_{T_1}^{T_2} \frac{dT}{T}$$

or $$s_2 - s_1 = c_v \log_e \frac{T_2}{T_1} \qquad ...(4.31)$$

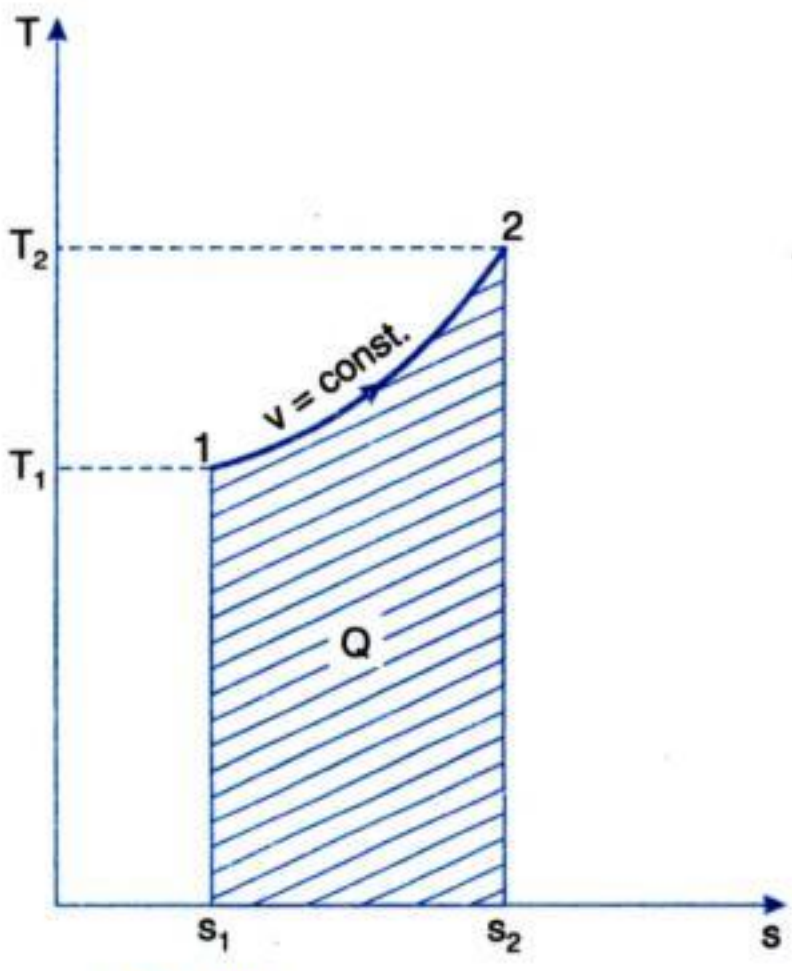

Fig. 4.25. *T-s* diagram : Constant volume process

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

$$= c_v \log_e \frac{T_2}{T_1} - (c_p - c_v) \times \left(\frac{1}{n-1}\right) \log_e \frac{T_2}{T_1} \qquad [\because \quad R = c_p - c_v]$$

$$= c_v \log_e \frac{T_2}{T_1} - (\gamma \,.\, c_v - c_v) \times \left(\frac{1}{n-1}\right) \log_e \frac{T_2}{T_1} \qquad [\because \quad c_p = \gamma \,.\, c_v]$$

$$= c_v\left[1 - \left(\frac{\gamma - 1}{n-1}\right)\right] \log_e \frac{T_2}{T_1} = c_v\left[\frac{(n-1)-(\gamma-1)}{(n-1)}\right] \log_e \frac{T_2}{T_1}$$

$$= c_v\left(\frac{n-1-\gamma+1)}{n-1}\right) \log_e \frac{T_2}{T_1}$$

$$= c_v \,.\, \left(\frac{n-\gamma}{n-1}\right) \log_e \frac{T_2}{T_1} \text{ per kg of gas}$$

$\therefore$ $$s_2 - s_1 = c_v \left(\frac{n-\gamma}{n-1}\right) \log_e \frac{T_2}{T_1} \text{ per kg of gas} \qquad ...(4.35)$$

4.17.7. Approximation for Heat Absorbed

The curve *LM* shown in Fig. 4.30 is obtained by heating 1 kg of gas from initial state *L* to final state *M*. Let temperature during heating increases from T_1 to T_2. Then heat absorbed by the gas will be given by the area (shown shaded) under curve *LM*.

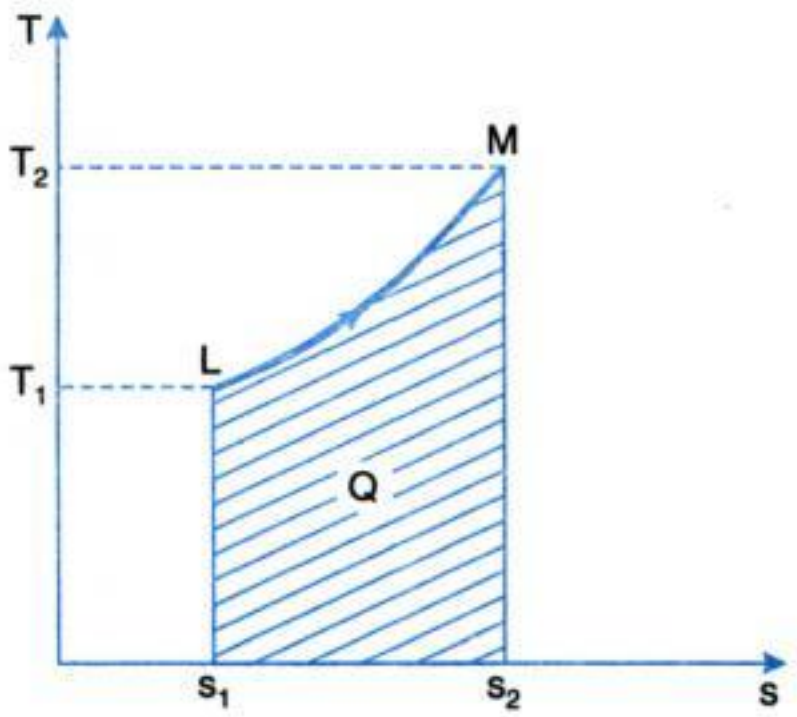

Fig. 4.30

As the curve on *T-s* diagram which represents the heating of the gas, usually has very slight curvature, it can be assumed a straight line for a small temperature range. Then,

Heat absorbed = Area under the curve *LM*

$$= (s_2 - s_1)\left(\frac{T_1 + T_2}{2}\right) \qquad ...(4.36)$$

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Solution. Change in entropy during constant volume process

$$= m\, c_v \ln \left(\frac{T_2}{T_1}\right) \qquad ...(i)$$

Change in entropy during polytropic process (pv^n = constant)

$$= m\, c_v \left(\frac{\gamma - n}{n-1}\right) \ln \left(\frac{T_2}{T_1}\right) \qquad ...(ii)$$

For the same entropy, equating (*i*) and (*ii*), we have

$$\frac{\gamma - n}{n-1} = 1, \quad \text{or} \quad (\gamma - n) = (n-1) \quad \text{or} \quad 2n = \gamma + 1$$

$$\therefore \qquad \mathbf{n = \frac{\gamma + 1}{2}}. \quad \textbf{(Ans).}$$

Example 4.23. *Air at 20°C and 1.05 bar occupies 0.025 m^3. The air is heated at constant volume until the pressure is 4.5 bar, and then cooled at constant pressure back to original temperature. Calculate :*

(i) The net heat flow from the air.

(ii) The net entropy change.

Sketch the process on T-s diagram.

Solution. The processes are shown on a *T-s* diagram in Fig. 4.31.

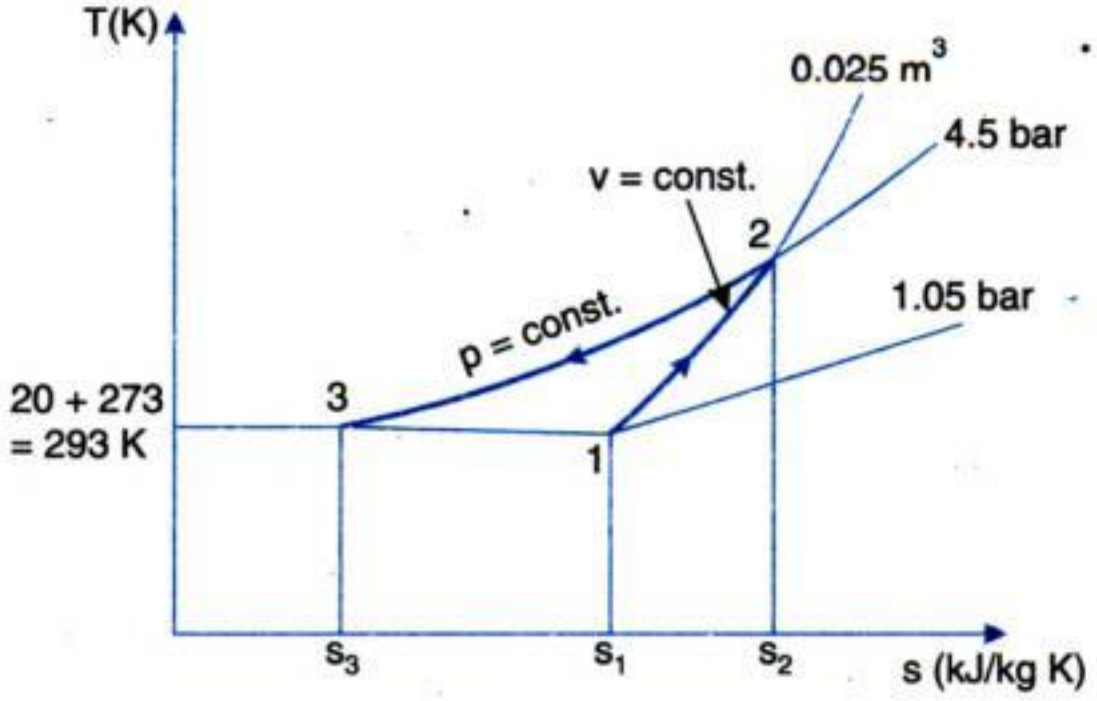

Fig. 4.31

For air :

Temperature, $T_1 = 20 + 273 = 293$ K

Volume, $V_1 = V_3 = 0.025$ m³

Pressure, $p_1 = 1.05$ bar $= 1.05 \times 10^5$ N/m²

Pressure, $p_2 = 4.5$ bar $= 4.5 \times 10^5$ N/m².

(*i*) Net heat flow :

For a perfect gas (corresponding to point 1 of air),

$$m = \frac{p_1 V_1}{R T_1} = \frac{1.05 \times 10^5 \times 0.025}{0.287 \times 10^3 \times 293} = 0.0312 \text{ kg}$$

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$$\Delta U = 0,\ Q = W$$

i.e.,
$$Q = \int_{v_1}^{v_2} p \, . \, dv$$

$$= \int_{v_1}^{v_2} \frac{RT}{v} \, . \, dv \qquad \left[\because \ pv = RT \text{ and } p = \frac{RT}{v} \right]$$

$$= RT \log_e \frac{v_2}{v_1}$$

$$\therefore \qquad \frac{Q}{T} = R \log_e \frac{v_2}{v_1}$$

But this is the expression for change in entropy of the system. Entropy being the property of the system, its change is same whether it is reversible or irreversible process.

∴ For the given process,

$$\Delta s = R \log_e \left(\frac{v_2}{v_1} \right)$$

$$= 287 \log_e (2) \qquad [\because \ v_2 = 2v_1 \text{ (given)}]$$

$$= 198.9 \text{ kJ/kg K}$$

Hence **change in entropy = 198.9 kJ/kg K.** **(Ans.)**

Example 4.26. *0.04 kg of carbon dioxide (molecular weight = 44) is compressed from 1 bar, 20°C, until the pressure is 9 bar, and the volume is then 0.003 m^3. Calculate the change of entropy.*

Take c_p for carbon dioxide as 0.88 kJ/kg K, and assume carbon dioxide to be a perfect gas.

Solution. Mass of carbon dioxide, m = 0.04 kg

Molecular weight, $M = 44$

Initial pressure, $p_1 = 1 \text{ bar} = 1 \times 10^5 \text{ N/m}^2$

Initial temperature, $T_1 = 20 + 273 = 293 \text{ K}$

Final pressure, $p_2 = 9 \text{ bar}$

Final volume, $V_2 = 0.003 \text{ m}^3$

c_p for carbon dioxide $= 0.88 \text{ kJ/kg K}$

Change of entropy :

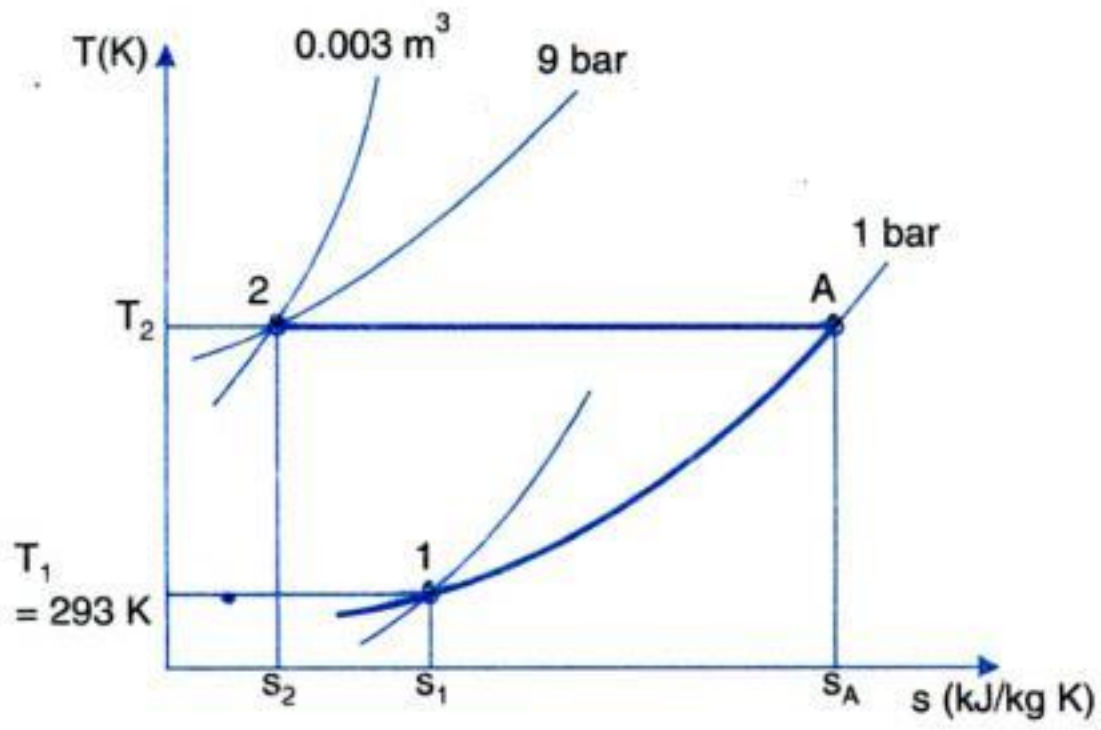

Fig. 4.34

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$$\therefore \qquad T_2 = 733 \times \left(\frac{1.012}{7}\right)^{\frac{0.4}{1.4}} = 733 \times (0.1446)^{0.286} = 421.6 \text{ K}$$

$$= 421.6 - 273 = 148.6°\text{C}.$$

But the actual temperature is 160°C at the pressure of 1.012 bar, *hence the process is irreversible.* **Proved.**

(*ii*) **Change of entropy per kg of air :**

The change of entropy $s_2' - s_1$, can be found by considering a reversible constant pressure process between 2 and 2′.

$$\therefore \qquad s_2' - s_2 = c_p \log_e \frac{T_2'}{T_2} = 1.005 \log_e \frac{433}{421.6} = 0.02681 \text{ kJ/kg K}$$

i.e., **Increase of entropy, $s_2' - s_1 = 0.02681$ kJ/kg K.** **(Ans.)**

☞**Example 4.29.** *A fluid undergoes a reversible adiabatic compression from 4 bar, 0.3 m^3 to 0.08 m^3 according to the law, $pv^{1.25}$ = constant.*

Determine : (*i*) *Change in enthalpy ;* (*ii*) *Change in internal energy ;*

(*iii*) *Change in entropy ;* (*iv*) *Heat transfer ;*

(*v*) *Work transfer.*

Solution. Refer Fig. 4.38.

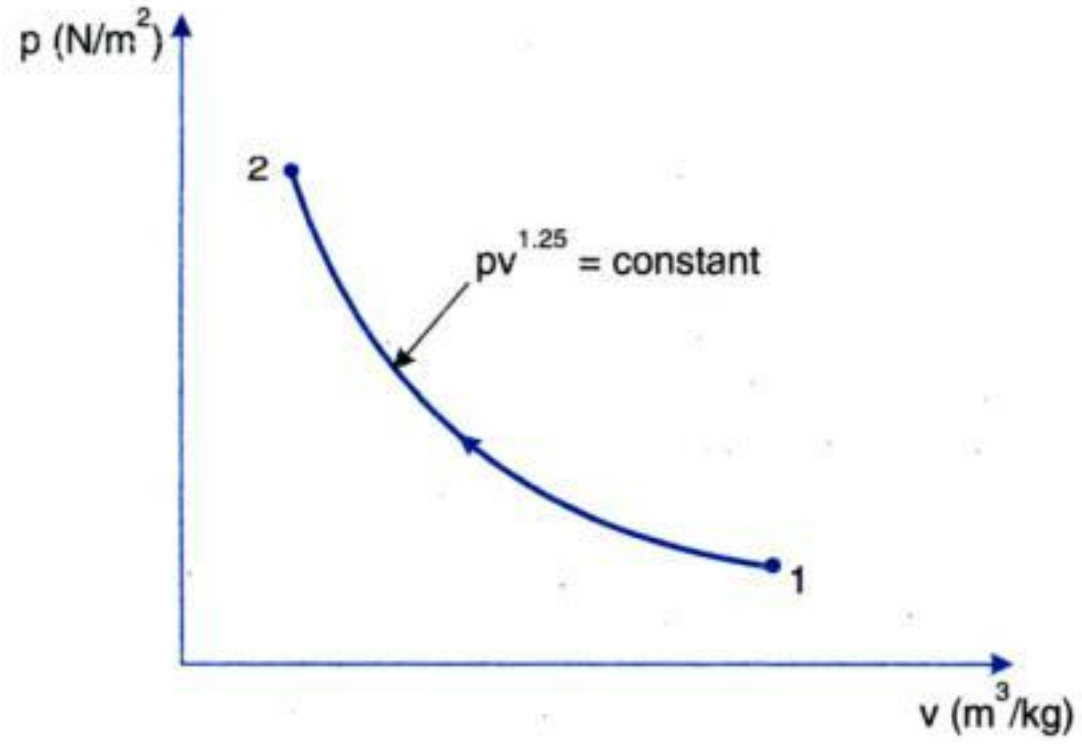

Fig. 4.38

Initial volume, $V_1 = 0.3 \text{ m}^3$

Initial pressure, $p_1 = 4 \text{ bar} = 4 \times 10^5 \text{ N/m}^2$

Final volume, $V_2 = 0.08 \text{ m}^3$

Law of compression : $pv^{1.25}$ = constant.

For reversible adiabatic process,

$$p_1 V_1^{\,n} = p_2 V_2^{\,n}$$

or $$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^n$$

$$\therefore \qquad p_2 = p_1 \times \left(\frac{V_1}{V_2}\right)^n = 4 \times \left(\frac{0.3}{0.08}\right)^{1.25} = 20.87 \text{ bar.}$$

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$$s_3 - s_1 = (s_3 - s_2) + (s_2 - s_1)$$
$$= (s_3 - s_2) + 0 = (s_3 - s_2)$$
$$= c_v \log_e \frac{T_3}{T_2} = 1.25 \log_e \left(\frac{530 - 25}{530 - 165}\right)$$
$$= 1.25 \log_e \left(\frac{505}{365}\right) = \mathbf{0.4058\ kJ/kg\ K.}\ \textbf{(Ans.)}$$

Example 4.32. *A heat pump operates between two identical bodies which are at temperature T_1 and cools one of the bodies to a temperature T_2 ($T_2 < T_1$). Prove that for this operation the minimum work required by the heat pump is given by*

$$W = c_p \left(\frac{T_1^2}{T_2^2} + T_2 - 2T_1\right)$$

where c_p is the specific heat which is same for both the bodies.

Solution. The arrangement is shown in Fig. 4.40.

For the minimum work absorbed by the heat pump, the heat pump must be reversed Carnot cycle engine and the required condition is

$$\oint \left(\frac{dQ}{T}\right) = 0$$

For infinitely small changes, we can write

$$c_p \frac{dT_1}{T_1} + c_p \frac{dT_2}{T_2} = 0$$

If T_1' is the final temperature of the high level reservoir, then the above equation can be written as

$$c_p \int_{T_1}^{T_1'} \left(\frac{dT_1}{T_1}\right) + c_p \int_{T_1}^{T_2} \left(\frac{dT_2}{T_2}\right) = 0$$

$$\therefore \quad \log_e \left(\frac{T_1'}{T_1}\right) + \log_e \left(\frac{T_2}{T_1}\right) = 0 = \log_e (1)$$

$$\therefore \quad \log_e \left(\frac{T_1' T_2}{T_1^2}\right) = \log_e (1)$$

$$\therefore \quad T_1' = \frac{T_1^2}{T_2}$$

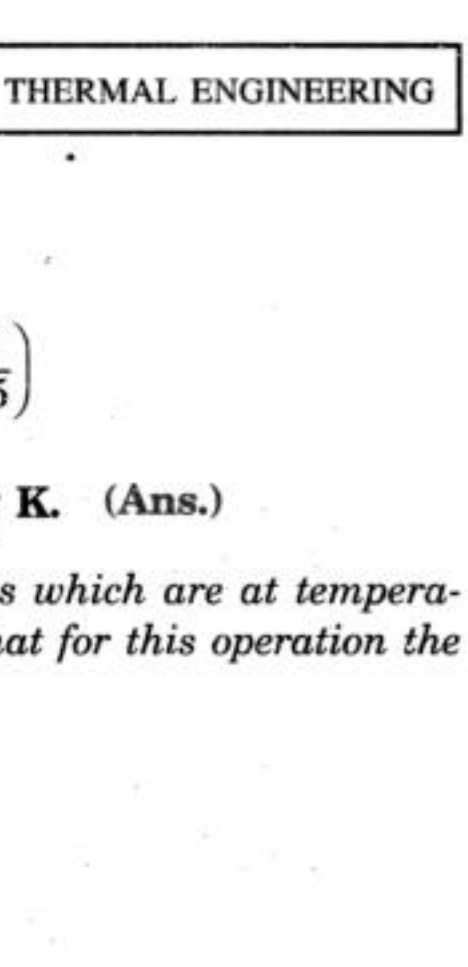

Fig. 4.40

Now the work given to the heat pump

= Heat rejected at higher level temperature
– heat picked up at lower level temperature

$$\therefore \quad W = c_p \int_{T_1}^{T_1'} (dT) - c_p \int_{T_2}^{T_1} (dT)$$
$$= c_p [(T_1' - T_1) - (T_1 - T_2)] = c_p (T_1' + T_2 - 2T_1)$$

Now substituting the value of T_1' in the above equation in terms of T_1 and T_2

$$W = c_p \left[\frac{T_1^2}{T_2} + T_2 - 2T_1\right] \textbf{. Proved.}$$

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$$\therefore \quad s_B - s_A = 1.005 \log_e \frac{T_B}{T_A} - 0.287 \log_e \frac{110}{140}$$

$$= 1.005 \log_e \left(\frac{15+273}{60+273}\right) - 0.287 \log_e \frac{110}{140}$$

$$= -0.1459 + 0.0692 = -0.0767 \text{ kJ/kg K}$$

$$(\Delta s)_{\text{system}} = -0.0767 \text{ kJ/kg K}$$

Since the duct is insulated, $(\Delta s)_{\text{surroundings}} = 0$

$\therefore \quad (\Delta s)_{\text{universe}} = -0.0767$ kJ/kg K.

This is impossible.

So, **the flow is from B to A.** **(Ans.)**

Example 4.36. *3 kg of water at 80°C is mixed with 4 kg of water at 15°C in an isolated system. Calculate the change of entropy due to mixing process.*

Solution. Fig. 4.44 shows the isolated system before mixing. When barrier is removed, the water from two compartments mix each other. Let t_m is the final equilibrium temperature after mixing.

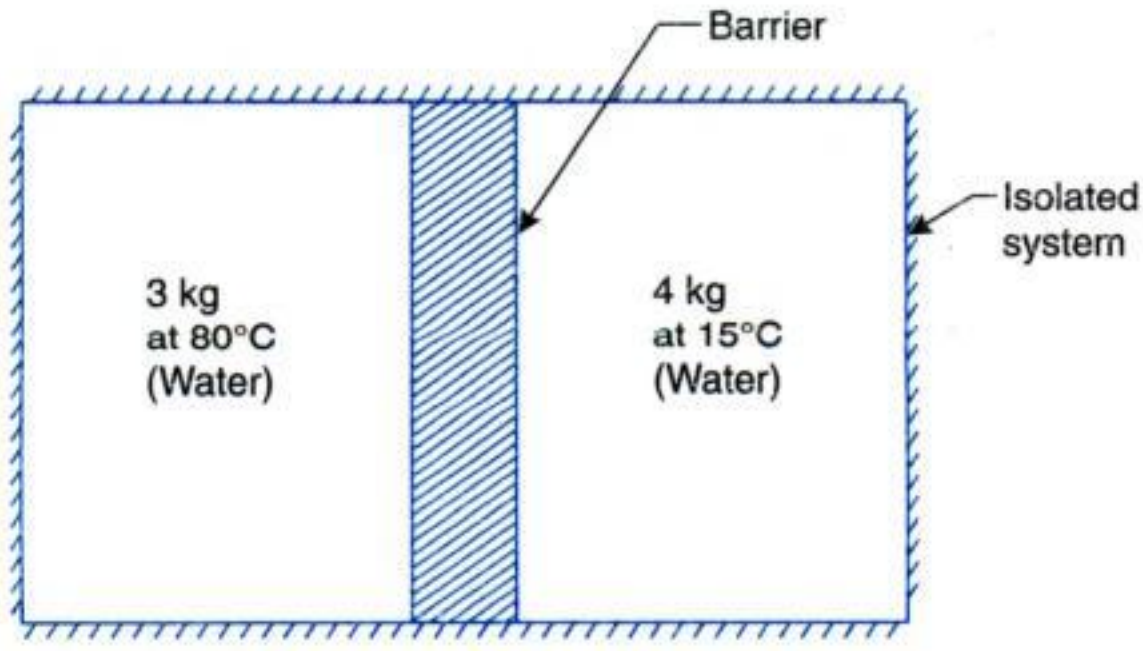

Fig. 4.44

Applying first law of thermodynamics to the isolated system :

Total energy before mixing = Total energy after mixing

$$\therefore \quad 3c_{pw}(80-0) + 4c_{pw}(15-0) = 7\,c_{pw}(t_m - 0)$$

[c_{pw} = Specific heat of water at constant pressure]

or $$240c_{pw} + 60c_{pw} = 7c_{pw}\,t_m$$

or $$240 + 60 = 7\,t_m$$

$$\therefore \quad t_m = \frac{300}{7} = 42.85°\text{C}$$

Initial entropy of the system,

$$= 3c_{pw} \log_e \left(\frac{80+273}{273}\right) + 4c_{pw} \log_e \left(\frac{15+273}{273}\right)$$

$$= 0.7709c_{pw} + 0.2139\,c_{pw} = 0.9848\,c_{pw}$$

Final entropy of the system

$$= (3+4)\,c_{pw} \log_e \left(\frac{42.85+273}{273}\right) = 1.0205\,c_{pw}$$

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(*ii*) **Minimum amount of work necessary to convert the water back into ice at – 5°C, $W_{min.}$:**

To convert 1 kg of water at 25°C to ice at – 5°C, 448.46 kJ of heat have to be removed from it, and the system has to be brought from state 4 to state 1 (Fig. 4.47). A refrigerator cycle, as shown in Fig. 4.48, is assumed to accomplish this. The entropy change of the system would be the same, *i.e.*, $s_4 - s_1$, with the only difference that its sign will be negative, because heat is removed from the system (Fig. 4.47).

$$(\Delta s)_{system} = s_1 - s_4 \ (negative)$$

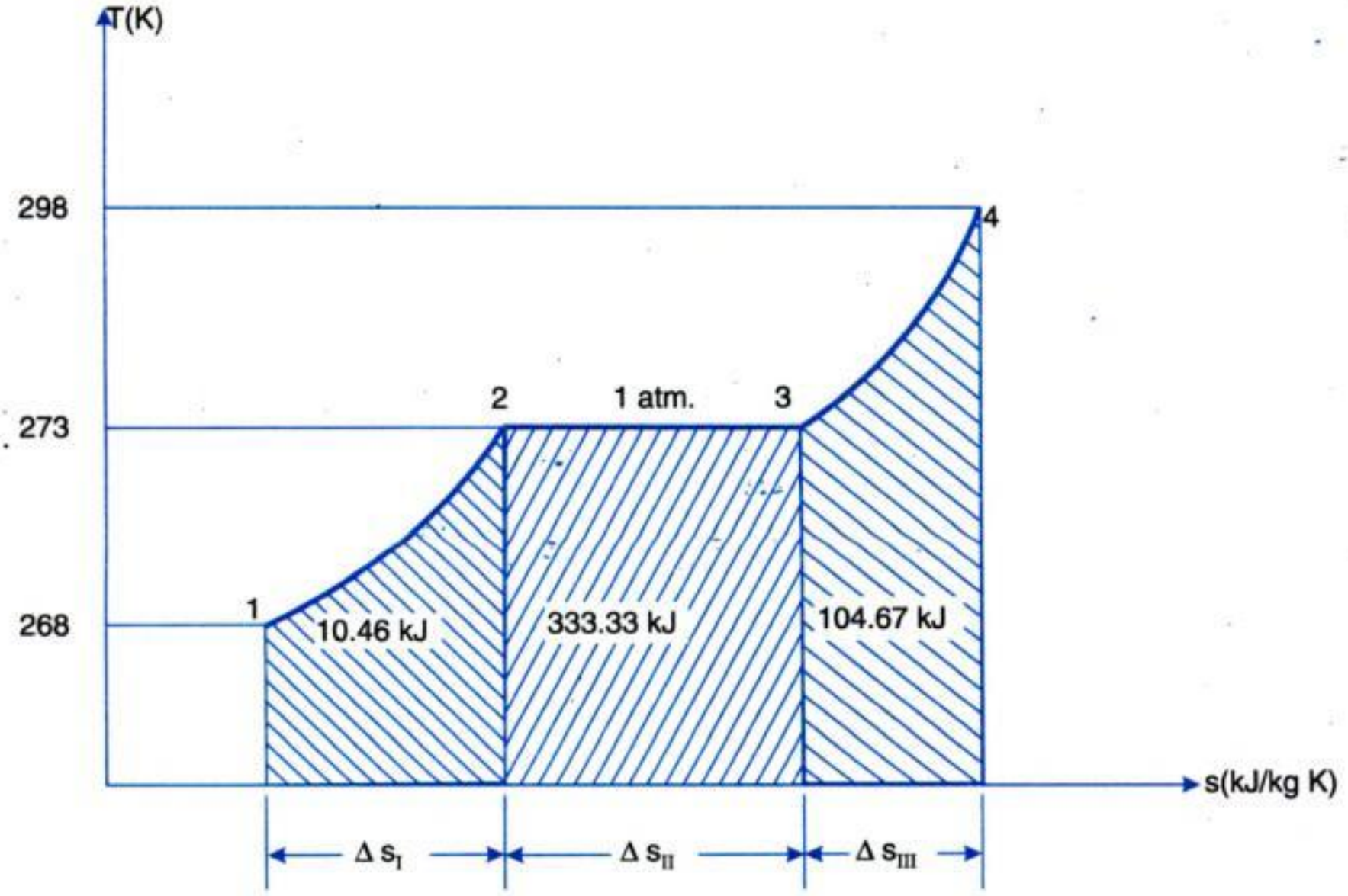

Fig. 4.47

The entropy change of the working fluid in the refrigerator would be zero, since it is operating in a cycle, *i.e.*,

$$(\Delta s)_{refrigerator} = 0.$$

The entropy change of the atmosphere (*positive*),

$$(\Delta s)_{atm.} = \frac{Q + W}{T}$$

∴ Entropy change of the universe

$$(\Delta s)_{univ.} = (\Delta s)_{system} + (\Delta s)_{refrigerator} + (\Delta s)_{atm.}$$

$$= (s_1 - s_4) + 0 + \frac{Q + W}{T} = (s_1 - s_4) + \frac{Q + W}{T}$$

By the principle of increase of entropy

$$(\Delta s)_{univ. \text{ or } isolated \ system} \geq 0$$

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or, $\text{Entropy change} = \dfrac{\text{Heat transferred}}{\text{Mean absolute temperature}}$

(*b*) *Given :* $m = 1$ kg ; $T_1 = 290$ K ; $pv^{1.3} =$ constant ; $r = 16$; $\gamma = 1.4$; $c_v = 0.718$ kJ/kg K

For a polytropic process : $\dfrac{T_2}{T_1} = \left(\dfrac{v_1}{v_2}\right)^{n-1}$

or, $T_2 = 290 \times (16)^{1.3-1} = 666.2$ K

Now, $s_2 - s_1 = c_v \left(\dfrac{n-\gamma}{n-1}\right) \log_e \left(\dfrac{T_2}{T_1}\right)$ ... per kg

$= 0.718 \left(\dfrac{1.3-1.4}{1.3-1}\right) \log_e \left(\dfrac{666.2}{290}\right) =$ **– 0.199 kJ/kg K.** **(Ans.)**

The –ve sign indicates *decrease* in entropy.

Heat transferred during the process is given by,

$$Q = \frac{\gamma - n}{\gamma - 1} \times W = \frac{\gamma - n}{\gamma - 1} \times \frac{R(T_1 - T_2)}{n-1} = c_v \left(\frac{\gamma - n}{\gamma - 1}\right)(T_1 - T_2) \quad \text{...per kg}$$

$$= 0.718 \left(\frac{1.4 - 1.3}{1.3 - 1}\right)(290 - 666.2) = -90.04 \text{ kJ/kg} \qquad \left(\because c_v = \frac{R}{\gamma - 1}\right)$$

Mean absolute temperature, $T_{mean} = \dfrac{T_1 + T_2}{2} = \dfrac{290 + 666.2}{2} = 478.1$ K

Approximate change of entropy $= \dfrac{Q}{T_{mean}} = \dfrac{-90.04}{478.1} = -0.188$ kJ/kg K

$\therefore$ *%age error* $= \dfrac{0.199 - 0.188}{0.199} \times 100 =$ **5.53%.** **(Ans.)**

- The approximate value of entropy change is lower, because in the relation $Q = T_{av} \times dS$ actual value of heat transferred is substituted instead of approximate value (*i.e.*, Area under the straight line) which is higher.

Example 4.43. *1.2 m^3 of air is heated reversibly at constant pressure from 300 K to 600 K, and is then cooled reversibly at constant volume back to initial temperature. If the initial pressure is 1 bar, calculate :*

(i) The net heat flow.

(ii) The overall change in entropy.

Represent the processes on T-S plot.

Take $c_p = 1.005$ kJ/kg K and $R = 0.287$ kJ/kg K

Solution. *Given :* $V_1 = 1.2$ m^3 ; $p_1 = p_2$; $T_1 = 300$ K ; $T_2 = 600$ K ; $p_1 = 1$ bar ;

$c_p = 1.005$ kJ/kg K ; $R = 0.287$ kJ/kg K

Fig. 4.50 shows the *T-S* plot of the processes.

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● Derivation of the formula $T^{b}\, v^{a-b} e^{kT}$ = constant for the *adiabatic expansion of gas* :

We know that, $ds = (a-b)\dfrac{dv}{v} + (b+kT)\dfrac{dT}{T}$

$$\left.\begin{aligned} s_2 - s_1 &= a \log_e v - b \log_e v + b \log_e T + kT \\ &= a \log_e v + b \log_e \left(\frac{T}{v}\right) + kT \\ &= a \log_e v + b \log_e \left(\frac{p}{R}\right) + kT \\ &= a \log_e v + \log_e \left(\frac{p}{a-b}\right)^{b} + kT \end{aligned}\right\} = 0 \text{ for adiabatic expansion}$$

This gives : $v^a\, p^b\, e^{kT}$ = constant

$p v^{a-b}\, e^{kT}$ = constant

$T^b\, v^{a-b} e^{kT}$ = constant

These expressions can be obtained by taking kT on right-side and taking the antilog of the resulting expressions.

Example 4.46. *Determine the entropy change of 4 kg of a perfect gas whose temperature varies from 127°C to 227°C during a constant volume process. The specific heat varies linearly with absolute temperature and is represented by the relation :*

$$c_v = (0.48 + 0.0096\ T)\ kJ/kg\ K.$$

Solution. *Given :* $m = 4$ kg ; $T_1 = 127 + 273 = 400$ K ; $T_2 = 227 + 273 = 500$ K ;

$c_v = (0.48 + 0.0096\ T)$ kJ/kg K.

Entropy variation for a constant volume process is given by :

$$dS = mc_v \frac{dT}{T}, \quad \text{or,} \quad dS = 4 \times (0.48 + 0.0096T)\frac{dT}{T}$$

Integrating both sides, we get

$$S_2 - S_1 = 4 \times 0.48 \int_{T_1}^{T_2} dT + 4 \times 0.0096 \int_{T_1}^{T_2} dT$$

$$= 1.92 \log_e \left(\frac{T_2}{T_1}\right) + 0.0384\ (T_2 - T_1)$$

$$= 1.92 \log_e \left(\frac{500}{400}\right) + 0.0384(500 - 400) = 4.268 \text{ kJ/K}$$

i.e., $\mathbf{S_2 - S_1 = 4.268\ kJ.}$ **(Ans.)**

Example 4.47. *The specific heats of a gas vary linearly with absolute temperature according to the following relations :*

$c_p = (0.85 + 0.00025\ T)\ kJ/kg\ K$, *and*

$c_v = (0.56 + 0.00025\ T)\ kJ/kg\ K$

If the entropy of the gas at 1 bar pressure and 273 K is zero, find the entropy of the gas at 25 bar and 750 K temperature.

Solution. *Given :* $c_p = (0.85 + 0.00025\ T)$ kJ/kg K ; $c_v = (0.56 + 0.00025\ T)$ kJ/kg K ;

$p_1 = 1$ bar ; $T_1 = 273$ K ; $p_2 = 25$ bar ; $T_2 = 750$ K.

We know that, $ds = c_v \dfrac{dT}{T} + \dfrac{p}{T} dv = c_v \dfrac{dT}{T} + R \dfrac{dv}{v}$

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8. If the temperature of the source is increased, the efficiency of the Carnot engine
(*a*) decreases (*b*) increases
(*c*) does not change (*d*) will be equal to the efficiency of a practical engine
(*e*) depends on other factors.

9. The efficiency of an ideal Carnot engine depends on
(*a*) working substance (*b*) on the temperature of the source only
(*c*) on the temperature of the sink only
(*d*) on the temperatures of both the source and the sink
(*e*) on the construction of engine.

10. The efficiency of a Carnot engine using an ideal gas as the working substance is
(*a*) $\frac{T_1 - T_2}{T_1}$ (*b*) $\frac{T_1}{T_1 - T_2}$ (*c*) $\frac{T_1 T_2}{T_1 - T_2}$
(*d*) $\frac{T_1 - T_2}{T_1 T_2}$ (*e*) $\frac{T_2(T_1 - T_2)}{T_1(T_1 + T_2)}$.

11. In a reversible cycle, the entropy of the system
(*a*) increases (*b*) decreases
(*c*) does not change (*d*) first increases and then decreases
(*e*) depends on the properties of working substance.

12. A frictionless heat engine can be 100% efficient only if its exhaust temperature is
(*a*) equal to its input temperature (*b*) less than its input temperature
(*c*) 0°C (*d*) 0°K (*e*) – 100°C.

13. Kelvin-Planck's law deals with
(*a*) conservation of energy (*b*) conservation of heat (*c*) conservation of mass
(*d*) conversion of heat into work (*e*) conversion of work into heat.

14. Which of the following statements is *correct* according to Clausius statement of second law of thermodynamics ?
(*a*) It is impossible to transfer heat from a body at a lower temperature to a body at a higher temperature
(*b*) It is impossible to transfer heat from a body at a lower temperature to a body at a higher temperature, without the aid of an external source.
(*c*) It is possible to transfer heat from a body at a lower temperature to a body at a higher temperature by using refrigeration cycle
(*d*) None of the above.

15. According to Kelvin-Planck's statement of second law of thermodynamics
(*a*) It is impossible to construct an engine working on a cyclic process, whose sole purpose is to convert heat energy into work
(*b*) It is possible to construct an engine working on a cyclic process, whose sole purpose is to convert the heat energy into work
(*c*) It is impossible to construct a device which while working in a cyclic process produces no effect other than the transfer of heat from a colder body to a hotter body
(*d*) When two dissimilar metals are heated at one end and cooled at the other, the e.m.f. developed is proportional to the difference of their temperatures at the two end.
(*e*) None of the above.

The property of a working substance which increases or decreases as the heat is supplied or removed in a ·rsible manner is known as
·lpy (*b*) internal energy
(*d*) external energy.

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20. 1 m^3 of air is heated reversibly at constant pressure from 15°C to 300°C, and is then cooled reversibly at constant volume back to the initial temperature. The initial pressure is 1.03 bar. Calculate the net heat flow and overall change of entropy, and sketch the process on a *T-s* diagram. [**Ans.** 101.5 kJ ; 0.246 kJ/K]

21. 1 kg of air is allowed to expand reversibly in a cylinder behind a piston in such a way that the temperature remains constant at 260°C while the volume is doubled. The piston is then moved in, and heat is rejected by the air reversibly at constant pressure until the volume is the same as it was initially. Calculate the net heat flow and the overall change of entropy. Sketch the processes on a *T-s* diagram.

[**Ans.** – 161.9 kJ/kg ; – 0.497 kJ/kg K]

22. 1 kg of air at 1.013 bar, 17°C, is compressed according to a law $pv^{1.3}$ = constant, until the pressure is 5 bar. Calculate the change of entropy and sketch the process on a *T-s* diagram, indicating the area, which represents the heat flow. [**Ans.** – 0.0885 kJ/kg K]

23. 0.06 m^3 of ethane (molecular weight 30), at 6.9 bar and 60°C, is allowed to expand isentropically in a cylinder behind a piston to a pressure of 1.05 bar and a temperature of 107°C. Calculate γ, R, c_p, c_v for ethane, and calculate the work done during the expansion. Assume ethane to be a perfect gas.

The same mass of ethane at 1.05 bar, 107°C, is compressed to 6.9 bar according to a law $pv^{1.4}$ = constant. Calculate the final temperature of the ethane and the heat flow to or from the cylinder walls during the compression. Calculate also the change of entropy during the compression, and sketch both processes on *p-v* and *T-s* diagrams.

[**Ans.** 1.219 ; 0.277 kJ/kg K ; 1.542 kJ/kg K ; 1.265 kJ/kg K ; 54.2 kJ ; 378°C ; 43.4 kJ ; 0.0867 kJ/K]

24. In a reversible process the rate of heat transfer to the system per unit temperature rise is given by $\frac{dQ}{dT}$ = 0.5 kJ/°C. Find the change in entropy of the system if its temperature rises from 500 K to 800 K.

[**Ans.** 0.235 kJ/kg°C]

25. In a reversible cycle 100 kJ of heat is received at 500 K ; then an adiabatic expansion occurs to 400 K, at which temperature 50 kJ of heat is received, then a further adiabatic expansion to 300 K at which temperature 100 kJ of heat is rejected :

(*i*) Find the change in entropy which occurs as the system is restored to its initial state in the remainder of the cycle.

(*ii*) If during the remainder of the cycle heat is transferred only at 400 K, how much heat is transferred and in what direction ? [**Ans.** 0.008 kJ/K ; + 3.2 kJ]

26. 1 kg of air is compressed according to the law $pv^{1.25}$ = constant from 1.03 bar and 15°C to 16.45 bar. Calculate the change in entropy. [**Ans.** 0.255 kJ/kg K]

27. A quantity of gas (mean molecular weight 36.2) is compressed according to the law pv^n = constant, the initial pressure and volume being 1.03 bar and 0.98 m^3 respectively. The temperature at the start of compression is 17°C and at the end it is 115°C. The amount of heat rejected during compression is 3.78 kJ, c_p = 0.92. Calculate :

(*i*) Value of n, (*ii*) Final pressure, (*iii*) Change in entropy.

[**Ans.** (*i*) 1.33 ; (*ii*) 1.107 bar ; (*iii*) 0.228 kJ/kg K]

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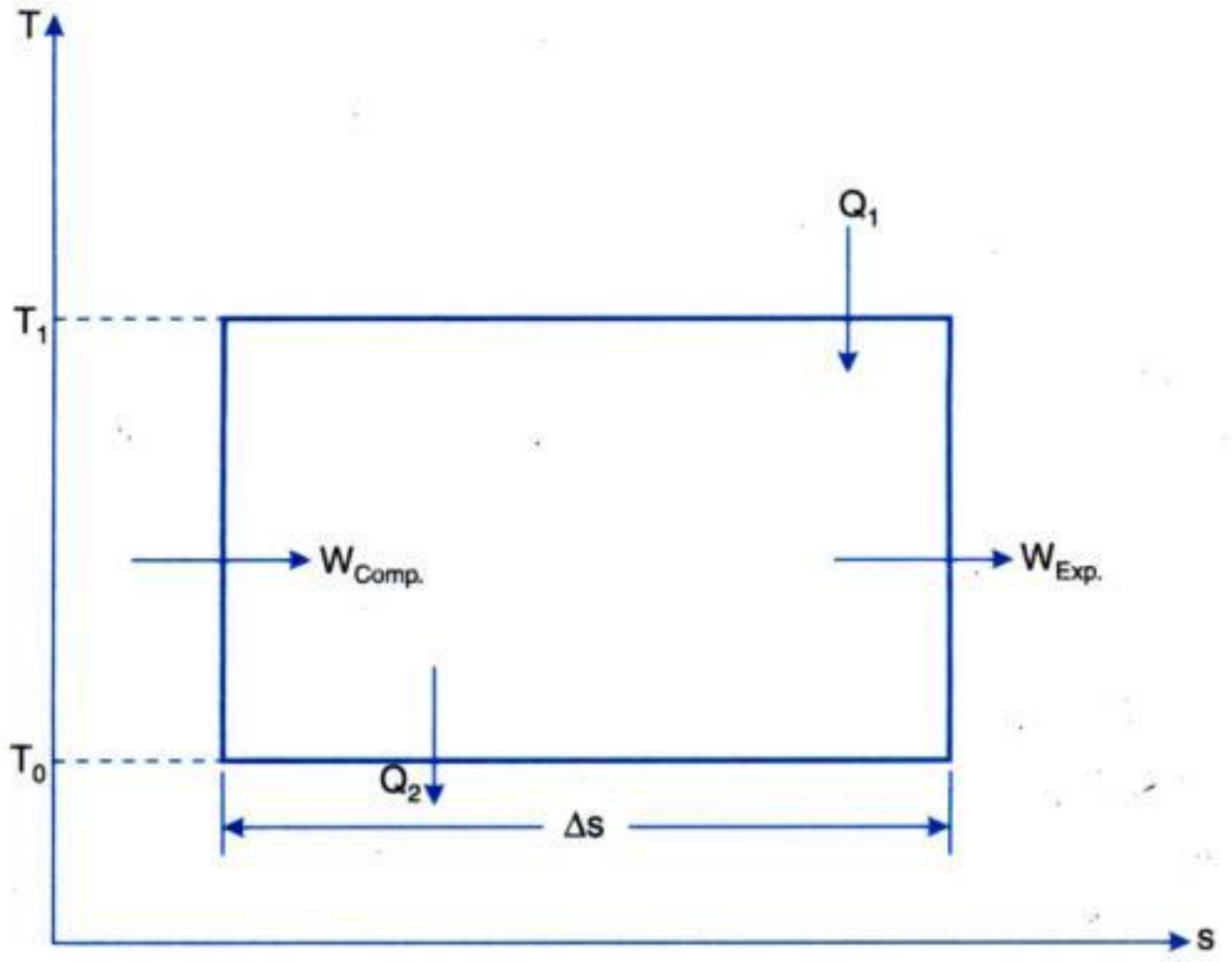

Fig. 5.4. Carnot cycle.

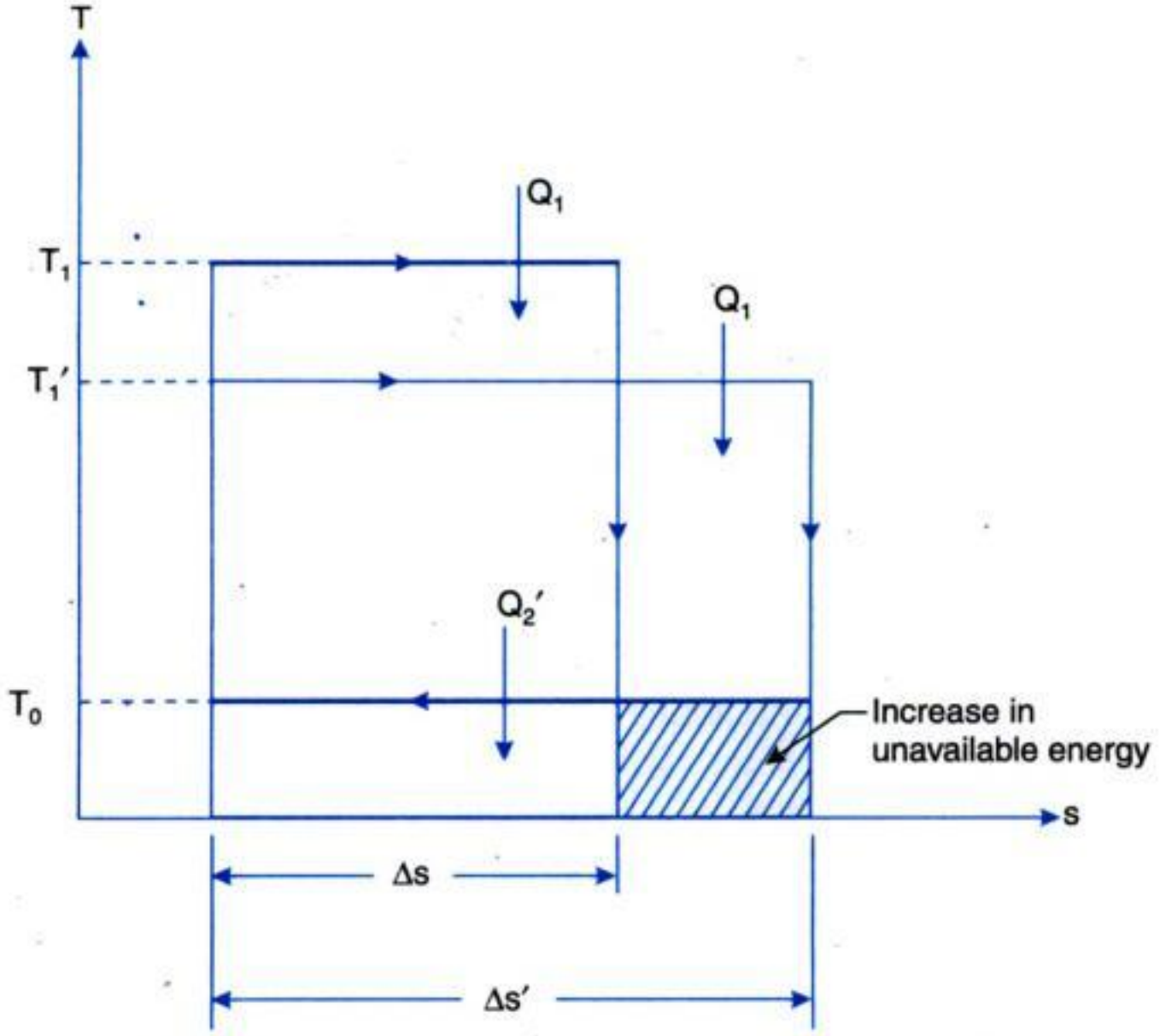

Fig. 5.5. Increase in unavailable energy due to heat transfer through a finite temperature difference.

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$$= T_0 (s_2 - s_1) - Q$$
$$= T_0 (\Delta s)_{system} + T_0 (\Delta s)_{surr.}$$

i.e., $$i = T_0 (\Delta s_{system} + \Delta s_{surr.})$$

The same expression for irreversibility applies to both flow and non-flow processes.

The quantity $T_0 (\Delta s_{system} + \Delta s_{surr.})$ represents (per unit mass) an increase in unavailable energy (or energy).

5.8. EFFECTIVENESS

Effectiveness *is defined as the ratio of actual useful work to the maximum useful work.* The useful output of a system is given by the increase of availability of the surroundings.

Effectiveness, $$\epsilon = \frac{\text{Increase of availability of surroundings}}{\text{Loss of availability of the system}} \qquad ...(5.10)$$

For a compression or heating process the effectiveness is given by

$$\epsilon = \frac{\text{Increase of availability of the system}}{\text{Loss of availability of the surroundings}}$$

or $$\epsilon = \frac{W_{useful}}{W_{max\,.\,useful}} \qquad ...[5.10\ (a)]$$

The *effectiveness of an actual process is always less than unity*. Thus effectiveness of a process is the measure of the extent to which advantage has been taken of an opportunity to obtain useful work.

Example 5.1. *One kg of air is compressed polytropically from 1 bar pressure and temperature of 300 K to a pressure of 6.8 bar and temperature of 370 K. Determine the irreversibility if the sink temperature is 293 K. Assume $R = 0.287$ kJ/kg K, $c_p = 1.004$ kJ/kg K and $c_v = 0.716$ kJ/kg/K.* **(U.P.S.C.)**

Solution. Irreversibility $I = W_{max} - W_{act}$

$$-W_{max} = \text{Change in internal energy} - T_0 \times \text{Change in entropy}$$

or $$-W_{max} = (u_2 - u_1) - T_0(s_2 - s_1) = W_{rev}$$

or $$-W_{max} = c_v(T_2 - T_1) - T_0[c_p \ln (T_2/T_1) - R \ln (p_2/p_1)]$$
$$= 0.716(370 - 300) - 293 \times [1.005 \ln (370/300) - 0.287 \ln (6.8/1)]$$

or $$W_{max} = -149.53 \text{ kJ/kg} = W_{rev}$$

(negative sign indicates that work is done on air)

The index of compression 'n' is given by

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{[(n-1)/n]}$$

or $$\frac{n-1}{n} = \frac{\ln (T_2/T_1)}{\ln (p_2/p_1)} = \frac{\ln (370/300)}{\ln (6.8/1)}$$

or $$n = 1.123$$

$$W_{actual} = \frac{mR(T_1 - T_2)}{n-1} = \frac{1 \times 0.287(300 - 370)}{1.123 - 1} = -163.33 \text{ kJ/kg}$$

$$I = W_{rev} - W_{act} = -149.53 - (-163.33) = \mathbf{13.8\ kJ/kg.}$$ **(Ans.)**

Example 5.2. *A system at 500 K receives 7200 kJ/min from a source at 1000 K. The temperature of atmosphere is 300 K. Assuming that the temperatures of system and source remain constant during heat transfer find out :*

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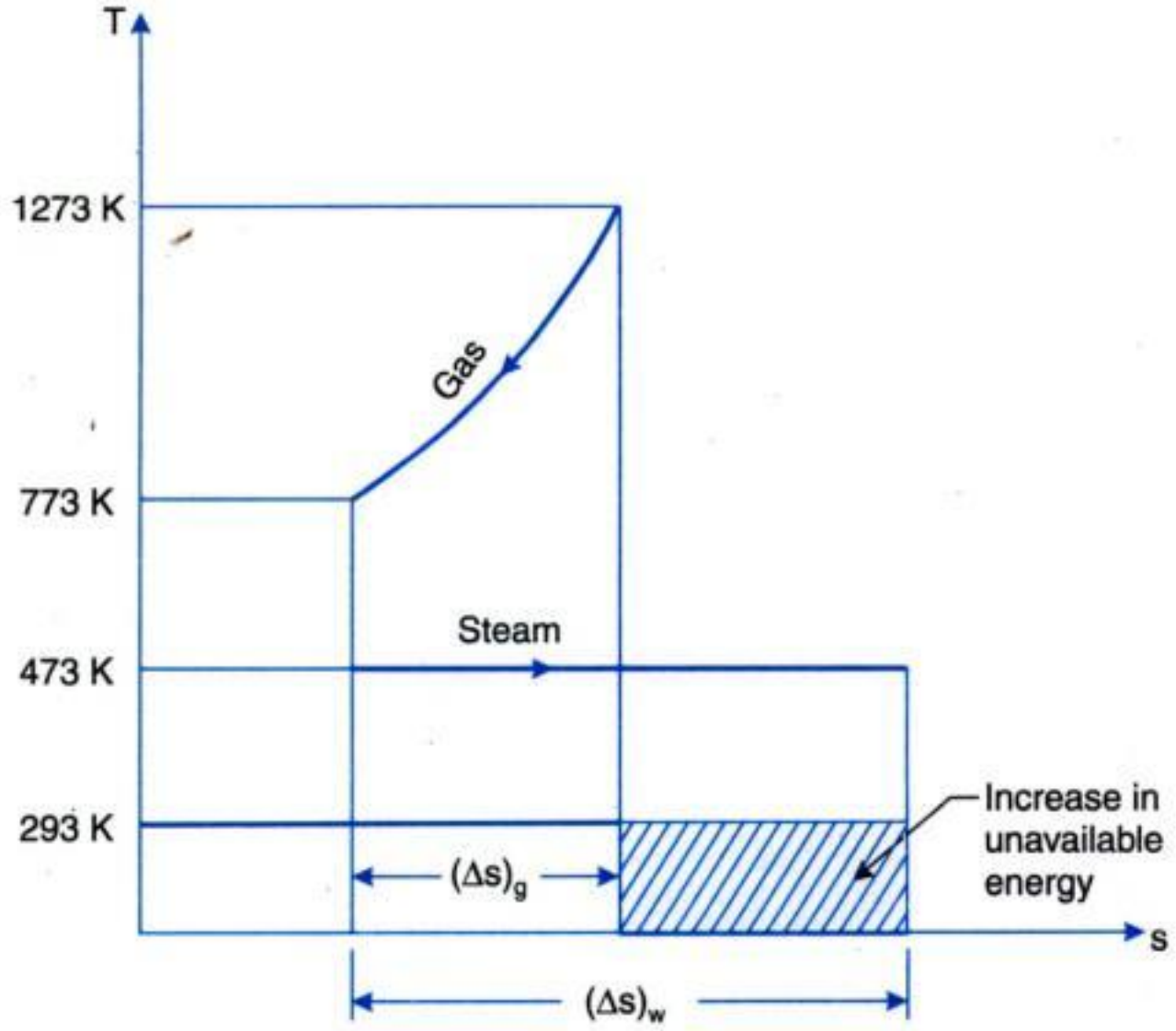

Fig. 5.8

Change of entropy of water (per kg) when it is converted into steam,

$$(\Delta s)_w = \frac{h_{fg}}{T_s} = \frac{1940.7}{(200 + 273)} = 4.103 \text{ kJ/kg K.}$$

Net change in entropy due to heat transfer

$$= -1.935 + 4.103 = \textbf{2.168 kJ/K.} \quad \textbf{(Ans.)}$$

Increase in unavailable energy due to heat transfer

$$= 293 \times 2.168, \textit{ i.e.,} \text{ cross hatched area}$$

$$= \textbf{635.22 kJ per kg of steam formed.} \quad \textbf{(Ans.)}$$

Example 5.5. *3 kg of gas (c_v = 0.81 kJ/kg K) initially at 2.5 bar and 400 K receives 600 kJ of heat from an infinite source at 1200 K. If the surrounding temperature is 290 K, find the loss in available energy due to above heat transfer.*

Solution. Refer Fig. 5.9.

Mass of gas, $m_g = 3$ kg

Initial pressure of gas = 2.5 bar

Initial temperature, $T_1' = 400$ K

Quantity of heat received by gas, $Q = 600$ kJ

Specific heat of gas, $c_v = 0.81$ kJ/kg K

Surrounding temperature = 290 K

Temperature of infinite source, $T_1 = 1200$ K

Heat received by the gas is given by,

$$Q = m_g c_v (T_2' - T_1')$$

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(*ii*) Heat transferred during cooling

$$Q = m \times c_p \times (T_1 - T_0)$$
$$= 5 \times 1.005 \times (550 - 290)$$
$$= 1306.5 \text{ kJ} \quad \text{..... heat lost by the system}$$

Change of entropy during cooling

$$\Delta S = m \times c_p \times \log_e \left(\frac{T_1}{T_0}\right)$$
$$= 5 \times 1.005 \times \log_e \left(\frac{550}{290}\right) = 3.216 \text{ kJ/K}$$

Unavailable portion of this energy

$$= T_0 \,.\, (\Delta S) = 290 \times 3.216 = 932.64 \text{ kJ}$$

$\therefore$ **Available energy** $= 1306.5 - 932.64 =$ **373.86 kJ.** (Ans.)

Effectiveness, $\epsilon = \dfrac{\text{Available energy}}{\text{Availability of the system}} = \dfrac{373.86}{576.7}$

$=$ **0.648 or 64.8%.** (Ans.)

Example 5.9. *Air at the rate of 25 kg/min is compressed in a centrifugal air compressor from 1 bar to 2 bar. The temperature increases from 15°C to 100°C during compression. Determine actual and minimum power required to run the compressor. The surrounding air temperature is 15°C.*

Neglect the heat interaction between the compressor and surroundings and changes in potential and kinetic energy.

Take for air, c_p = 1.005 kJ/kg K, R = 0.287 kJ/kg K.

Solution. Rate of flow of air, m = 25 kg/min.

Initial pressure, $p_1 = 1$ bar

Final pressure, $p_2 = 2.0$ bar

Initial temperature, $T_1 = T_0 = 15 + 273 = 288$ K

Final temperature, $T_2 = 100 + 273 = 373$ K.

Applying energy equation to compressor,

$$W_{actual} = h_2 - h_1 \qquad [\text{as } Q = 0,\ \Delta PE = 0,\ \Delta KE = 0]$$
$$= c_p\,(T_2 - T_1) = 1.005\,(373 - 288) = 85.4 \text{ kJ/kg}$$

Total actual work done/min

$$= 25 \times 85.4 = 2135 \text{ kJ/min}$$
$$= \frac{2135}{60} = 35.58 \text{ kJ/s} = 35.58 \text{ kW}$$

The *minimum work required is given by the increase in availability of the air stream.*

$$W_{min} = b_2 - b_1 = (h_2 - h_1) - T_0\,(s_2 - s_1)$$
$$s_2 - s_1 = c_p \log_e \left(\frac{T_2}{T_1}\right) - R \log_e \left(\frac{p_2}{p_1}\right) \qquad \text{...per unit mass}$$
$$= 1.005 \log_e \left(\frac{373}{288}\right) - 0.287 \log_e \left(\frac{2.0}{1}\right)$$
$$= 0.2599 - 0.1989 = 0.061 \text{ kJ/kg K}$$

$\therefore$ $W_{min} = (h_2 - h_1) - T_0(s_2 - s_1)$
$= 85.4 - 288 \times 0.061 = 67.8$ kJ/kg

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Loss in availability :

Consider one kg of oil.

Heat lost by oil = Heat gained by water

$$m_0 \times c_{p0} \times (T_{0_1} - T_{0_2}) = m_w \times c_{pw} \times (T_{w_2} - T_{w_1})$$

where c_{p0} = Specific heat of oil (2.6 kJ/kg K),

c_{pw} = Specific heat of water (4.18 kJ/kg K), and

m_0 = Mass of oil (= 1 kg),

m_w = Mass of water (= ?)

$\therefore \quad 1 \times 2.6 \times (513 - 363) = m_w \times 4.18 \times (343 - 323)$

or $390 = 83.6\ m_w$ or $m_w = 4.66$ kg

Entropy change of water

$$= m_w\ c_{pw} \log_e \frac{T_{w_2}}{T_{w_1}} = 4.66 \times 4.18 \times \log_e \left(\frac{343}{323}\right) = 1.17 \text{ kJ/K}$$

Entropy change of oil

$$= m_0 c_{p0} \log_e \left(\frac{T_{0_2}}{T_{0_1}}\right) = 1 \times 2.6 \log_e \left(\frac{363}{513}\right) = -0.899 \text{ kJ/K}$$

Change in availability of water

$$= m_w[c_{pw}(T_{w_2} - T_{w_1})] - T_0\ (\Delta S)_w$$

$$= 4.66[(4.18\ (343 - 323)] - 300 \times 1.17 = 38.57 \text{ kJ}$$

+ve sign indicates an *increase in availability*

Change in availability of oil

$$= m_0[c_{p0}(T_{0_2} - T_{0_1})] - T_0\ (\Delta S)_0]$$

$$= 1[2.6(363 - 513)] - 300 \times (-0.899) = -120.3 \text{ kJ/K}$$

$\therefore$ **Loss in availability**

$$= -120.3 + 38.57 = \mathbf{-81.73\ kJ.}$$ (Ans.)

(–ve sign indicates the **loss**).

Example 5.13. *1 kg of ice at 0°C is mixed with 12 kg of water at 27°C. Assuming the surrounding temperature as 15°C, calculate the net increase in entropy and unavailable energy when the system reaches common temperature :*

Given : Specific heat of water = 4.18 kJ/kg K ; specific heat of ice = 2.1 kJ/kg K and enthalpy of fusion of ice (latent heat) = 333.5 kJ/kg.

Solution. Mass of ice, $m_{ice} = 1$ kg

Temperature of ice, $T_{ice} = 0 + 273 = 273$ K

Mass of water, $m_{water} = 12$ kg

Temperature of water, $T_{water} = 27 + 273 = 300$ K

Surrounding temperature, $T_0 = 15 + 273 = 288$ K

Specific heat of water = 4.18 kJ/kg K

Specific heat of ice = 2.1 kJ/kg K

Latent heat of ice = 333.5 kJ/kg

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$$= 0.6[50.25 - 42.38] = 4.722 \text{ kJ/kg}$$

$$\therefore \quad \textit{Effectiveness} = \frac{\text{Increase of availability of system}}{\text{Loss of availability of surroundings}}$$

$$= \frac{1.465}{4.722} = \mathbf{0.31} \quad \text{or} \quad \mathbf{31\%.} \quad \text{(Ans.)}$$

The *low* figure for the effectiveness is an indication of the *highly irreversible nature of the mixing process.*

☞**Example 5.17.** *2.5 kg of air at 6 bar, 90°C expands adiabatically is a closed system until its volume is doubled and its temperature becomes equal to that of the surroundings which is at 1 bar, 5°C. For this process determine :*

(i) The maximum work ;

(ii) The change in availability ;

(iii) The irreversibility.

For air take : c_v = 0.718 kJ/kg K, R = 0.287 kJ/kg K.

Solution. Mass of air, $m = 2.5$ kg

Initial pressure of air, $p_1 = 6 \text{ bar} = 6 \times 10^5 \text{ N/m}^2$

Ratio of final to initial volume, $\frac{V_2}{V_1} = 2$

Initial temperature of air, $T_1 = 90 + 273 = 363$ K

Final pressure of air, $p_2 = 1 \text{ bar} = 1 \times 10^5 \text{ N/m}^2$

Final temperature of air, $T_2 = T_0 = 5 + 273 = 278$ K

From the property relation

$$TdS = dU + pdV$$

or
$$dS = \frac{dU}{T} + \frac{pdV}{T}$$

$$dS = \frac{mc_v dT}{T} + \frac{mRdV}{V} \quad \left[\because u = c_v\, dT \text{ and } pV = mRT \text{ or } \frac{p}{T} = \frac{mR}{V}\right]$$

∴ The entropy change of air between the initial and final states is

$$\int_1^2 dS = \int_1^2 \frac{mc_v dT}{T} + \int_1^2 \frac{mRdV}{V}$$

or
$$S_2 - S_1 = mc_v \log_e \frac{T_2}{T_1} + mR \log_e \frac{V_2}{V_1}$$

(*i*) The maximum work, W_{max} :

Also, $W_{max} = (U_1 - U_2) - T_0(S_1 - S_2)$

$$= m\left[c_v(T_1 - T_2) + T_0\left(c_v \log_e \frac{T_2}{T_1} + R \log_e \frac{v_2}{v_1}\right)\right]$$

$$= 2.5\left[0.718(363 - 278) + 278\left(0.718 \log_e \left(\frac{278}{363}\right) + 0.287 \log_e 2\right)\right]$$

$$= 2.5[61.03 + 278(-0.1915 + 0.1989)] = 157.7 \text{ kJ}$$

Hence, **maximum work = 157.7 kJ.** (Ans.)

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Example 5.20. *A flywheel whose moment of inertia is 0.62 kg m² rotates at a speed 2500 r.p.m. in a large heat insulated system, the temperature of which is 20°C.*

(i) If the K.E. of the flywheel is dissipated as frictional heat at the shaft bearings which have a water equivalent of 1.9 kg, find the rise in the temperature of the bearings when the flywheel has come to rest.

(ii) Calculate the greatest possible amount of the above heat which may be returned to the flywheel as high-grade energy, showing how much of the original K.E. is now unavailable. What would be the final r.p.m. of the flywheel, if it is set in motion with this available energy ?

Solution. Moment of inertia of the flywheel, $I = 0.62$ kg m²

Initial angular velocity of the flywheel, $\omega_1 = \dfrac{2\pi N_1}{60} = \dfrac{2\pi \times 2500}{60}$

$= 261.8$ rad/s.

Temperature of insulated system, $T_0 = 20 + 273 = 293$ K

Water equivalent of shaft bearings $= 1.9$ kg

(i) Initial available energy of the flywheel,

$$(\text{K.E.})_{initial} = \frac{1}{2} I \omega_1^{\,2}$$

$$= \frac{1}{2} \times 0.62 \times (261.8)^2 = 2.12 \times 10^4 \text{ Nm} = 21.2 \text{ kJ}.$$

When this K.E. is dissipated as frictional heat, if Δt is the temperature rise of the bearings, we have

Water equivalent of bearings × rise in temperature = 21.2

i.e., $(1.9 \times 4.18)\, \Delta t = 21.2$

or $$\Delta t = \frac{21.2}{1.9 \times 4.18} = 2.67°\text{C}$$

Hence, **rise in temperature of bearings = 2.67°C.** (Ans.)

∴ **Final temperature of the bearings = 20 + 2.67 = 22.67°C.**

(ii) The maximum amount of energy which may be returned to the flywheel as high-grade energy is,

$$\text{A.E.} = 1.9 \times 4.18 \int_{293}^{295.67} \left(1 - \frac{293}{T}\right) dT$$

$$= 1.9 \times 4.18 \left[(295.67 - 293) - 293 \log_e \left(\frac{295.67}{293}\right)\right] = \mathbf{0.096\ kJ.} \quad \text{(Ans.)}$$

The amount of energy rendered unavailable is

$$\text{U.E.} = (\text{A.E.})_{\text{initial}} - (\text{A.E.})_{\text{returnable as high grade energy}}$$

$$= 21.2 - 0.096 = 21.1 \text{ kJ}.$$

Since the amount of energy returnable to the flywheel is 0.096 kJ, if ω_2 is the final angular velocity, and the flywheel is set in motion with this energy, then

$$0.096 \times 10^3 = \frac{1}{2} \times 0.62 \times \omega_2^{\,2}$$

∴ $$\omega_2^{\,2} = \frac{0.096 \times 10^3 \times 2}{0.62} = 309.67 \quad \text{or} \quad \omega_2 = 17.59 \text{ rad/s}.$$

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3. Energy is said to be *degraded* each time it flows through a finite temperature difference. That is, why the second law of thermodynamics is sometimes called the *law of the degradation of energy*, and energy is said to '*run down hill*'.
4. In non-flow systems :
Maximum work available,

$$W_{max} = (u_1 - u_0) - T_0(s_1 - s_0) - p_0(v_0 - v_1)$$
$$= (u_1 + p_0v_1 - T_0s_1) - (u_0 + p_0v_0 - T_0s_0)$$
$$= a_1 - a_0 \quad \text{...per unit mass}$$

The property $a = u + p_0v - T_0s$ is called the *non-flow availability function*.
5. In steady-flow systems :
Maximum work available,

$$W_{max} = (h_1 - T_0s_1) - (h_0 - T_0s_0)$$
$$= b - b_0 \quad \text{...per unit mass}$$

The property, $b = h - T_0s$ is called the *steady-flow availability function*.
6. It may be noted that Gibb's function $g = (h - Ts)$ is a property of the system where availability function $a = u + p_0v - T_0s$ is a composite property of the system and surroundings.
Again,

$$a = u + p_0v - T_0s$$
$$b = u + pv - T_0s$$
$$g = u + pv - Ts$$

When state 1 proceeds to dead state (zero state)

$$a = b = g.$$

7. The actual work which a system does is always less than the idealized reversible work, and the difference between the two is called the *irreversibility of the process*. This is also sometimes referred to as *degradation* or *dissipation*.

Effectiveness is defined as the ratio of actual useful work to the maximum useful work.

OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. Gibb's function is expressed as
(a) $(u + pv - Ts)$ (b) $(u + pv - Tds)$
(c) $(u + pdv - Tds)$ (d) $(u + pv - sdT)$.
2. Availability function is expressed as
(a) $a = (u + p_0v - T_0s)$ (b) $a = (u + p_0dv + T_0ds)$
(c) $a = (du + p_0dv - T_0ds)$ (d) $a = (u + p_0v + T_0s)$.
3. To increase work capacity of energy transferred by heat transfer from high temperature to low temperature
(a) lower temperature should be lowered keeping temperature difference same
(b) higher temperature should be increased keeping temperature difference same
(c) temperature difference should be increased (d) temperature difference should be decreased.
4. Helmholtz function is expressed as
(a) $(u - Ts)$ (b) $(h - Ts)$
(c) $(-sdT + vdp)$ (d) $(u + pv)$.
5. If a heat source at temperature T_1 transfers heat to a system at temperature $T_2 (T_1 > T_2)$, state which of the following statements is *not true* ?
(a) Δ_{system} decreases (b) Δ_{source} decreases
(c) $(\Delta_{system} + \Delta_{source})$ decreases (d) $(\Delta_{system} + \Delta_{source})$ increases.

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6 Thermodynamic Relations

6.1. General aspects. 6.2. Fundamentals of partial differentiation. 6.3. Some general thermodynamic relations. 6.4. Entropy equations (*Tds* equations). 6.5. Equations for internal energy and enthalpy. 6.6. Measurable quantities—Equation of state—Co-efficient of expansion and compressibility—Specific heats—Joule-Thomson co-efficient 6.7. Clausius-Claperyon equation—Highlights—Objective Type Questions—Exercises.

6.1. GENERAL ASPECTS

In this chapter, some important thermodynamic relations are deduced ; principally those which are useful when tables of properties are to be compiled from limited experimental data, those which may be used when calculating the work and heat transfers associated with processes undergone by a liquid or solid. It should be noted that the relations only apply to a substance in the solid phase when the stress, *i.e.* the pressure, is uniform in all directions ; if it is not, a single value for the pressure cannot be alloted to the system as a whole.

Eight properties of a system, namely *pressure* (p), *volume* (v), *temperature* (T), *internal energy* (u), *enthalpy* (h), *entropy* (s), *Helmholtz function* (f) and *Gibbs function* (g) have been introduced in the previous chapters. h, f and g are sometimes referred to as **thermodynamic potentials.** Both f and g are useful when considering chemical reactions, and the former is of fundamental importance in statistical thermodynamics. The Gibbs function is also useful when considering processes involving a change of phase.

Of the above eight properties only the first three, *i.e.*, p, v and T are directly measurable. We shall find it convenient to introduce other combination of properties which are relatively easily measurable and which, together with measurements of p, v and T, enable the values of the remaining properties to be determined. These combinations of properties might be called **'thermodynamic gradients'** ; they are all defined as the rate of change of one property with another while a third is kept constant.

6.2. FUNDAMENTALS OF PARTIAL DIFFERENTIATION

Let three variables are represented by x, y and z. Their functional relationship may be expressed in the following forms :

$$f(x, y, z) = 0 \qquad ...(i)$$

$$x = x(y, z) \qquad ...(ii)$$

$$y = y(x, z) \qquad ...(iii)$$

$$z = z(x, y) \qquad ...(iv)$$

Let x is a function of two independent variables y and z

$$x = x(y, z) \qquad ...(6.1)$$

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Consider the first quantity $\left(\frac{\partial u}{\partial T}\right)_v$. During a process at constant volume, the first law informs us that an increase of internal energy is equal to heat supplied. If a calorimetric experiment is conducted with a known mass of substance at constant volume, the quantity of heat Q required to raise the temperature of unit mass by ΔT may be measured. We can then write : $\left(\frac{\Delta u}{\Delta T}\right)_v = \left(\frac{Q}{\Delta T}\right)_v$. The quantity obtained this way is known as the mean *specific heat at constant volume* over the temperature range ΔT. It is found to vary with the conditions of the experiment, *i.e.,* with the temperature range and the specific volume of the substance. As the temperature range is reduced the value approaches that of $\left(\frac{\partial u}{\partial T}\right)_v$, and the *true specific heat at constant volume* is defined by $c_v = \left(\frac{\partial u}{\partial T}\right)_v$. This is a property of the substance and in general its value varies with the state of the substance, *e.g.,* with temperature and pressure.

According to first law of thermodynamics the heat supplied is equal to the increase of enthalpy during a reversible constant pressure process. Therefore, a calorimetric experiment carried out with a substance at constant pressure gives us, $\left(\frac{\Delta h}{\Delta T}\right)_p = \left(\frac{Q}{\Delta T}\right)_p$ which is the mean specific heat at constant pressure. As the range of temperature is made infinitesimally small, this becomes the rate of change of enthalpy with temperature at a particular state defined by T and p, and this is true *specific heat at constant pressure* defined by $c_p = \left(\frac{\partial h}{\partial T}\right)_p$. c_p also varies with the state, *e.g.,* with pressure and temperature.

The description of experimental methods of determining c_p and c_v can be found in texts on physics. When solids and liquids are considered, it is not easy to measure c_v owing to the stresses set up when such a substance is prevented from expanding. However, a relation between c_p, c_v, β and K can be found as follows, from which c_v may be obtained if the remaining three properties have been measured.

The First Law of Thermodynamics, for a reversible process states that

$$dQ = du + p\,dv$$

Since we may write $u = \phi(T, v)$, we have

$$du = \left(\frac{du}{\partial T}\right)_v dT + \left(\frac{\partial u}{\partial v}\right)_T dv$$

$$\therefore \quad dQ = \left(\frac{\partial u}{\partial T}\right)_v dT + \left\{p + \left(\frac{\partial u}{\partial v}\right)_T\right\} dv = c_v\,dT + \left\{p + \left(\frac{\partial u}{\partial v}\right)_T\right\} dv$$

This is true for any reversible process, and so, for a reversible constant pressure process,

$$dQ = c_p(dT)_p = c_v(dT)_p + \left\{p + \left(\frac{\partial u}{\partial v}\right)_T\right\} (dv)_p$$

Hence $$c_p - c_v = \left\{p + \left(\frac{\partial u}{\partial v}\right)_T\right\} \left(\frac{\partial v}{\partial T}\right)_p$$

Also $$\left(\frac{\partial p}{\partial T}\right)_v = \left(\frac{\partial s}{\partial v}\right)_T = \frac{1}{T}\left\{p + \left(\frac{\partial u}{\partial v}\right)_T\right\}$$, and therefore

$$c_p - c_v = T\left(\frac{\partial p}{\partial T}\right)_v \left(\frac{\partial v}{\partial T}\right)_p$$

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or
$$\left(\frac{\partial v}{\partial T}\right)_p = \frac{R}{p} = \frac{v}{T}$$

$$\therefore \quad \mu = \frac{1}{c_p}\left(T \times \frac{v}{T} - v\right) = 0.$$

Therefore, if an ideal gas is throttled, there will not be any change in temperature.

Let $h = f(p, T)$

Then
$$dh = \left(\frac{\partial h}{\partial p}\right)_T dp + \left(\frac{\partial h}{\partial T}\right)_p dT \qquad ...(6.47)$$

But $\left(\frac{\partial h}{\partial T}\right)_p = c_p$

$$\therefore \quad dh = \left(\frac{\partial h}{\partial p}\right)_T dp + c_p\, dT$$

For throttling process, $dh = 0$

$$\therefore \quad 0 = \left(\frac{\partial h}{\partial p}\right)_T \left(\frac{\partial p}{\partial T}\right)_h + c_p \qquad ...(6.48)$$

or
$$c_p = -\frac{1}{\mu}\left(\frac{\partial h}{\partial p}\right)_T \qquad ...(6.49)$$

$\left(\frac{\partial h}{\partial p}\right)_T$ is known as the *constant temperature co-efficient.*

6.7. CLAUSIUS-CLAPERYON EQUATION

Clausius-Claperyon equation is a relationship between the *saturation pressure*, *temperature*, the *enthalpy of evaporation*, and the *specific volume* of the *two phases* involved. This equation provides a basis for calculations of properties in a two-phase region. It gives the *slope* of a curve separating the two phases in the p-T diagram.

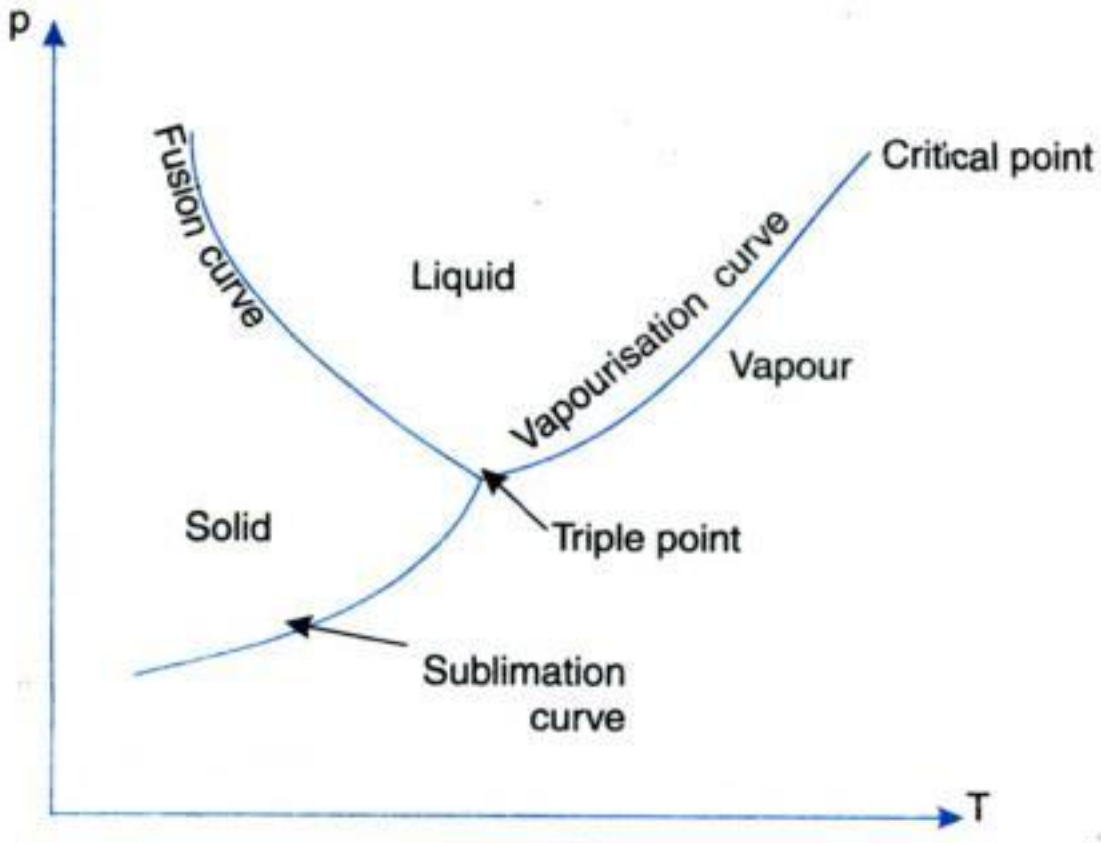

Fig. 6.4. p-T diagram.

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or $$\beta = \frac{1}{v}\left[-\frac{\frac{R}{v-b}}{\frac{-RT}{(v-b)^2}+\frac{2a}{v^3}}\right] \cdot \frac{\mathbf{Rv^2\,(v-b)}}{\mathbf{RTv^3 - 2a(v-b)^2}} \cdot \quad \textbf{(Ans.)}$$

Also, $$K = -\frac{1}{v}\left(\frac{\partial v}{\partial p}\right)_T = -\frac{1}{v}\left[\frac{1}{\frac{2a}{v^3}-\frac{RT}{(v-b)^2}}\right] = \frac{\mathbf{v^2\,(v-b)^2}}{\mathbf{RTv^3 - 2a(v-b)^2}} \cdot \textbf{(Ans.)}$$

Example 6.3. *Prove that the internal energy of an ideal gas is a function of temperature alone.*

Solution. The equation of state for an ideal gas is given by

$$p = \frac{RT}{v}$$

But $$\left(\frac{\partial u}{\partial v}\right)_T = T\left(\frac{\partial p}{\partial T}\right)_v - p \qquad \text{[Eqn. (6.29)]}$$

$$= T\,\frac{R}{v} - p = p - p = 0.$$

Thus, if the temperature remains constant, there is no change in internal energy with volume (and therefore also with pressure). Hence internal energy (u) is a function of temperature (T) alone. **...Proved.**

Example 6.4. *Prove that specific heat at constant volume (c_v) of a Van der Waals' gas is a function of temperature alone.*

Solution. The Van der Waals equation of state is given by,

$$p = \frac{RT}{v-b} - \frac{a}{v^2}$$

or $$\left(\frac{\partial p}{\partial T}\right)_v = \frac{R}{v-b}$$

or $$\left(\frac{\partial^2 p}{\partial T^2}\right)_v = 0$$

Now $$\left(\frac{dc_v}{dv}\right)_T = T\left(\frac{\partial^2 p}{\partial T^2}\right)_v$$

Hence $$\left(\frac{\partial c_v}{\partial v}\right)_T = 0$$

Thus c_v of a Van der Waals gas is independent of volume (and therefore of pressure also). Hence it is a function of temperature alone.

☞ **Example 6.5.** *Determine the following when a gas obeys Van der Waals' equation,*

$$\left(p + \frac{a}{v^2}\right)(v - b) = RT$$

(i) Change in internal energy ; *(ii) Change in enthalpy ;*
(iii) Change in entropy.

Solution. (*i*) **Change in internal energy :**

The change in internal energy is given by

$$du = c_v dT + \left[T\left(\frac{\partial p}{\partial T}\right)_v - p\right] dv$$

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Equating the co-efficients of dT in the two equations of ds, we have

$$\frac{c_v}{T} = \left(\frac{\partial s}{\partial T}\right)_v$$

$$c_v = T\left(\frac{\partial s}{\partial T}\right)_v$$

$$\left(\frac{\partial c_v}{\partial v}\right)_T = T\,\frac{\partial^2 s}{\partial T \partial v}$$

From eqn. (6.20),

$$\left(\frac{\partial s}{\partial v}\right)_T = \left(\frac{\partial p}{\partial T}\right)_v$$

$$\frac{\partial^2 s}{\partial v \partial T} = \left(\frac{\partial^2 p}{\partial T^2}\right)_v$$

$$\left(\frac{\partial c_v}{\partial v}\right)_T = T\left(\frac{\partial^2 p}{\partial T^2}\right)_v$$

Also $$p = \frac{RT}{v} \qquad \text{...(Given)}$$

$$\left(\frac{\partial p}{\partial T}\right)_v = \frac{R}{v}$$

$$\left(\frac{\partial^2 p}{\partial T^2}\right)_v = 0 \qquad \text{or} \qquad \left(\frac{\partial c_v}{\partial v}\right)_T = 0$$

This shows that c_v *is a function of T alone,* or c_v *is independent of pressure.*

Also, $$c_p = T\left(\frac{\partial s}{\partial T}\right)_p$$

$$\left(\frac{\partial c_p}{\partial p}\right)_T = T\,\frac{\partial^2 s}{\partial T \partial p}$$

From eqn. (6.21), $$\left(\frac{\partial s}{\partial p}\right)_T = -\left(\frac{\partial v}{\partial T}\right)_p$$

$$\frac{\partial^2 s}{\partial p \partial T} = -\left(\frac{\partial^2 v}{\partial T^2}\right)_p$$

$$\left(\frac{\partial c_p}{\partial p}\right)_T = -T\left(\frac{\partial^2 v}{\partial T^2}\right)_p$$

Again, $$v = \frac{R}{p} \qquad \text{...(Given)}$$

$$\left(\frac{\partial v}{\partial T}\right)_p = \frac{R}{p}$$

and $$\left(\frac{\partial^2 v}{\partial T^2}\right)_p = 0\ ;\ \left(\frac{\partial c_p}{\partial p}\right)_T = 0$$

This shows that c_p *is a function of T alone,* or c_p *is independent of pressure.*

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Now substituting this in eqn. (*i*), we get

$$ds = c_p \frac{dT}{T} - \left(\frac{\partial s}{\partial T}\right)_p . dp \qquad ...(ii)$$

But $$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_p$$

Substituting this in eqn. (*ii*), we get

$$\boxed{ds = c_p \frac{dT}{T} - \beta v dp}$$ **(Ans.)**

Example 6.11. *Derive the following relations :*

(*i*) $\left(\frac{\partial T}{\partial p}\right)_s = \frac{Tv\beta}{c_p}$ (*ii*) $\left(\frac{\partial T}{\partial v}\right)_s = -\frac{T\beta}{c_v K}$.

where β = *Co-efficient of cubical expansion, and*

K = *Isothermal compressibility.*

Solution. (*i*) Using the Maxwell relation (6.19), we have

$$\left(\frac{\partial T}{\partial p}\right)_s = \left(\frac{\partial v}{\partial s}\right)_p = \left(\frac{\partial v}{\partial T}\right)_p \left(\frac{\partial T}{\partial s}\right)_p$$

Also $$c_p = T\left(\frac{\partial s}{\partial T}\right)_p$$

From eqn. (6.34), $$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_p$$

$$\left(\frac{\partial T}{\partial p}\right)_s = \frac{\beta v T}{c_p}$$

i.e., $$\left(\frac{\partial T}{\partial p}\right)_s = \frac{\mathbf{Tvb}}{\mathbf{c_p}} .$$ **(Ans.)**

(*ii*) Using the Maxwell relation (6.18)

$$\left(\frac{\partial T}{\partial v}\right)_s = -\left(\frac{\partial p}{\partial s}\right)_v = -\left(\frac{\partial p}{\partial T}\right)_v \left(\frac{\partial T}{\partial s}\right)_v$$

Also $$c_v = T\left(\frac{\partial s}{\partial T}\right)_v$$ [Eqn. (6.23)]

$$K = -\frac{1}{v}\left(\frac{\partial v}{\partial p}\right)_T$$ [Eqn. (6.36)]

Then $$\left(\frac{\partial T}{\partial v}\right)_s = -\frac{T}{c_v}\left(\frac{\partial p}{\partial T}\right)_v$$

Also $$\left(\frac{\partial p}{\partial v}\right)_T \left(\frac{\partial v}{\partial T}\right)_p \left(\frac{\partial T}{\partial p}\right)_v = -1$$

i.e., $$\left(\frac{\partial p}{\partial T}\right)_v = -\left(\frac{\partial p}{\partial v}\right)_T \left(\frac{\partial v}{\partial T}\right)_p = -\left(\frac{1}{-vK}\right)\beta v = \frac{\beta}{K}$$

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With the aid of eqn. (ii) show that

$$\left(\frac{\partial u}{\partial p}\right)_T = -T\left(\frac{\partial v}{\partial T}\right)_p - p\left(\frac{\partial v}{\partial p}\right)_T$$

The quantity $c_p\left(\frac{\partial T}{\partial p}\right)_h$ *is known as Joule-Thomson cooling effect. Show that this cooling effect for a gas obeying the equation of state* $(v - b) = \frac{RT}{p} - \frac{C}{T^2}$ *is equal to* $\left(\frac{3C}{T^2}\right) - b$.

Solution. (*i*) **We know that**

$$\left(\frac{\partial h}{\partial p}\right)_T = -\mu c_p \qquad \text{...[Eqn. (6.44)]}$$

Also

$$\mu = \frac{1}{c_p}\left[T\left(\frac{\partial v}{\partial T}\right)_p - v\right] \qquad \text{...[Eqn. (6.46)]}$$

$\therefore$

$$\left(\frac{\partial h}{\partial p}\right)_T = -\left[T\left(\frac{\partial v}{\partial T}\right)_p - v\right] = v - T\left(\frac{\partial v}{\partial T}\right)_p \qquad \textbf{... Proved.}$$

Also

$$\mu = \left(\frac{\partial T}{\partial p}\right)_h$$

$\therefore$

$$\left(\frac{\partial h}{\partial p}\right)_T = -c_p\left(\frac{\partial T}{\partial p}\right)_h .$$

(*ii*) **Let**

$$u = f(T, v)$$

$$du = \left(\frac{\partial u}{\partial T}\right)_v dT + \left(\frac{\partial u}{\partial v}\right)_T dv$$

$$= c_v\, dT + \left(\frac{\partial u}{\partial v}\right)_T dv \qquad \text{...(i)}$$

Also

$$du = Tds - pdv$$

Substituting the value of Tds **[from eqn. (6.24)], we get**

$$du = c_v\, dT + T\left(\frac{\partial p}{\partial T}\right)_v dv - pdv$$

$$= c_v\, dT + \left[T\left(\frac{\partial p}{\partial T}\right)_v - p\right] dv \qquad \text{...(ii)}$$

From (*i*) **and** (*ii*), **we get**

$$\left(\frac{\partial u}{\partial v}\right)_T = T\left(\frac{\partial p}{\partial T}\right)_v - p \qquad \textbf{...Proved.}$$

Also

$$\left(\frac{\partial u}{\partial p}\right)_T = \left(\frac{\partial u}{\partial v}\right)_T \left(\frac{\partial v}{\partial p}\right)_T$$

or

$$\left(\frac{\partial u}{\partial p}\right)_T = \left(\frac{\partial v}{\partial p}\right)_T \left[T\left(\frac{\partial p}{\partial T}\right)_v - p\right]$$

or

$$\left(\frac{\partial u}{\partial p}\right)_T = T\left(\frac{\partial p}{\partial T}\right)_v \left(\frac{\partial v}{\partial p}\right)_T - p\left(\frac{\partial v}{\partial p}\right)_T \qquad \textbf{...Proved.}$$

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Differentiating both sides, we get

$$\frac{1}{2.302p} \cdot \frac{dp}{dT} = \frac{3276.6}{T^2} - \frac{0.652}{2.302\,T}$$

$$\frac{dp}{dT} = 2.302 \times 3276.6 \times \frac{p}{T^2} - 0.652\ \frac{p}{T} \qquad ...(ii)$$

From (i) and (ii), we have

or $$\frac{h_{fg}}{v_g T} = 2.302 \times 3276.6 \times \frac{p}{T^2} - 0.652\ \frac{p}{T} \qquad ...(iii)$$

We know that $$\log_{10} p = 7.0323 - \frac{3276.6}{T} - 0.652 \log_{10} T \qquad ...(given)$$

At $p = 0.1$ bar,

$$\log_{10} (0.1) = 7.0332 - \frac{3276.6}{T} - 0.652 \log_{10} T$$

$$-1 = 7.0323 - \frac{3276.6}{T} - 0.652 \log_{10} T$$

or $$0.652 \log_{10} T = 8.0323 - \frac{3276.6}{T}$$

or $$\log_{10} T = 12.319 - \frac{5025.4}{T}$$

Solving by hit and trial method, we get

$$T = 523 \text{ K}$$

Substituting this value in eqn. (iii), we get

$$\frac{294.54 \times 10^3}{v_g \times 523} = 2.302 \times 3276.6 \times \frac{0.1 \times 10^5}{(523)^2} - 0.652 \times \frac{0.1 \times 10^5}{523}$$

$$\frac{563.17}{v_g} = 275.75 - 12.46$$

i.e., $$\mathbf{v_g = 2.139\ m^3/kg.}\quad \textbf{(Ans.)}$$

HIGHLIGHTS

1. Maxwell relations are given by

$$\left(\frac{\partial T}{\partial v}\right)_s = -\left(\frac{\partial p}{\partial s}\right)_v ; \left(\frac{\partial T}{\partial p}\right)_s = \left(\frac{\partial v}{\partial s}\right)_p$$

$$\left(\frac{\partial p}{\partial T}\right)_v = \left(\frac{\partial s}{\partial v}\right)_T ; \left(\frac{\partial v}{\partial T}\right)_p = -\left(\frac{\partial s}{\partial p}\right)_T .$$

2. The specific heat relations are

$$c_p - c_v = \frac{vT\beta^2}{K} ; c_v = T\left(\frac{\partial s}{\partial T}\right)_v ; c_p = T\left(\frac{\partial s}{\partial T}\right)_p .$$

3. Joule-Thomson co-efficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial p}\right)_h .$$

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If p is the absolute pressure of the gas and V is the volume occupied by the gas, then

$$V \propto \frac{1}{p}$$

or pV = Constant, so long as the temperature is constant ...(7.1)

Fig. 7.1 shows the graphical representation of Boyle's law. The curves are rectangular hyperbolas asymptotic to the p-v axis. Each curve corresponds to a different temperature. For any two points on the curve,

$$\frac{p_1}{p_2} = \frac{V_2}{V_1} \quad ...(7.2)$$

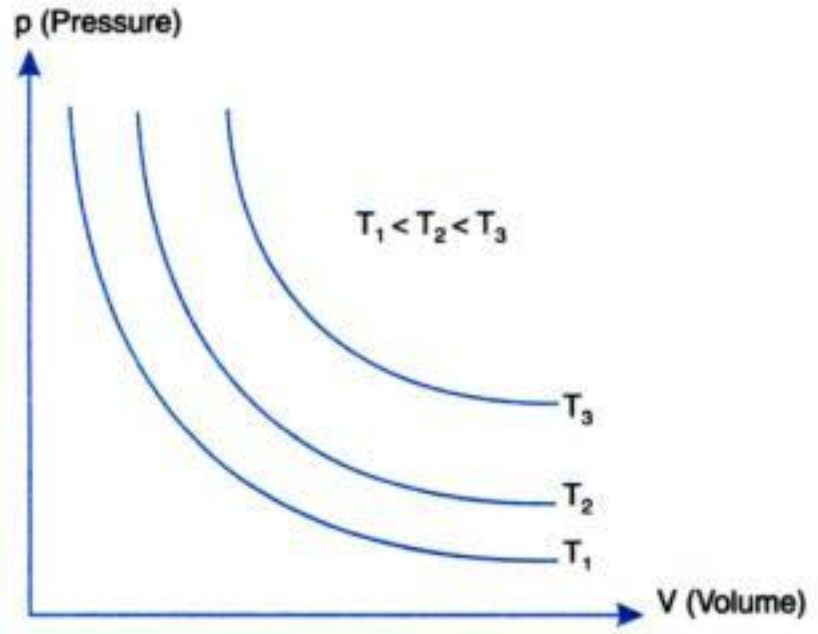

Fig. 7.1 p-V relation of a perfect gas at constant temperature.

Charle's law. It states that *if any gas is heated at constant pressure, its volume changes directly as its absolute temperature.*

In other words, $V \propto T$

or $$\frac{V}{T} = \text{Constant, so long as pressure is constant} \quad ...(7.3)$$

If a gas changes its volume from V_1 to V_2 and absolute temperature from T_1 to T_2 without any change of pressure, then

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \quad ...(7.4)$$

Fig. 7.2 gives the graphical representation of Charle's law.

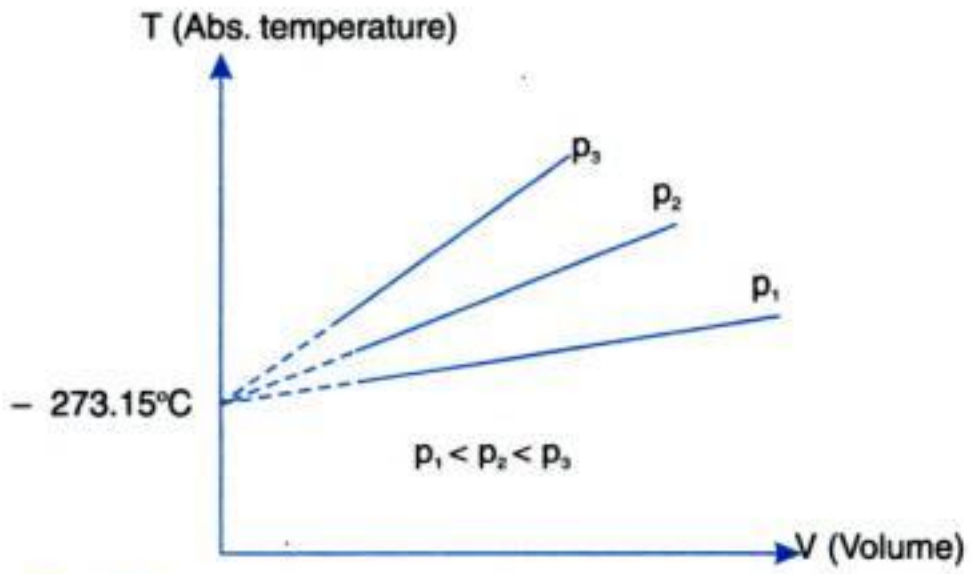

Fig. 7.2 T-v relation of a perfect gas constant pressure.

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The form of curve given by equation (7.33) is shown in Fig. 7.9. The pressure is zero when $v = \frac{3b}{2}$, or infinity. These values of v determine the limiting values of the temperature of inversion, as it is only between these limits that p is positive. Substituting these limits of v in equation (7.34) the limiting values of the temperature of inversion are $\frac{2}{9}\frac{a}{bR}$ and $\frac{2a}{bR}$, or from (7.27), $\frac{3}{4}T_c$ and $\frac{27}{4}T_c$.

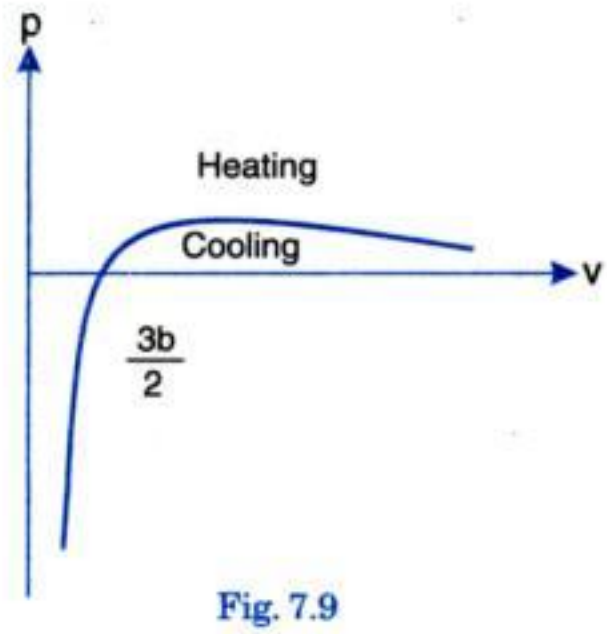

Fig. 7.9

The equation (7.33) being quadratic there are two values of v for a constant value of p at which inversion occurs, as may also be seen by reference to Fig. 7.9. Consequently by equation (7.34) there are two temperatures for a constant value of p at which inversion occurs. As the temperature increases through the lower of these values the change is from a heating to a cooling effect, and as it increases through the higher of these values the change is from a cooling to a heating effect.

The inversion will occur when the maximum value of p is $\frac{a}{3b^2}$, when $v = 3b$. For any value of p less than this there is a cooling effect provided the condition of the substance is represented by a point inside the area enclosed by the curve and the axis of volume, Fig. 7.9, and for any greater value of p there is a heating effect as indicated by equations (7.31) and (7.32) respectively.

Let us take the case of *hydrogen*. In the experiments of Joule and Thomson the pressure used was 4.7 atmospheres. The critical temperature and pressure are 35 K and 15 atmospheres.

From equation (7.33) we can find the values of $\frac{b}{v}$ corresponding to the pressure used by Joule and Thomson, and by substitution in equation (7.34) find the two temperatures at which inversion occurs at this pressure. Equation (7.33) can be written as :

$$p = 27p_c\left[2\frac{b}{v} - 3\left(\frac{b}{v}\right)^2\right]$$

Hence
$$\frac{b}{v} = \frac{2 \pm \sqrt{4 - \frac{12p}{27\,p_c}}}{6} = 0.6608 \text{ or } 0.0058$$

by substitution of the above values for p and p_c.

Writing equation (7.34) in the form

$$T = \frac{27}{4}\,T_c\left(1 - \frac{b}{v}\right)^2,$$

we have by substitution for $\frac{b}{v}$: $T = 233.5$ K or 27.2 K

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$$p_1V_1 = m_1RT_1 \qquad \left(\text{or } m_1 = \frac{p_1V_1}{RT_1}\right)$$

$$p_2V_2 = m_2RT_2 \qquad \left(\text{or } m_2 = \frac{p_2V_2}{RT_2}\right)$$

Mass of air removed during the process = $(m_1 - m_2)$ kg

$$(m_1 - m_2) = \frac{p_1V_1}{RT_1} - \frac{p_2V_2}{RT_2}$$

$$= \frac{1}{R}\left(\frac{p_1V_1}{T_1} - \frac{p_2V_2}{T_2}\right) = \frac{1}{287}\left[\frac{(1\times10^5)\times40}{298} - \frac{(0.4\times10^5)\times40}{278}\right]$$

$$= \mathbf{26.71\ kg.} \quad \textbf{(Ans.)}$$

Volume of this mass of gas at 1 bar and 25°C is given by

$$V = \frac{mRT}{p} = \frac{26.71\times287\times298}{1\times10^5} = \mathbf{22.84\ m^3.} \quad \textbf{(Ans.)}$$

Example 7.2. *A steel flask of 0.04 m^3 capacity is to be used to store nitrogen at 120 bar, 20°C. The flask is to be protected against excessive pressure by a fusible plug which will melt and allow the gas to escape if the temperature rises too high.*

(i) How many kg of nitrogen will the flask hold at the designed conditions ?

(ii) At what temperature must the fusible plug melt in order to limit the pressure of a full flask to a maximum of 150 bar ?

Solution. Capacity of the steel flask, $V = 0.04\ m^3$

Pressure, $p = 120$ bar

Temperature, $T = 20 + 273 = 293$ K

(*i*) kg of nitrogen the flask can hold :

Now, R for nitrogen (molecular weight, $M = 28$)

$$= \frac{R_0}{M} = \frac{8314}{28} = 296.9\ \text{J/kg K}$$

Assuming nitrogen to be a perfect gas, we get

Mass of nitrogen in the flask at designed condition

$$= m = \frac{pV}{RT} = \frac{120\times10^5\times0.04}{296.9\times293} = \mathbf{5.51\ kg.} \quad \textbf{(Ans.)}$$

(*ii*) Temperature at which fusible plug should melt, t :

When the fusible plug is about to melt

$$p = 150\ \text{bar} ;\ V = 0.04\ m^3 ;\ m = 5.51\ \text{kg}$$

Therefore, temperature t at which fusible plug must melt is given by

$$T = \frac{pV}{mR} = \frac{150\times10^5\times0.04}{5.51\times296.9} = 366.7\ \text{K}$$

$$\therefore \qquad \mathbf{t = 366.7 - 273 = 93.7°C.} \quad \textbf{(Ans.)}$$

Example 7.3. *A balloon of spherical shape 6 m in diameter is filled with hydrogen gas at a pressure of 1 bar abs. and 20°C. At a later time, the pressure of gas is 94 per cent of its original pressure at the same temperature :*

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But $$\gamma = \frac{c_p}{c_v} = 1.4 \text{ (given)} \qquad ...(i)$$

and $$c_p - c_v = R \text{ (= 0.287 kJ/kg K for air)} \qquad ...(ii)$$

Solving for c_v between (i) and (ii),

$$c_v = 0.717 \text{ kJ/kg K}$$

$$\therefore \quad \Delta s = 0.717 \log_e \left(\frac{290.8}{373}\right) + 0.287 \log_e \left(\frac{0.4637}{0.1338}\right)$$

$$= -0.1785 + 0.3567 = 0.1782 \text{ kJ/kg K}$$

i.e., Increase in entropy, Δs = **0.1782 kJ/kg K.** **(Ans.)**

(*ii*) **Work done and heat transfer :**

The work done in a polytropic process is given by,

$$W = \frac{p_1 v_1 - p_2 v_2}{n-1} = \frac{R(T_1 - T_2)}{n-1}$$

$$= \frac{0.287\,(373 - 290.8)}{(1.2 - 1)} = 117.96 \text{ kJ/kg}$$

i.e., **Work done = 117.96 kJ/kg.** **(Ans.)**

Heat transfer, $Q = \Delta u + W$

where $\Delta u = c_v(T_2 - T_1)$

$$= 0.717\,(290.8 - 373) = -58.94 \text{ kJ/kg}$$

$$\therefore \quad Q = -58.94 + 117.96 = 59.02 \text{ kJ/kg}$$

Hence **heat transfer = 59.02 kJ/kg.** **(Ans.)**

(*b*) (*i*) Though the process is assumed now to be irreversible and adiabatic, the end states are given to be the same as in (*a*). Therefore, **all the properties at the end of the process are the same as in (a).** **(Ans.)**

(*ii*) As the process is adiabatic, Q (heat transfer) = **0.** **(Ans.)**

$$\Delta u = \Delta u \text{ in } (a)$$

Applying first law for this process

$$Q = \Delta u + W$$

$$0 = \Delta u + W$$

or $$W = -\Delta u$$

$$= -(-58.94) = 58.94$$

$$\therefore \quad \textbf{Work done = 58.94 kJ/kg.} \quad \textbf{(Ans.)}$$

Example 7.6. *Two spheres each 2.5 m in diameter are connected to each other by a pipe with a valve as shown in Fig. 7.11. One sphere contains 16 kg of air and other 8 kg of air when the valve is closed. The temperature of air in both sphere is 25°C. The valve is opened and the whole system is allowed to come to equilibrium conditions. Assuming there is no loss or gain of energy, determine the pressure in the spheres when the system attains equilibrium.*

Neglect the volume of the pipe.

Solution. Volume of each sphere $= \frac{4}{3}\pi R^3 = \frac{4}{3}\pi \times \left(\frac{2.5}{2}\right)^3 = 8.18 \text{ m}^3$

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$$2.857 = \left(\frac{8}{p_3}\right)^{0.2857}$$

$$p_3 = \frac{8}{(2.857)^{1/0.2857}} = \frac{8}{(2.857)^{3.5}} = 0.2 \text{ bar}$$

Now, $p_3V_3 = mRT_3$

$\therefore$ $0.2 \times 10^5 \times V_3 = 0.1764 \times 287 \times 553$

$\therefore$ $V_3 = \dfrac{0.1764 \times 287 \times 553}{0.2 \times 10^5} = 1.399 \text{ m}^3$

(*i*) The heat received in the cycle :

Applying first law to the constant pressure *process 1-2*,

$$Q = \Delta U + W$$

$$W = \int_1^2 pdV \qquad \text{(as the process is reversible)}$$

$$= p(V_2 - V_1)$$

$$= 8 \times 10^5 (0.1 - 0.035)$$

$$= 52000 \text{ J or } 52 \text{ kJ (work done by air)}$$

$\therefore$ $Q = m \times c_v(T_2 - T_1) + 52$

$= 0.1764 \times 0.71(1580 - 553) + 52 = 180.6$ kJ

i.e., Heat received = 180.6 kJ

Applying first law to reversible polytropic *process 2-3*

$$Q = \Delta U + W$$

But $$W = \frac{p_2V_2 - p_3V_3}{n-1} = \frac{mR(T_2 - T_3)}{n-1}$$

$$= \frac{0.1764 \times 0.287\,(1580 - 553)}{1.4 - 1} = 129.98 \text{ kJ (work done } \textbf{by air)}$$

$\therefore$ $Q = mc_v(T_3 - T_2) + 129.98$

$= 0.1764 \times 0.71\,(553 - 1580) + 129.98$

$= -\,128.6 + 129.98 = 1.354$ kJ (heat received)

$\therefore$ **Total heat received in the cycle** = 180.6 + 1.354 = **181.954 kJ.** (Ans.)

(*ii*) The heat rejected in the cycle :

Applying first law to reversible isothermal *process 3-1*,

$$Q = \Delta U + W$$

$$W = p_3V_3 \log_e \left(\frac{V_1}{V_3}\right)$$

$$= 0.2 \times 10^5 \times 1.399 \times \log_e \left(\frac{0.035}{1.399}\right) \times 10^{-3}$$

$$= -\,103.19 \text{ kJ (work done } \textbf{on} \text{ the air)}$$

$\therefore$ $Q = mc_v(T_1 - T_3) + W$

$= 0 - 103.19 = -\,103.19$ kJ $\qquad (\because \quad T_1 = T_3)$

Hence heat rejected in the cycle = 103.19 kJ. (Ans.)

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Example 7.13. *Calculate the density of N_2 at 260 bar and 15°C by using the compressibility chart.*

Solution. Pressure, $p = 260$ bar

Temperature, $T = 15 + 273 = 288$ K

Density, ρ **= ?**

For N_2 (from Table 7.3) : $p_c = 33.94$ bar

$T_c = 126.2$ K

Now, $$p_r = \frac{p}{p_c} = \frac{260}{33.94} = 7.6$$

and $$T_r = \frac{T}{T_c} = \frac{288}{126.2} = 2.28$$

From the compressibility chart for $p_r = 7.6$ and $T_r = 2.28$, $Z \simeq 1.08$

Also $$Z = \frac{pv}{RT} = \frac{p}{\rho RT}$$, where ρ stands for density

or $$\rho = \frac{p}{ZRT} = \frac{260 \times 10^5}{1.08 \times \frac{8314}{28} \times 288} = \mathbf{281.5\ kg/m^3.}$$ (Ans.)

Example 7.14. *What should be the temperature of 1.3 kg of CO_2 gas in a container at a pressure of 200 bar to behave as an ideal ?*

Solution. Pressure, $p = 200$ bar

Temperature, **T = ?**

For CO_2 (from Table 7.3) $p_c = 73.86$ bar

$T_c = 304.2$ K

As the gas behaves like an ideal gas, $Z = 1$

$$p_r = \frac{p}{p_c} = \frac{200}{73.86} = 2.7$$

From compressibility chart for $Z = 1$, $p_r = 2.7$

$T_r = 2.48$

$\therefore$ $\mathbf{T = T_r \times T_c = 2.48 \times 304.2 = 754.4\ K.}$ (Ans.)

Example 7.15. *A spherical shaped balloon of 12 m diameter contains H_2 at 30°C and 1.21 bar. Find the mass of H_2 in the balloon using real gas equation.*

Solution. Diameter of spherical balloon = 12 m

$\therefore$ Volume, $V = 4/3\ \pi \times (6)^3 = 904.78\ m^3$

Temperature, $T = 30 + 273 = 303$ K

Pressure, $p = 1.21$ bar

Mass of H_2 in the balloon, m :

For H_2 (from Table 7.3) $p_c = 12.97$ bar

$T_c = 33.3$ K

Now, $$p_r = \frac{p}{p_c} = \frac{1.21}{12.97} = 0.093$$

$$T_r = \frac{T}{T_c} = \frac{303}{33.3} = 9.1$$

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UNSOLVED EXAMPLES

IDEAL GASES

1. A vessel of 0.03 m^3 capacity contains gas at 3.5 bar pressure and 35°C temperature. Determine the mass of the gas in the vessel. If the pressure of this gas is increased to 10.5 bar while the volume remains constant, what will be the temperature of the gas ?
   For the gas take R = 290 J/kg K. [Ans. 0.118 kg, 650°C]
2. The tyre of an automobile contains a certain volume of air at a gauge pressure of 2 bar and 20°C. The barometer reads 75 cm of Hg. The temperature of air in the tyre rises to 80°C due to running of automobile for two hours. Find the new pressure in the tyre.
   Assume that the air is an ideal gas and tyre does not stretch due to heating. [Ans. 2.62 bar]
3. A tank made of metal is designed to bear an internal gauge pressure of 7 bar. The tank is filled with a gas at a pressure of 5.5 bar abs., and 15°C. The temperature in the tank may reach to 50°C when the tank stands in the sun.
   (*i*) If the tank does not expand with temperature, will the design pressure be exceeded on a day when atmospheric pressure is 1 bar and air in the tank reaches 50°C when exposed to hot sun.
   (*ii*) What temperature would have to be reached to raise the air pressure to the design limit ?
   [Ans. (*i*) 6.16 bar, (*ii*) 147°C]
4. A vessel of spherical shape is 1.5 m in diameter and contains air at 40°C. It is evacuated till the vacuum inside the vessel is 735 mm of mercury. Determine :
   (*i*) The mass of air pumped out ;
   (*ii*) If the tank is then cooled to 10°C what is the pressure in the tank ?
   The barometer reads 760 mm of mercury. Assume that during evacuation, there is no change in temperature of air. [Ans. (*i*) 1.91 kg, (*ii*) 3 kPa]
5. A balloon of spherical shape is 8 m in diameter and is filled with hydrogen at a pressure of 1 bar abs. and 15°C. At a later time, the pressure of gas is 95 per cent of its original pressure at the same temperature.
   (*i*) What mass of original gas must have escaped if the dimensions of the balloon are not changed ?
   (*ii*) Find the amount of heat to be removed to cause the same drop in pressure at constant volume.
   [Ans. (*i*) 5 per cent, (*ii*) 3.26 MJ]
6. Find the molecular weight and gas constant for the gas whose specific heats are as follows :
   c_p = 1.967 kJ/kg K, c_v = 1.507 kJ/kg K. [Ans. 180.461 kJ/kg K]
7. A constant volume chamber of 0.3 m^3 capacity contains 1 kg of air at 20°C. Heat is transferred to the air until its temperature is 200°C. Find :
   (*i*) Heat transferred ;
   (*ii*) Change in entropy and enthalpy. [Ans. (*i*) 128.09 kJ, (*ii*) 0.339 kJ/kg K, 180.8 kJ]
8. 1 kg of air at 20°C occupying a volume of 0.3 m^3 undergoes a reversible constant pressure process. Heat is transferred to the air until its temperature is 200°C. Determine :
   (*i*) The work and heat transferred.
   (*ii*) The change in internal energy, enthalpy and entropy.
   [Ans. (*i*) 51.5 kJ, 180.8 kJ ; (*ii*) 128.09 kJ, 180.8 kJ, 0.479 kJ/kg K]
9. A balloon of spherical shape, 10 m in diameter is filled with hydrogen at 20°C and atmospheric pressure. The surrounding air is at 15°C and barometer reads 75 mm of Hg. Determine the load lifting capacity of the balloon. [Ans. 587.2 kg]
10. Air expands in a cylinder in a reversible adiabatic process from 13.73 bar to 1.96 bar. If the final temperature is to be 27°C, what would be the initial temperature ?
    Also calculate the change in specific enthalpy, heat and work transfers per kg of air.
    [Ans. 524 K, 224.79 kJ/kg, zero, 160.88 kJ/kg]
11. 1 kg mole of N_2 is contained in a vessel of volume 2.5 m^3 at 100°C.
    (*i*) Find the mass, the pressure and the specific volume of the gas.
    (*ii*) If the ratio of the specific heats is 1.4, evaluate the values of c_p and c_v.

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Mean molecular weight of air = 28.96

Gas constant $R = 0.287$ kJ/kg K

For *approximate* calculations the air is said to be composed of oxygen and 'atmospheric nitrogen'.

Constituent	*Molecular weight*	*Volumetric analysis %*	*Gravimetric analysis %*
Oxygen	32	21	23.3
Atmospheric nitrogen	28	79	76.7
Nitrogen/Oxygen	—	3.76 : 1	3.29 : 1

Note. Volumetric analysis is the analysis by **volume** ; gravimetric analysis is the analysis by **weight or mass**.

8.3. VOLUMETRIC ANALYSIS OF A GAS MIXTURE

It is usual practice to quote the analysis of a mixture by volume as this is the most convenient for practical determinations. In article 10.15, the analysis of exhaust or flue gases by means of the Orsat apparatus is described. The volume of the gas sample is measured at atmospheric pressure, and the temperature is held constant by means of a water jacket round the gas sample. The constituents are absorbed chemically one by one, and the remainder of the sample is measured after each absorption ; the difference in volume gives the partial volume occupied by the constituent in the mixture.

Let us consider a volume V of a gaseous mixture at a temperature T, consisting of three constituents A, B and C [Fig. 8.2 (a)]. Let us further assume that each of the constituents is compressed to a pressure p equal to the total pressure of the mixture, and let the *temperature remain constant*. The partial volumes then occupied by the constituents will be V_A, V_B and V_C.

$$m = m_A + m_B + m_C = \Sigma m_i$$

$$p = p_A + p_B + p_C = \Sigma p_i$$

$$n = n_A + n_B + n_C = \Sigma n_i$$

(a)

V_A	V_B	V_C
p	p	p
m_A	m_B	m_C
n_A	n_B	n_C

(b)

Fig. 8.2

Now using the eqn. $pV = mRT$, we get

$$m_A = \frac{p_A V}{R_A T}$$ ... referring to Fig. 8.2 (a)

and

$$m_A = \frac{pV_A}{R_A T}$$ ... referring to Fig. 8.2 (b)

Now equating the two values for m_A, we have

$$\frac{p_A V}{R_A T} = \frac{pV_A}{R_A T} \quad i.e., \quad p_A V = pV_A$$

or

$$V_A = \frac{p_A}{p} V$$

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8.5. SPECIFIC HEATS OF A GAS MIXTURE

As per Gibbs-Dalton law, the internal energy of a mixture of gases is given by

$$mu = \Sigma\, m_i u_i \qquad ...(i) \quad \text{[from eqn. (8.5)]}$$

Also $$u = c_v T \qquad ...(ii) \quad \text{[from perfect gas equation]}$$

Now from (i) and (ii), we have

$$mc_v T = \Sigma\, m_i c_{vi} T$$

$$\therefore \quad mc_v = \Sigma\, m_i c_{vi}$$

or
$$c_v = \sum \frac{m_i}{m}\, c_{vi} \qquad ...(8.19)$$

Similarly from equations, $mh = \Sigma\, m_i h_i$

and $h = c_p T$, we get

$$mc_p T = \Sigma\, m_i c_{pi} T$$

$$\therefore \quad m_{cp} = \Sigma\, m_i c_{pi}$$

or
$$c_p = \sum \frac{m_i}{m}\, c_{pi} \qquad ...(8.20)$$

From eqns. (8.18) and (8.19),

$$c_p - c_v = \sum \frac{m_i}{m}\, c_{pi} - \sum \frac{m_i}{m}\, c_{vi} = \sum \frac{m_i}{m}\, (c_{pi} - c_{vi})$$

Also $c_{pi} - c_{vi} = R_i$, therefore,

$$c_p - c_v = \sum \frac{m_i}{m}\, R_i$$

Also from eqn. (8.12), $R = \sum \frac{m_i}{m}\, R_i$, therefore for the mixture

$$c_p - c_v = R$$

The following equations can be applied to a mixture of gases

$$\gamma = \frac{c_p}{c_v}; \quad c_v = \frac{R}{\gamma - 1}; \quad c_p = \frac{\gamma R}{\gamma - 1}$$

It should be borne in mind that γ *must be determined from the eqn.* $\gamma = \frac{c_p}{c_v}$; *there is no weighted mean expression as there is for* R, c_v *and* c_p.

In problems on mixtures it is often convenient to work in moles and the specific heats can be expressed in terms of the mole. These are known as *molar heats*, and are denoted by C_p and C_v.

Molar heats are defined as follows :

$$C_p = Mc_p \quad \text{and} \quad C_v = Mc_v \qquad ...(8.21)$$

But $$c_p - c_v = R$$

$$\therefore \quad C_p - C_v = Mc_p - Mc_v = M(c_p - c_v) = MR$$

But $$MR = R_0$$

$$\therefore \quad C_p - C_v = R_0 \qquad ...(8.22)$$

Also
$$U = mc_v T = \frac{mMc_v T}{M}$$

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Also, the characteristic gas equation is given by

$$pV = mRT \qquad ...(ii)$$

$$\therefore \quad pV = \frac{\dot{m}R_0T}{M} \qquad ...(iii)$$

i.e.,

$$p = \frac{mR_0T}{MV} \qquad ...(iv)$$

Hence, for a constituent,

$$p_i = \frac{m_iR_0T}{M_iV} \qquad ...(v)$$

Substituting the values, we get the partial pressures as follows :

(*i*) **Partial pressures :**

For O_2, $p_{O_2} = \dfrac{0.233 \times (8.314 \times 10^3) \times 293}{32 \times 0.35 \times 10^5}$ = **0.5068 bar.** **(Ans.)**

For N_2, $p_{N_2} = \dfrac{0.767 \times (8.314 \times 10^3) \times 293}{28 \times 0.35 \times 10^5}$ = **1.9065 bar.** **(Ans.)**

For CO, $p_{CO} = \dfrac{0.40 \times (8.314 \times 10^3) \times 293}{28 \times 0.35 \times 10^5}$ = **0.9943 bar.** **(Ans.)**

(*ii*) **Total pressure in the vessel, p :**

$$p = \Sigma p_i = p_{O_2} + p_{N_2} + p_{CO}$$

$$= 0.5068 + 1.9065 + 0.9943 = \textbf{3.4076 bar.} \quad \textbf{(Ans.)}$$

Example 8.2. *The gravimetric analysis of air and other data are as follows :*

Constituent	*Percentage*	*Molecular weight*
Oxygen	*23.14*	*32*
Nitrogen	*75.53*	*28*
Argon	*1.28*	*40*
Carbon dioxide	*0.05*	*44*

Calculate : (i) Gas constant for air ; (ii) Apparent molecular weight.

Solution. Using the relation, $R = \dfrac{R_0}{M}$ ...(*i*)

$$R_{O_2} = \frac{8.314}{32} = 0.2598 \text{ kJ/kg K}$$

$$R_{N_2} = \frac{8.314}{28} = 0.2969 \text{ kJ/kg K}$$

$$R_{Ar} = \frac{8.314}{40} = 0.2078 \text{ kJ/kg K}$$

$$R_{CO_2} = \frac{8.314}{44} = 0.1889 \text{ kJ/kg K}$$

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Solution. Composition of mixture by volume : H_2 = 78%, CO = 22%

Final composition desired : H_2 = 52%, CO = 48%

Since the pressure and temperature remain constant, then the number of moles in the vessel remain the same throughout.

∴ Moles of mixture removed = Moles of CO added.

Let x kg of mixture be removed and y kg of CO be added.

For the mixture, $$M = \sum \frac{V_i}{V} M_i$$

∴ $$M = 0.78 \times 2 + 0.22 \times 28 = 7.72$$

Also from equation, $n = \frac{m}{M}$, we have

Moles of mixture removed = $\frac{x}{7.72}$ = moles of CO added = $\frac{y}{28}$

From equation, $\frac{V_i}{V} = \frac{n_i}{n}$, we have

Moles of H_2 in the mixture removed

$$= 0.78 \times \frac{x}{7.72} = 0.101\, x$$

and Moles of H_2 initially = $0.78 \times 1 = 0.78$

Hence, Moles of H_2 remaining in vessel = $0.78 - 0.101\, x$

But 1 mole of the new mixture is 52% H_2 and 48% CO, therefore

$$0.78 - 0.101\, x = 0.52$$

∴ $$0.101\, x = 0.26 \quad \text{or} \quad x = 2.57$$

i.e., **Mass of mixture removed = 2.57 kg.** **(Ans.)**

Also since $$\frac{x}{7.72} = \frac{y}{28}$$

∴ $$y = \frac{28}{7.72} \times x = \frac{28}{7.72} \times 2.57 = 9.32 \text{ kg}$$

i.e., **Mass of CO added = 9.32 kg.** **(Ans).**

☞**Example 8.7.** *In an engine cylinder a gas has a volumetric analysis of 13% CO_2, 12.5% O_2, and 74.5% N_2. The temperature at the beginning of expansion is 950°C and the gas mixture expands reversibly through a volume ratio of 8 : 1, according to the law $pv^{1.2}$ = constant. Calculate per kg of gas :*

(i) The work done ;

(ii) The heat flow ;

(iii) Change of entropy per kg of mixture.

The values of c_p for the constituents CO_2, O_2 and N_2 are 1.235 kJ/kg K, 1.088 kJ/kg K and 1.172 kJ/kg K respectively.

Solution. From equation $m_i = n_i M_i$, the conversion from volume fraction to mass fraction is as follows :

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$$x_{CO_2} = \frac{\frac{6}{44}}{\frac{4}{28}+\frac{6}{44}} = \frac{0.1364}{0.1428+0.1364} = \mathbf{0.488.}$$ **(Ans.)**

(*ii*) **The equivalent molecular weight of the mixture, M :**

$$M = 0.511 \times 28 + 0.488 \times 44 = \mathbf{35.78\ kg/kg\ mole.}$$ **(Ans.)**

(*iii*) **The equivalent gas constant of the mixture, R_{mix} :**

Total mass, $m = m_{N_2} + m_{CO_2} = 4 + 6 = 10$ kg

$$\therefore \quad R_{mix} = \frac{m_{N_2} R_{N_2} + m_{CO_2} R_{CO_2}}{m}$$

$$= \frac{4 \times \left(\frac{8.314}{28}\right) + 6 \times \left(\frac{8.314}{44}\right)}{10} \qquad \left[\because\ R = \frac{R_0}{M}\right]$$

$$= \mathbf{0.232\ kJ/kg\ K.}$$ **(Ans.)**

(*iv*) **The partial pressures and partial volumes :**

$$P_{N_2} = x_{N_2} \times p = 0.511 \times 4 = \mathbf{2.044\ bar.}$$ **(Ans.)**

$$P_{CO_2} = x_{CO_2} \times p = 0.488 \times 4 = \mathbf{1.952\ bar.}$$ **(Ans.)**

$$V_{N_2} = \frac{m_{N_2} R_{N_2} T}{p} = \frac{4 \times \frac{8.314}{28} \times 293 \times 10^3}{4 \times 10^5} = \mathbf{0.87\ m^3.}$$ **(Ans.)**

$$V_{CO_2} = \frac{m_{CO_2} R_{CO_2} T}{p} = \frac{6 \times \frac{8.314}{44} \times 293 \times 10^3}{4 \times 10^5} = \mathbf{0.83\ m^3.}$$ **(Ans.)**

(*v*) **The volume and density of the mixture :**

Total volume of the mixture,

$$V = \frac{mRT}{p} = \frac{m_{N_2} R_{N_2} T}{p_{N_2}} = \frac{m_{CO_2} R_{CO_2} T}{p_{CO_2}}$$

$$\therefore \quad V = \frac{10 \times 0.232 \times 293 \times 10^3}{4 \times 10^5} = \mathbf{1.699\ m^3.}$$ **(Ans.)**

Density of the mixture,

$$\rho_{mix} = \rho_{N_2} + \rho_{CO_2} = \frac{m}{V} = \frac{10}{1.699} = \mathbf{5.88\ kg/m^3.}$$ **(Ans.)**

(*vi*) **c_p and c_v of the mixture :**

$$c_{p_{N_2}} - c_{v_{N_2}} = R_{N_2}$$

$$\therefore \quad c_{v_{N_2}} = \frac{R_{N_2}}{\gamma - 1} = \frac{8.314}{28(1.4-1)} = 0.742\ \text{kJ/kg K.}$$

and $$c_{p_{N_2}} = 1.4 \times 0.742 = 1.039\ \text{kJ/kg K}$$

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$$16 \times 10^5 \times V_A = 0.6 \times 8.314 \times 10^3 \times (55 + 273)$$

$$\therefore \quad V_A = 1.022 \text{ m}^3$$

The mass of gas in vessel A

$$m_A = n_A M_A = 0.6 \times 28 = 16.8 \text{ kg}$$

Characteristic gas constant R of nitrogen

$$R = \frac{8.314}{28} = 0.297 \text{ kJ/kg K}$$

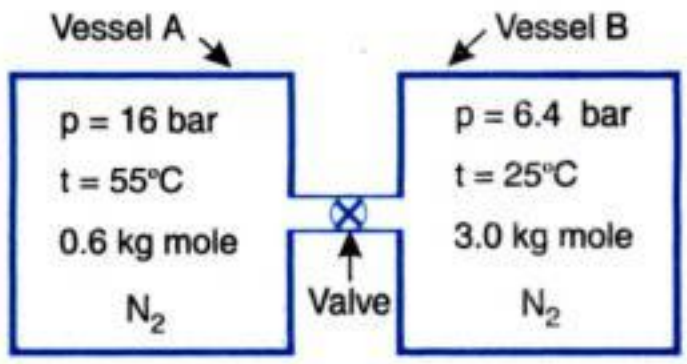

Fig. 8.9

Vessel B :

$$p_B V_B = m_B R T_B$$

$$6.4 \times 10^5 \times V_B = 3.0 \times 0.297 \times 10^3 \times (25 + 273)$$

$$\therefore \quad V_B = 0.415 \text{ m}^3$$

Total volume of A and B

$$V = V_A + V_B = 1.022 + 0.415 = 1.437 \text{ m}^3$$

Total mass of gas

$$m = m_A + m_B = 16.8 + 3 = 19.8 \text{ kg}$$

Final temperature after mixing

$$T = 30 + 273 = 303 \text{ K}$$

(a) (i) **Final equilibrium pressure, p :**

$$pV = mRT$$

or $$p \times 1.437 = 19.8 \times 0.297 \times 303 \times 10^3 = 12.4 \times 10^5 \text{ N/m}^2$$

i.e., **p = 12.4 bar. (Ans.)**

Also $$c_v = \frac{R}{\gamma - 1} = \frac{0.297}{1.4 - 1} = 0.743 \text{ kJ/kg K.}$$

(ii) **Amount of heat transferred, Q :**

As there is no work transfer, the amount of heat transfer,

Q = change of internal energy

$= U_2 - U_1$

Measuring the internal energy above the datum of absolute zero (at $T = 0$ K, $u = 0$ kJ/kg).

Internal energy U_1 (before mixing)

$$= m_A c_v T_A + m_B c_v T_B$$

$$= 16.8 \times 0.743 \times 328 + 3.0 \times 0.743 \times 298 = 4758.5 \text{ kJ}$$

Final internal energy U_2 (after mixing)

$$= mc_v T = 19.8 \times 0.743 \times 303 = 4457.5 \text{ kJ}$$

$\therefore$ **Q** $= U_2 - U_1 = 4457.5 - 4758.5 = -$ **301 kJ. (Ans.)**

(b) **If the vessel were insulated :**

(i) **Final temperature, t_2 :**

If the vessel were insulated

$$Q = U_2 - U_1 = 0$$

i.e., $$U_1 = U_2$$

$$m_A c_v T_A + m_B c_v T_B = mc_v T$$

$$\therefore \quad T = \frac{m_A c_v T_A + m_B c_v T_B}{mc_v} = \frac{m_A T_A + m_B T_B}{m}$$

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or $$p_{CO_2} + 12 = 18$$

$$p_{CO_2} = 6 \text{ bar}$$

Now $$p_{CO_2} = \frac{m_{CO_2} R_{CO_2} T_{CO_2}}{V}$$

$$\therefore \quad m_{CO_2} = \frac{p_{CO_2}}{R_{CO_2} T_{CO_2}} = \frac{6 \times 10^5 \times 0.6}{\left(\frac{8.314}{44}\right) \times 298 \times 10^3} = 6.39 \text{ kg}$$

i.e., **Mass of CO_2 = 6.39 kg.** (Ans.)

Example 8.17. *A vessel of 6 m^3 capacity contains two gases A and B in proportion of 45 per cent and 55 per cent respectively at 30°C. If the value of R for the gases is 0.288 kJ/kg K and 0.295 kJ/kg K and if the total weight of the mixture is 2 kg, calculate :*

(i) The partial pressure ; *(ii) The total pressure ;*

(iii) The mean value of R for the mixture.

Solution. Capacity of the vessel, $V = 6 \text{ m}^3$

%age content of gas $A = 45\%$

%age content of gas $B = 55\%$

R for gas A, $R_A = 0.288$ kJ/kg K

R for gas B, $R_B = 0.295$ kJ/kg K

Total weight of the mixture = 2 kg

Temperature, $T = 30 + 273 = 303$ K.

(i) **The partial pressures, p_A, p_B :**

$$m_A = 2 \times 0.45 = 0.9 \text{ kg}$$

$$m_B = 2 \times 0.55 = 1.1 \text{ kg}$$

Now, $$p_A V = m_A R_A T_A$$

$$\therefore \quad \mathbf{p_A} = \frac{m_A R_A T_A}{V} = \frac{0.9 \times 0.288 \times 303 \times 10^3}{6 \times 10^5} = \mathbf{0.13 \text{ bar.}} \text{ (Ans.)}$$

and $$\mathbf{p_B} = \frac{m_B R_B T_B}{V} = \frac{1.1 \times 0.295 \times 303 \times 10^3}{6 \times 10^5} = \mathbf{0.164 \text{ bar.}} \text{ (Ans.)}$$

(ii) **The total pressure, p :**

$$p = p_A + p_B = 0.13 + 0.164 = \mathbf{0.294 \text{ bar.}} \text{ (Ans.)}$$

(iii) **The mean value of R for the mixture, R_m :**

$$R_m = \frac{m_A R_A + m_B R_B}{m_A + m_B}$$

$$= \frac{0.9 \times 0.288 + 1.1 \times 0.295}{(0.9 + 1.1)} = 0.292 \text{ kJ/kg K}$$

i.e., **Mean value of R for the mixture = 0.292 kJ/kg K.** (Ans.)

Example 8.18. *The pressure and temperature of mixture of 4 kg of O_2 and 6 kg of N_2 are 4 bar and 27°C respectively. For the mixture determine the following :*

(i) The mole fraction of each component ; *(ii) The average molecular weight ;*

(iii) The specific gas constant ; *(iv) The volume and density ;*

(v) The partial pressures and partial volumes.

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Now entropy change in the mixing process is given by

$$\Delta S = m_{O_2}\,\Delta_{s_{(O_2)}} + m_{N_2}\,\Delta_{s_{(N_2)}}$$

where $\Delta_s = c_v \log_e \frac{T_2}{T_1} + R \log_e \frac{V_2}{V_1}$

Now to find initial and final volumes of O_2 and N_2 using the relation,

$$V = \frac{nR_0T}{p}$$

$\therefore$ The initial volume of $O_2 = \frac{1 \times 8.314 \times 293 \times 10^3}{2.5 \times 10^5} = 9.74\ m^3$

and Initial volume of $N_2 = \frac{2 \times 8.314 \times 301 \times 10^3}{1.5 \times 10^5} = 33.4\ m^3$

Final volume of the mixture = 9.74 + 33.4 = 43.14 m^3

Thus
$$\Delta S = \left[(1 \times 32)\left\{0.39 \times \log_e \frac{298.4}{293} + \frac{8.314}{32} \log_e \frac{43.14}{9.74}\right\}\right]$$
$$+ \left[(2 \times 28)\left\{0.446 \times \log_e \frac{298.4}{301} + \frac{8.314}{28} \log_e \frac{43.14}{33.4}\right\}\right]$$
$$= 12.60 + 4.04 = 16.64\ kJ$$

i.e., **Entropy change in the mixing process = 16.64 kJ.** (Ans.)

☞**Example 8.21.** *A tank of capacity 0.45 m^3 is insulated and is divided into two sections through a partition. One section initially contains H_2 at 3 bar and 130°C and has a volume of 0.3 m^3 and the other section initially holds N_2 at 6 bar and 30°C. The gases are then allowed to mix after removing the adiabatic partition. Determine :*

(i) The temperature of the equilibrium mixture ;

(ii) The pressure of the mixture ;

(iii) The change in entropy for each component and total value.

Assume : $c_{v_{(N_2)}} = 0.744\ kJ/kg\ K$, $c_{v_{(H_2)}} = 10.352\ kJ$

$c_{p_{(N_2)}} = 1.041\ kJ/kg\ K$, $c_{p_{(H_2)}} = 14.476\ kJ/kg\ K.$

Solution. Total capacity of the tank, $V = 0.45\ m^3$

$V_{H_2} = 0.3\ m^3$; $T_{H_2} = 130 + 273 = 403\ K$

$p_{H_2} = 3$ bar ; $V_{N_2} = 0.15\ m^3$ (*i.e.,* $0.45 - 0.3 = 0.15\ m^3$)

$p_{N_2} = 6$ bar ; $T_{N_2} = 30 + 273 = 303$ K.

(*i*) Temperature of equilibrium mixture, T_2 :

Now $p_{H_2} V_{H_2} = m_{H_2} R_{H_2} T_{H_2}$

$\therefore$ $m_{H_2} \frac{3 \times 10^5 \times 0.3}{\left(\frac{8.314}{2}\right) \times 403 \times 10^3} = 0.0537\ kg$ $\left[\because R = \frac{R_0}{M}\right]$

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3. In a mixture of gases, the partial pressure p_i of any constituent gas can be found by
   (*a*) $p_i = n_i RT/V$ (*b*) $p_i = m_i RT/V_i$
   (*c*) $p_i = n_i R_0 T/V_i$ (*d*) $p_i = n_i R_0 T/V$
   where, R = Characteristic gas constant,
   R_0 = Universal gas constant,
   V = Volume of the mixture,
   V_i = Volume of the ith constituent, and
   T = Temperature of the mixture.
4. Mole fraction of a component of gas mixture is equal to
   (*a*) $1/f$ (*b*) f^2
   (*c*) f (*d*) f/p
   where, f = Volume fraction, and
   p = Pressure of the mixture.
5. In a gaseous mixture the specific volume of each component is given by
   (*a*) V/m (*b*) V_i/m_i
   (*c*) V/m_i (*d*) none of the above.
   where, V = Volume of the mixture,
   V_i = Volume of the ith component,
   m = Mass of mixture, and
   m_i = Mass of the ith component.

ANSWERS

1. (*b*) 2. (*d*) 3. (*d*) 4. (*c*) 5. (*c*).

THEORETICAL QUESTIONS

1. Define the following terms :
   (*i*) Partial pressure (*ii*) Mole fraction
   (*iii*) Volume fraction of a gas constituent in a mixture.
2. Explain briefly Dalton's law and Gibbs-Dalton law.
3. State and explain Amagat's law or Leduc's law.
4. Prove that the molar analysis is identical with the volumetric analysis, and both are equal to the ratio of the partial pressure to the total pressure.
5. Prove the following relation

$$M = \frac{\Sigma n_i M_i}{\Sigma n_i} \cdot \frac{1}{\sum \frac{m_{fi}}{M_i}}$$

   where, M = Molecular weight of the mixture,
   n_i = Number of moles of an any constituent,
   m_{fi} = Mass fraction of the constituent, and
   M_i = Molecular weight of the constituent.

UNSOLVED EXAMPLES

1. 0.45 kg of carbon monoxide (28) and 1 kg of air at 15°C are contained in a vessel of volume 0.4 m³. Calculate the partial pressure of each constituent and the total pressure in the vessel. The gravimetric analysis of air is to be taken as 23.3% oxygen (32) and 76.7% nitrogen (28).

   [**Ans.** $p_{O_2} = 0.4359$ bar ; $p_{N_2} = 1.64$ bar, $p_{CO} = 0.962$ bar]

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Enthalpy of moist air

It is the sum of enthalpy of dry air and enthalpy of water vapour associated with dry air. It is expressed in kJ/kg of dry air

$$h = h_{air} + W \cdot h_{vapour}$$
$$= c_p t_{db} + W \cdot h_{vapour}$$

where h = Enthalpy of mixture/kg of dry air,

h_{air} = Enthalpy of 1 kg of dry air,

h_{vapour} = Enthalpy of 1 kg of vapour obtained from steam tables,

W = Specific humidity in kg/kg of dry air, and

c_p = Specific heat of dry air normally assumed as 1.005 kJ/kg K.

Also $$h_{vapour} = h_g + c_{ps} (t_{db} - t_{dp})$$

where h_g = Enthalpy of saturated steam at dew point temperature,

and c_{ps} = 1.88 kJ/kg K.

$\therefore$ $$h = c_p t_{db} + W[h_g + c_{ps}(t_{db} - t_{dp})] \quad ...(9.11)$$
$$= (c_p + c_{ps} W)\, t_{db} + W(h_g - c_{ps} t_{dp})$$
$$= c_{pm} t_{db} + W(h_g - c_{ps} t_{dp}) \quad ...[9.11(a)]$$

where $c_{pm} = (c_p + c_{ps} W)$ is the *specific heat of humid air or humid specific heat.*

The value of c_{pm} is taken as 1.021 kJ/kg dry air per K. It is the heat capacity of (1 + W) kg of moisture per kg of dry air.

$h_{vapour} \simeq h_g$ at dry bulb temperature. So,

$$h = c_p t_{db} + W h_g. \quad ...(9.12)$$

However, a better approximation is given by the following relationship :

$$h_{vapour} = 2500 + 1.88 t_{db} \text{ kJ/kg of water vapour} \quad ...[9.12\ (a)]$$

where t_{db} is dry bulb temperature in °C, and the datum state is liquid water at 0°C.

$\therefore$ $$h = 1.005\ t_{db} + W(2500 + 1.88\ t_{db}) \text{ kJ/kg dry air.} \quad ...[9.12\ (b)]$$

Adiabatic saturation process

In an insulated chamber when unsaturated air flows over a long sheet of water (Fig. 9.1), the water evaporates, and the specific humidity of the air increases. As the evaporation takes place

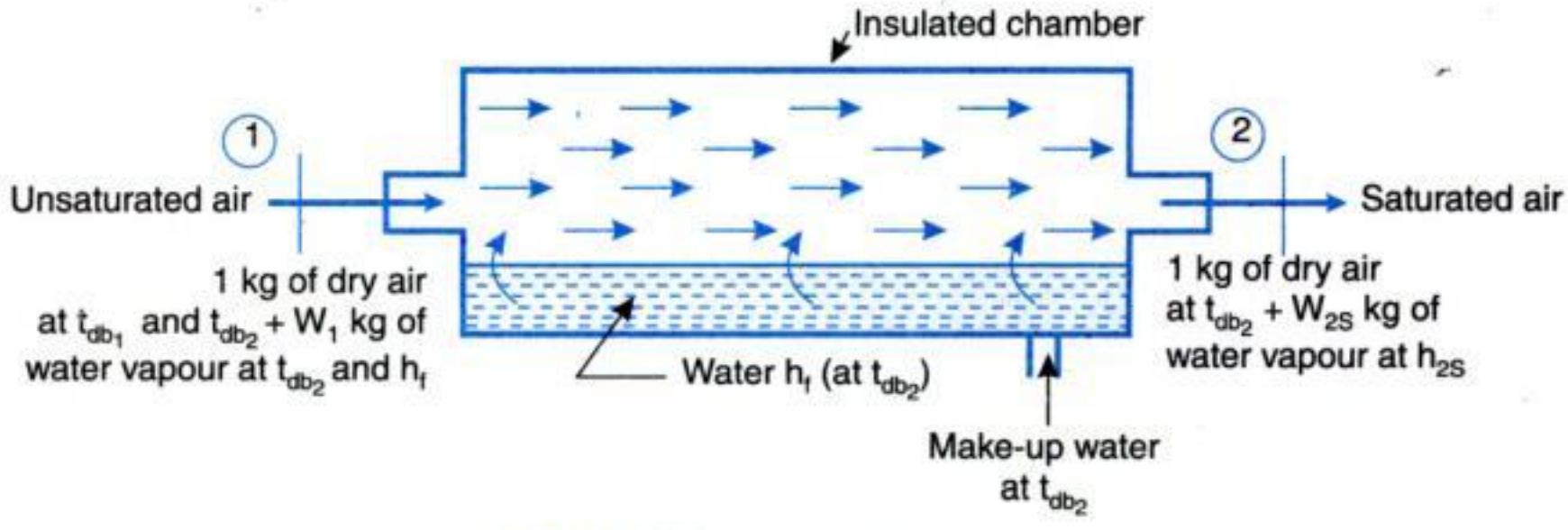

Fig. 9.1. Adiabatic saturation process.

both the air and water are cooled. *The process continues until the energy transferred from the air to the water is equal to the energy required to vapourise the water.* When this point is reached,

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$$0.009625 - 0.0095\, p_v = 0.622\, p_v$$

$$p_v = \mathbf{0.01524\ bar.} \quad \textbf{(Ans.)}$$

(*ii*) **Relative humidity ϕ :**

Corresponding to 20°C, from steam tables,

$$p_{vs} = 0.0234 \text{ bar}$$

$\therefore$ **Relative humidity,** $$\phi = \frac{p_v}{p_{vs}} = \frac{0.01524}{0.0234} = \mathbf{0.65\ or\ 65\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Dew point temperature, t_{dp} :**

The dew point temperature is the saturation temperature of water vapour at a pressure of 0.01524 bar,

t_{dp} [from steam tables by interpolation]

$$= 13 + \frac{(14 - 13)}{(0.01598 - 0.0150)} \times [0.01524 - 0.0150]$$

$$= 13 + \frac{0.00024}{0.00098} = \mathbf{13.24°C.} \quad \textbf{(Ans.)}$$

Example 9.2. *The air supplied to a room of a building in winter is to be at 17°C and have a relative humidity of 60%. If the barometric pressure is 1.01325 bar, find : (i) The specific humidity ; (ii) The dew point under these conditions.*

Solution. Dry bulb temperature, t_{db} = 17°C

Relative humidity, ϕ = 60%

Barometric or total pressure, p_t = 1.01325 bar

Specific humidity, W :

Corresponding to 17°C, from steam tables,

$$p_{vs} = 0.0194 \text{ bar}$$

Also, $$\phi = \frac{p_v}{p_{vs}}$$

i.e., $$0.6 = \frac{p_v}{0.0194}$$

$\therefore$ $$p_v = 0.6 \times 0.0194 = 0.01164 \text{ bar.}$$

Specific humidity, $$\mathbf{W} = \frac{0.622\, p_v}{p_t - p_v} = \frac{0.622 \times 0.01164}{1.01325 - 0.01164}$$

$$= \mathbf{0.00723\ kg/kg\ of\ dry\ air.} \quad \textbf{(Ans.)}$$

Dew point temperature, t_{dp} :

If the air is cooled at constant pressure the vapour will begin to condense at the saturation temperature corresponding to 0.01164 bar. By interpolation from steam tables, the dew point temperature, t_{dp} is then

$$\mathbf{t_{dp}} = 9 + (10 - 9) \times \frac{0.01164 - 0.01150}{0.01230 - 0.01150} = \mathbf{9.18°C.} \quad \textbf{(Ans.)}$$

Example 9.3. *0.004 kg of water vapour per kg of atmospheric air is removed and temperature of air after removing the water vapour becomes 20°C. Determine :*

(i) Relative humidity *(ii) Dew point temperature.*

Assume that condition of atmospheric air is 30°C and 55% R.H. and pressure is 1.0132 bar.

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(*ii*) **Heat added to air per minute :**

Enthalpy, $h_2 = c_p t_{db} + W h_{\text{vapour}}$

$= 1.005 \times 30 + 0.0109 \times [h_g + 1.88\,(t_{db} - t_{dp})]$

$= 1.005 \times 30 + 0.0109\,[2556.3 + 1.88\,(30 - 15.5)]$

$= 58.31$ kJ/kg of dry air

Mass of dry air in 90 m^3 of air supplied

$$m_a = \frac{pV}{RT} = \frac{(p_t - p_v)V}{RT}$$

$$= \frac{(1.0132 - 0.01755) \times 10^5 \times 90}{287 \times (273 + 20)} = 106.5 \text{ kg/min.}$$

Amount of heat added per minute

$= 106.5\,(h_2 - h_1) = 106.5\,(58.31 - 47.85) \simeq$ **1114 kJ.** **(Ans.)**

Example 9.7. Sensible cooling : *40 m^3 of air at 35°C DBT and 50% R.H. is cooled to 25°C DBT maintaining its specific humidity constant. Determine :*

(i) Relative humidity (R.H.) of cooled air ;

(ii) Heat removed from air.

Solution. ***For air at 35°C DBT and 50% R.H. :***

$p_{vs} = 0.0563$ bar (At 35°C, from steam tables)

$$\phi = \frac{p_v}{p_{vs}}$$

$\therefore$ $p_v = \phi \times p_{vs} = 0.5 \times 0.0563 = 0.02815$ bar

$$W_1 = \frac{0.622\,p_v}{p_t - p_v} = \frac{0.622 \times 0.02815}{1.0132 - 0.02815} = 0.0177 \text{ kg/kg of dry air}$$

$h_1 = c_p\, t_{db_1} + W_1\,[h_{g_1} + 1.88\,(t_{db_1} - t_{dp_1}]$

$t_{dp_1} \simeq 23$°C (corresponding to 0.02815 bar)

$\therefore$ $h_1 = 1.005 \times 35 + 0.0177\,[2565.3 + 1.88\,(35 - 23)] = 80.98$ kJ/kg of dry air

For air at 25°C DBT :

(*i*) **R.H. of cooled air :**

Since the specific humidity remains constant the vapour pressure in the air remains constant.

$$\phi = \frac{p_v}{p_{vs}} = \frac{0.02815}{0.0317} = 0.888 \text{ or } 88.8\%$$

i.e., Relative humidity of the cooled air = **88.8%.** **(Ans.)**

(*ii*) **Heat removed from air :**

$h_2 = c_p\, t_{db_2} + W_2\,[h_{g_2} + 1.88\,(t_{db_2} - t_{dp_2})]$

$= 1.005 \times 25 + 0.0177\,[2547.2 + 1.88\,(25 - 23)]$

$= 70.27$ kJ/kg of dry air.

To find mass of dry air (m_a), using the relation :

$p_a v_a = m_a R_a T_a$

$[\because\ W_1 = W_2 = 0.0177$ kg/kg of dry air, $t_{dp_2} = t_{dp_1} = 23$°C since p_v does not change$]$

$\therefore$ $$m_a = \frac{p_a v_a}{R_a T_a} = \frac{(1.0132 - 0.02815) \times 10^5 \times 40}{287 \times (273 + 35)} = 44.57 \text{ kg}$$

$\therefore$ Heat removed from 40 m^3 of air

$= m_a\,(h_1 - h_2) = 44.57\,(80.98 - 70.27) =$ **477.3 kJ.** **(Ans.)**

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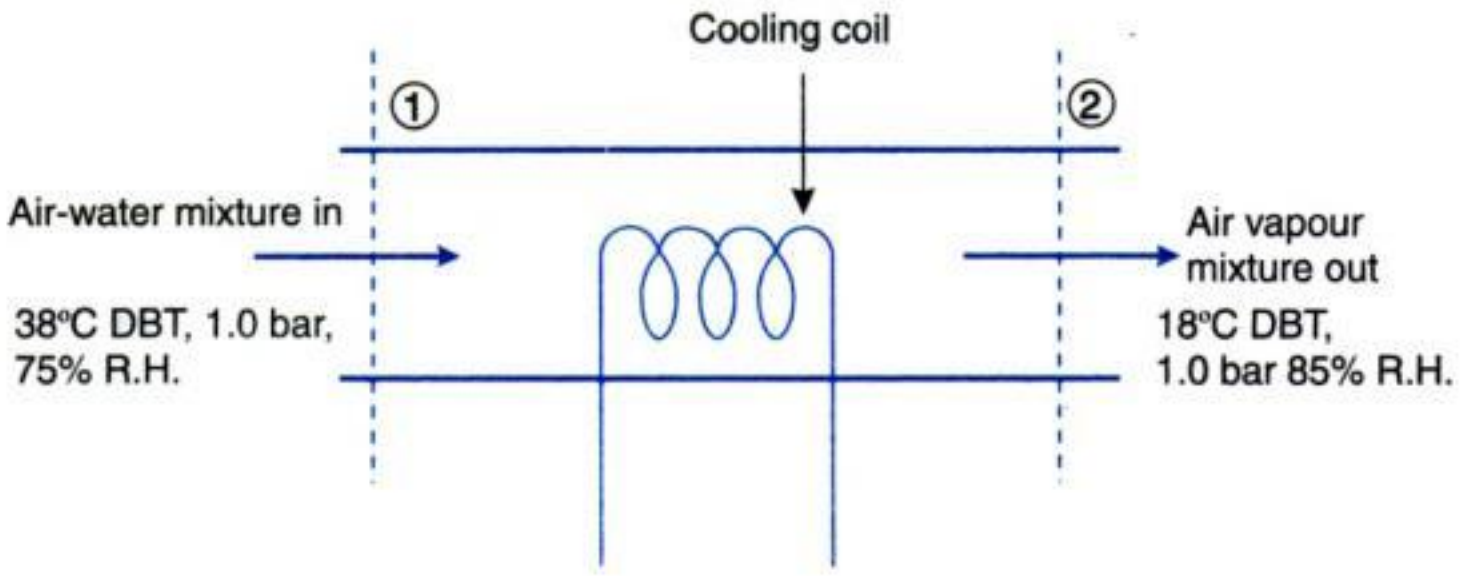

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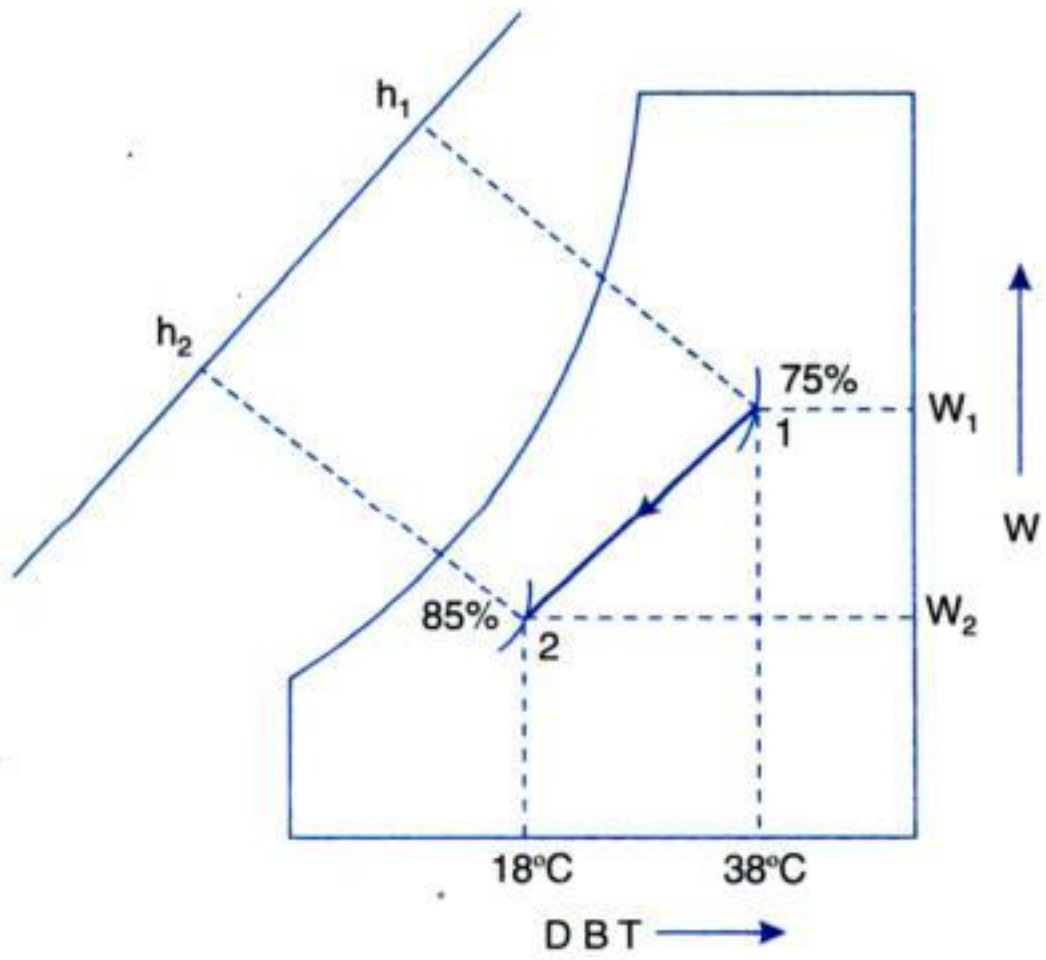

(*b*)

Fig. 9.16

Heat transfer rate,

$$q = (W_2 h_{g_2} - W_1 h_{g_1}) + c_p (t_{db_2} - t_{db_1}) + (W_1 - W_2) h_{f_2}$$

$$= (0.01108 \times 2534.4 - 0.0325 \times 2570.7) + 1.005 (18 - 38) + (0.0325 - 0.01108) \times 75.6$$

$$= - 55.46 - 20.1 + 1.62 = - \mathbf{73.94\ kJ/kg\ of\ dry\ air.}\ \textbf{(Ans.)}$$

Example 9.12. Evaporative Cooler : *Atmospheric air at 38°C and 25% relative humidity passes through an evaporator cooler. If the final temperature of air is 18°C, how much water is added per kg of dry air and what is the final relative humidity ?*

Solution. At 38°C :

$$p_{vs} = 0.0663 \text{ bar}, \quad h_{g_1} = 2570.7 \text{ kJ/kg}$$

and $$p_v = \phi \times p_{vs} = 0.25 \times 0.0663 = 0.01657 \text{ bar}$$

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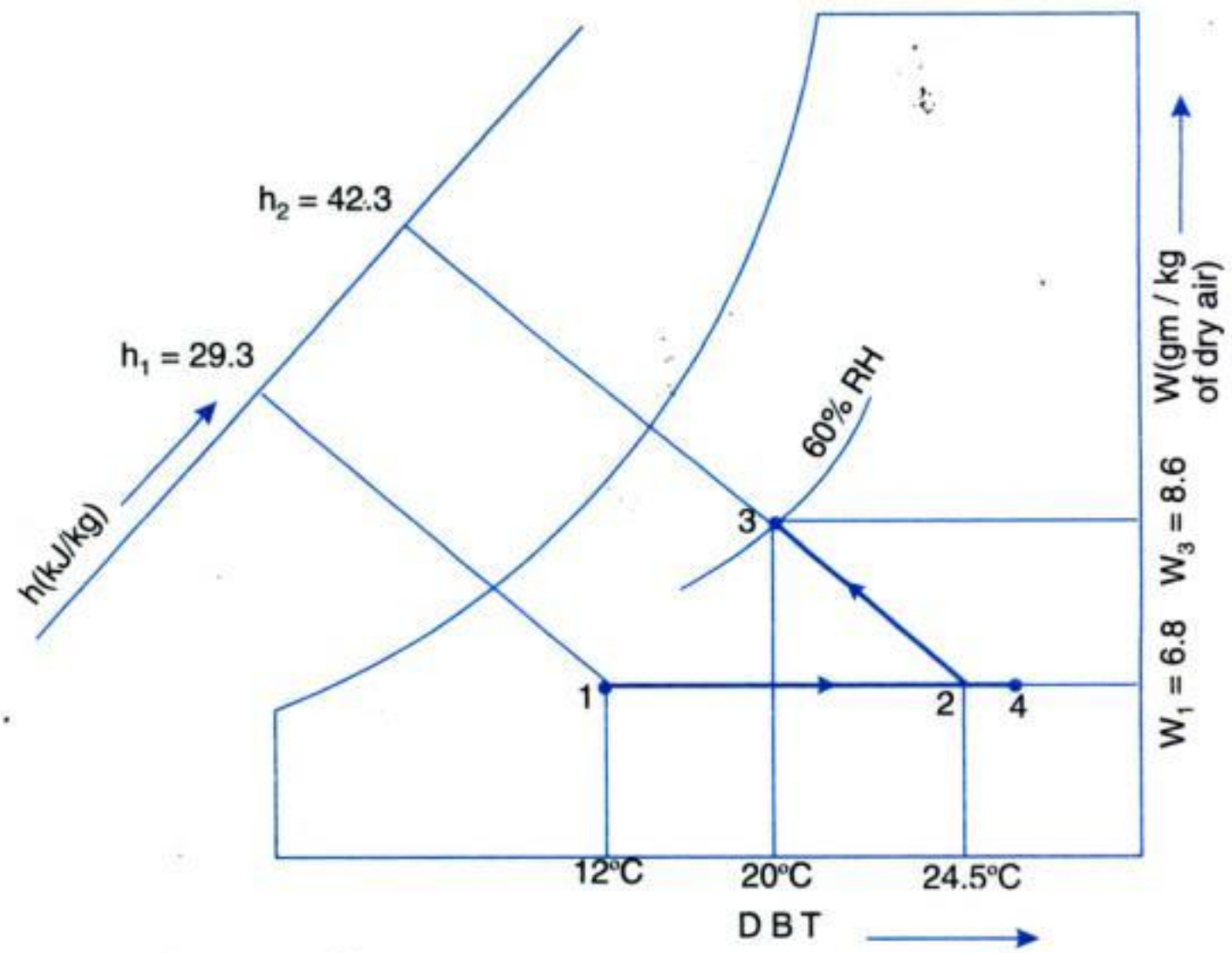

Fig. 9.22

From the psychrometric chart :

$$h_1 = 29.3 \text{ kJ/kg}, \; h_2 = h_3 = 42.3 \text{ kJ/kg}$$

$$t_{db_2} = 24.5^\circ\text{C}, \; v_{s_1} = 0.817 \text{ m}^3/\text{kg}$$

The mass of air circulated per minute,

$$m_a = \frac{0.30 \times 60}{0.817} = 22.03 \text{ kg/min.}$$

(*i*) **Heating capacity of the heating coil**

$$= m_a(h_2 - h_1) = 22.03 \; (42.3 - 29.3) = 286.4 \text{ kJ/min.}$$

$$= \mathbf{4.77 \text{ kJ/s or } 4.77 \text{ kW.}} \quad \textbf{(Ans.)}$$

The by-pass factor (*BF*) of heating coil is given by :

$$BF = \frac{t_{db_4} - t_{db_2}}{t_{db_4} - t_{db_1}}$$

$$0.4 = \frac{t_{db_4} - 24.5}{t_{db_4} - 12}$$

$$\therefore \qquad 0.4 \; (t_{db_4} - 12) = t_{db_4} - 24.5$$

i.e., t_{db_4} *(coil surface temperature)* $= \mathbf{32.8^\circ C.}$ **(Ans.)**

(*ii*) **The capacity of the humidifier**

$$= \frac{m_a \; (W_3 - W_1)}{1000} \times 60 \text{ kg/h} = \frac{22.03 \; (8.6 - 6.8)}{1000} \times 60 = \mathbf{2.379 \text{ kg/h.}} \quad \textbf{(Ans.)}$$

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10. The relative humidity, during heating and humidification,
(*a*) increases (*b*) decreases
(*c*) may increase or decrease (*d*) remains constant.

11. The relative humidity, during cooling and dehumidification of moist air
(*a*) increases (*b*) decreases
(*c*) can increase or decrease (*d*) remains constant.

ANSWERS

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

THEORETICAL QUESTIONS

1. Define the following terms :
(*i*) Saturated air (*ii*) Dry bulb temperature
(*iii*) Dew point temperature (*iv*) Relative humidity
(*v*) Specific humidity.

2. State 'Dalton's law of partial pressure'.

3. Derive the following relations :

(*i*) Specific humidity, $W = 0.622 \frac{p_v}{p_t - p_v}$.

(*ii*) Degree of saturation, $\mu = \frac{p_v}{p_s}\left[\frac{\left(1 - \frac{p_s}{p_t}\right)}{\left(1 - \frac{p_v}{p_t}\right)}\right]$.

4. Explain briefly with a neat sketch a 'sling psychrometer'.

5. Describe briefly any two of the following processes :
(*i*) Sensible heating (*ii*) Cooling and dehumidification
(*iii*) Heating and humidification (*iv*) Heating and dehumidification.

6. Write a short note on 'by-pass factor'.

UNSOLVED EXAMPLES

1. The atmospheric conditions are 30°C and specific humidity of 0.0215 kg/kg of air. Determine :
(*i*) Partial pressure of air (*ii*) Relative humidity
(*iii*) Dew point temperature.
Atmospheric pressure = 756 mm Hg. [**Ans.** (*i*) 14.89 mm of Hg, (*ii*) 46.8%, (*iii*) 17°C]

2. A mixture of air and water vapour at 1 bar and 25°C has a dew point temperature of 15°C. Determine the relative humidity and specific humidity. [**Ans.** 53.8%, 0.01078 kg/kg of dry air]

3. An air-water vapour mixture at 1.24 bar has a temperature of 38°C and relative humidity of 60%. Calculate the kg-mass of water vapour per kg of air and per kg of mixture. Also find the dew point.
[**Ans.** 0.0203 kg/kg of dry air, 0.0199 kg/kg of mixture, 28.5°C]

4. In a house, the temperature of the windows on a day in winter is 5°C. When the temperature in the room is 23°C, and the barometric pressure is 748.8 mm Hg, what would be maximum relative humidity that could be maintained in the room without condensation on the window panes ? Under these conditions, find the partial pressure of the water vapour and air, the specific humidity, and the density of the mixture.
[**Ans.** 30.7%, 0.00872 bar, 0.989 bar, 0.00548 kg/kg of dry air, 1.164 kg/m^3]

5. Atmospheric air enters a heater at 4.5°C and 60% relative humidity and leaves at a temperature of 21°C. Calculate :
(*i*) The heat supplied to the air ;
(*ii*) The final relative humidity. [**Ans.** 16.8 kJ/kg, 29.2%]

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Briquettes. These are prepared from fine coal or coke by compressing the material under high pressure.

10.4. LIQUID FUELS

The chief source of liquid fuels is *petroleum* which is obtained from wells under the earth's crust. These fuels have proved *more advantageous in comparison to solid fuels* in the following respects.

Advantages :

1. Require less space for storage.
2. Higher calorific value.
3. Easy control of consumption.
4. Staff economy.
5. Absence of danger from spontaneous combustion.
6. Easy handling and transportation.
7. Cleanliness.
8. No ash problem.
9. Non-deterioration of the oil in storage.

Petroleum. There are different opinions regarding the origin of petroleum. However, now it is accepted that petroleum has originated probably from organic matter like fish and plant life etc., by bacterial action or by their distillation under pressure and heat. It consists of a mixture of gases, liquids and solid hydrocarbons with small amounts of nitrogen and sulphur compounds. In India, the main sources of Petroleum are Assam and Gujarat.

Heavy fuel oil or crude oil is imported and then refined at different refineries. The refining of crude oil supplies the most important product called *petrol*. Petrol can also be made by polymerization of refinery gases.

Other liquid fuels are kerosene, fuels oils, colloidal fuels and alcohol.

10.5. GASEOUS FUELS

Natural gas. The main constituents of natural gas are *methane* (CH_4) and *ethane* (C_2H_6). It has calorific value nearly 21000 kJ/m^3. Natural gas is used alternately or simultaneously with oil for internal combustion engines.

Coal gas. Mainly consists of *hydrogen, carbon monoxide* and *hydrocarbons*. It is prepared by carbonisation of coal. It finds its use in boilers and sometimes used for commercial purposes.

Coke-oven gas. It is obtained during the production of coke by heating the bituminous coal. The volatile content of coal is driven off by heating and major portion of this gas is utilised in heating the ovens. This gas *must be thoroughly filtered before using in gas engines.*

Blast furnace gas. It is obtained from smelting operation in which air is forced through layers of coke and iron ore, the example being that of pig iron manufacture where this gas is produced as by product and contains about 20% carbon monoxide (CO). After filtering it may be blended with richer gas or used in gas engines directly. The heating value of this gas is very low.

Producer gas. It results from the partial oxidation of coal, coke or peat when they are burnt with an insufficient quantity of air. It is produced in specially designed retorts. It has low heating value and in general is suitable for large installations. It is also used in steel industry for firing open hearth furnaces.

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Typical values of ΔH_f for different substances at 25°C (298 K) in **kJ/mole** are given below :

S. No.	Substance	Formula	State	ΔH_f
1.	Oxygen	O	gas	249143
		O_2	gas	zero
2.	Water	H_2O	liquid	–285765
			vapour	–241783
3.	Carbon	C	gas	714852
			diamond	1900
			graphite	zero
4.	Carbon monoxide	CO	gas	–111508
5.	Carbon dioxide	CO_2	gas	–393443
6.	Methane	CH_4	gas	–74855
7.	Methyl alcohol	CH_3OH	vapour	–240532
8.	Ethyl alcohol	C_2H_5OH	vapour	–281102
9.	Ethane	C_2H_6	gas	–83870
10.	Ethene	C_2H_4	gas	51780
11.	Propane	C_3H_8	gas	–102900
12.	Butane	C_4H_{10}	gas	–125000
13.	Octane	C_8H_{18}	liquid	–247600

10.18. CALORIFIC OR HEATING VALUES OF FUELS

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

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

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

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

Thus :
$$(HHV)_p = (LHV)_p + m\, h_{fg} \quad ...(10.26)$$
$$(HHV)_v = (LHV)_v + m(u_g - u_f) \quad ...(10.27)$$

where m = Mass of water formed by combustion,

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

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

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

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

10.19. DETERMINATION OF CALORIFIC OR HEATING VALUES

The calorific value of fuels can be determined either from chemical analysis or in the laboratory.

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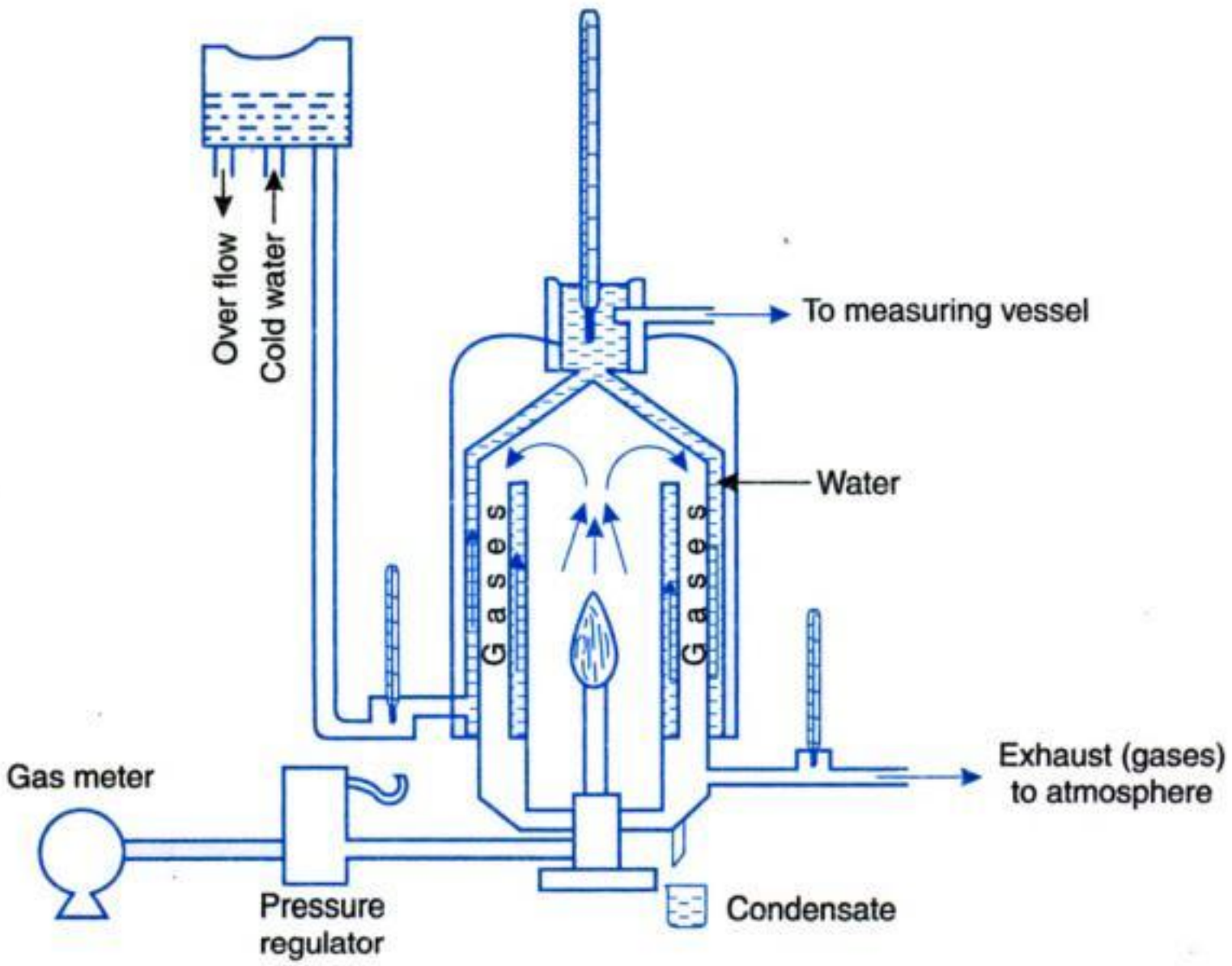

Fig. 10.6. Junker's gas calorimeter.

Note. 1. The volume of gas used during the experiment should be converted to volume at S.T.P. (standard temperature and pressure *i.e.*, 15°C, 760 mm respectively).

2. The gross calorific value is obtained by dividing the heat given out by corrected volume of gas. The net or lower calorific value of the gas is obtained by subtracting from total heat the heat associated with condensed water (which is obtained by multiplying the weighed condensate by latent heat of vapourisation of water.).

Example 10.2. *Following results were obtained when a sample of gas was tested by Junker's gas calorimeter :*

Gas burnt in the calorimeter = 0.08 m^3, Pressure of gas supply = 5.2 cm of water, Barometer = 75.5 cm of Hg. Temperature of gas = 13°C, Weight of water heated by gas = 28 kg, Temperature of water at inlet = 10°C, Temperature of water at outlet = 23.5°C, Steam condensed = 0.06 kg.

Determine the higher and lower calorific values per m^3 of the gas at a temperature of 15°C and barometric pressure of 76 cm of Hg.

Solution. The volume of the gas is measured at a temperature of 13°C and pressure of 5.2 cm of water. Let us reduce this volume to S.T.P. by using the general gas equation,

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

$$p_1 = 75.5 + \left(\frac{5.2}{13.6}\right) = 75.882 \text{ cm of Hg} ; T_1 = 273 + 13 = 286 \text{ K}$$

$$V_1 = 0.08 \text{ m}^3 ; p_2 = 76 \text{ cm of Hg}$$

$$V_2 = ? ; \; T_2 = 273 + 15 = 288 \text{ K}$$

$$\therefore \quad \frac{75.882 \times 0.08}{286} = \frac{76 \times V_2}{288}$$

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☞**Example 10.8.** *A sample of fuel has the following percentage composition : Carbon = 86 per cent ; Hydrogen = 8 per cent ; Sulphur = 3 per cent ; Oxygen = 2 per cent ; Ash = 1 per cent.*

For an air-fuel ratio of 12 : 1, calculate :

(i) Mixture strength as a percentage rich or weak.

(ii) Volumetric analysis of the dry products of combustion.

Solution.

Element, wt. (kg)	*O_2 reqd. (kg)*
C = 0.86	$0.86 \times \frac{8}{3} = 2.29$
$H_2 = 0.08$	$0.08 \times 8 = 0.64$
S = 0.03	$0.03 \times \frac{1}{1} = 0.03$
$O_2 = 0.02$	
	Total $O_2 = 2.96$

Weight of oxygen to be supplied per kg of fuel = 2.96 – 0.02 = 294 kg.

Weight of minimum air required for complete combustion = $\frac{2.94 \times 100}{23}$ = 12.78 kg

Hence "*correct*" fuel-air ratio = $\frac{1}{12.78}$: 1

But actual ratio is $\frac{1}{12}$: 1.

(*i*) **Mixture strength** = $\frac{12.78}{12} \times 100 = 106.5\%$

This show that mixture is **6.5% rich. (Ans.)**

Deficient amount of air = 12.78 – 12 = 0.78 kg

Amount of air saved by burning 1 kg of C to CO instead of CO_2

$$= \text{Oxygen saved} \times \frac{100}{23}$$

$$= \left[\frac{8}{3}(CO_2) - \frac{4}{3}(CO)\right] \times \frac{100}{23} = 5.8 \text{ kg}$$

Hence $\frac{0.78}{5.8}$ = 0.134 kg of carbon burns to CO and as such 0.86 – 0.134 = 0.726 kg of carbon burns to CO_2.

$\therefore$ CO formed = $0.134 \times \frac{7}{3} = 0.313$ kg

CO_2 formed = $0.726 \times \frac{11}{3} = 2.662$ kg

N_2 supplied = 12 × 0.77 = 9.24 kg

SO_2 formed = 0.03 × 2 = 0.06 kg.

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Air applied on the basis of conditions at exit

$$= \frac{80.6 \times 80}{33(7.9 + 0)} = 24.73 \text{ kg}$$

$\therefore$ *Air leakage* = 24.73 – 23.45 = **1.28 kg of air per kg of fuel.** **(Ans.)**

(*ii*) For each kg of fuel burnt, the ash collected is 15% *i.e.*, 0.15 kg.

$\therefore$ Weight of fuel passing up the chimney = 1 – 0.15 = 0.85 kg

$\therefore$ Total weight of products

= Weight of air supplied per kg of fuel
+ Weight of fuel passing through chimney per kg of fuel

= 23.45 + 0.85 = 24.3 kg

Heat in flue gases per kg of coal

= Weight of flue gases × Specific heat × Temperature rise above 0°C

= 24.3 × 1.05 × (410 – 0) = 10461 kJ

Heat in leakage air = Weight of leakage air × Specific heat
× Temperature rise of air above 0°C

= 1.28 × 1.005 × (20 – 0) = 25.73 kJ.

We can still consider, in the mixture, the gas and the air as separate and having their own specific heats, but sharing a common temperature t.

For heat balance :

$$(1.005 \times 1.28 + 24.3 \times 1.05)\, t = 10461 + 25.73$$

$$26.8\, t = 10486.73$$

$$\therefore \quad t = 391.3°C$$

$\therefore$ *Fall in temperature as a result of the air leakage into the economiser*

= 410 – 391.3 = **18.7°C.** **(Ans.)**

AIR-FUEL RATIO AND ANALYSIS OF PRODUCTS OF COMBUSTION

Example 10.12. *The chemical formula for alcohol is C_2H_6O. Calculate the stoichiometric air/fuel ratio by mass and the percentage composition of the products of combustion per kg of C_2H_6O.* **(GATE 1998)**

Solution. Chemical equation for complete combustion of given fuel can be written as follows :

$$C_2H_6O + 3O_2 \longrightarrow 2CO_2 + 3H_2O$$
$$(1 \times 46) + (3 \times 32) = (2 \times 44) + (3 \times 18)$$

For complete combustion of 1 kg of C_2H_6O, oxygen required

$$= \frac{3 \times 32}{46} = 2.087 \text{ kg of oxygen}$$

$$= 2.087 \times \frac{100}{23} = 9.074 \text{ kg of air}$$

A : F ratio for complete combustion = **9.074 : 1.** **Ans.**

Also 46 kg of fuel produces products of combustion = 88 + 54 = 142 kg

$$\left[\therefore \quad 1 \text{ kg of fuel produces } \frac{142}{46} = 3.087 \text{ kg of products of combustion (}i.e.,\ CO_2 \text{ and } H_2O\text{)}\right]$$

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and $$b = 2 - a = 2 - 1.3 = 0.7$$

Thus the combustion equation becomes :

$$C_2H_6 + (0.9)\,(3.5)\,O_2 + (0.9)\,(3.5)\left(\frac{79}{21}\right) N_2$$

$$\longrightarrow 1.3CO_2 + 0.7CO + 3H_2O + (0.9)\,(3.5)\left(\frac{79}{21}\right) N_2$$

Total number of moles of *dry products* of combustion

$$= 1.3 + 0.7 + (0.9)\,(3.5)\left(\frac{79}{21}\right)$$

$$= 1.3 + 0.7 + 11.85 = 13.85 \text{ moles/mole of fuel}$$

Volumetric analysis of dry products of combustion is as follows :

$$\mathbf{CO_2} = \frac{1.3}{13.85} \times 100 = \mathbf{9.38\%.} \quad \textbf{(Ans.)}$$

$$\mathbf{CO} = \frac{0.7}{13.85} \times 100 = \mathbf{5.05\%.} \quad \textbf{(Ans.)}$$

$$\mathbf{N_2} = \frac{11.85}{13.85} \times 100 = \mathbf{85.56\%.} \quad \textbf{(Ans.)}$$

☞ **Example 10.18.** *Methane (CH_4) is burned with atmospheric air. The analysis of the products on a 'dry' basis is as follows :*

$$CO_2 = 10.00\%,\ O_2 = 2.37\%,\ CO = 0.53\%,\ N_2 = 87.10\%.$$

(i) Determine the combustion equation ; (ii) Calculate the air-fuel ratio ;

(iii) Percent theoretical air.

Solution. *(i)* **Combustion equation :**

From the analysis of the products, the following equation can be written, keeping in mind that this analysis is on a *dry basis*.

$$x\ CH_4 + y\ O_2 + z\ N_2 \longrightarrow 10.0\ CO_2 + 0.53\ CO + 2.37\ O_2 + a\ H_2O + 87.1\ N_2$$

To determine all the unknown co-efficients let us find balance for each of the elements.

Nitrogen balance : $z = 87.1$

Since all the nitrogen comes from the air,

$$\frac{z}{y} = \frac{79}{21}\ ;\ y = \frac{87.1}{(79/21)} = 23.16$$

Carbon balance : $x = 10.00 + 0.53 = 10.53$

Hydrogen balance : $a = 2x = 2 \times 10.53 = 21.06$

Oxygen balance. All the unknown co-efficients have been solved for, and in this case the oxygen balance provides a check on the accuracy. Thus, y can also be determined by an oxygen balance

$$y = 10.00 + \frac{0.53}{2} + 2.37 + \frac{21.06}{2} = 23.16$$

Substituting these values for x, y, z and a, we have,

$$10.53\ CH_4 + 23.16\ O_2 + 87.1\ N_2 \longrightarrow 10.0\ CO_2 + 0.53\ CO + 2.37\ O_2 + 21.06\ H_2O + 87.1\ N_2$$

Dividing both sides by 10.53, we get the **combustion equation** per mole of fuel,

$$\mathbf{CH_4 + 2.2\ O_2 + 8.27\ N_2 \longrightarrow 0.95\ CO_2 + 0.05\ CO + 2H_2O + 0.225\ O_2 + 8.27\ N_2.} \quad \textbf{(Ans.)}$$

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Solution. The example is solved by a tabular method ; a specimen calculation is given below :

For CH_4 :

$$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$$

i.e., 1 mole CH_4 + 2 moles O_2 ⟶ 1 mole CO_2 + 2 moles H_2O

There are 0.2 moles of CH_4 per mole of the coal gas, hence

0.2 moles CH_4 + 0.2 × 2 moles O_2 ⟶ 0.2 moles CO_2 + 0.2 × 2 moles H_2O

∴ O_2 required for the CH_4 in the coal gas = 0.4 moles per mole of coal gas.

The oxygen in the fuel (0.004 moles) is included in column 4 as a *negative quantity*.

Product	*Moles/mole fuel*	*Combustion equation*	*O_2 moles / mole fuel*	*Products CO_2*	*H_2O*
1	*2*	*3*	*4*	*5*	*6*
H_2O	0.504	$2H_2 + O_2 \longrightarrow 2H_2O$	0.252	—	0.504
CO	0.17	$2CO + O_2 \longrightarrow 2CO_2$	0.085	0.17	—
CH_4	0.20	$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$	0.400	0.20	0.40
C_4H_8	0.02	$C_4H_8 + 6O_2 \longrightarrow 4CO_2 + 4H_2O$	0.120	0.08	0.08
O_2	0.004	—	−0.004	—	—
N_2	0.062	—	—	—	—
CO_2	0.04	—	—	0.04	—
			Total = 0.853	0.49	0.984

(*i*) **Stoichiometric A/F ratio :**

$$\text{Air required} = \frac{0.853}{0.21} = 4.06 \text{ moles/mole of fuel}$$

(where air is assumed to contain 21% O_2 by volume)

∴ **Stoichiometric A/F ratio = 4.06/1 by volume.** **(Ans.)**

(*ii*) **Wet and dry analyses of the products of combustion if the actual mixture is 30% weak :**

Actual A/F ratio with 30% weak mixture

$$= 4.06 + \frac{30}{100} \times 4.06 = 1.3 \times 4.06 = 5.278/1$$

Associated N_2 = 0.79 × 5.278 = 4.17 moles/mole fuel

Excess oxygen = 0.21 × 5.278 − 0.853 = 0.255 moles

Total moles of N_2 in products = 4.17 + 0.062 = 4.232 moles/mole fuel.

Analysis by volume of wet and dry products :

Product	*Moles/mole fuel*	*% by vol. (dry)*	*% by vol. (wet)*
CO_2	0.490	9.97	8.31
H_2O	0.984	—	16.68
O_2	0.255	5.19	4.32
N_2	4.170	84.84	70.69
	Total wet = 5.899	100.00	100.00
	$-H_2O$ = 0.984		**(Ans.)**
	Total dry = 4.915		

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If the air-fuel ratio is 7/1 by volume, calculate the analysis of the dry products of combustion. It can be assumed that the stoichiometric A/F ratio is less than 7/1.

Solution. Since it is given that the actual *A/F ratio is greater than the stoichiometric,* therefore it follows that *excess air has been supplied.* The products will therefore consist of CO_2, H_2O, O_2 and N_2.

The combustion equation can be written as follows :

$$0.506H_2 + 0.1CO + 0.26CH_4 + 0.04C_4H_8 + 0.004O_2 + 0.03CO_2 + 0.06N_2 + 0.21 \times 7O_2 + 0.79 \times 7N_2 \longrightarrow a\ CO_2 + b\ H_2O + c\ O_2 + d\ N_2$$

Then,

Carbon balance : $0.1 + 0.26 + 4 \times 0.04 + 0.03 = a$ $\quad\therefore\quad a = 0.55$

Hydrogen balance : $2 \times 0.506 + 4 \times 0.26 + 8 \times 0.04 = 2b$ $\quad\therefore\quad b = 1.186$

Oxygen balance : $0.1 + 2 \times 0.004 + 2 \times 0.03 + 0.21 \times 7 \times 2 = 2a + b + 2c$ $\quad\therefore\quad c = 0.411$

Nitrogen balance : $2 \times 0.06 + 2 \times 0.79 \times 7 = 2d$ $\quad\therefore\quad d = 5.59$

$\therefore$ Total moles of dry products = $0.55 + 0.411 + 5.59 = 6.65$

Then analysis by volume is :

$$\mathbf{CO_2} = \frac{0.55}{6.55} \times 100 = \mathbf{8.39\%.} \quad \text{(Ans.)}$$

$$\mathbf{O_2} = \frac{0.411}{6.55} \times 100 = \mathbf{6.27\%.} \quad \text{(Ans.)}$$

$$\mathbf{N_2} = \frac{5.59}{6.55} \times 100 = \mathbf{85.34\%.} \quad \text{(Ans.)}$$

Example 10.25. *The following is the analysis (by weight) of a chemical fuel :*

Carbon = 60 per cent ; Hydrogen = 20 per cent ; Oxygen = 5 per cent ; Sulphur = 5 per cent and Nitrogen = 10 per cent.

Find the stoichiometric amount of air required for complete combustion of this fuel.

Solution. On the basis of 100 kg fuel let us assume an equivalent formula of the form :

$$C_aH_bO_cN_dS_e$$

From the given analysis by weight, we can write

$12a = 60$ or $a = 5$

$1b = 20$ or $b = 20$

$16c = 5$ or $c = 0.3125$

$14d = 10$ or $d = 0.7143$

$32e = 5$ or $e = 0.1562$

Then the formula of the fuel can be written as

$$C_5H_{20}O_{0.3125}N_{0.7143}S_{0.1562}$$

The combustion equation is

$$C_5H_{20}O_{0.3125}N_{0.7143}S_{0.1562} + x\ O_2 + x\left(\frac{79}{21}\right) N_2 \longrightarrow p\ CO_2 + q\ H_2O + r\ SO_2 + s\ N_2$$

Then,

Carbon balance : $5 = p$ $\quad\therefore\quad p = 5$

Hydrogen balance : $20 = 2q$ $\quad\therefore\quad q = 10$

Sulphur balance : $0.1562 = r$ $\quad\therefore\quad r = 0.1562$

Oxygen balance : $0.3125 + 2x = (2p + q + 2r)$

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☞ **Example 10.29.** *The exhaust from an engine running on benzole was measured with the help of Orsat apparatus. Orsat analysis showed a CO_2 content of 12%, but no CO. Assuming that the remainder of the exhaust contains only oxygen and nitrogen, calculate the air-fuel ratio of the engine.*

The ultimate analysis of benzole is C = 88% and H_2 = 12%.

Solution. 1 kg of fuel, consisting of 0.88 kg C and 0.12 kg H_2, can be written as 0.88/2 moles C and 0.12/2 moles H_2. Therefore, considering 1 mole of *dry exhaust gas* (D.E.G.) we can write the combustion equation as follows :

$$X\left(\frac{0.88}{12}C+\frac{0.12}{2}H_2\right)+Y\,O_2+\frac{79}{21}\,Y\,N_2\longrightarrow 0.12CO_2+a\,O_2+(0.88-a)\,N_2+b\,H_2O$$

[Let the D.E.G. contain a moles of O_2. The moles of CO_2 in 1 mole of D.E.G. are 0.12.
Therefore the D.E.G. contains $(1-a-0.12)=(0.88-a)$ moles of N_2.]

where, X = Mass of fuel per mole D.E.G.,
Y = Moles of O_2 per mole D.E.G.,
a = Moles of *excess* O_2 per mole D.E.G., and
b = Moles of H_2O per mole D.E.G.

Now,

Carbon balance : $\frac{0.88}{12}X=0.12$ $\quad\therefore\quad X=1.636$

Hydrogen balance : $0.06X=b$ $\quad\therefore\quad b=0.06\times1.636=0.098$

Oxygen balance : $2Y=2\times0.12+2a+b$

or $2Y=0.24+2a+0.098$ $\quad\therefore\quad Y=0.169+a$

Nitrogen balance : $\frac{79}{21}\,Y=(0.88-a)$ $\quad\therefore\quad Y=0.234-0.266a$

Equating the expressions for Y gives

$0.234-0.266a=0.169+a$ $\quad\therefore\quad a=0.0513$

i.e., $Y=0.169-0.0513=0.2203$

$\therefore$ O_2 supplied = 0.2203×32 kg/mole E.G.

i.e., Air supplied = $\frac{0.2203\times32}{0.233}$ = 30.26 kg/mole D.E.G.

Since X = 1.636, then, the fuel supplied per mole D.E.G. is 1.636 kg

$\therefore$ **A/F ratio** = $\frac{30.26}{1.636}$ = **18.5/1.** **(Ans.)**

Example 10.30. *The analysis of the dry exhaust from an internal combustion engine is as follows :*

Carbon dioxide (CO_2) = 15 per cent — *Carbon monoxide (CO) = 3 per cent*
Methane (CH_4) = 3 per cent — *Hydrogen (H_2) = 1 per cent*
Oxygen (O_2) = 2 per cent — *Nitrogen (N_2) = 76 per cent*

Calculate the proportions by mass of carbon to hydrogen in the fuel, assuming it to be a pure hydrocarbon.

Solution. Let 1 kg of fuel contain x kg of carbon (C) and y kg hydrogen (H_2). Then considering 1 mole of D.E.G. and introducing X and Y, we can write

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Given that the enthalpies of combustion at 25°C are :

$$C_6H_6 = -3169500 \text{ kJ/mole}$$

$$C_8H_{18} = -5116200 \text{ kJ/mole}$$

Both the above figures are for the case where the water in the products is in the vapour phase.

Solution. (*i*) **Air and benzene vapour :**

For benzene, the combustion equation is as follows :

$$C_6H_6(g) + 7.5O_2(g) + 7.5\left(\frac{79}{21}\right)N_2(g) = 6CO_2(g) + 3H_2O(g) + 7.5\left(\frac{79}{21}\right)N_2(g)$$

Since the water in the products is in *vapour phase*, therefore, the given value of enthalpy of combustion corresponds to the *lower heating* value at constant pressure.

i.e., $$(LHV)_p = 3169500 \text{ kJ/mole}$$

$$\textbf{(LHV)}_{\textbf{v}} \textbf{ per kg of mixture} = \frac{3169500}{(12\times 6+6\times 1)+(7.5\times 32)+7.5\left(\frac{79}{21}\right)(28)}$$

$$= \frac{3169500}{78+240+790} = \textbf{2861 kJ/kg.} \quad \text{(Ans.)}$$

Now, $$(HHV)_p = (LHV)_p + mh_{fg}$$

where $(HHV)_p$ = Higher heating value at constant pressure,

$(LHV)_p$ = Lower heating value at constant pressure,

m = Mass of water formed by combustion.

= 3 × 18 = 54 kg/kg mole of fuel, and

h_{fg} = Latent heat of vapourisation at given temperature per unit mass of water

= 2442 kJ/kg at 25°C.

$\therefore$ $$(HHV)_p = 3169500 + 54 \times 2442 = 3301368 \text{ kJ/mole}$$

Thus, $$\textbf{(HHV)}_{\textbf{p}} \textbf{ per kg of mixture} = \frac{3301368}{78+240+790} = \textbf{2980 kJ/kg.} \quad \text{(Ans.)}$$

(*ii*) **Air and octane vapour :**

$$(LHV)_p = 5116200 \text{ kJ/mole of } C_8H_{18}$$

For octane, the combustion equation is written as follows :

$$C_8H_{18}(g) + 12.5O_2(g) \longrightarrow 8CO_2(g) + 9H_2O(g) + 12.5\left(\frac{79}{21}\right)N_2(g)$$

$$\textbf{(LHV)}_{\textbf{p}} \textbf{ per kg of mixture} = \frac{5116200}{(12\times 8+18\times 1)+12.5\times 32+12.5\times\frac{79}{21}\times 28}$$

$$= \frac{5116200}{114+400+1317} = \textbf{2794 kJ/kg.} \quad \text{(Ans.)}$$

$$(HHV)_p = (LHV)_p + mh_{fg}$$

$m = 9 \times 18 = 162$ kJ/kg mole of fuel

$\therefore$ $$(HHV)_p = 5116200 + 162 \times 2442 = 5511804$$

Hence, $$\textbf{(HHV)}_{\textbf{p}} \textbf{ per kg of mixture} = \frac{5511804}{114+400+1317} = \textbf{3010 kJ/kg.} \quad \text{(Ans.)}$$

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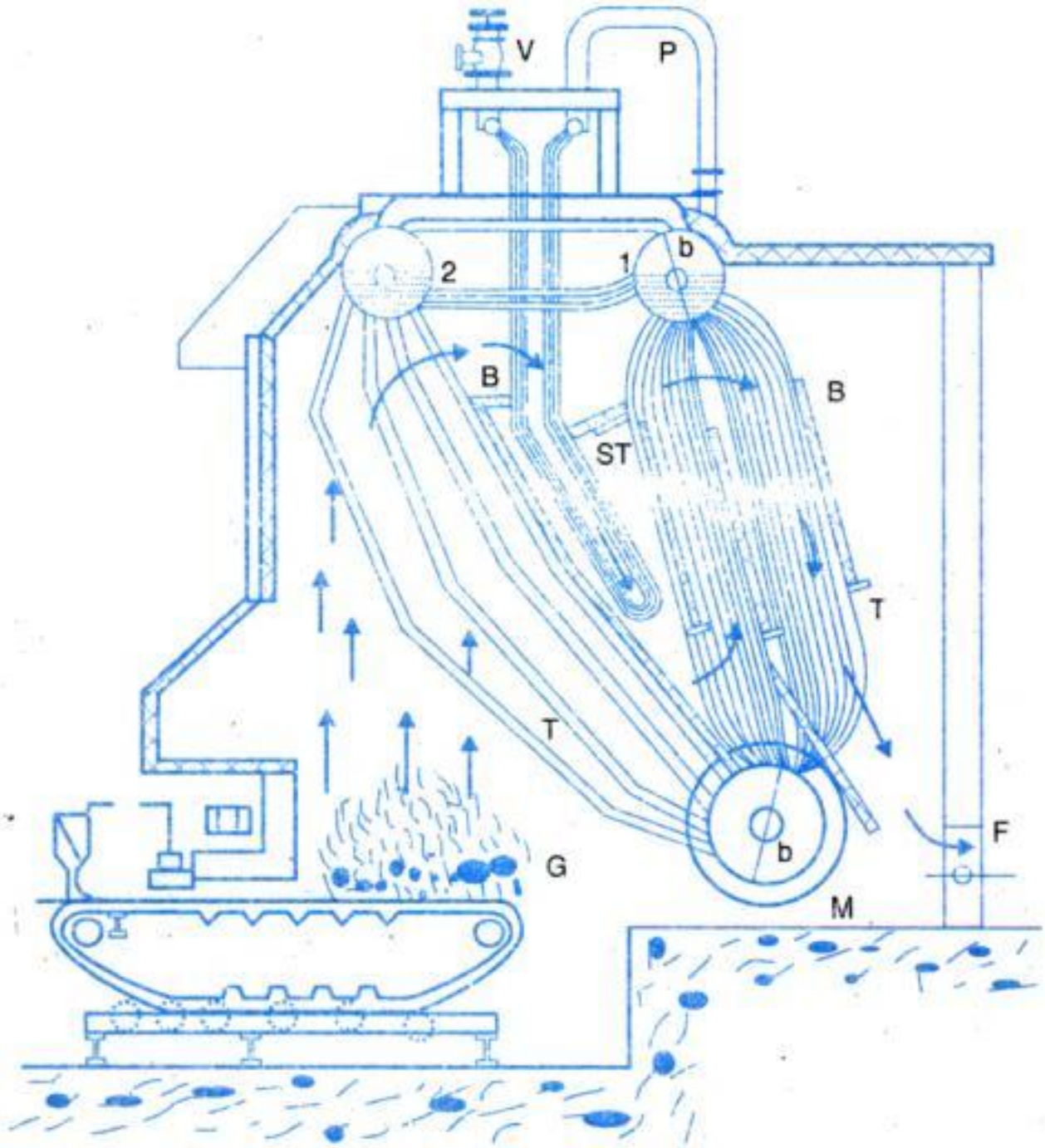
The various mountings used on the boiler are shown in Fig. 11.7.

A Babcock Wilcox water tube boiler with cross draw differs from longitudinal drum boiler in a way that how drum is placed with reference to the axis of the water tubes of the boiler. The longitudinal drum restricts the number of tubes that can be connected to one drum circumferentially and limits the capacity of the boiler. In the cross drum there is no limitation of the number of connecting tubes.

The pressure of steam in case of cross drum boiler may be as high as 100 bar and steaming capacity upto 27000 kg/h.

11.8.2. Stirling Boiler

Stirling water tube boiler is an example of *bent tube* boiler. The main elements of a bent type water tube boiler are essentially drum or drums and headers connected by bent tubes. For large central power stations these boilers are very popular. They have steaming capacities as high as 50000 kg/h and pressure as high as 60 bar.



V = Stop valve
P = Steam pipe
b = Water baffle
B = Baffle wall
ST = Superheater tubes
T = Water tubes
G = Grate
M = Mud drum

Fig. 11.8. Stirling boiler

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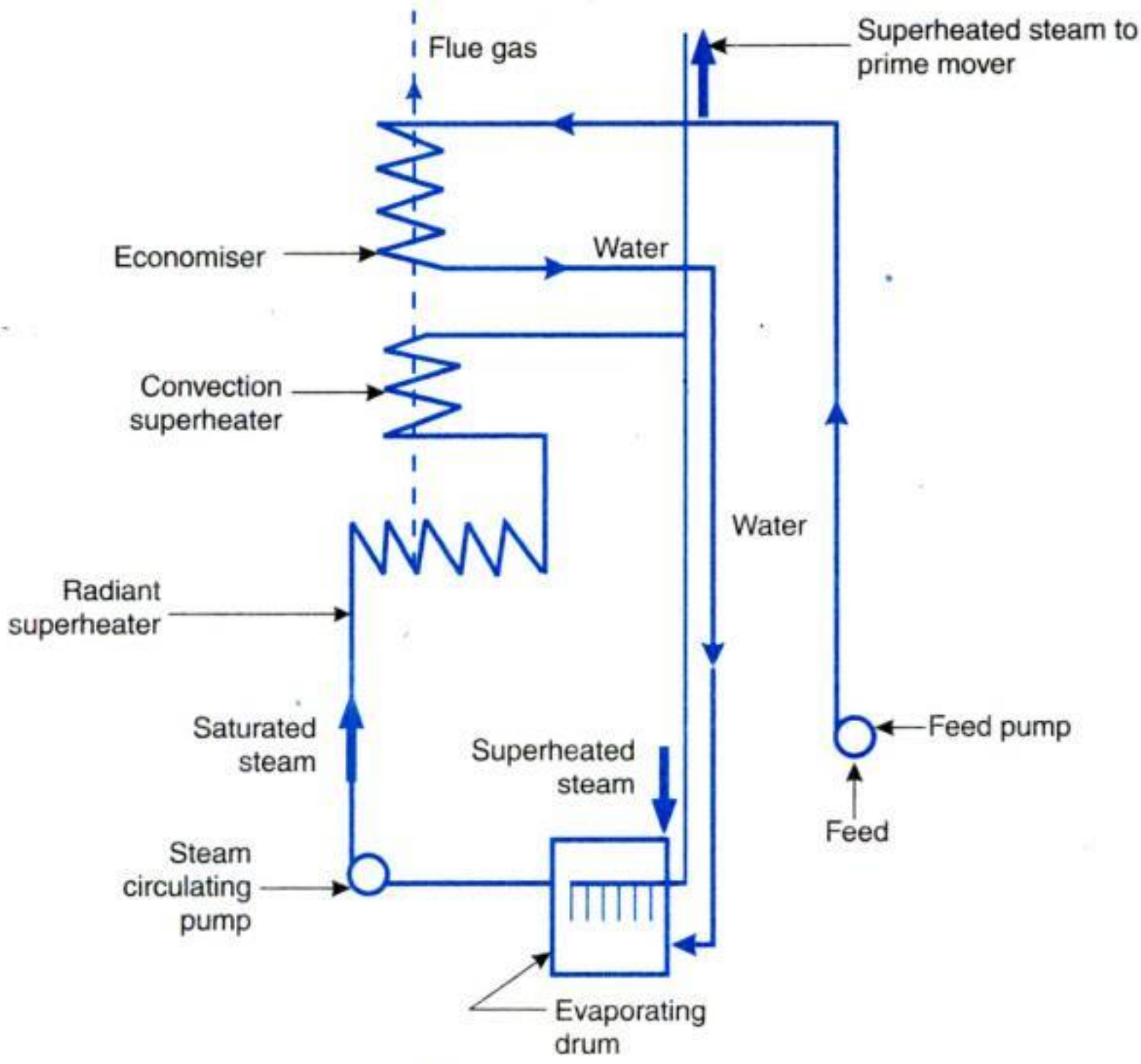

Fig. 11.10. Loeffler boiler.

11.9.6. Benson Boiler

In the LaMont boiler, the main difficult experienced *is the formation and attachment of bubbles on the inner surfaces of the heating tubes. The attached bubbles to the tube surfaces reduce the heat flow and steam generation as it offers high thermal resistance than water film. Benson in 1922 argued that if the boiler pressure was raised to critical pressure (225 atm.), the steam and water have the same density and therefore, the danger of bubble formation can be easily eliminated.* The first high pressure Benson boiler was put into operation in 1927 in West Germany.

This boiler too makes use of forced circulation and uses oil as fuel. It chief novel *principle is that it eliminates the latent heat of water by first compressing the feed to a pressure of 235 bar, it is then above the critical pressure and its latent heat is zero.*

Fig. 11.11 shows a schematic diagram of a Benson boiler. This boiler does not use any drum. The feed water after circulation through the economic tubes flows through the radiant parallel tube section to evaporate partly. The steam water mixture produced then moves to the transit section where this mixture is converted into steam. The steam is now passed through the convection superheater and finally supplied to the prime mover.

Boilers having as high as 650°C temperature of steam had been put into service. The maximum working pressure obtained so far from commercial Benson boiler is 500 atm. The Benson boilers of 150 tonnes/h generating capacity are in use.

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Since so many different principles are used in combustion equipment, Table 11.1 gives the more important manufactured types of stokers and burners.

Table 11.1. Important Types of Stokers and Burners

- Fluid fuels (Burners)
  - Gas
    - Multiple jet
    - Fan mix
    - Pre-mix
      - Multiple burner
      - Replaceable tip
      - Wide range tip
  - Oil
    - Pressure atomizing
      - Replaceable tip
      - Wide range tip
    - Steam atomizing
      - Outside mix
      - Inside mix
    - Rotary cup
      - Inside mix
    - Vaporiser
      - Wick
      - Hot plate
  - Crushed and finely sized coal
    - Pulverized coal
      - Short flame
      - Long flame
      - Tangential
    - High turbulence furnace (cyclone)
- Lump coal stokers
  - Overfeed
    - Spreader stoker
      - Mechanical throw
        - Overthrow
          - Dump grate
          - Travelling grate
        - Underthrow
          - Dump grate
          - Travelling grate
      - Jet throw
        - Air jet
        - Steam jet
    - Conveyor stoker
      - Travelling grate
        - Natural draft
        - Forced draft
      - Chain grate
        - Natural draft
        - Forced draft
  - Underfeed
    - Horizontal retort
      - Single
        - Ram feed
          - Grates stationary
          - Grates agitated
        - Screw feed
          - Grates stationary
          - Grates agitated
      - Twin
        - Ram feed
          - Grates stationary
          - Grates agitated
        - Screw feed
          - Grates stationary
          - Grates agitated
    - Sloping retort—Multiple report, large capacity.

The fuels are mainly bituminous coal, fuel oil, and natural gas mentioned in order of importance. All are composed of hydrocarbons, and coal has, as well much fixed carbon and little sulphur. To burn these fuels to the desired and products, CO_2 and H_2O, requires (*i*) *air in sufficient proportions,* (*ii*) *a good mixing of the fuel and air,* (*iii*) *a turbulence or relative motion between fuel and air.* The combustion equipment *must fulfill these requirements and, in addition, be capable of close regulation of rate of firing the fuel, for boilers ordinarily operate on variable load.* Coal-firing equipment *must also have a means for holding and discharging the ash residue.*

The basic requirements of combustion equipment are :

1. Thorough mixing of fuel and air.
2. Optimum fuel-air ratios leading to most complete combustion possible maintained over full load range.
3. Ready and accurate response of rate fuel feed to load demand (usually as reflected in boiler steam pressure).
4. Continuous and reliable ignition of fuel.

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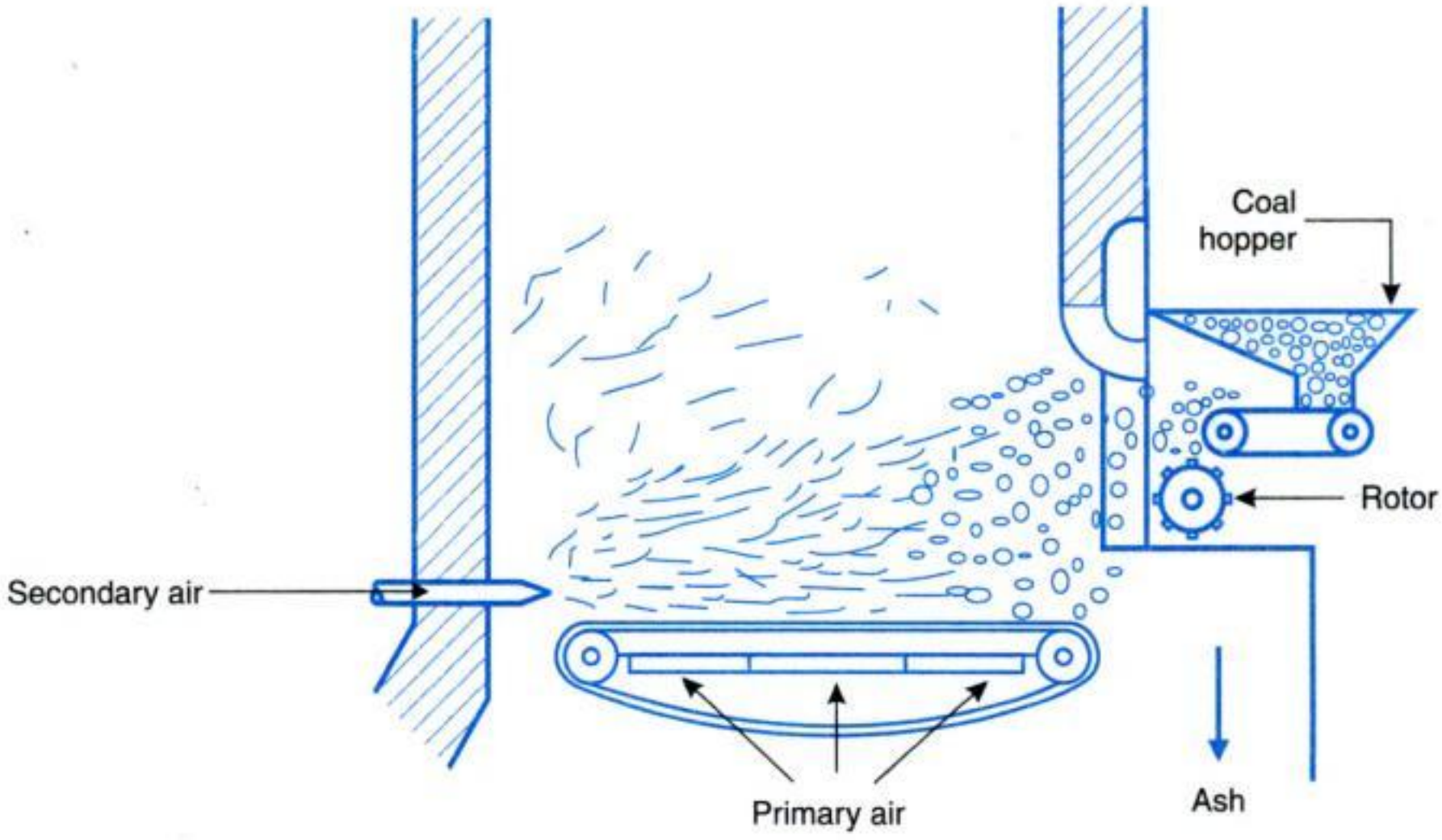

Fig. 11.15. Spreader stoker.

Disadvantages :

1. It is always difficult to operate spreader with varying sizes of coal with varying moisture content.
2. Fly-ash is much more.

2. Underfeed stokers

Refer Fig. 11.16. In these stokers, the coal is fed into the furnace below the point of air admission. Coal from the hopper is pushed into the retort by means of reciprocating plunger. When coal gets heated, all the volatiles in it are distilled and when coal reaches the zone of active combustion, it is in the form of coke and ash. The ash discharge plates are at the back of the furnace and by the time coal is pushed down on to those, all the combustion has been completed. Air is admitted into the furnace through holes in the sides of the retort. The coal is continuously agitated by the plunger and also by three pusher plates along the bottom of the retort. Due to this, the fuel bed remains porous and free from clinkers.

Underfeed stokers are suitable for *non-clinkering, high volatile coals having coking properties and low ash content*.

Advantages :

1. Give higher thermal efficiency compared with chain grate stokers.
2. Combustion rate is considerably higher.
3. The grate is self cleaning.
4. Different variety of coals can be used.
5. Grate bars, tuyeres and retorts are not subjected to high temperature as they remain always in contact with fresh coal.
6. Smokeless operation is possible even at very light load.

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Oil Burning

Fig. 11.19 shows the simple method of oil burning.

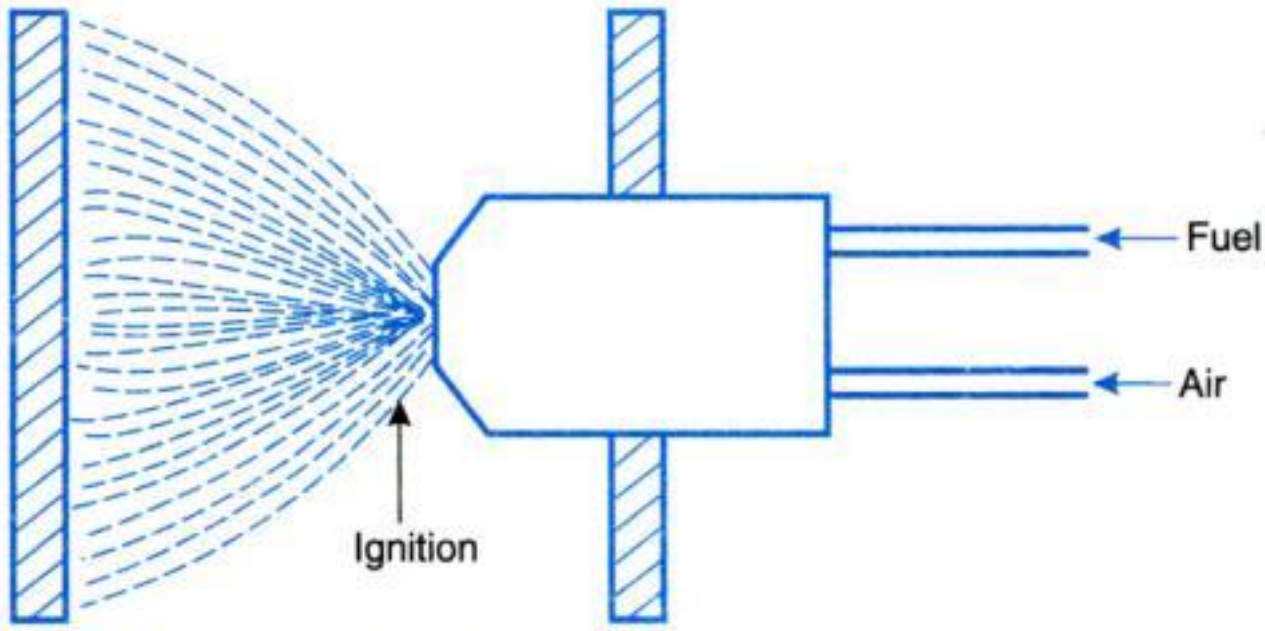

Fig. 11.19. Principle of oil firing.

The functions of the oil burner are to mix the fuel and air in the proper proportion and to prepare the fuel for combustion. Following are the two ways to achieve it :

1. The oil may be vaporised or gasified by heating within the burner.
2. The oil may be atomised by the burner and its vaporisation occurs in the combustion space.

Vaporising burners find little use as there are limited range of fuels they can handle. In the second arrangement, the atomisation of the oil is done in three basic ways :

(*i*) The oil is broken into small droplets by using air or steam under pressure.

(*ii*) Forcing oil under pressure through a suitable nozzle.

(*iii*) Tearing an oil film into drops by centrifugal force.

Gas burning. Gas burning is much simpler as the fuel is ready for combustion and requires no preparation. The remaining parts of the job *i.e.,* proportioning, mixing and burning can be achieved in many ways. The most simple and familiar gas burner is the atmospheric burner. In this the momentum of the incoming low pressure gas stream is used to draw in or aspirate the air needed for combustion. Gas and air together pass through a tube leading to the burner ports, mixing in the process. The mixture burns at the ports or the openings in the burner head. Secondary air is drawn into the flame from the surrounding atmosphere. Larger counterparts of this type having ring or sectional burning heads with many ports are used to fire small boilers.

Example 11.1. *Sketch and describe the working of a once-through boiler. Discuss its special features.* **(U.P.S.C., 1997)**

Solution.

- The **once-through boiler** or steam generator is also called the *forced circulation, Benson or universal pressure boiler* because it is *applicable to all temperature and pressures, although economically it is suited to large sizes and pressures in the high subcritical and supercritical range.*
- In a once through boiler, in contrast to the drum type [Fig. 11.20], the feedwater goes through the economiser, furnace walls, and superheat sections, changing sequentially to saturated water, saturated steam, and superheated steam in *one continuous pass* [Fig. 11.21]. *No. steam drum is required to separate saturated steam from boiling water and no water circulation takes place.* Reheat of steam after it is expanded in the high-pressure turbine is accomplished by a *reheater* in the usual manner.

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12 Boiler Mountings and Accessories

12.1. Introduction. 12.2. Boiler mountings—Water level indicator—Pressure gauge—Safety valves—High steam and low water safety valve—Fusible plug—Blow-off cock—Feed check valve—Junction or stop valve. 12.3. Accessories—Feed pumps—Injector—Economiser—Air preheater—Superheater—Steam separator—Steam trap—Highlights—Objective Type Questions—Theoretical Questions.

12.1. INTRODUCTION

Boiler Mountings. These are different fittings and devices which are *necessary for the operation and safety of a boiler*. Usually these devices are *mounted over boiler shell*.

In accordance with the Indian boiler regulation the following *mountings* should be fitted to the boilers.

- Two safety valves
- Two water level indicators
- A pressure gauge
- A steam stop valve
- A feed check valve
- A blow-off cock
- An attachment for inspector's test gauge
- A man hole
- Mud holes or sight holes.

Boilers of Lancashire and Cornish type should be fitted with a *high pressure and low water safety valve.*

All land boilers should have a *fusible plug* in each furnace.

Boiler Accessories. These are *auxiliary plants required for steam boilers for their proper operation and for the increase of their efficiency.* Commonly used boiler *accessories* are :

- Feed pumps
- Injector
- Economiser
- Air preheater
- Superheater
- Steam separator
- Steam trap.

12.2. BOILER MOUNTINGS

The various boiler mountings are discussed as follows :

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12.2.3. Safety Valves

The function of a safety valve is to release the excess steam when the pressure of steam inside the boiler exceeds the rated pressure. As soon as the pressure of steam inside the boiler exceeds the rated pressure the safety valve automatically opens and excess steam rushes out into the atmosphere till the pressure drops down to the normal value. A safety valve is generally mounted on the top of the shell.

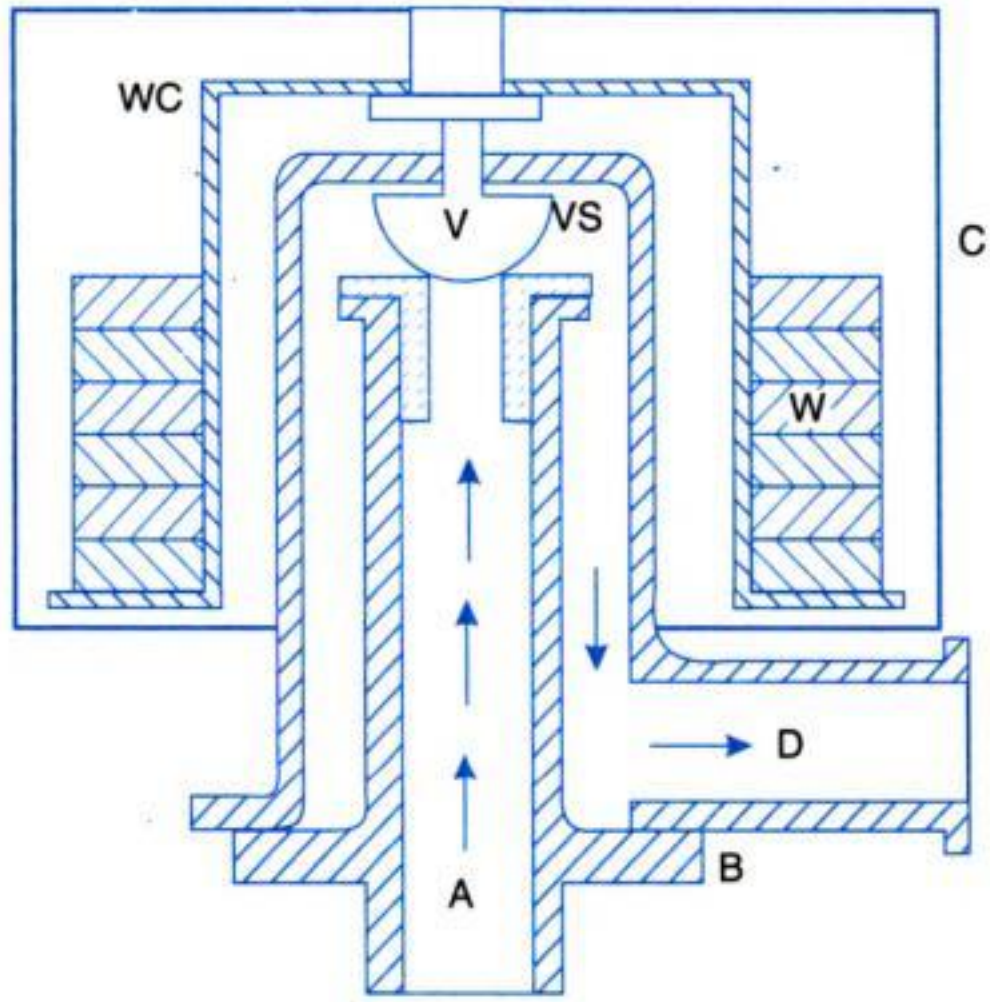
As per boiler regulations every boiler must be fitted at least with two safety valves.

The various types of safety valves are enumerated and discussed as follows :

1. Dead weight safety valve.
2. Lever safety valve.
3. Spring loaded safety valve.
4. High steam and low water safety valve.

12.2.3.1. Dead Weight Safety Valve

Fig. 12.4 shows a dead weight safety valve. *A* is the vertical cast iron pipe through which steam pressure acts. *B* is the bottom flange directly connected to seating block on the boiler shell communicating to the steam space. *V* is the gun metal valve and *VS* is the gun metal valve seat. *D* is another cast iron pipe for discharge of excess steam from the boiler. *W* are the weights in the form of cylindrical disc of cast iron. *WC* is the weight carrier carrying the weights *W*. The cover plate *C* covers these weights. The steam pressure acts in the upward direction and is balanced by the force of the dead weights *W*. The total dead-weights consist of the sum of the weights *W*, weight of the valve *V*, weight of the weight carrier and weight of the cover plate *C*.



A = Cast iron pipe
B = Bottom flange
V = Gun metal valve
W = Cast iron weights
WC = Weight carrier
D = Discharge pipe
VS = Gun metal valve seat
C = Cover plate

Fig. 12.4. Dead weight safety valve.

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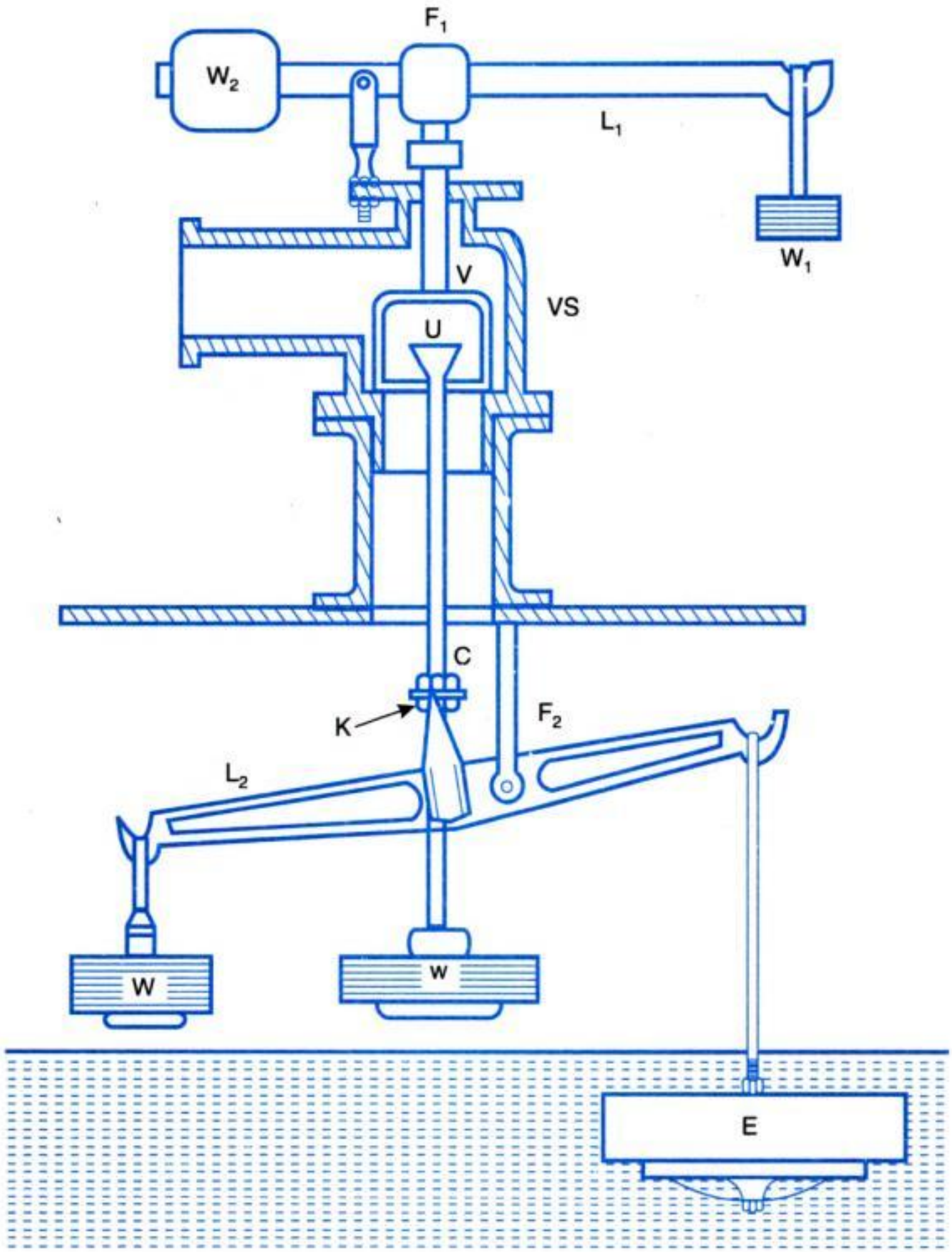

W_1 = Main weight
W_2 = Counter weight
L_1, L_2 = Levers
V = Valve
U = Hemispherical valve
VS = Valve seat
F_1 = Fulcrum (upper lever L_1)
F_2 = Fulcrum (lower level L_2)
E = Float
K = Knife edge
C = Spindle
w = Dead weight
W = Balance weight

Fig. 12.8. High steam and low water safety valve.

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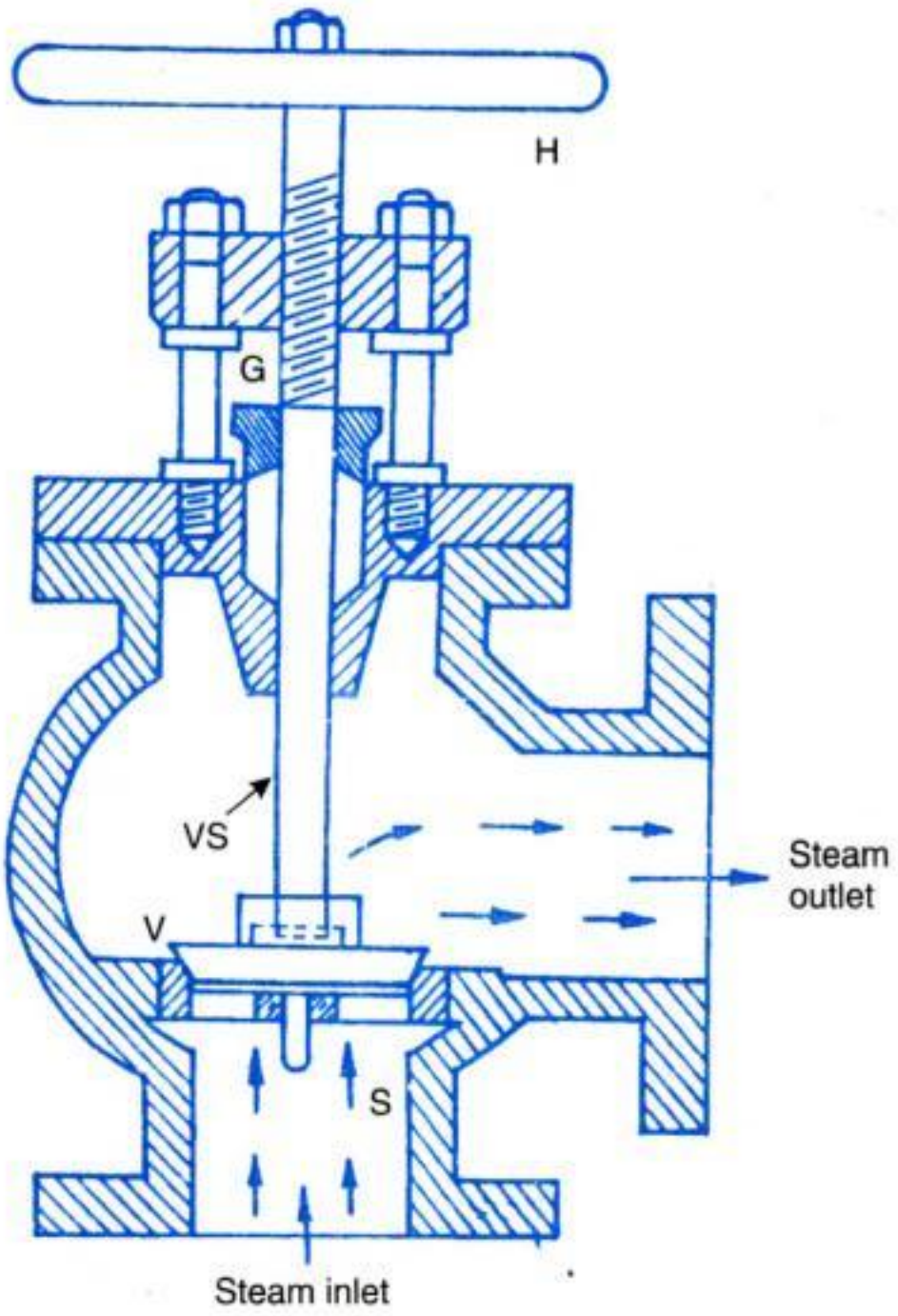

H = Hand wheel
VS = Valve spindle
G = Gland
V = Valve
S = Valve seat

Fig. 12 12. Stop valve.

Note. In a locomotive boiler the supply of steam is regulated by means of a regulator which is placed inside the boiler shell.

12.3. ACCESSORIES

Commonly used accessories are discussed as follows :

12.3.1. Feed Pumps

The feed pump is a pump which is used to *deliver feed water to the boiler.* It is desirable that the quantity of water supplied should be *at least equal to that evaporated and supplied to the engine*. Two types of pumps which are commonly used as feed pumps are (*i*) reciprocating pump and (*ii*) rotary pump.

The reciprocating pump consists of a pump cylinder and a piston. Inside the cylinder reciprocates a piston which displaces water. The reciprocating pump may be of two types :

1. Single acting pump
2. Double acting pump

In a single acting pump the water is displaced by one side of the piston only and so the water is discharged in alternate strokes.

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Fig. 12.15 shows an independent type vertical tube economiser (called Green's economiser). It is employed for boilers of medium pressure range upto about 25 bar. It consists of a large number of vertical cast iron pipes P which are connected with two horizontal pipes, one at the top and the other at the bottom. A is the bottom pipe through which the feed water is pumped into the economiser. The water comes into the top pipe B from the bottom pipe (*via* vertical pipes) and finally flows to the boiler. The flue gases move around the pipes in the direction opposite to the flow of water. Consequently, heat transfer through the surfaces of the pipes takes place and water is thereby heated.

A blow-off cock is provided at the back end of vertical pipes to remove sediments deposited in the bottom boxes. The soot of the flue gases which gets deposited on the pipes reduces the efficiency of the economiser. To prevent the soot deposit, the scrapers S move up and down to keep the external surface of the pipe clean (for better heat transfer).

By-pass arrangement (Fig. 12.16) *enables to isolate or include the economiser in the path of flue gases.*

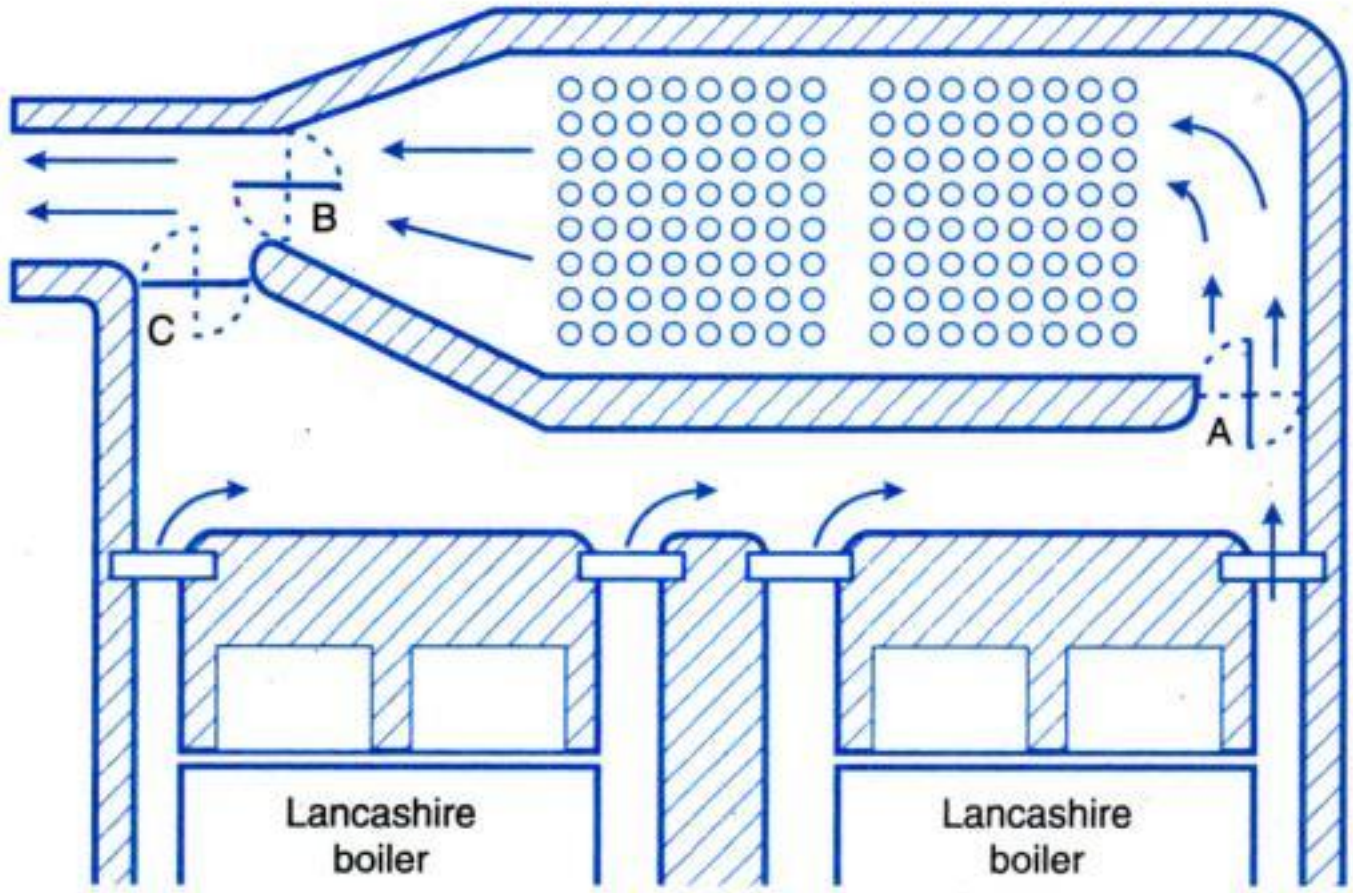

Fig. 12.16. By-pass arrangement of flues.

The use of an economiser entails the following *advantages* :

1. The *temperature range* between various parts of the boiler is *reduced* which results in *reduction of stresses* due to unequal expansion.
2. If the boiler is fed with cold water it may result in *chilling* the boiler metal. Hot feed water *checks* it.
3. *Evaporative capacity* of the boiler is *increased.*
4. *Overall efficiency* of the plant is *increased.*

12.3.4. Air Preheater

The function of the air pre-heater is to increase the temperature of air before it enters the furnace. It is generally *placed after the economiser* ; so the flue gases pass through the economiser and then to the air preheater.

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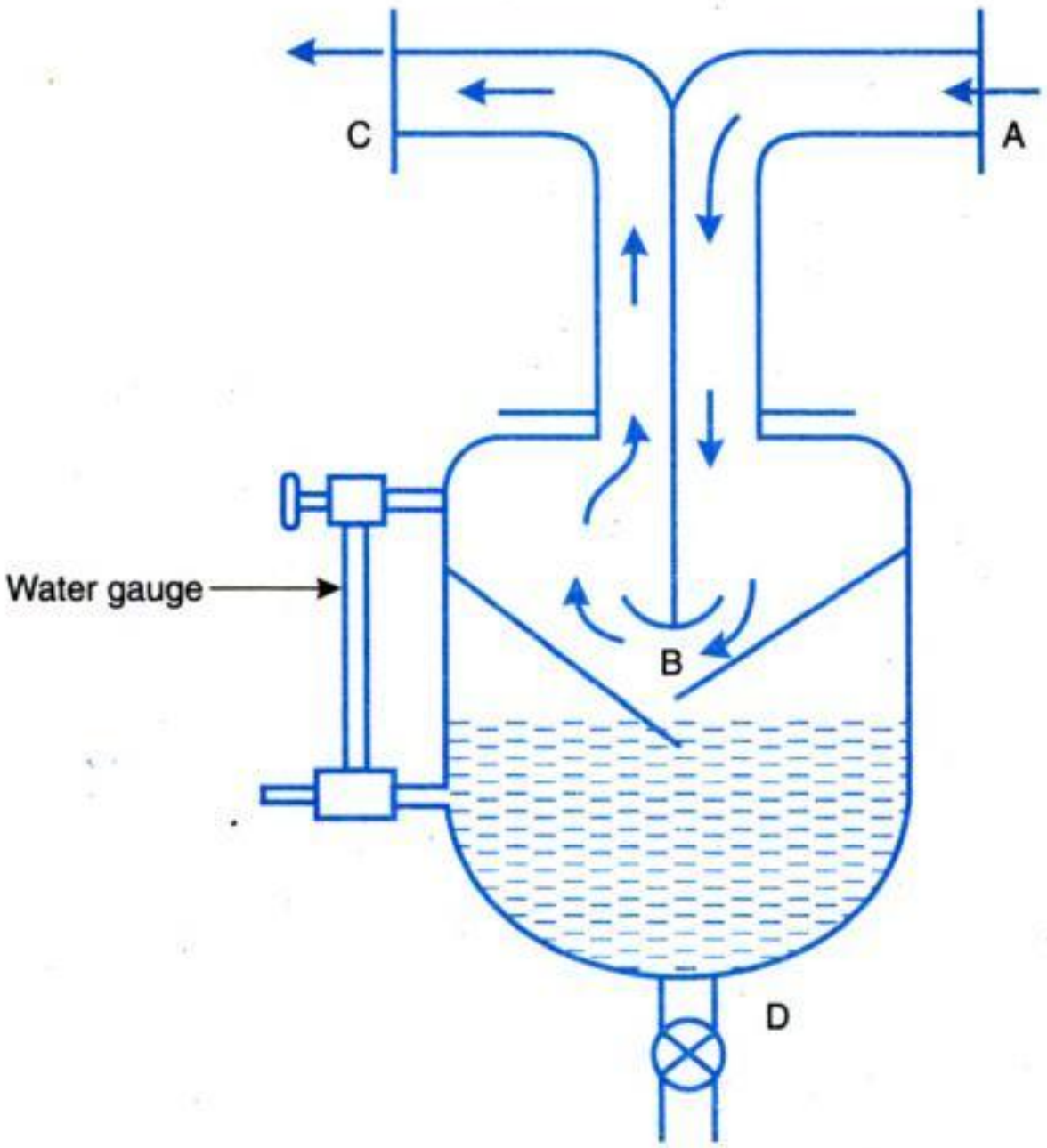
besides involving some risk ; hence it is usual to endeavour to separate any water that may be present from the steam before the latter enters the engine. This is accomplished by the use of a *steam separator*. Thus *the function of a steam separator is to remove the entrained water particles from the steam conveyed to the steam engine or turbine.* It is installed as close to the steam engine as possible on the main steam pipe from the boiler.

According to the principle of operation the steam separators are classified as follows :

1. Impact or baffle type
2. Reverse current type
3. Centrifugal type.



A, *C* = Flanges *D* = Drain pipe
B = Baffles

Fig. 12.20. Baffle plate steam separator.

Fig. 12.20 shows *baffle plate steam separator.* The steam enters the flange *A* and flows down. In its passage it strikes the baffles *B* ; as a result it gets deflected, but water particles having greater density and greater inertia fall to the bottom of the separator. The drier steam discharges through the flange *C*. To see the level of water collected a water gauge is provided. The water collected in the vessel is removed at intervals through the drain pipe *D*.

12.3.7. Steam Trap

The function of a steam trap is to drain away automatically the condensed steam from the steam pipes, steam jackets and steam separators without permitting any steam to escape.

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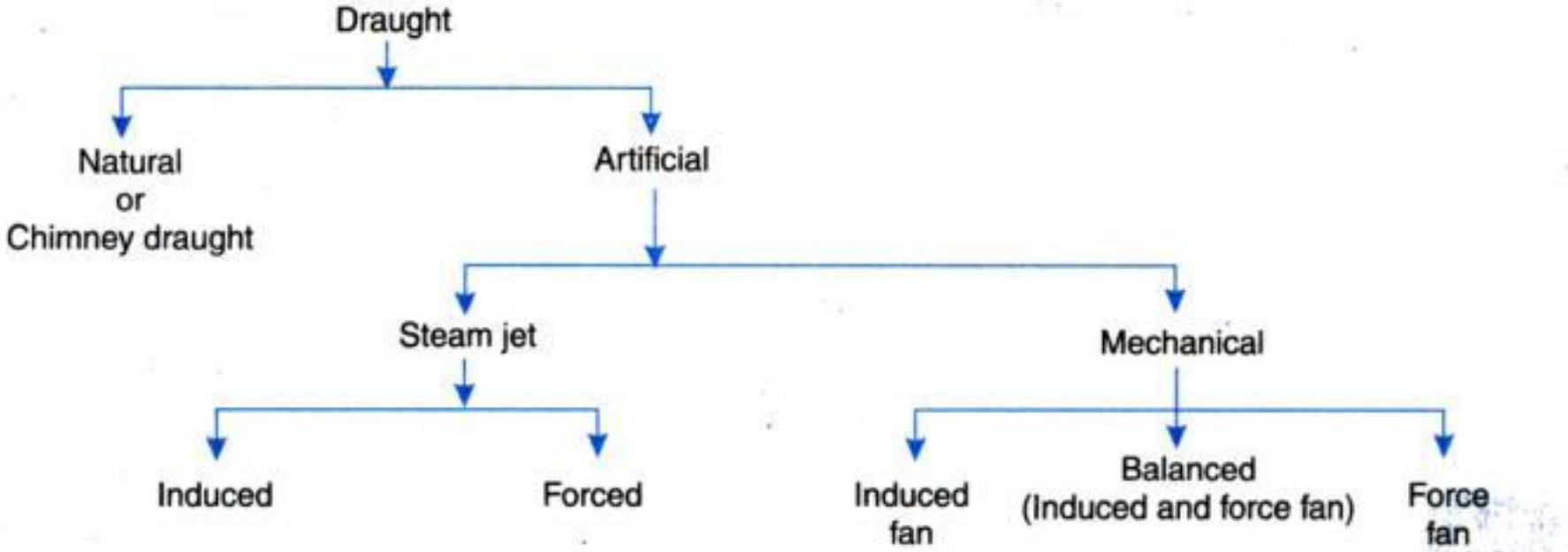
13 Draught

13.1. Definition and classification of draught. 13.2. Natural draught—Chimney. 13.3. Chimney height and diameter. 13.4. Condition for maximum discharge through a chimney. 13.5. Efficiency of a chimney. 13.6. Draught losses. 13.7. Artificial draught—Forced draught—Induced draught—Balanced draught—Advantages of mechanical draught—Power required to drive fan—Steam jet draught——Worked Examples— Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

13.1. DEFINITION AND CLASSIFICATION OF DRAUGHT

The small pressure difference which causes a flow of gas to take place is termed as a ***draught***. The function of the draught, in case of a boiler, is to force air to the fire and to carry away the gaseous products of combustion. In a boiler furnace proper combustion takes place only when sufficient quantity of air is supplied to the burning fuel.

The draught may be classified as :



13.2. NATURAL DRAUGHT—CHIMNEY

Natural draught is obtained by the use of a chimney. The chimney in a boiler installation performs one or more of the following functions : (*i*) *It produces the draught whereby the air and gas are forced through the fuel bed, furnace, boiler passes and settings ; (ii) It carries the* products of combustion to such a height before discharging them that they will not be objectionable or injurious to surroundings. A *chimney* is vertical tubular structure built either of masonry, concrete or steel. *The draught produced by the chimney is due to the density difference between the column of hot gases inside the chimney and the cold air outside.*

Fig. 13.1 shows a diagrammatic arrangement of a chimney of height '*H*' metres above the grate.

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The value of m_g will be *maximum*, if

$$\frac{dm_g}{dT_g} = 0 \text{ as } T_a \text{ and } m_a \text{ are fixed quantities}$$

$$\therefore \quad \frac{d}{dT_g}\left[\frac{K}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1}\right] = 0$$

or
$$\frac{d}{dT_g}\left[\frac{1}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a}-1}\right] = 0$$

or
$$\frac{d}{dT_g}\left[\frac{(ZT_g-1)^{1/2}}{T_g}\right] = 0,$$

where $Z = \dfrac{m_a}{m_a+1}\cdot\dfrac{1}{T_a}$

or
$$\frac{d}{dT_g}\left[(ZT_g-1)^{1/2}\times T_g^{-1}\right] = 0$$

or
$$(Z\,.\,T_g-1)^{1/2}\,.\,(-1)(T_g)^{-2} + T_g^{-1}\times\tfrac{1}{2}(ZT_g-1)^{-1/2}\times Z = 0$$

or
$$\frac{-(ZT_g-1)^{1/2}}{T_g^{\,2}} + \frac{Z}{2T_g(ZT_g-1)^{1/2}} = 0$$

or
$$\frac{-2(ZT_g-1)+ZT_g}{2(T_g)^3(ZT_g-1)^{1/2}} = 0$$

or
$$-2(ZT_g-1)+ZT_g = 0$$

or
$$-2ZT_g+2+ZT_g = 0$$

or
$$ZT_g = 2$$

or
$$\frac{m_a}{m_a+1}\cdot\frac{T_g}{T_a} = 2$$

i.e.,
$$\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right) \qquad ...(13.13)$$

Thus we see that the *absolute temperature of the chimney gases bears a certain ratio to the absolute temperature of the outside air.*

Putting the value of $\dfrac{T_g}{T_a}$ in eqn. (13.6), we get

$$(H_1)_{max} = H\left[\left(\frac{m_a}{m_a+1}\right)\times 2\left(\frac{m_a+1}{m_a}\right)-1\right]$$

$$= H\,(2-1) = H$$

i.e.,
$$(H_1)_{max} = H \qquad ...(13.14)$$

The draught in mm of water column for maximum discharge can be evaluated by inserting the value of T_g/T_a in eqn. (13.7)

$$\therefore \quad (h_w)_{max} = 353\,H\left(\frac{1}{T_a}-\frac{1}{2T_a}\right) = \frac{176.5H}{T_a} \text{ mm of water} \qquad ...(13.15)$$

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$$= \frac{M(m_a + 1)}{\rho_g} = \frac{M(m_a + 1)T_g V_o m_a}{273\,(m_a + 1)} = \frac{M m_a T_g V_o}{273}$$

Substituting in eqn. (13.18), we get

$$\text{Power of I.D. fan,} \quad P_{ID} = \frac{2.725 \times 10^{-6} \times h M m_a T_g V_o}{273 \times \eta_f}$$

$$= 0.998 \times 10^{-8} \frac{h V m_a M T_g}{\eta_f} \qquad \text{...(13.20)}$$

Assuming *same efficiency*, for the *same draught* (neglecting leakage), we have

$$\frac{\text{Power of I.D. fan}}{\text{Power of F.D. fan}} = \frac{T_g}{T_a} \qquad \text{...(13.21)}$$

13.7.6. Steam Jet Draught

Steam jet draught is a simple and easy method of producing artificial draught. It may be forced or induced type depending upon where the steam jet to produce draught is located. If the steam jet is *directed into the smoke box* near the stack, the air is *induced* through the flues, the grate and ash pit to the smoke box. If the jet is *located before the grate,* air is *forced* through the fuel bed, furnace and flues to the chimney.

The steam jet draught entails the following *advantages* :

(*i*) Very simple and economical.

(*ii*) Occupies minimum space.

(*iii*) Requires very little attention.

(*iv*) In forced type steam jet draught, the steam keeps the fire bars cool and prevents the adhering of clinker to the fire bars.

(*v*) Several classes of low grade fuels can be used with this system.

WORKED EXAMPLES

Example 13.1. *Calculate the height of chimney required to produce a draught equivalent to 1.7 cm of water if the flue gas temperature is 270°C and ambient temperature is 22°C and minimum amount of air per kg of fuel is 17 kg.*

Solution. Draught in mm of H_2O, $h_w = 1.7 \text{ cm} = 17 \text{ mm}$

Flue gas temperature, $T_g = 270 + 273 = 543 \text{ K}$

Ambient temperature, $T_a = 22 + 273 = 295 \text{ K}$

Minimum amount of air per kg of fuel, $m_a = 17 \text{ kg}$

Height of chimney, H :

Using the relation,

$$h_w = 353\, H \left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a}\right)\right] \text{ mm of } H_2O$$

$$17 = 353\, H \left(\frac{1}{295} - \frac{1}{543} \times \frac{18}{17}\right) \text{ mm of } H_2O$$

$$= 353\, H\, (0.00339 - 0.00195) = 0.508\, H \text{ or } H = 33.46 \text{ m}$$

Hence, *height of chimney* = **33.46 m. (Ans.)**

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Temperature of flue gases, $T_g = 310 + 273 = 583$ K.

Height of chimney, H :

Draught required is equivalent to overcome the losses and velocity head

$= 7 + 7 + 3 + 3 + 1.3 = 21.3$ mm of water.

Actual draught to be produced,

$$h_w = \frac{21.3}{0.8} = 26.62 \text{ mm of water}$$

$$h_w = 353\, H \left(\frac{1}{T_a} - \frac{1}{T_g} \cdot \frac{m_a + 1}{m_a} \right)$$

$$26.62 = 353\, H \left(\frac{1}{308} - \frac{1}{583} \times \frac{14}{13} \right)$$

$$= H\,(1.146 - 0.652) = 0.494\, H \text{ or } H = 53.88 \text{ m}$$

Diameter of the chimney, D :

$$\rho_g = \frac{353}{T_g} \left(\frac{m_a + 1}{m_a} \right) = \frac{353}{583} \times \frac{14}{13} = 0.652 \text{ kg/m}^3$$

$$\text{Flue gases formed per second} = \frac{1800 \times 14}{3600} = 7 \text{ kg}$$

$$m_g = A \times C \times \rho_g \qquad \ldots(i)$$

But $C = \sqrt{2gH_1}$

where H_1 is the equivalent velocity expressed in m of gas

$$H_1 \rho_g = h_w \rho_w$$

where h_w is the water head equivalent to velocity head responsible for giving velocity to the gas,

$$\therefore \quad H_1 = \frac{h_w \rho_w}{\rho_g} = \frac{1.3 \times 1000}{1000 \times 0.652} = 1.993 \text{ m}$$

$$\therefore \quad C = \sqrt{2 \times 9.81 \times 1.993} = 6.25 \text{ m/s}$$

Substituting this value in eqn. (*i*), we get

$$7 = \frac{\pi}{4} D^2 \times 6.25 \times 0.625 \text{ or } D^2 = \frac{7 \times 4}{\pi \times 6.25 \times 0.652}$$

$$\therefore \quad D = \mathbf{1.478 \text{ m.}} \quad \textbf{(Ans.)}$$

Example 13.7. *How much air is used per kg of coal burnt in a boiler having chimney of 32.3 m height to create a draught of 19 mm of water column when the temperature of flue gases in the chimney is 370°C and the temperature of the boiler house is 29.5°C ?*

(AMIE Winter, 2000)

Solution. Height of the chimney, $H = 32.3$ m

Draught of water, $h_w = 19$ mm

Temperature of flue gases in the chimney, $T_g = 370 + 273 = 643$ K

Ambient temperature, $T_a = 29.5 + 273 = 302.5$ K

Air used per kg of coal burnt, $\mathbf{m_a}$:

We know that $h_w = 353\, H \left[\frac{1}{T_a} - \frac{1}{T_g} \left(\frac{m_a + 1}{m_a} \right) \right]$

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(*c*) remains the same irrespective of outside air temperature

(*d*) may increase or decrease with increase in outside air temperature.

5. The draught produced by chimney of given height at given outside temperature

(*a*) decreases if the chimney gas temperature increases

(*b*) increases if the chimney gas temperature increases

(*c*) remains same irrespective of chimney gas temperature

(*d*) may increase or decrease.

6. For forced draught system, the function of chimney is mainly

(*a*) to produce draught to accelerate the combustion of fuel

(*b*) to discharge gases high up in the atmosphere to avoid hazard

(*c*) to reduce the temperature of the hot gases discharged

(*d*) none of the above.

7. Artificial draught is produced by

(*a*) induced fan (*b*) forced fan

(*c*) induced and forced fan (*d*) all of the above.

8. The draught in locomotive boilers is produced by

(*a*) forced fan (*b*) chimney

(*c*) steam jet (*d*) only motion of locomotive.

9. For the same draught produced the power of induced draught fan as compared to forced draught fan is

(*a*) less (*b*) more

(*c*) same (*d*) not predictable.

10. Artificial draught is produced by

(*a*) air fans (*b*) steam jet

(*c*) fan or steam jet (*d*) all of the above.

11. The artificial draught normally is designed to produce

(*a*) less smoke (*b*) more draught

(*c*) less chimney gas temperature (*d*) all of the above.

12. For the induced draught the fan is located

(*a*) near bottom of chimney (*b*) near bottom of furnace

(*c*) at the top of the chimney (*d*) anywhere permissible.

13. The pressure at the furnace is minimum in case of

(*a*) forced draught system (*b*) induced draught system

(*c*) balanced draught system (*d*) natural draught system.

14. For maximum discharge of hot gases through the chimney the height of hot gas column producing draught is

(*a*) twice the height of chimney (*b*) equal to the height of chimney

(*c*) half the height of chimney (*d*) none of the above.

15. The efficiency of chimney is approximately

(*a*) 80% (*b*) 40%

(*c*) 20% (*d*) 0.25%.

16. In balanced draught system the pressure at force fan inlet

(*a*) is greater than pressure at chimney outlet

(*b*) is less than pressure at chimney outlet

(*c*) approximately same as that at chimney outlet.

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The evaporation rate of the boiler is also sometimes given in terms of *kg of steam / kg of fuel.*

The presently accepted standard of expressing the capacity of a boiler is in terms of the total heat added per hour.

14.3. FACTOR OF EVAPORATION

It is defined as *the ratio of heat received by 1 kg of water under working conditions to that received by 1 kg of water evaporated from and at 100°C.* It is denoted by F_e.

$$\therefore \quad \text{Factor of evaporation,} \qquad F_e = \frac{h - h_{f_1}}{2257} \qquad \ldots(14.2)$$

14.4. BOILER EFFICIENCY

'Boiler efficiency' is the *ratio of heat actually utilised in generation of steam to the heat supplied by the fuel in the same period.*

$$i.e., \qquad \text{Boiler efficiency} = \frac{m_a (h - h_{f_1})}{C} \qquad \ldots(14.3)$$

where, m_a = Mass of water actually evaporated into steam per kg of fuel at the working pressure, and

C = Calorific value of the fuel in kJ/kg.

If the boiler, economiser, and superheater are considered as a single unit, then the *boiler efficiency is termed as overall efficiency of the boiler plant.*

The following are the *factors* on which the boiler efficiency depends :

1. Fixed factors.
2. Variable factors.

1. **Fixed Factors.** These are :

(*i*) *Boiler design.* It includes the arrangement and effectiveness of the heating surfaces, the shape and volume of the furnace, the arrangement of flues, the arrangement of steam and water circulation.

(*ii*) *Heat recovery equipment.* It includes the economiser, superheater, air preheater and feed water heater.

(*iii*) *Built in losses.* It includes the heat transfer properties of the settings and construction materials, flue gas and ash heat losses.

(*iv*) Rated rate of firing, the furnace volume and heating surface.

(*v*) Properties and characteristics of fuel burnt.

2. Variable Factors. These are :

(*i*) Actual firing rate.

(*ii*) Fuel condition as it is fired.

(*iii*) The conditions of heat absorbing surfaces.

(*iv*) Excess air fluctuations.

(*v*) Incomplete combustion and combustibles in the refuse.

(*vi*) Change in draught from the rated, due to atmospheric conditions.

(*vii*) Humidity and temperature of the combustion air.

14.5. HEAT LOSSES IN A BOILER PLANT

The following *heat losses* occur in a boiler plant :

1. Heat lost to flue gases.
2. Heat lost due to incomplete combustion.

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Solution. Mean steam pressure, p = 12 bar
Mass of steam generated = 4000 kg
Mean dryness fraction, x = 0.85
Mean feed water temperature = 30°C
Coal used = 4000 kg
Calorific value of coal, C = 33400 kJ/kg
From steam tables, *corresponding to 12 bar :*

$$h_f = 798.4 \text{ kJ/kg}, \; h_{fg} = 1984.3 \text{ kJ/kg}$$

Now, $$h = h_f + x\, h_{fg} = 798.4 + 0.85 \times 1984.3 = 2485.05 \text{ kJ/kg}$$

Heat of feed water, $$h_{f_1} = 1 \times 4.18 \times (30 - 0) = 125.4 \text{ kJ/kg}$$

Total net heat given to produce 1 kg of steam

$$= h - h_{f_1} = 2485.05 - 125.4 = 2359.65 \text{ kJ/kg.}$$

(*i*) **Factor of equivalent evaporation, F_e :**

$$F_e = \frac{h - h_{f_1}}{2257} = \frac{2359.65}{2257} = \mathbf{1.045.} \quad \textbf{(Ans.)}$$

(*ii*) **Equivalent evaporation from and at 100°C, m_e :**

$$m_e = \frac{m_a\,(h - h_{f_1})}{2257}$$

But, $$m_a = \frac{40000}{4000} = 10 \text{ kg/kg of fuel}$$

$\therefore$ $$m_e = \frac{10 \times 2359.65}{2257} = \mathbf{10.45 \text{ kg of steam/kg of coal.}} \quad \textbf{(Ans.)}$$

(*iii*) **Efficiency of boiler, η_{boiler} :**

$$\eta_{boiler} = \frac{m_a\,(h - h_{f_1})}{C} = \frac{10 \times 2359.65}{33400} = \mathbf{0.7065 \text{ or } 70.65\%.} \quad \textbf{(Ans.)}$$

Example 14.5. *A steam generator evaporates 18000 kg/h of steam at 12.5 bar and a quality of 0.97 from feed water at 105°C, when coal is fired at the rate of 2040 kg/h. If the higher calorific value of the coal is 27400 kJ/kg, find :*

(*i*) *The heat rate of boiler in kJ/h ;*
(*ii*) *The equivalent evaporation ;*
(*iii*) *The thermal efficiency.*

Solution. Steam generated, m = 18000 kg/h
Steam pressure, p = 12.5 bar
Quality of steam, x = 0.97
Feed water temperature = 105°C
Rate of coal firing, m_f = 2040 kg/h
Higher calorific value (H.C.V.) of coal, C = 27400 kJ/kg
(*i*) **Heat rate of boiler** :
At 12.5 bar : From steam tables,

$$h_f = 806.7 \text{ kJ/kg}, \; h_{fg} = 1977.4 \text{ kJ/kg}$$

$\therefore$ $$h = h_f + x h_{fg} = 806.7 + 0.97 \times 1977.4 = 2724.78 \text{ kJ/kg}$$

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Dryness fraction of steam = *0.95*
Coal consumption = *600 kg/hour*
Calorific value of coal = *30400 kJ/kg*
Feed water supplied to boiler = *4800 kg/hour*
Determine equivalent evaporation of boiler from and at 100°C. **(P.U.)**

Solution. Given : p = 5 bar, x = 0.95, C = 30400 kJ/kg

From steam tables, corresponding to 5 bar :

h_f = 640.1 kJ/kg, h_{fg} = 2107.4 kJ/kg

Now, $h = h_f + x h_{fg}$

$= 640.1 + 0.95 \times 2107.4 = 2642.13$ kJ/kg

Heat of feed water, $h_{f_1} = 1 \times 4.18 \times (50 - 0)$

$= 209$ kJ/kg

Equivalent evaporation from and at 100°C, m_e :

$$m_e = \frac{m_a (h - h_{f_1})}{2257}$$

But $m_a = \dfrac{4800}{600}$

$= 8$ kg/kg of fuel

$$\therefore \quad m_e = \frac{8(2642.13 - 209)}{2257}$$

= 8.624 kg of steam/kg of coal. **(Ans.)**

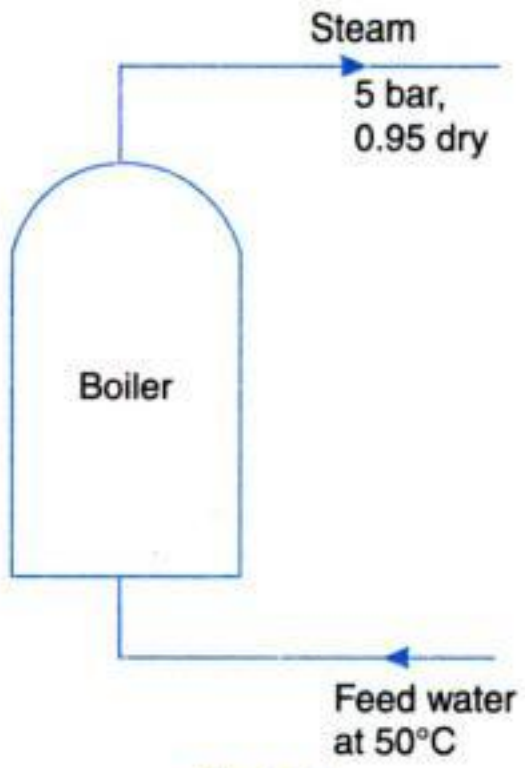

Fig. 14.1

HIGHLIGHTS

1. The *evaporative capacity* of a boiler may be expressed in terms of :
   (*i*) kg of steam/h ; (*ii*) kg of steam/h-m^2 of heating surface ; and (*iii*) kg of steam/kg of fuel fired.
2. *Equivalent evaporation* may be defined as the amount of water evaporated from water at 100°C to dry and saturated steam at 100°C.

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15 Basic Steam Power Cycles

15.1. Carnot cycle. 15.2. Rankine cycle. 15.3. Modified Rankine cycle. 15.4. Regenerative cycle. 15.5. Reheat cycle. 15.6. Binary vapour cycle—Additional/Typical Worked Examples—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

15.1. CARNOT CYCLE

Fig. 15.1 shows a Carnot cycle on T-s and p-V diagrams. It consists of (i) two constant pressure operations (4-1) and (2-3) and (ii) two frictionless adiabatics (1-2) and (3-4). These operations are discussed below :

1. **Operation (4-1).** 1 kg of boiling water at temperature T_1 is heated to form wet steam of dryness fraction x_1. Thus heat is absorbed at constant temperature T_1 and pressure p_1 during this operation.

2. **Operation (1-2).** During this operation steam is expanded isentropically to temperature T_2 and pressure p_2. The point '2' represents the condition of steam after expansion.

3. **Operation (2-3).** During this operation heat is rejected at constant pressure p_2 and temperature T_2. As the steam is exhausted it becomes wetter and cooled from 2 to 3.

4. **Operation (3-4).** In this operation the wet steam at '3' is compressed isentropically till the steam regains its original state of temperature T_1 and pressure p_1. Thus cycle is completed.

Refer T-s diagram :

Heat supplied at constant temperature T_1 [operation (4-1)] = area 4-1-b-a = $T_1 (s_1 - s_4)$ or $T_1 (s_2 - s_3)$.

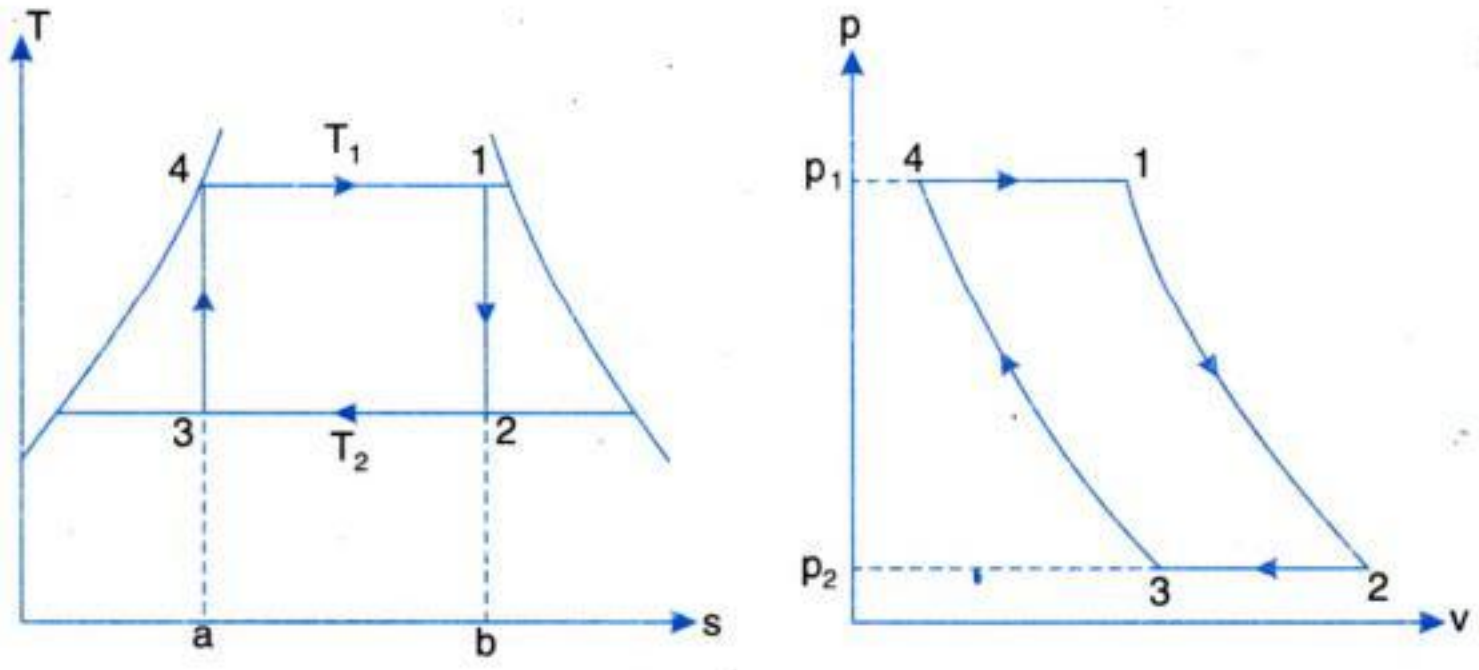

Fig. 15.1. Carnot cycle on T-s and p-V diagrams.

Heat rejected at constant temperature T_2 (operation 2-3) = area 2-3-a-b = $T_2 (s_2 - s_3)$.

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Effect of Operating Conditions on Rankine Cycle Efficiency

The Rankine cycle efficiency can be *improved* by :

(*i*) *Increasing the average temperature at which heat is supplied.*

(*ii*) *Decreasing / reducing the temperature at which heat is rejected.*

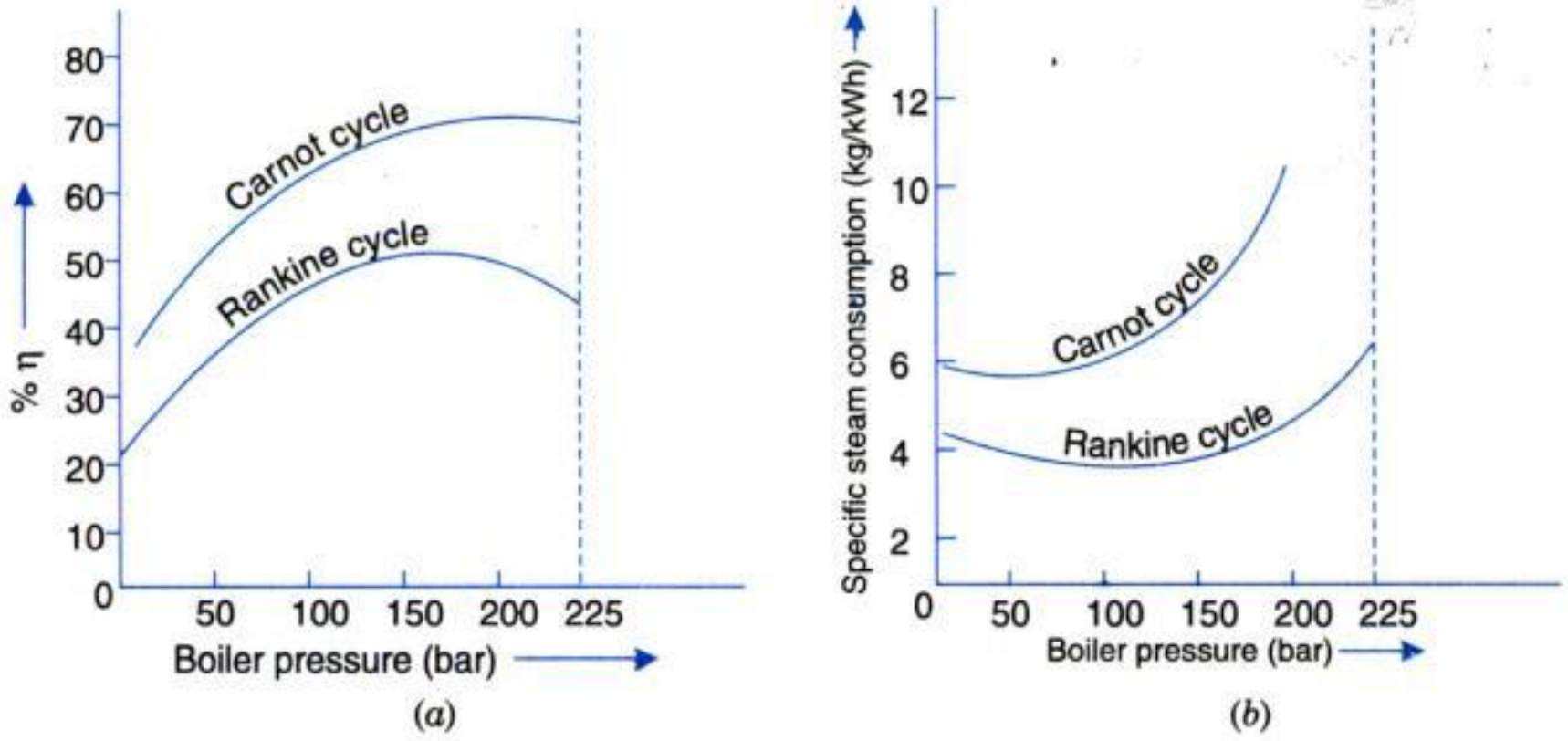


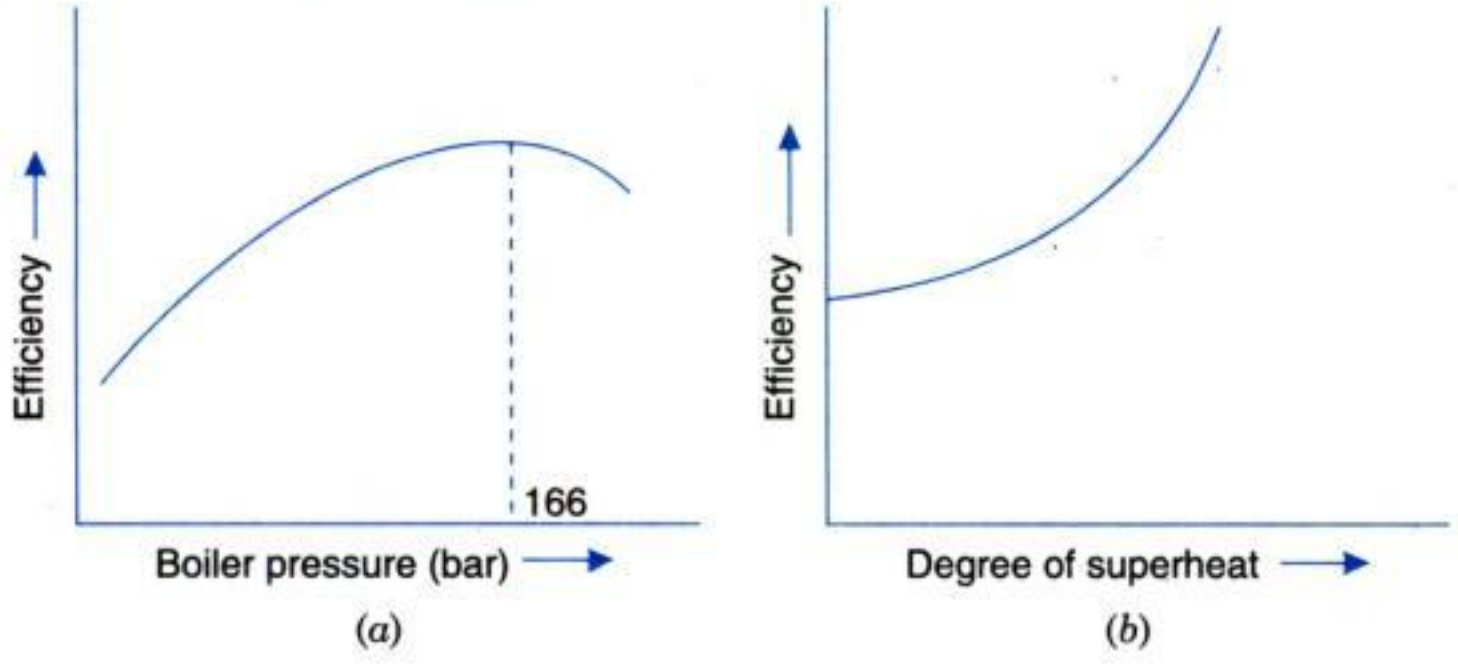
Fig. 15.4

This can be achieved by making suitable changes in the conditions of steam generation or condensation, as discussed below :

1. ***Increasing boiler pressure.*** It has been observed that by increasing the boiler pressure (other factors remaining the same) the cycle tends to rise and reaches a maximum value at a boiler pressure of about 166 bar [Fig. 15.5 (*a*)].

2. ***Superheating.*** All other factors remaining the same, if the steam is superheated before allowing it to expand the Rankine cycle efficiency may be increased [Fig. 15.5 (*b*)]. The use of superheated steam also ensures longer turbine blade life because of the absence of erosion from high velocity water particles that are suspended in wet vapour.

3. ***Reducing condenser pressure.*** The thermal efficiency of the cycle can be amply improved by reducing the condenser pressure [Fig. 15.5 (*c*)] (hence by reducing the temperature at which heat is rejected), especially in high vacuums. But the increase in efficiency is obtained at the *increased cost of condensation apparatus*.



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Example 15.3. *In a steam turbine steam at 20 bar, 360°C is expanded to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. Assume ideal processes, find per kg of steam the net work and the cycle efficiency.*

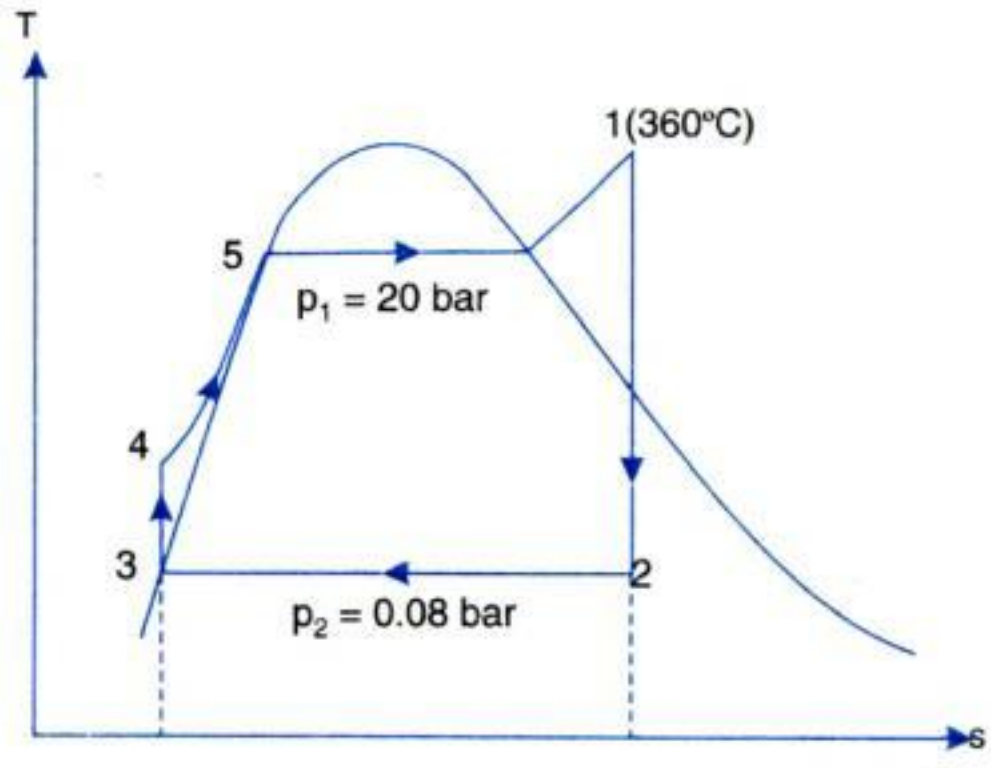

Fig. 15.7

Solution. Boiler pressure, $p_1 = 20$ bar (360°C)

Condenser pressure, $p_2 = 0.08$ bar

From steam tables :

At 20 bar (p_1), 360°C : $h_1 = 3159.3$ kJ/kg

$s_1 = 6.9917$ kJ/kg K

At 0.08 bar (p_2) : $h_3 = h_{f(p_2)} = 173.88$ kJ/kg,

$s_3 = s_{f(p_2)} = 0.5926$ kJ/kg K

$h_{fg(p_2)} = 2403.1$ kJ/kg, $s_{g(p_2)} = 8.2287$ kJ/kg K

$v_{f(p_2)} = 0.001008$ m^3/kg $\therefore$ $s_{fg(p_2)} = 7.6361$ kJ/kg K

Now $s_1 = s_2$

$6.9917 = s_{f(p_2)} + x_2\ s_{fg(p_2)} = 0.5926 + x_2 \times 7.6361$

$\therefore$ $$x_2 = \frac{0.69917 - 0.5926}{7.6361} = 0.838$$

$\therefore$ $h_2 = h_{f(p_2)} + x_2\ h_{fg(p_2)}$

$= 173.88 + 0.838 \times 2403.1 = 2187.68$ kJ/kg.

Net work, W_{net} :

$W_{net} = W_{turbine} - W_{pump}$

$W_{pump} = h_{f_4} - h_{f(p_2)}\ (= h_{f_3}) = v_{f(p_2)}\ (p_1 - p_2)$

$= 0.00108$ (m^3/kg) $\times$ (20 − 0.08) $\times$ 100 kN/m^2

$= 2.008$ kJ/kg

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From steam tables :

At 35 bar : $h_1 = h_{g_1} = 2802$ kJ/kg, $s_{g_1} = 6.1228$ kJ/kg K

At 0.26 bar : $h_f = 251.5$ kJ/kg, $h_{fg} = 2358.4$ kJ/kg,

$v_f = 0.001017$ m³/kg, $s_f = 0.8321$ kJ/kg K, $s_{fg} = 7.0773$ kJ/kg K.

(*i*) **The pump work :**

Pump work $= (p_4 - p_3)\, v_f = (35 - 0.2) \times 10^5 \times 0.001017$ J or 3.54 kJ/kg

$$\left[\begin{array}{l}\text{Also } h_{f_4} - h_{f_3} = \text{Pump work} = 3.54 \\ \therefore \quad h_{f_4} = 251.5 + 3.54 = 255.04 \text{ kJ/kg}\end{array}\right]$$

Now power required to drive the pump

$= 9.5 \times 3.54$ kJ/s or **33.63 kW.** **(Ans.)**

(*ii*) **The turbine work :**

$$s_1 = s_2 = s_{f_2} + x_2 \times s_{fg_2}$$

$$6.1228 = 0.8321 + x_2 \times 7.0773$$

$$\therefore \quad x_2 = \frac{6.1228 - 0.8321}{7.0773} = 0.747$$

$$\therefore \quad h_2 = h_{f_2} + x_2\, h_{fg_2} = 251.5 + 0.747 \times 2358.4 = 2013 \text{ kJ/kg}$$

$$\therefore \quad \textit{Turbine work} = \dot{m}\ (h_1 - h_2) = 9.5\ (2802 - 2013) = \mathbf{7495.5\ kW.} \quad \textbf{(Ans.)}$$

It may be noted that pump work (33.63 kW) is very small as compared to the turbine work (7495.5 kW).

(*iii*) **The Rankine efficiency :**

$$\eta_{\text{rankine}} = \frac{h_1 - h_2}{h_1 - h_{f_2}} = \frac{2802 - 2013}{2802 - 251.5} = \frac{789}{2550.5} = \mathbf{0.3093 \text{ or } 30.93\%.} \quad \textbf{(Ans.)}$$

(*iv*) **The condenser heat flow :**

The condenser heat flow $= \dot{m}\ (h_2 - h_{f_3}) = 9.5\ (2013 - 251.5) = \mathbf{16734.25\ kW.}$ **(Ans.)**

(*v*) **The dryness at the end of expansion, x_2 :**

The dryness at the end of expansion,

$$x_2 = \mathbf{0.747 \text{ or } 74.7\%.} \quad \textbf{(Ans.)}$$

Example 15.7. *The adiabatic enthalpy drop across the primemover of the Rankine cycle is 840 kJ/kg. The enthalpy of steam supplied is 2940 kJ/kg. If the back pressure is 0.1 bar, find the specific steam consumption and thermal efficiency.*

Solution. Adiabatic enthalpy drop, $h_1 - h_2 = 840$ kJ/kg

Enthalpy of steam supplied, $h_1 = 2940$ kJ/kg

Back pressure, $p_2 = 0.1$ bar

From steam tables, corresponding to 0.1 bar : $h_f = 191.8$ kJ/kg

Now, $$\eta_{\text{rankine}} = \frac{h_1 - h_2}{h_1 - h_{f_2}} = \frac{840}{2940 - 191.8} = 0.3056 = \mathbf{30.56\%.} \quad \textbf{(Ans.)}$$

Useful work done per kg of steam = 840 kJ/kg

$$\therefore \quad \textit{Specific steam consumption} = \frac{1}{840} \text{ kg/s} = \frac{1}{840} \times 3600 = \mathbf{4.286\ kg/kWh.} \quad \textbf{(Ans.)}$$

Example 15.8. *A 35 kW (I.P.) system engines consumes 284 kg/h at 15 bar and 250°C. If condenser pressure is 0.14 bar, determine :*

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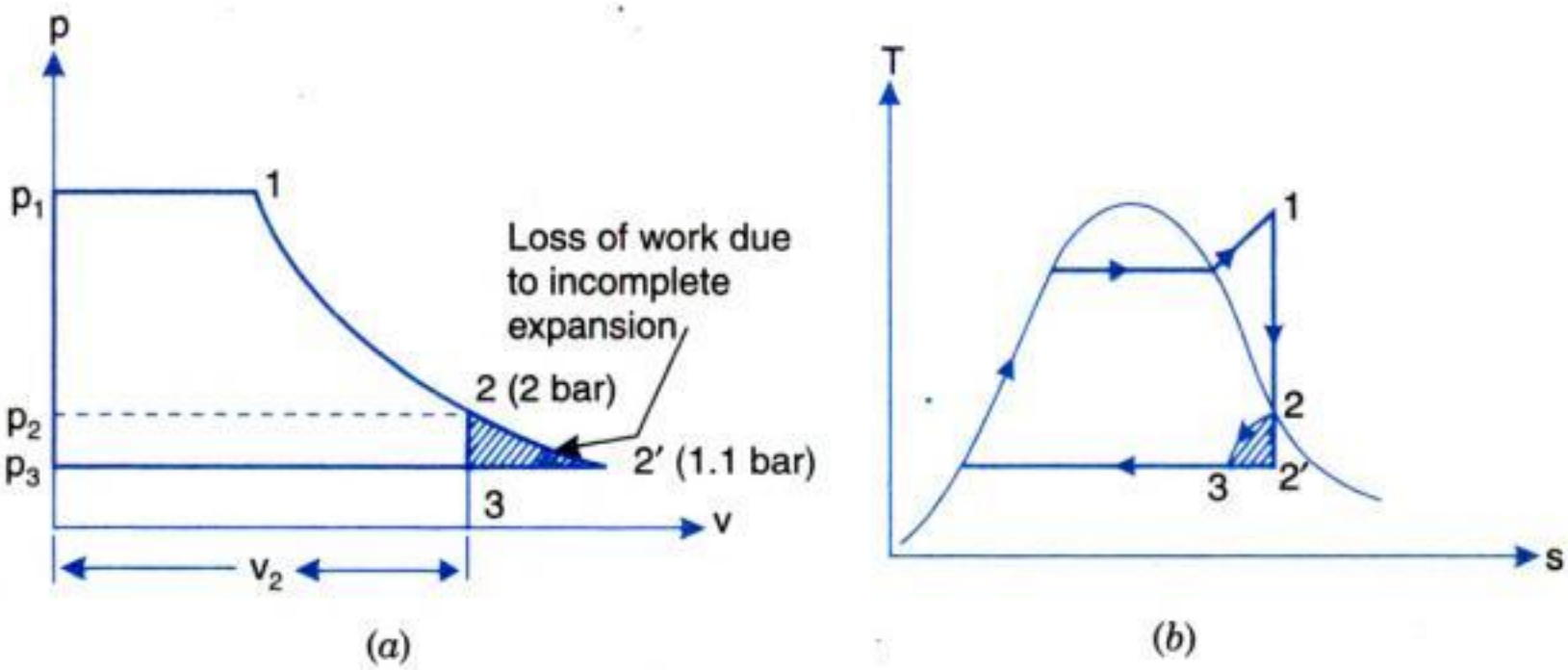

Fig. 15.13. *p-v* and *T-s* diagrams.

During isentropic expansion 1–2, we have

$$s_1 = s_2$$

$$6.918 = s_{f_2} + x_2\, s_{fg_2} = 1.5301 + x_2 \times 5.5967$$

$$\therefore \qquad x_2 = \frac{6.918 - 1.5301}{5.5967} = 0.96.$$

Then

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 504.7 + 0.96 \times 2201.6 = 2618.2 \text{ kJ/kg}$$

$$v_2 = x_2\, v_{g_2} + (1 - x_2)\, v_{f_2}$$

$$= 0.96 \times 0.885 + (1 - 0.96) \times 0.00106 = 0.849 \text{ m}^3\text{/kg}.$$

(*i*) **Ideal work :**

Ideal work or modified Rankine engine work/kg,

$$W = (h_1 - h_2) + (p_2 - p_3)\, v_2$$

$$= (3037.6 - 2618.2) + (2 - 1.1) \times 10^5 \times 0.849/1000$$

$$= 419.4 + 76.41 = \mathbf{495.8\ kJ/kg.} \quad \textbf{(Ans.)}$$

(*ii*) **Rankine engine efficiency :**

$$\eta_{\text{rankine}} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{495.8}{(h_1 - h_{f_3})}$$

$$= \frac{495.8}{3037.6 - 428.8} = \mathbf{0.19}\ \text{or}\ \mathbf{19\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Indicated and brake work per kg :**

Indicated work/kg, $\qquad W_{\text{indicated}} = \dfrac{\text{I.P.}}{\dot{m}}$

$$= \frac{1 \times 3600}{12.8} = \mathbf{281.25\ kJ/kg.} \quad \textbf{(Ans.)}$$

Brake work/kg, $\qquad W_{\text{brake}} = \dfrac{\text{B.P.}}{\dot{m}} = \dfrac{\eta_{\text{mech.}} \times \text{I.P.}}{\dot{m}}$

$$= \frac{0.8 \times 1 \times 3600}{12.8} = \mathbf{225\ kJ/kg.} \quad \textbf{(Ans.)}$$

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Energy/Heat balance equation for H.P. heater :

$$m_1 (h_1 - h_{f_6}) = (1 - m_1) (h_{f_6} - h_{f_5})$$

or $$m_1[(h_1 - h_{f_6}) + (h_{f_6} - h_{f_5})] = (h_{f_6} - h_{f_5})$$

or $$m_1 = \frac{h_{f_6} - h_{f_5}}{h_1 - h_{f_5}} \qquad ...(15.8)$$

Energy/Heat balance equation for L.P. heater :

$$m_2 (h_2 - h_{f_5}) = (1 - m_1 - m_2) (h_{f_5} - h_{f_3})$$

or $$m_2[(h_2 - h_{f_5}) + (h_{f_5} - h_{f_3})] = (1 - m_1) (h_{f_5} - h_{f_3})$$

or $$m_2 = \frac{(1 - m_1) (h_{f_5} - h_{f_3})}{(h_2 - h_{f_3})} \qquad ...(15.9)$$

All enthalpies may be determined ; therefore m_1 and m_2 may be found. The maximum temperature to which the water can be heated is dictated by that of bled steam. The condensate from the bled steam is added to feed water.

Neglecting pump work :

The heat supplied externally in the cycle

$$= (h_0 - h_{f_6})$$

Isentropic work done $$= m_1 (h_0 - h_1) + m_2 (h_0 - h_2) + (1 - m_1 - m_2) (h_0 - h_3)$$

The thermal efficiency of regenerative cycle is

$$\eta_{thermal} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{m_1 (h_0 - h_1) + m_2 (h_0 - h_2) + (1 - m_1 - m_2)(h_0 - h_3)}{(h_0 - h_{f_6})} \qquad ...(15.10)$$

[The work done by the turbine may also be calculated by summing up the products of the steam flow and the corresponding heat drop in the turbine stages.

i.e., Work done $= (h_0 - h_1) + (1 - m_1) (h_1 - h_2) + (1 - m_1 - m_2) (h_2 - h_3)$]

Advantages of Regenerative cycle over Simple Rankine cycle :

1. The heating process in the boiler tends to become reversible.

2. The thermal stresses set up in the boiler are minimised. This is due to the fact that temperature ranges in the boiler are reduced.

3. The thermal efficiency is improved because the average temperature of heat addition to the cycle is increased.

4. Heat rate is reduced.

5. The blade height is less due to the reduced amount of steam passed through the low pressure stages.

6. Due to many extractions there is an improvement in the turbine drainage and it reduces erosion due to moisture.

7. A small size condenser is required.

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(*i*) **Efficiency of cycle, η_{cycle} :**

$$\eta_{cycle} = \frac{W_T}{Q_1} = \frac{934.8}{2590.8} = \textbf{0.3608 or 36.08\%.} \quad \text{(Ans.)}$$

$$\textbf{Steam rate} = \frac{3600}{934.8} = \textbf{3.85 kg/kWh.} \quad \text{(Ans.)}$$

(*ii*) $$T_{m_1} = \frac{h_1 - h_{f_7}}{s_1 - s_7} = \frac{2590.8}{6.921 - 1.8604} = 511.9 \text{ K} = 238.9°\text{C}.$$

T_{m_1} (without regeneration)

$$= \frac{h_1 - h_{f_4}}{s_1 - s_4} = \frac{3230.9 - 191.8}{6.921 - 0.649} = \frac{3039.1}{6.272} = 484.5 \text{ K} = 211.5°\text{C}.$$

Increase in T_{m_1} due to regeneration

$$= 238.9 - 211.5 = \textbf{27.4°C.} \quad \text{(Ans.)}$$

W_T (without regeneration)

$$= h_1 - h_3 = 3230.9 - 2192.2 = 1038.7 \text{ kJ/kg}$$

Steam rate without regeneration

$$= \frac{3600}{1038.7} = 3.46 \text{ kg/kWh}$$

$\therefore$ **Increase in steam rate due to regeneration**

$$= 3.85 - 3.46 = \textbf{0.39 kg/kWh.} \quad \text{(Ans.)}$$

$$\eta_{cycle} \text{ (without regeneration)} = \frac{h_1 - h_3}{h_1 - h_{f_4}} = \frac{1038.7}{3230.9 - 191.8} = \textbf{0.3418 or 34.18\%.} \quad \text{(Ans.)}$$

Increase in cycle efficiency due to regeneration

$$= 36.08 - 34.18 = \textbf{1.9\%.} \quad \text{(Ans.)}$$

Example 15.14. *Steam is supplied to a turbine at a pressure of 30 bar and a temperature of 400°C and is expanded adiabatically to a pressure of 0.04 bar. At a stage of turbine where the pressure is 3 bar a connection is made to a surface heater in which the feed water is heated by bled steam to a temperature of 130°C. The condensed steam from the feed heater is cooled in a drain cooler to 27°C. The feed water passes through the drain cooler before entering the feed heater. The cooled drain water combines with the condensate in the well of the condenser.*

Assuming no heat losses in the steam, calculate the following :

(i) Mass of steam used for feed heating per kg of steam entering the turbine ;

(ii) Thermal efficiency of the cycle.

Solution. Refer Fig. 15.18.

From steam tables :

At 3 bar: $t_s = 133.5°\text{C}$, $h_f = 561.4$ kJ/kg.

At 0.04 bar : $t_s = 29°\text{C}$, $h_f = 121.5$ kJ/kg.

From Mollier chart :

$h_0 = 3231$ kJ/kg (at 30 bar, 400°C)

$h_1 = 2700$ kJ/kg (at 3 bar)

$h_2 = 2085$ kJ/kg (at 0.04 bar).

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$$\therefore \qquad \eta_{overall} = \frac{1039.55}{2775.6 + 526.1} \times 100 = \mathbf{31.48\%.} \quad \textbf{(Ans.)}$$

(*iv*) **Specific steam consumption :**

$$\text{Specific steam consumption} = \frac{4.155 \times 10^5}{120 \times 10^3} = \mathbf{3.4625\ kg/kWh.} \quad \textbf{(Ans.)}$$

15.5. REHEAT CYCLE

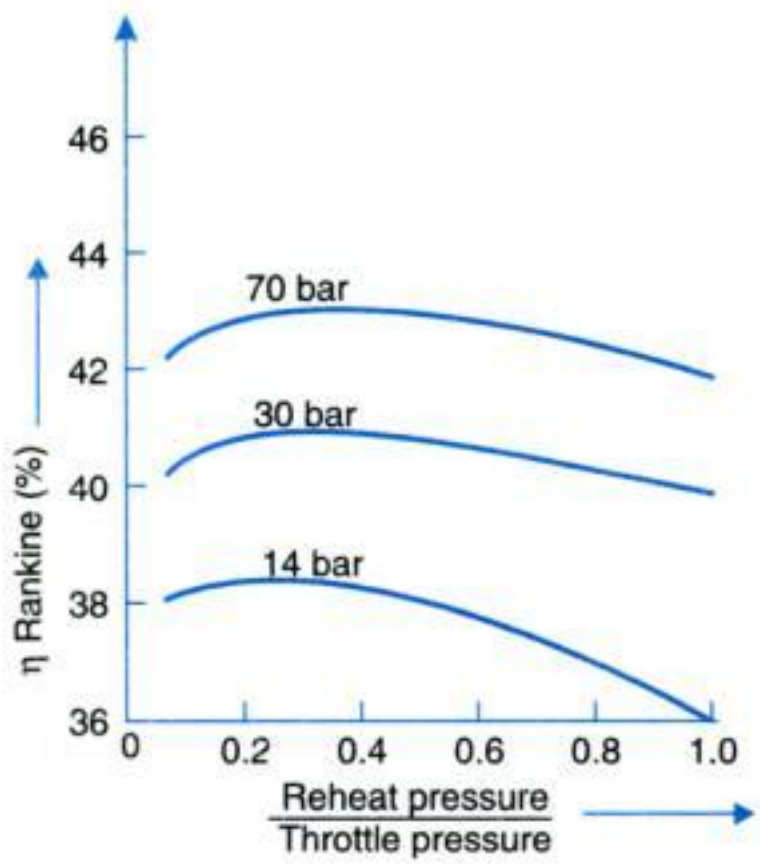
For attaining greater thermal efficiencies when the initial pressure of steam was raised beyond 42 bar it was found that resulting condition of steam after, expansion was increasingly wetter and exceeded in the safe limit of 12 per cent condensation. It, therefore, became necessary to *reheat* the steam after part of expansion was over so that the resulting condition after complete expansion fell within the region of permissible wetness.

The reheating or resuperheating of steam is now universally used when high pressure and temperature steam conditions such as 100 to 250 bar and 500°C to 600°C are employed for throttle. For plants of *still higher pressures and temperatures, a double reheating may be used.*

In actual practice reheat *improves* the cycle efficiency by about 5% for a 85/15 bar cycle. A *second reheat* will give a *much less gain* while the initial cost involved would be so high as to prohibit use of two stage reheat except in case of very high initial throttle conditions. The cost of reheat equipment consisting of boiler, piping and controls may be 5% to 10% more than that of the conventional boilers and this additional expenditure is justified only if gain in thermal efficiency is sufficient to promise a return of this investment. *Usually a plant with a base load capacity of 50000 kW and initial steam pressure of 42 bar would economically justify the extra cost of reheating.*

The improvement in thermal efficiency due to reheat is greatly dependent upon the *reheat pressure* with respect to the original pressure of steam.

Fig. 15.23 shows the reheat pressure selection on cycle efficiency.



Condenser pressure : 12.7 mm Hg
Temperature of throttle and heat : 427°C

Fig. 15.23. Effect of reheat pressure selection on cycle efficiency.

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(*c*) *In a 15 MW steam power plant operating on ideal reheat cycle, steam enters the H.P. turbine at 150 bar and 600°C. The condenser is maintained at a pressure of 0.1 bar. If the moisture content at the exit of the L.P. turbine is 10.4%, determine :*

(*i*) *Reheat pressure ;* (*ii*) *Thermal efficiency ;* (*iii*) *Specific steam consumption ;* and (*iv*) *Rate of pump work in kW. Assume steam to be reheated to the initial temperature.* (AMIE)

Solution. (*a*) The erosion of the moving blades is *caused by the presence of water particles in (wet) steam in the L.P. stages.* The water particles strike the leading surface of the blades. Such impact, if sufficiently heavy, produces *severe local stresses in the blade material causing the surface metal to fail and flake off.*

The erosion, if any, is more likely to occur in the region where the steam is wettest, *i.e.*, in the last one or two stages of the turbine. Moreover, the water droplets are concentrated in the outer parts of the flow annuals where the velocity of impact is highest.

Erosion difficulties due to moisture in the steam may be avoided by reheating (see Fig. 15.31). The whole of steam is taken from the turbine at a suitable point 2, and a further supply of heat is given to it along 2-3 after which the steam is readmitted to the turbine and expanded along 3-4 to condenser pressure.

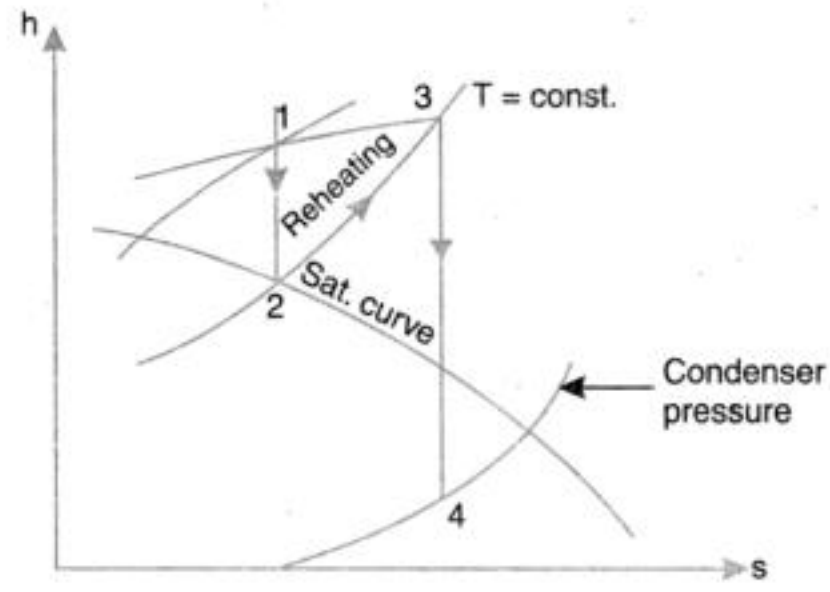

Fig. 15.31

Erosion *may also be reduced by using steam traps in between the stages to separate moisture from the steam.*

(*b*) TTD means *"Terminal temperature difference"*. It is the *difference between temperatures of bled steam/condensate and the feed water at the two ends of the feed water heater.*

The required temperature-path-line diagram of a closed feed water heater is shown in Fig. 15.32.

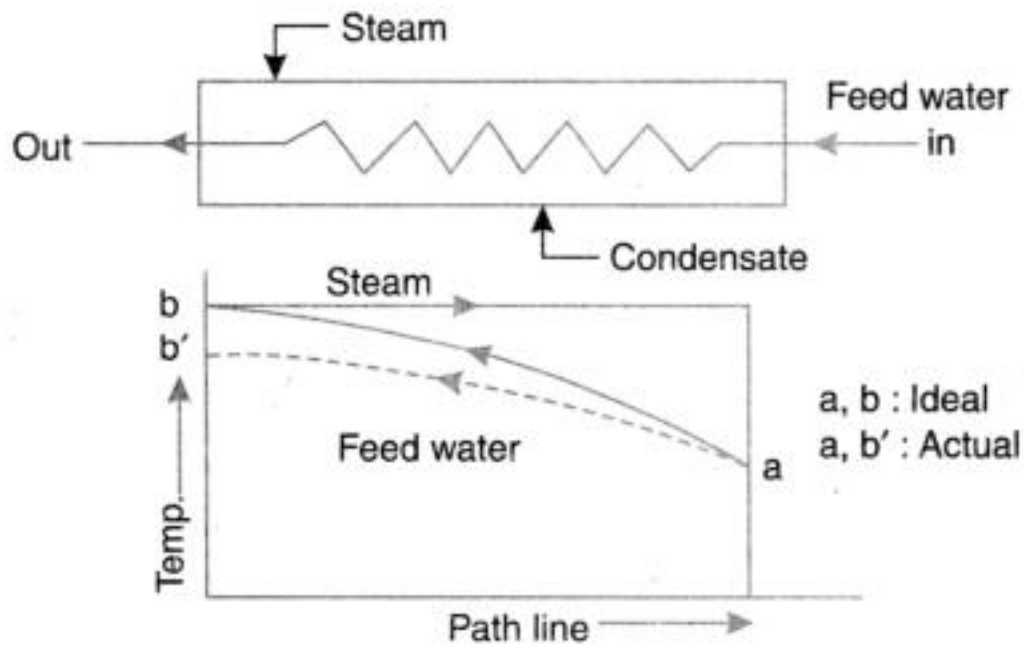

Fig. 15.32

(*c*) The cycle is shown on *T-s* and *h-s* diagrams in Figs. 15.33 and 15.34 respectively. The following values are read from the Mollier diagram :

h_1 = 3580 kJ/kg, h_2 = 3140 kJ/kg, h_3 = 3675 kJ/kg, and h_4 = 2335 kJ/kg

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Total work done in the binary cycle is given by

$$W_t = m\ W_{hg} + W_s \qquad ...(15.17)$$

∴ Overall efficiency of the binary cycle is given by

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{W_t}{h_t} = \frac{mW_{hg} + W_s}{mh_{hg_1}} \qquad ...(15.18)$$

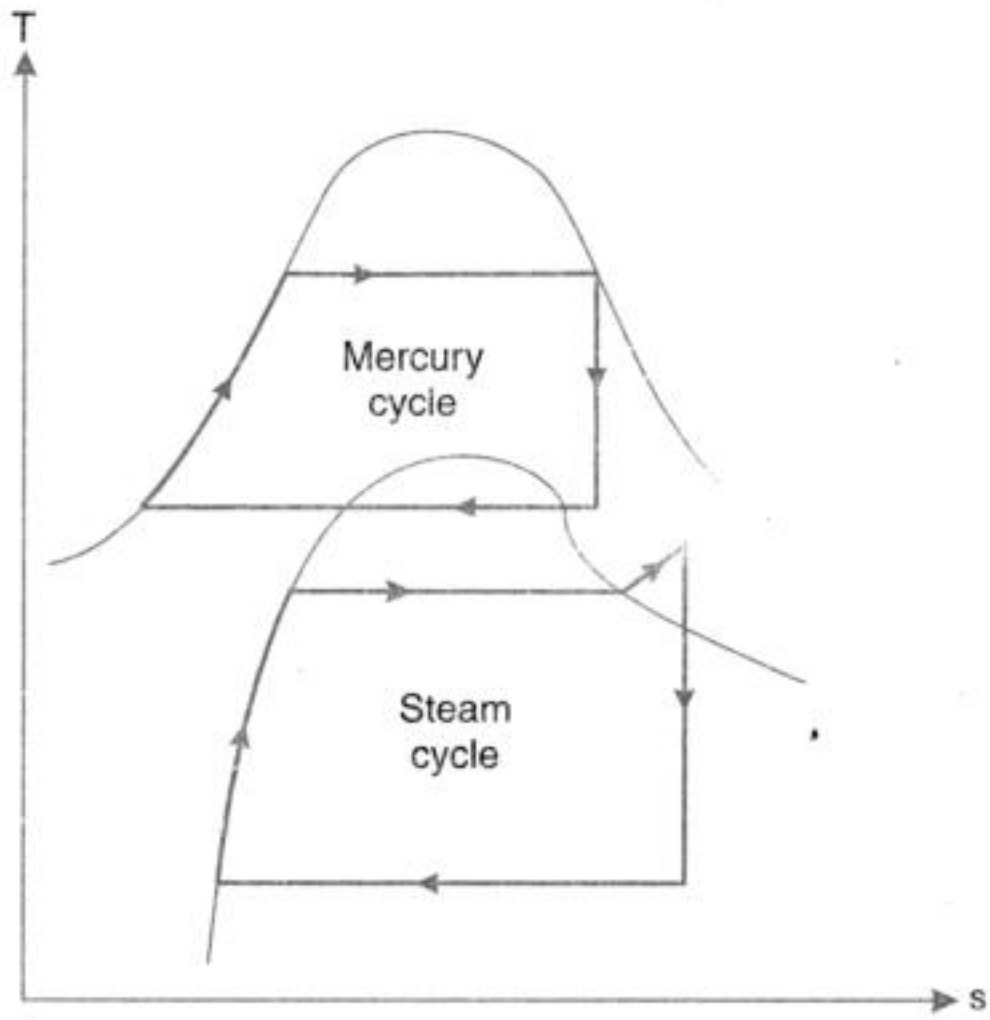

Fig. 15.36. Binary vapour cycle on T-s diagram.

Thermal efficiency of the mercury cycle is given by

$$\eta_{hg} = \frac{mW_{hg}}{mh_{hg_1}}$$

$$= \frac{W_{hg}}{h_{hg_1}} = \frac{h_{hg_1} - h_{hg_2}}{h_{hg_1}} = 1 - \frac{h_{hg_2}}{h_{hg_1}} \qquad ...(15.19)$$

$$= \frac{mh_{hg_1} - h_s}{mh_{hg_1}} = 1 - \frac{1}{m} \cdot \frac{h_s}{h_{hg_1}} \qquad ...(15.20)$$

Heat lost by mercury vapour = Heat gained by steam

$$\therefore \qquad m h_{hg_2} = 1 \times h_s \qquad ...(15.21)$$

Substituting the value of m from eqn. (15.21) into eqn. (15.20), we get

$$\eta_{hg} = 1 - \frac{h_{hg_2}}{h_{hg_1}} \qquad ...(15.22)$$

The thermal efficiency of the steam cycle is given by

$$\eta_s = \frac{W_s}{h_s} = \frac{h_{s_1} - h_{s_2}}{h_{s_1}} = \frac{h_{s_1} - h_{s_2}}{mh_{hg_2}} \qquad ...(15.23)$$

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$= 838.46$ kJ/kg

$(W_T)_{total} = 11.06 \times 84.92 + 838.46 = 1777.67$ kJ/kg

Neglecting pump work,

$$\eta_{overall} = \frac{1777.67}{3853.26} = \mathbf{0.4613 \text{ or } 46.13\%.} \quad \textbf{(Ans.)}$$

ADDITIONAL/TYPICAL WORKED EXAMPLES

Example 15.23. *The following data relate to a regenerative steam power plant generating 22500 kW energy, the alternator directly coupled to steam turbine :*

Condition of steam supplied to the steam turbine	*... 60 bar, 450°C*
Condenser vacuum	*... 707.5 mm*
Pressure at which steam is bled from the steam turbine	*... 3 bar*
Turbine efficiency of each portion of expansion	*... 87 percent*
Boiler efficiency	*... 86 percent*
Alternator efficiency	*... 94 percent*
Mechanical efficiency from turbine to generator	*... 97 percent*

Neglecting the pump work in calculating the input to the boiler, determine :

(i) The steam bled per kg of steam supplied to the turbine.

(ii) The steam generated per hour if the 9 percent of the generator output is used to run the pumps.

(iii) The overall efficiency of the plant.

Solution. The schematic arrangement of the steam power plant is shown in Fig. 15.38 (*a*), while the conditions of the fluid passing through the components are represented on *T-s* and *h-s* diagrams as shown in Fig. 15.38 (*b*) and (*c*). The conditions of the fluid entering and leaving the pump are shown by the same point as the rise in temperature due to pump work is neglected.

Given : Power generated = 22500 kW ;

$$p_1 = 60 \text{ bar} ; t_1 = 450°C ; p_2 (= p_2') = 3 \text{ bar} ;$$

$$p_3 (= p_3') = \frac{760 - 707.5}{760} \times 1.013 = 0.07 \text{ bar} ; \eta_{turbine\ (each\ portion)} = 87\% ;$$

$$\eta_{boiler} = 86\% ; \eta_{alt.} = 94\%, \eta_{mech.} = 97\%$$

- Locate point 1 corresponding to the conditions : $p_1 = 60$ bar ; $t_1 = 450°C$ on the *h-s* chart *(Mollier chart)*.

  From *h-s* chart ; we find : $h_1 = 3300$ kJ/kg.

- Draw vertical line through point 1 till it cuts the 3 bar pressure line, then locate point 2.

  $\therefore \quad h_2 = 2607$ kJ/kg

  Now, $\eta_{turbine} = 0.87 = \dfrac{h_1 - h_2'}{h_1 - h_2}$ or, $0.87 = \dfrac{3300 - h_2'}{3300 - 2607}$

  $\therefore \quad h_2' = 2697$ kJ/kg

- Locate the point 2 on the *h-s* chart as enthalpy and pressure are known and then draw a vertical line through the point 2 till it cuts the 0.07 bar pressure line and then locate the point 3.

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(*ii*) **The boiler generating capacity :**

If m_s is the mass of steam supplied to the power plant per second, then the work developed is given by :

$$m_s(h_1 - h_2) + m_s(1 - m)\,(h_3 - h_4) = 110 \times 10^3$$

or, $$m_s(2985 - 2520) + m_s(1 - 0.172)\,(3170 - 2555) = 110 \times 10^3$$

or, $$m_s(465 + 509.22) = 110 \times 10^3$$

$\therefore$ $$m_s = 112.91 \text{ kg/s} \quad \text{or} \quad \textbf{406.48 tonnes/hour} \quad \text{(Ans.)}$$

(*iii*) **Thermal efficiency of the cycle, η_{thermal} :**

$$\eta_{\text{thermal}} = \frac{\text{Output / kg of steam}}{\text{Input / kg of steam}} = \frac{(h_1 - h_2) + (1 - m)\,(h_3 - h_4)}{(h_1 - h_{f_2}) + (1 - m)\,(h_3 - h_2)}$$

$$= \frac{(2985 - 2520) + (1 - 0.172)\,(3170 - 2555)}{(2985 - 697.1) + (1 - 0.172)\,(3170 - 2520)}$$

$$= \frac{974.22}{2826.1} = 0.3447 \quad \text{or} \quad \textbf{34.47\%.} \quad \text{(Ans.)}$$

Example 15.25. *A steam power plant equipped with regenerative as well as reheat arrangement is supplied with steam to the H.P. turbine at 80 bar 470°C. For feed heating, a part of steam is extracted at 7 bar and remainder of the steam is reheated to 350°C in a reheater and then expanded in L.P. turbine down to 0.035 bar. Determine :*

(i) Amount of steam bled-off for feed heating,

(ii) Amount of steam supplied to L.P. turbine,

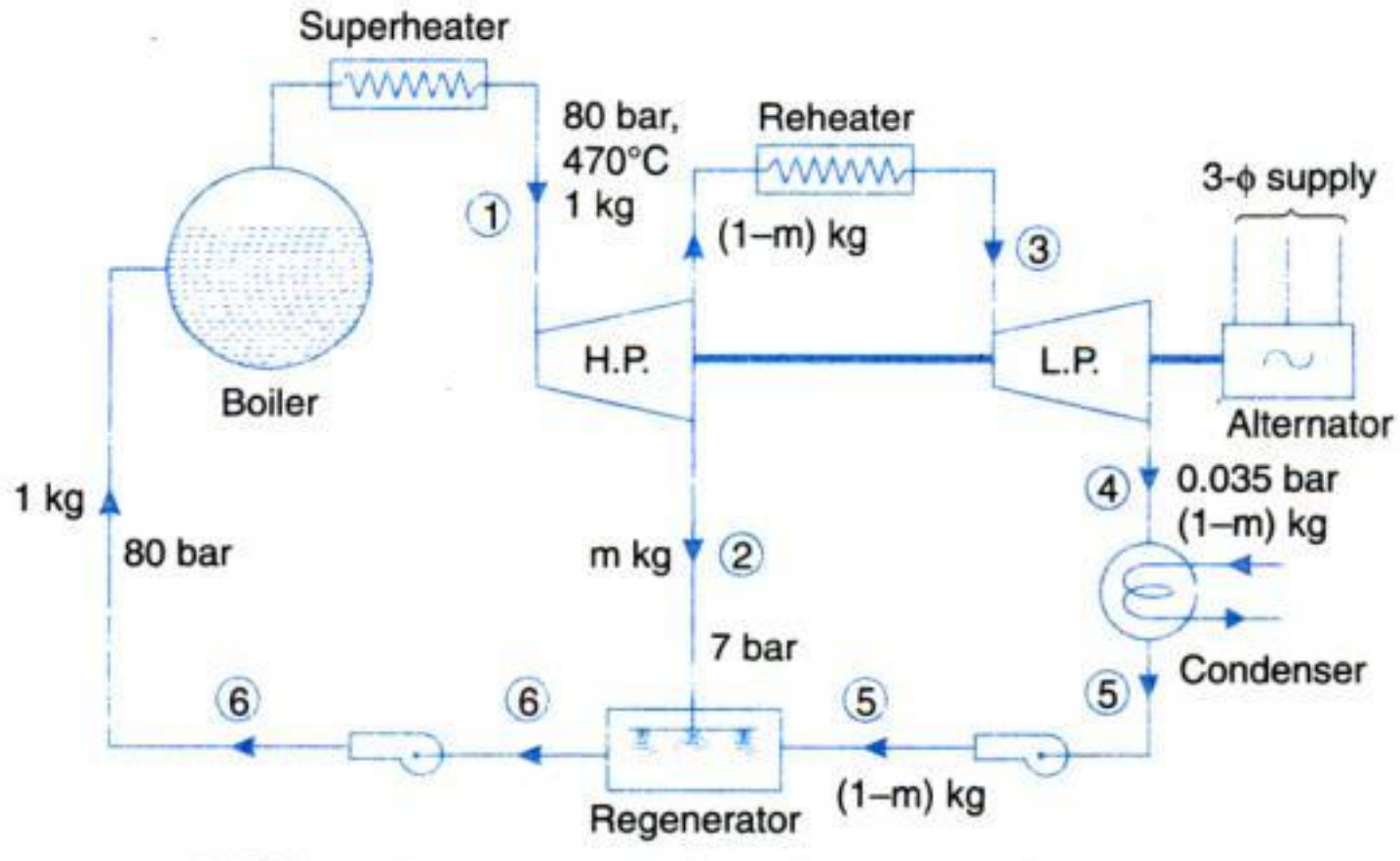
(iii) Heat supplied in the boiler and reheater

(iv) Cycle efficiency, and

(v) Power developed by the system.

The steam supplied by the boiler is 50 kg/s. **(B.U. Dec., 2005)**

Solution. The schematic arrangement of the steam power plant is shown in Fig. 15.40 (*a*) and the processes are represented on *h-s* diagram as shown in Fig. 15.40(*b*).



(*a*) Schematic arrangement of the steam power plant

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Work done by the pump P_2,

$$W_{P2} = v_{w2}\,(1 - m_1)(150 - 5) \times 10^5 \times 10^{-3} \text{ kJ/kg}$$

$$= \frac{1}{1000}\,(1 - 0.179)(150 - 5) \times 10^5 \times 10^{-3} = 11.9 \text{ kJ/kg}$$

Work done by pump P_3,

$$W_{P3} = v_{w3} \times m_1 \times (150 - 40) \times 10^5 \times 10^{-3}$$

$$= \frac{1}{1000} \times 0.179\,(150 - 40) \times 10^5 \times 10^{-3} = 1.97 \text{ kJ/kg}$$

Total pump work $= W_{P1} + W_{P2} + W_{P3}$

$= 0.338 + 11.9 + 1.97 = 14.21$ kJ/kg of steam supplied by boiler

$\therefore$ Net work done by the turbine per kg of steam supplied by the boiler,

$$W_{net} = 1456.94 - 14.21 = 1442.73 \text{ kJ/kg}$$

Heat of feed water extering the boiler

$$= (1 - m_1) \times 1611 + m_1 \times 1611 = 1611 \text{ kJ/kg}$$

Heat supplied by the boiler per kg of steam,

$$Q_{s1} = h_1 - 1610 = 3578 - 1610 = 1968 \text{ kJ/kg}$$

Q_{s2} = Heat supplied in the reheater

$= (1 - m_1)(h_3 - h_2) = (1 - 0.179)(3678 - 3140)$

$= 441.7$ kJ/kg of steam supplied by the boiler

Q_{st} (Total heat supplied) $= Q_{s1} + Q_{s2} = 1968 + 441.7 = 2409.7$ kJ/kg

$$\therefore \quad \eta_{\text{thermal}} = \frac{W_{net}}{Q_{st}} = \frac{1442.73}{2409.7} = 0.5987 \quad \text{or} \quad \mathbf{59.87\%.} \quad \textbf{(Ans.)}$$

Example 15.27. *Steam at 70 bar and 450°C is supplied to a steam turbine. After expanding to 25 bar in high pressure stages, it is reheated to 420°C at the constant pressure. Next ; it is expanded in intermediate pressure stages to an appropriate minimum pressure such that part of the steam bled at this pressure heats the feed water to a temperature of 180°C. The remaining steam expands from this pressure to a condenser pressure of 0.07 bar in the low pressure stage. The isentropic efficiency of H.P. stage is 78.5%, while that of the intermediate and L.P. stages is 83% each. From the above data, determine :*

(i) The minimum pressure at which bleeding is necessary.

(ii) The quantity of steam bled per kg of flow at the turbine inlet.

(iii) The cycle efficiency.

Neglect pump work. **(Roorkee University)**

Solution. The schematic arrangement of the plant is shown in Fig. 15.42 (*a*) and the processes are represented on *T-s* and *h-s* diagrams as shown in Fig. 15.42 (*b*) and (*c*) respectively.

(*i*) **The minimum pressure at which bleeding is necessary :**

It would be assumed that the feed water heater is an open heater. Feed water is heated to 180°C. So p_{sat} at 180°C $\simeq$ 10 bar is the pressure at which the heater operates.

Thus, *the pressure at which bleeding is necessary is* **10 bar.** **(Ans.)**

From the *h-s* chart (Mollier chart), we have :

$$h_1 = 3285 \text{ kJ/kg} \; ; \; h_2 = 2980 \text{ kJ/kg} \; ; \; h_3 = 3280 \text{ kJ/kg} \; ; \; h_4 = 3030 \text{ kJ/kg}$$

$$h_3 - h_4' = 0.83(h_3 - h_4) = 0.83(3280 - 3030) = 207.5 \text{ kJ/kg}$$

$$\therefore \quad h_4' = h_3 - 207.5 = 3280 - 207.5 = 3072.5 \text{ kJ/kg}$$

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UNSOLVED EXAMPLES

1. A simple Rankine cycle works between pressure of 30 bar and 0.04 bar, the initial condition of steam being dry saturated, calculate the cycle efficiency, work ratio and specific steam consumption.
[**Ans.** 35%, 0.997, 3.84 kg/kWh]

2. A steam power plant works between 40 bar and 0.05 bar. If the steam supplied is dry saturated and the cycle of operation is Rankine, find :
(*i*) Cycle efficiency (*ii*) Specific steam consumption.
[**Ans.** (*i*) 35.5%, (*ii*) 3.8 kg/kWh]

3. Compare the Rankine efficiency of a high pressure plant operating from 80 bar and 400°C and a low pressure plant operating from 40 bar 400°C, if the condenser pressure in both cases is 0.07 bar.
[**Ans.** 0.391 and 0.357]

4. A steam power plant working on Rankine cycle has the range of operation from 40 bar dry saturated to 0.05 bar. Determine :
(*i*) The cycle efficiency (*ii*) Work ratio
(*iii*) Specific fuel consumption. [**Ans.** (*i*) 34.64%, (*ii*) 0.9957, (*iii*) 3.8 kg/kWh]

5. In a Rankine cycle, the steam at inlet to turbine is saturated at a pressure of 30 bar and the exhaust pressure is 0.25 bar. Determine :
(*i*) The pump work (*ii*) Turbine work
(*iii*) Rankine efficiency (*iv*) Condenser heat flow
(*v*) Dryness at the end of expansion.
Assume flow rate of 10 kg/s. [**Ans.** (*i*) 30 kW, (*ii*) 7410 kW, (*iii*) 29.2%, (*iv*) 17900 kW, (*v*) 0.763]

6. In a regenerative cycle the inlet conditions are 40 bar and 400°C. Steam is bled at 10 bar in regenerative heating. The exit pressure is 0.8 bar. Neglecting pump work determine the efficiency of the cycle.
[**Ans.** 0.296]

7. A turbine with one bleeding for regenerative heating of feed water is admitted with steam having enthalpy of 3200 kJ/kg and the exhausted steam has an enthalpy of 2200 kJ/kg. The ideal regenerative feed water heater is fed with 11350 kg/h of bled steam at 3.5 bar (whose enthalpy is 2600 kJ/h). The feed water (condensate from the condenser) with an enthalpy of 134 kJ/kg is pumped to the heater. It leaves the heater dry saturated at 3.5 bar. Determine the power developed by the turbine. [**Ans.** 16015 kW]

8. A binary-vapour cycle operates on mercury and steam. Saturated mercury vapour at 4.5 bar is supplied to the mercury turbine, from which it exhaust at 0.04 bar. The mercury condenser generates saturated steam at 15 bar which is expanded in a steam turbine to 0.04 bar.
(*i*) Find the overall efficiency of the cycle.
(*ii*) If 50000 kg/h of steam flows through the steam turbine, what is the flow through the mercury turbine ?
(*iii*) Assuming that all processes are reversible, what is the useful work done in the binary vapour cycle for the specified steam flow ?
(*iv*) If the steam leaving the mercury condenser is superheated to a temperature of 300°C in a superheater located in the mercury boiler, and if the internal efficiencies of the mercury and steam turbines are 0.85 and 0.87 respectively, calculate the overall efficiency of the cycle. The properties of saturated mercury are given below :

p (bar)	t (°C)	h_f (kJ/kg)	h_g	s_f (kJ/kg K)	s_g	v_f (m^3/kg)	v_g
4.5	450	63.93	355.98	0.1352	0.5397	79.9×10^{-6}	0.068
0.04	216.9	29.98	329.85	0.0808	0.6925	76.5×10^{-3}	5.178

[**Ans.** (*i*) 52.94%, (*ii*) 59.35×10^4 kg/h, (*iii*) 28.49 MW, (*iv*) 46.2%]

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end with the steam chest by means of steam ports. The steam reaches the cylinder after passing through the steam ports. The exhaust port is located midway, the steam-ports which lead the used steam to the atmosphere or the condenser depending upon, whether the steam engine is *non-condensing* or *condensing*. The steam ports should be fairly large in cross-section to admit enough steam into the cylinder during the time the steam port is open otherwise steam will be throttled or wiredrawn.

One of the cylinder ends is fully covered by means of a cover known as **cover end** or **top end** or **front end.** The other end through which the piston rod passes is known as **bottom end** or **back end** or **crank end.** At this back-end cover, the piston rod passes through the *stuffing box* which makes the hole steam-tight and *prevents the leakage of steam*.

The steam engine cylinder is generally *steam jacketed in order to reduce the condensation of steam* as the steam cools during expansion. The condensation of steam is further reduced by covering the cylinder with non-conducting material known as **lagging.** The lagging is then covered with a plain thin sheet to give the cylinder a finished appearance.

In certain cases liners are fitted in the cylinder. The liner is an internal barrel made of steel or cast-iron. When the wear of the cylinder becomes too much, the whole of the cylinder block may have to be replaced which is decidedly a costly affair. In case the liner wears out too much, the replacement of liner alone is much cheaper than the whole cylinder block.

Steam chest. It is reservoir of steam from which the steam is admitted into the cylinder during the admission stroke of the piston. It is cast integral with the cylinder and is closed with a cover known as the steam chest cover through which the slide valve rod passes.

Piston and piston rings. *The function of a piston is to convert the pressure energy of steam into its reciprocating movement*. It moves from end to end of the cylinder. It is made of cast iron, cast steel or forged steel. The piston must form a steam-tight division between the two ends of a cylinder. This is accomplished by putting piston rings (generally made of rectangular section) on the piston. At least two rings should be fitted. The rings are generally of cast-iron. In addition to preventing leakage of steam, piston rings reduce considerably the friction between the piston and walls.

Solid type and hollow or box type pistons are shown in Figs. 16.3 and 16.4 respectively.

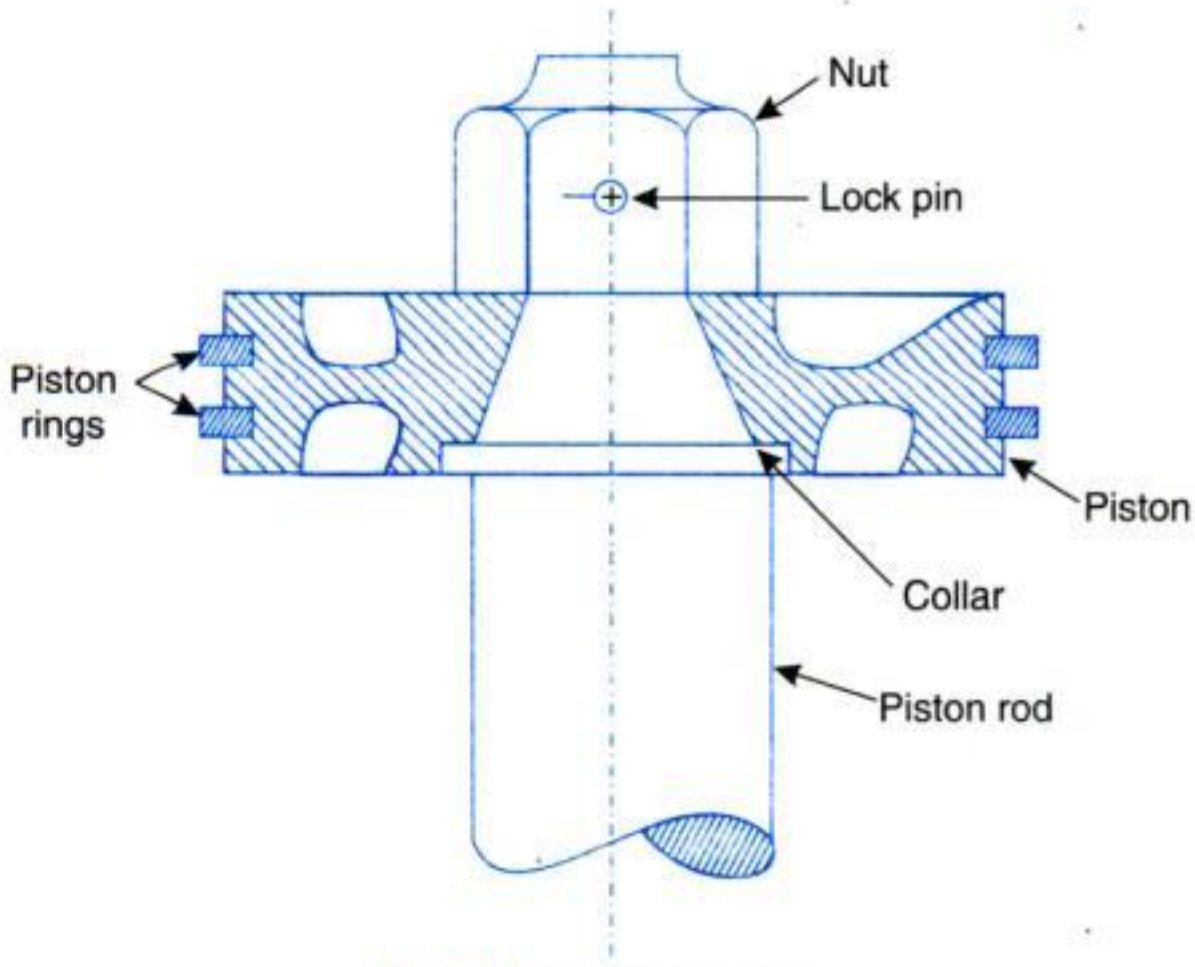

Fig. 16.3. Solid type piston.

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Case 1. Theoretical indicator diagram without clearance and compression.

Refer Fig. 16.10. The work done per cycle by an engine is equal to the area of the p-V diagram.

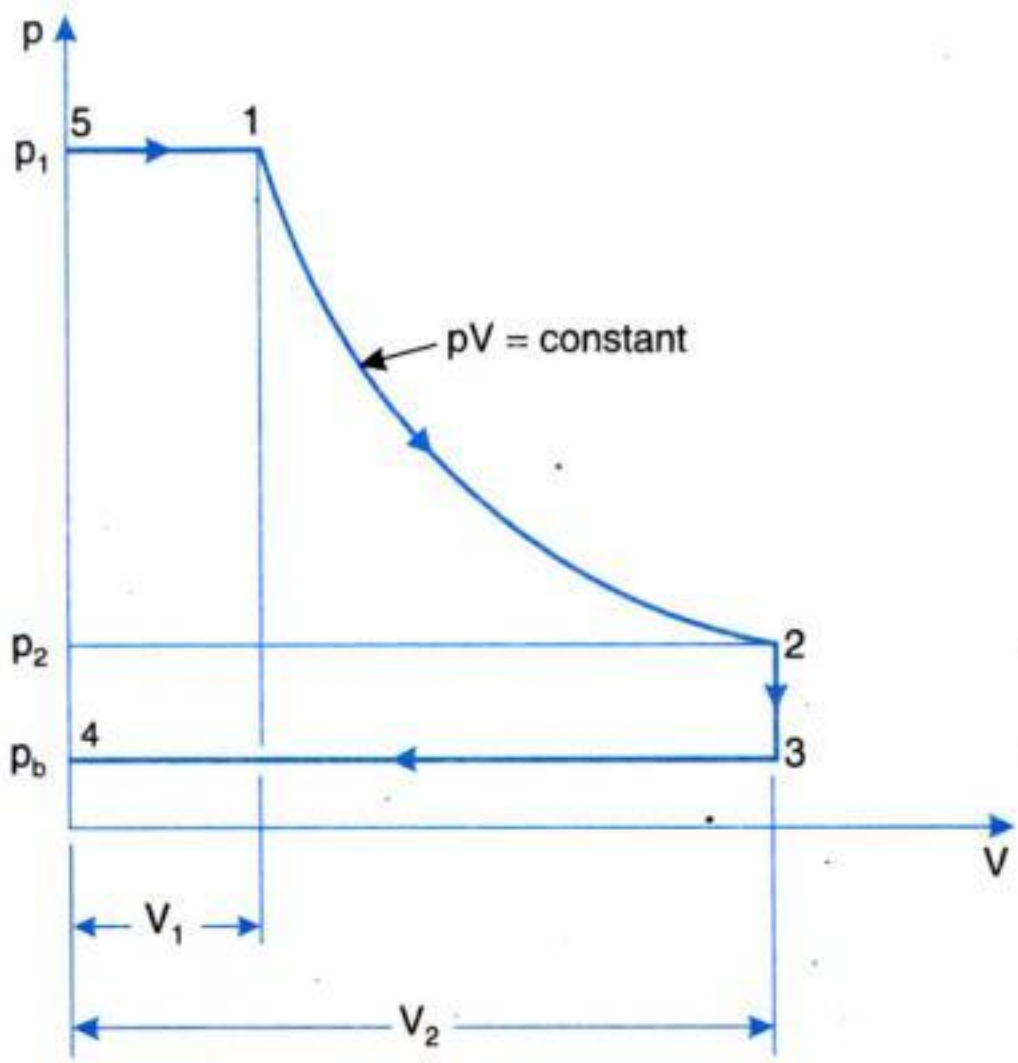

Fig. 16.10.

$\therefore$ Work done, W = Area of the theoretical indicator diagram

$$= p_1V_1 + p_1V_1 \log_e V_2/V_1 - p_bV_2$$

$$\text{Theoretical m.e.p.} = \frac{\text{Theoretical work done}}{\text{Stroke volume}}$$

$$= \frac{p_1V_1 + p_1V_1 \log_e V_2/V_1 - p_bV_2}{V_2} = \frac{p_1V_1}{V_2}\ (1 + \log_e V_2/V_1) - p_b$$

The expansion ratio, r, is defined as $\frac{V_2}{V_1}$, therefore,

$$\text{Theoretical/Hypothetical m.e.p., } p_m = \frac{p_1}{r}\ (1 + \log_e r) - p_b \qquad ...(16.3)$$

Note. (*i*) The ratio $\frac{V_1}{V_2} = \frac{1}{r}$ = cut-off ratio.

"Cut-off ratio is *the ratio of volume between the points of admission and cut-off to the swept volume*".

(*ii*) A hypothetical steam consumption can be estimated using the specific volume, v, of the inlet steam, and the volume induced per cycle, V_1,

i.e., Steam consumption/cycle = $\frac{V_1}{v}$ kg.

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An indicator is *used* :

1. To obtain a diagram from which conclusions may be drawn in respect of the following :
   (*i*) Behaviour of the steam in the cylinder ;
   (*ii*) The promptness of the steam admission ;
   (*iii*) The loss by fall of pressure between the boiler and the cylinder ;
   (*iv*) The loss by wire drawing ;
   (*v*) The extent and character of the expansion ;
   (*vi*) The efficiency of the arrangement for exhaust, including the extent of the back pressure, and
   (*vii*) The amount of compression.
2. To find the mean effective pressure exerted by the steam upon the piston and from there to calculate indicated power of the engine.
3. To determine whether the valves are set correctly by taking diagrams from each end of the cylinder and observing and comparing the respective positions of the points of admission, cut-off, release and compression.

16.10.2. Types of Indicators

Two simple types of indicators are :

1. Pencil indicator
2. Optical indicator.

The *pencil indicator* registers the pressure and volume by means of a *mechanism consisting of links and a spring.* The *essentials* of such an indicator are :

(*i*) *The rise of the 'Pencil' must be proportional to the rise of pressure in the cylinder of the engine.*

(*ii*) *The motion of the drum must be an exact copy of the motion of the engine piston to a reduced scale.*

The *Optical Indicator* registers the pressure and volume on *a photographic plate by means of a ray of light.* The main merit of this indicator is that the *diagram obtained is not affected by the inertia of the moving parts of any mechanism.*

16.10.3. Crosby Pencil Indicator

Fig. 16.13 shows a view of *crosby pencil indicator.* It consists of a small piston and cylinder assembly fitted on the top of the engine cylinder such that the cylinder pressure operates the indicator piston. The indicator piston is loaded with a precisely calibrated spring. The amount of movement of the piston is directly proportional to the engine cylinder pressure and this motion is amplified through a linkage mechanism, so that it is duplicated to a large scale by a pencil/stylus which moves over a drum. The drum executes a rotary motion proportional to the movement of the engine piston, obtained by means of a reducing mechanism attached to the piston rod or cross-head. As a result, the drum rotates back and forth in synchronism with the engine piston, while the stylus moves vertical distances that are proportional to the pressures in the engine cylinder.

The indicator may be placed in communication with either end of the cylinder or with the atmosphere through a three way cock.

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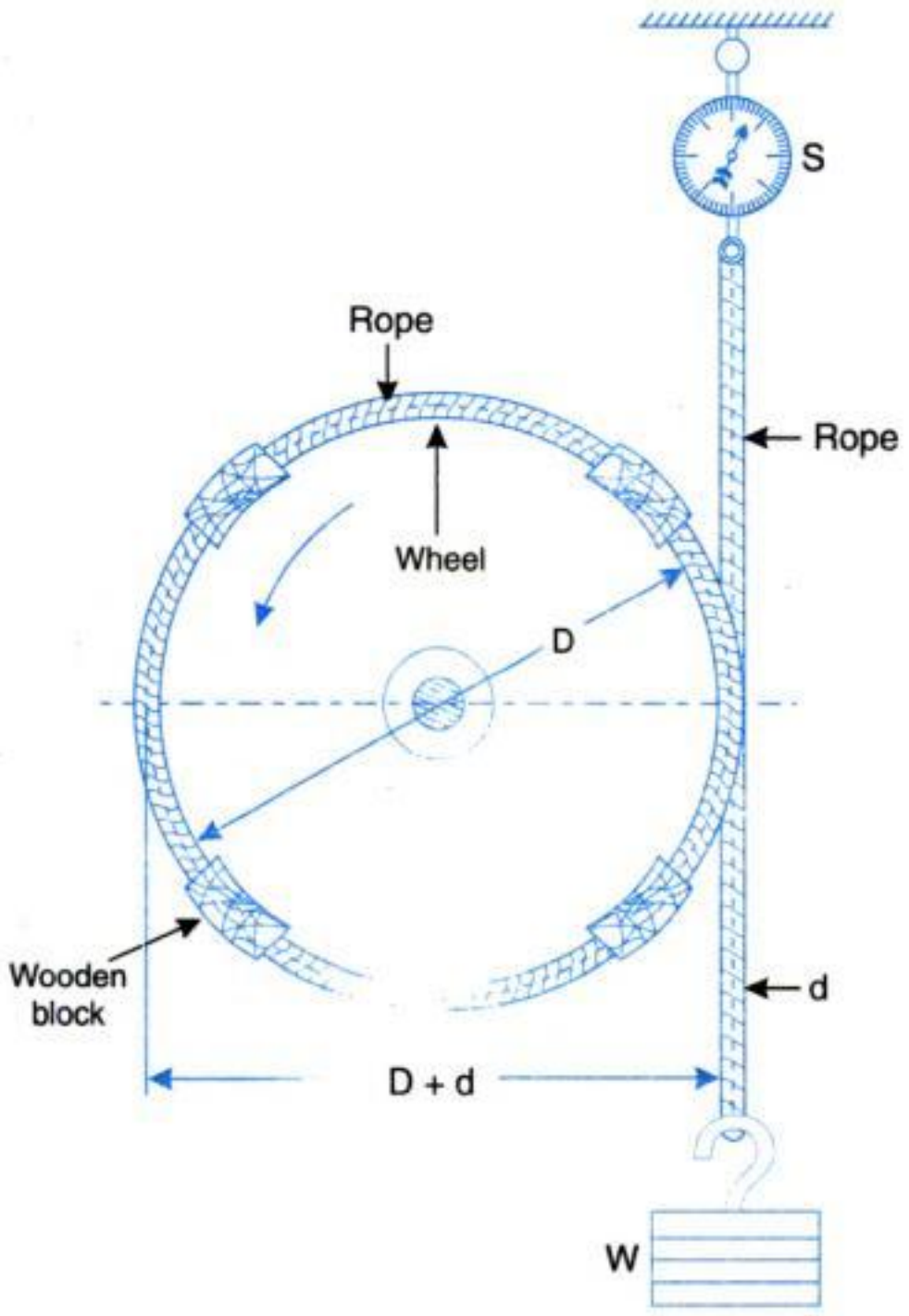

Fig. 16.14. Rope brake dynamometer.

$$\therefore \quad \text{Brake power, B.P.} = \frac{(W - S)\pi (D + d) N}{60 \times 10^3} \text{ kW} \qquad ...(16.16)$$

If diameter of the rope '*d*' is neglected, then

$$\text{B.P.} = \frac{(W - S)\pi DN}{60 \times 10^3} \text{ kW} \qquad (\because \quad d = 0) \qquad ...(16.17)$$

Also if T, braking torque is given,

Then $$\text{B.P.} = \frac{2\pi NT}{60 \times 10^3} \text{ kW.}$$

16.13. EFFICIENCIES OF STEAM ENGINE

1. **Mechanical efficiency** ($\eta_{mech.}$). It is the *ratio of brake power to indicated power.*

i.e., $$\eta_{mech.} = \frac{\text{Brake power}}{\text{Indicated power}} \qquad ...(16.18)$$

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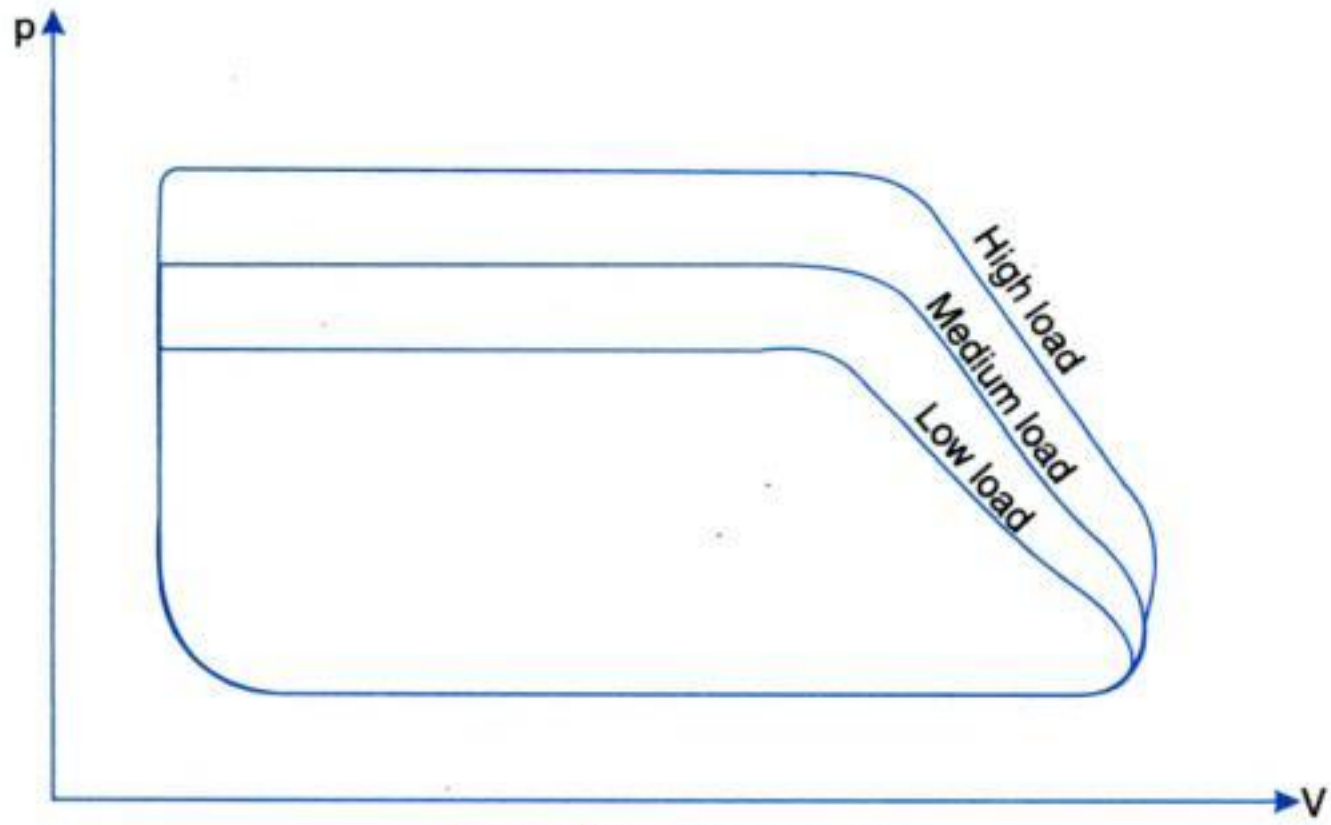

Fig. 16.17. Throttle governing.

Refer Fig. 16.18. During a test on a throttle governed engine, if the indicated power is varied by altering the load, and the rate of steam consumption is measured for all values of power, it will be found that the steam consumption is a straight line function of the indicated power. This is known as ***Willan's law*** ; *it holds good only for a throttle governed engine running at constant speed.*

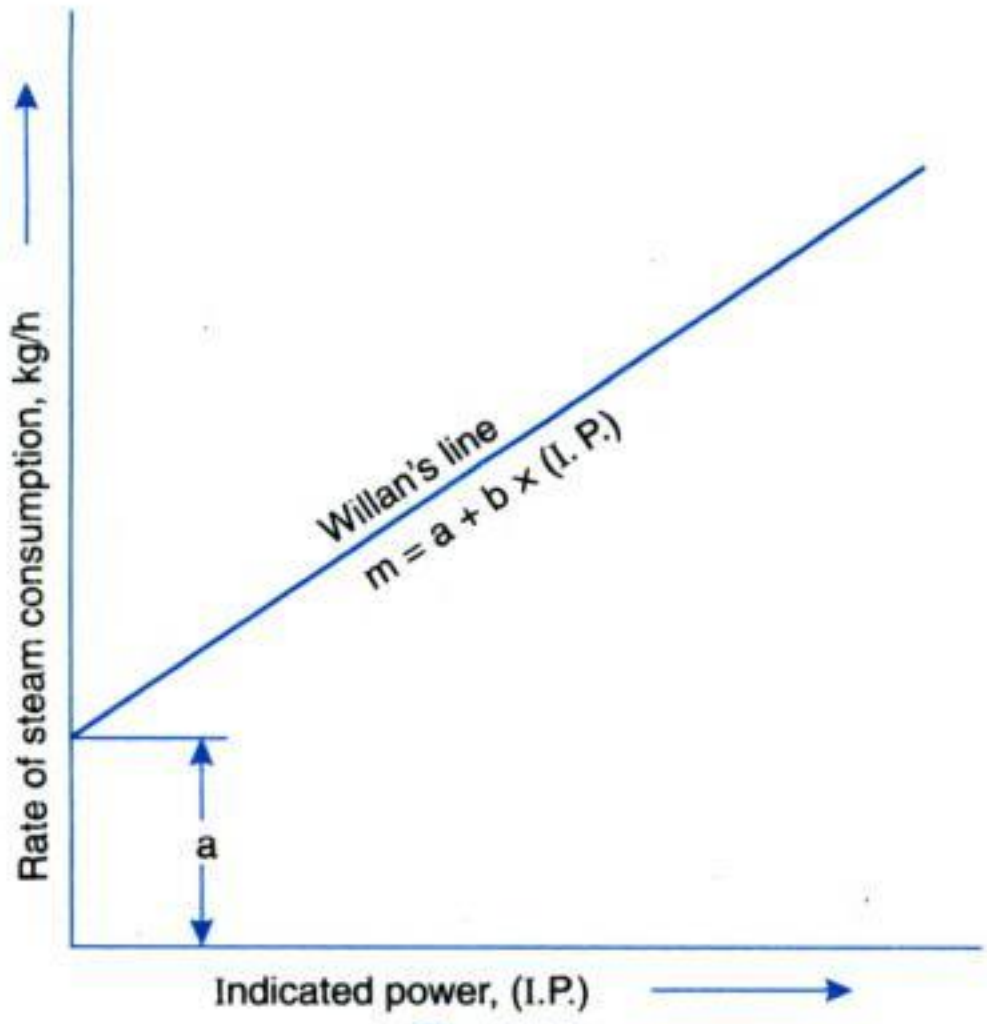

Fig. 16.18

Willan's line is a straight line and may therefore be expressed in the form as follows :

$$m = a + bx. \text{ (I.P.)} \qquad ...(16.25)$$

where,

m = Steam consumption in kg/h,

a = Steam consumption at no load in kg/h (constant), and

b = Slope of Willan's line (constant).

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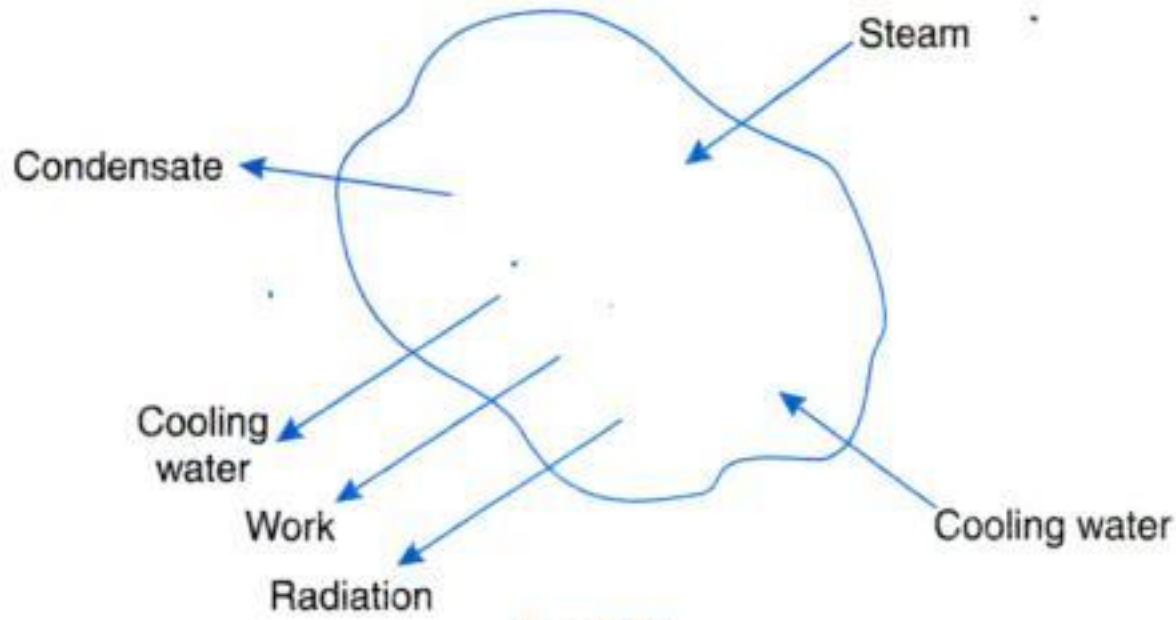

Fig. 16.27

$$m_{s_1} h_{s_1} + m_c h_{c_1} = m_{s_2} h_{s_2} + m_c h_{c_2} + W + Q_R \quad \text{...(16.26)}$$

$$m_{s_1} h_{s_1} = m_{s_2} h_{s_2} + m_c (h_{c_2} - h_{c_1}) + W + Q_R \quad \text{...(16.27)}$$

where, m = Mass, W = Work,

h = Specific enthalpy, Q = heat transferred

and suffices have the following meanings :

s_1 = Steam supplied, s_2 = Condensate discharged

c = Cooling water, R = Radiation.

The equation (16.27) in tabular form is expressed as follows :

Heat in		*Heat out*	
In steam	$m_{s_1} h_{s_1}$	Heat as shaft work	W
		Heat to condensate	$m_{s_2} h_{s_2}$
		Heat to coolant	$m_c (h_{c_2} - h_{c_1})$
		Heat to radiation	Q_R

The value of enthalpy may be obtained from the steam tables. The heat to cooling water is given by $m_c \times c_{pw} \times (t_{out} - t_{in})$ and the heat to radiation obtained by difference. In the absence of leakage, $m_{s_1} = m_{s_2}$.

The heat balance may therefore be written as follows :

Heat in	*Heat out*
Heat in steam	Heat as B.P.
	Heat to condensate.
	Heat to cooling water.
	Heat to radiation, by difference.

It is to be noted that friction terms are absent, since any work done against friction is manifest in heat to the condensate, cooling water and radiation loss.

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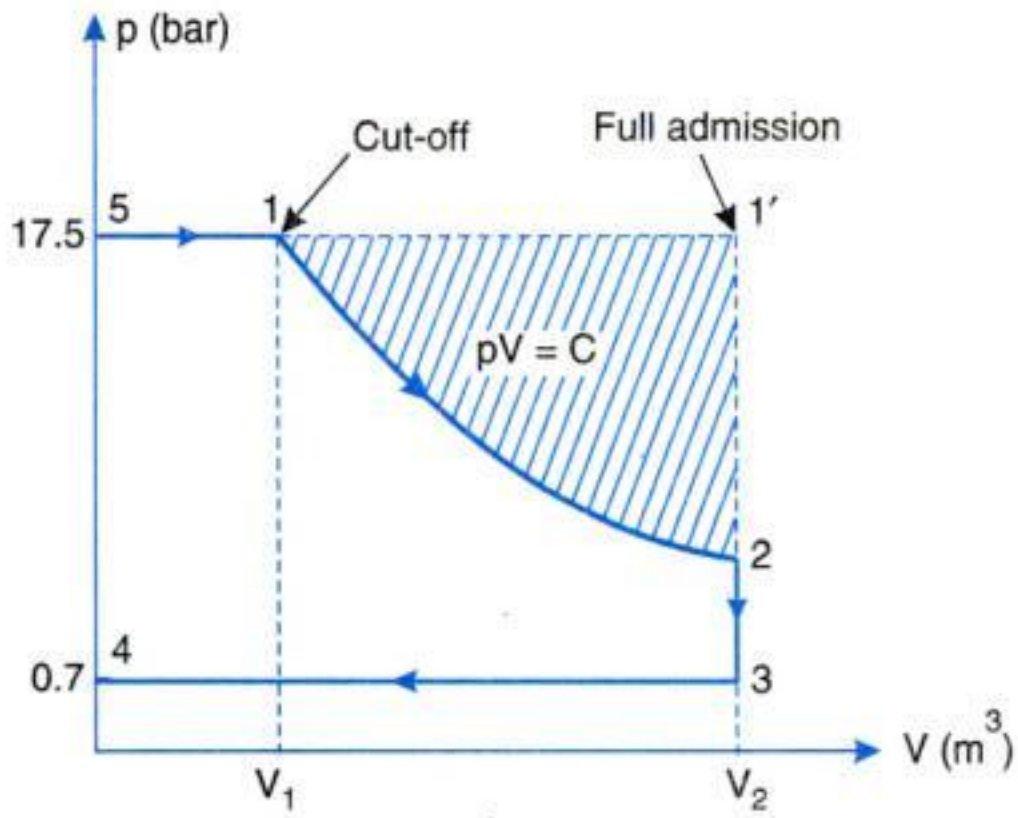

Fig. 16.30

Work obtained from the engine taking steam for $\frac{1}{5}$th of the stroke = area '512345'.

Work obtained from the engine taking steam for the whole stroke = area '51'345'.

Let us consider that 1 kg of steam is supplied to the engine during one full stroke. Hence for the engine taking steam for $\frac{1}{5}$th of stroke, only $\frac{1}{5}$th of the steam will be admitted.

Case 1. Engine taking steam for $\frac{1}{5}$th of stroke :

Considering the p-V diagram 51234, we have :

Mean effective pressure, $$p_m = \frac{p_1}{r}\ (1 + \log_e r) - p_b$$

Here expansion ratio, $$r = \frac{V_2}{V_1} = \frac{V_2}{\frac{V_2}{5}} = 5$$

$\therefore$ $$p_m = \frac{17.5}{5}\ (1 + \log_e 5) - 0.7 = 8.43 \text{ bar}$$

At 17.5 bar. Specific volume, $v = 0.113$ m³/kg.

Work done by $\frac{1}{5}$th of steam $= p_m \times v$

$$= \frac{8.43 \times 10^5 \times 0.113}{10^3} = 95.26 \text{ kJ/kg.}$$

Work done per kg of steam $= 95.26 \times 5 = 476.3$ kJ

Heat supplied per kg $= h_1 - h_{f_3}$ $\left[\begin{array}{l} \textit{At 17.5 bar. } h_1 = h_{g_1} = 2794.1 \text{ kJ/kg} \\ \textit{At 0.7 bar. } h_{f_3} = 376.8 \text{ kJ/kg.} \end{array}\right]$

$= 2794.1 - 376.8$

$= 2417.3$ kJ/kg

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(*ii*) Refer Fig. 16.32.

The theoretical mean effective pressure, when clearance considered, is given by

$$p_{m(th.)} = p_1 \left[\frac{1}{r} + \left(c + \frac{1}{r} \right) \log_e \left(\frac{c+1}{c+\frac{1}{r}} \right) \right] - p_b$$

Here, $r = 2.5$ or $\frac{1}{r} = 0.4$ and $c = \frac{10}{100} = 0.1$

$$\therefore \quad p_{m(th.)} = 7 \left[0.4 + (0.1 + 0.4) \log_e \left(\frac{0.1+1}{0.1+0.4} \right) \right] - 0.38$$

$$= 7(0.4 + 0.394) - 0.38 = 5.18 \text{ bar}$$

Assuming the same diagram factor,

$$p_{m(act.)} = 0.8 \times 5.18 = 4.14 \text{ bar}$$

Hence, *mean effective pressure (actual)* = **4.14 bar.** (Ans.)

Example 16.5. *A steam engine having swept volume 0.034 m^3 and clearance volume 20% of the piston displacement, consumes 2725 kg of steam per hour at 250 r.p.m. The engine is double acting and the pressure of steam near cut-off is 8.8 bar, when piston has traversed 30% of working stroke. If the compression commences at 65% of the return stroke when the steam remaining in the cylinder is dry at 0.14 bar, estimate the mass of cushion steam at this point. Also find the dryness fraction of the steam at cut-off.*

Solution. Swept volume, $V_s = 0.034 \text{ m}^3$

Clearance volume, $V_c = 0.2\ V_s$

Steam consumption $= 2725$ kg/h

Engine speed, $N = 250$ r.p.m.

Pressure of steam near cut-off, $p_1 = 8.8$ bar

Exhaust or back pressure, $p_b = 0.14$ bar.

(*i*) **Mass of cushion steam :**

$$V_c = 0.2V_s = 0.2 \times 0.034 = 0.0068 \text{ m}^3$$

Total volume, $V_2 = V_c + V_s = 0.0068 + 0.034 = 0.0408 \text{ m}^3$

$$V_4 = V_c + 0.35V_s$$

$$= 0.0068 + 0.35 \times 0.034 = 0.0187 \text{ m}^3$$

$$V_1 = V_c + 0.3V_s = 0.0068 + 0.3 \times 0.034$$

$$= 0.017 \text{ m}^3$$

Cylinder feed/stroke $= \frac{2725}{60 \times 250 \times 2} = 0.09083$ kg

Now, *mass of cushion steam* at point 4 is the mass of 0.0187 m^3 dry steam at 0.14 bar.

$= \frac{0.0187}{10.69} =$ **0.001749 kg.** (Ans.)

[From steam tables : Specific volume, v_g of steam at 0.14 bar = 10.69 m^3/kg]

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$$\alpha = \frac{V_4 - V_s}{V_s} = \frac{(0.2\,V_s + V_c) - V_c}{V_s} = \frac{(0.2\,V_s + 0.1\,V_s) - 0.1V_s}{V_s} = 0.2$$

$$\therefore \qquad p_m = 7\left[0.4 + (0.1 + 0.4)\log_e\left(\frac{0.1+1}{0.1+0.4}\right)\right]$$

$$- 1.2\left[(1-0.2) + (0.1+0.2)\log_e\left(\frac{0.2+0.1}{0.1}\right)\right]$$

$$= 5.56 - 1.35 = \textbf{4.21 bar.}\ \text{(Ans.)}$$

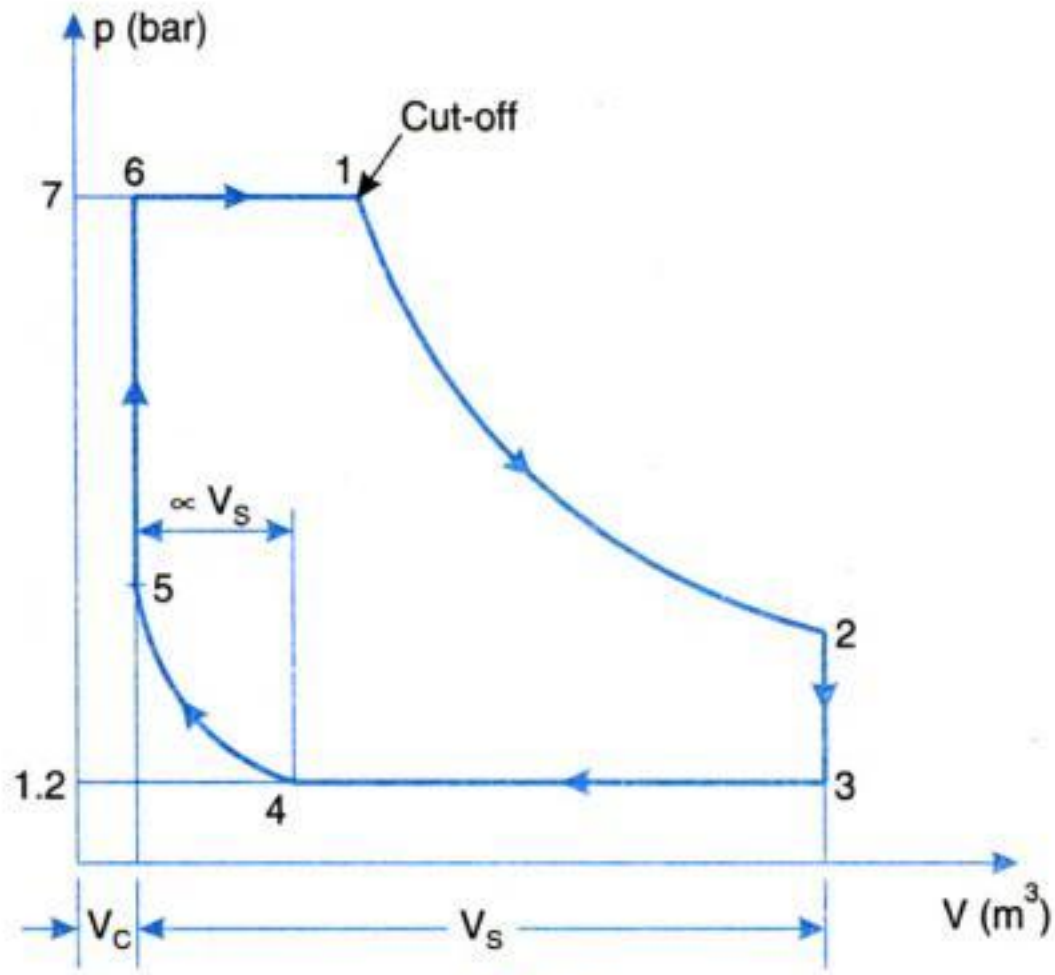

Fig. 16.35

(*ii*) **The diagram factor :**

If the clearance volume is included in the hypothetical or theoretical diagram the mean effective pressure p_m' is given by

$$p_m' = \frac{p_1}{r'}\,(1 + \log_e r') - p_b$$

$$\frac{1}{r'} = \frac{V_1}{V_s} = \frac{0.4\,V_s + V_c}{V_s} = \frac{0.4\,V_s + 0.1\,V_s}{V_s} = 0.5 \text{ and } r' = 2$$

$$\therefore \qquad p_m' = 7 \times 0.5\,(1 + \log_e 2) - 1.2 = 4.72 \text{ bar}$$

Hence diagram factor, D.F. $= \dfrac{p_m}{p_m'} = \dfrac{4.21}{4.72} = \textbf{0.89.}$ (Ans.)

(*iii*) **Power output :**

Since the volume occupied by the liquid is quite small compared with that of the vapour, it may be neglected and specific volume of vapour at cut-off may be written as $x_1 v_g$, x_1 being the quality of vapour, assumed to be 0.9.

Now, mass of steam at cut-off = Mass of cushion steam + mass admitted/stroke

$$= (\alpha V_s + V_c)/1.455 + 2700/(60 \times 2 \times 240)$$

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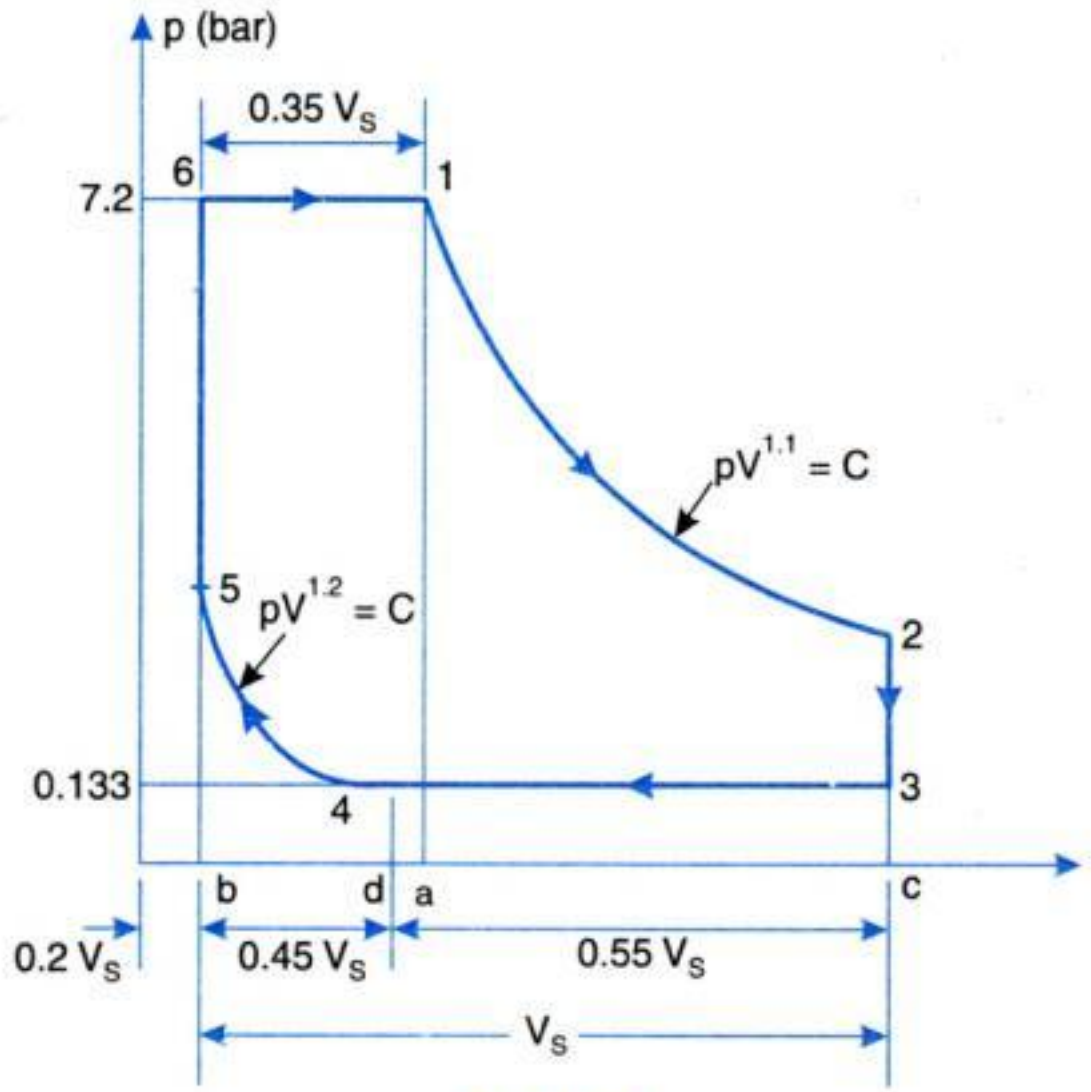

Fig. 16.36

Back pressure, $p_b = \dfrac{760 - 660}{760} \times 1.013 = 0.133$ bar

Considering *expansion curve*, $pV^{1.1}$ = constant

$$p_1 V_1^{1.1} = p_2 V_2^{1.1}$$

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^{1.1} = 7.2 \left[\frac{(0.35\,V_s + 0.2\,V_s)}{(V_s + 0.2\,V_s)}\right]^{1.1}$$

$$= 7.2 \left(\frac{0.55}{1.2}\right)^{1.1} = 3.05 \text{ bar}$$

Considering the *compression curve*, $pV^{1.2}$ = constant

$$p_4 V_4^{1.2} = p_5 V_5^{1.2}$$

$$p_5 = p_4 \times \left(\frac{V_4}{V_2}\right)^{1.2} = 0.133 \times \left[\frac{0.45\,V_s + 0.2\,V_s}{0.2\,V_s}\right]^{1.2}$$

$$= 0.133 \times \left(\frac{0.65}{0.2}\right)^{1.2} = 0.55 \text{ bar.}$$

Also, **mean effective pressure,** $p_m = \dfrac{\text{Area of the indicator diagram}}{\text{Stroke volume}}$

$$= \frac{\text{area '61}ab\text{6' + area '12}ca\text{1' − area '43}cd\text{4' − area '54}db\text{5'}}{V_s}$$

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Inserting the value of b in (i), we get

$$360 = a + 3.75 \times 60$$

$\therefore$ $$a = 135.$$

Steam consumption for developing 120 kW

$$= 135 + 3.75 \times 120 = \textbf{585 kg/h.} \quad \textbf{(Ans.)}$$

Heat supplied/kg $= h_1 - h_{f_2}$

From steam tables :

At 15 bar : $h_{f_1} = 844.7$ kJ/kg, $h_{fg_1} = 1945.2$ kJ/kg.

At 0.36 bar : $h_{f_2} = 307.1$ kJ/kg.

$\therefore$ Heat supplied/kg $= (844.7 + 0.96 \times 1945.2) - 307.1 = 2405$ kJ/kg.

Indicated thermal efficiency (developing 120 kW)

$$\eta_{th(I)} = \frac{\text{I.P.}}{\dot{m}_s (h_1 - h_{f_2})} = \frac{120}{\frac{585}{3600} \times 2405} = \textbf{0.3070 or 30.7\%.} \quad \textbf{(Ans.)}$$

Example 16.12. *The following readings were taken during the test at full load on a single-cylinder, double-acting condensing type throttle governed steam engine :*

Diameter of the cylinder	*400 mm*
Stroke of the engine	*600 mm*
Cut-off	*50% of stroke*
Pressure of steam supplied	*11 bar*
Back pressure	*0.8 bar*
Brake wheel diameter	*4.5 m*
Net load on the brake	*4900 N*
Speed of the engine	*150 r.p.m.*
Diagram factor	*0.82*

(i) Find the indicated power, brake power and mechanical efficiency of the engine at full load.

(ii) If the load on the brake is reduced to 50% of the full load, find the pressure of steam entering the cylinder. Assume the diagram factor at reduced load as 0.82 and mechanical efficiency 70% of the mechanical efficiency at full load.

Solution. Theoretical mean effective pressure,

$$p_{m(th.)} = \frac{p_1}{r} (1 + \log_e r) - p_b$$

$$= \frac{11}{2} (1 + \log_e 2) - 0.8 \qquad \left(r = \frac{1}{0.5} = 2\right)$$

$$= 8.51 \text{ bar}$$

Actual mean effective pressure,

$$p_{m(act.)} = (\text{D.F.}) \times p_{m(th.)} = 0.82 \times 8.51 = 6.98 \text{ bar}$$

(*i*) $\therefore$ $$\text{I.P.} = \frac{10\, p_m\, LAN}{3} = \frac{10 \times 6.98 \times 0.6 \times \pi / 4 \times 0.4^2 \times 150}{3}$$

$$= \textbf{263.14 kW.} \quad \textbf{(Ans.)}$$

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$$20 = \frac{p_m \times 0.20 \times \frac{\pi}{4} \times 0.15^2 \times 300 \times 10}{3} = 3.534\ p_m$$

$$\therefore \quad p_m = \frac{20}{3.534} = 5.659 \text{ bar}$$

$$(p_m)_{\text{th.}} = \frac{p_m}{\text{D.F.}} = \frac{5.659}{0.72} = 7.86 \text{ bar}$$

Also, $$(p_m)_{\text{th.}} = \frac{p_1}{r}\ [1 + \log_e r] - p_b,$$

where, $$r = \left(\text{Expansion ratio} = \frac{V_2}{V_1}\right) = \frac{1}{0.2} = 5$$

$$\therefore \quad 7.86 = \frac{p_1}{5}\ [1 + \log_e 5] - 0.28$$

or $$= 0.522\ p_1 - 0.28 \quad \text{or} \quad p_1 = \frac{(7.86 + 0.28)}{0.522} = \textbf{15.59 bar.}$$ **(Ans.)**

Indicated thermal efficiency, $\eta_{\text{th. (I)}}$:

From steam tables : $h_1 = 2793.4$ kJ/kg at 15.59 bar ($x = 1$) ; $h_{f_2} = 282.7$ kJ/kg at 2.8 bar

$$\text{Heat supplied to steam} = \frac{222}{3600}\ (h_1 - h_{f_2}) = \frac{222}{3600}\ (2793.4 - 282.7)$$

$$= 154.83 \text{ kJ/s}$$

$$\therefore \quad \eta_{\text{th.}(I)} = \frac{20}{154.83} = \textbf{0.1292 or 12.92\%.}$$ **(Ans.)**

Example 16.15. *A double-acting single-cylinder steam engine runs at 250 r.p.m. and develops 30 kW. The pressure limits of operation are 10 bar and 1 bar. Cut-off is 40% of the stroke. The L/D ratio is 1.25 and the diagram factor is 0.75. Assume dry saturated steam at inlet, hyperbolic expansion and negligible effect of piston rod. Find :*

(i) Mean effective pressure, *(ii) Cylinder dimensions, and*

(iii) Indicated thermal efficiency. **(AMIE Summer, 2001)**

Solution. Speed of the steam engine, $N = 250$ r.p.m.

Power developed, $P = 30$ kW

Pressure limits of operation : 10 bar (p_1), 1 bar (p_b)

Cut-off ratio, $r = \frac{1}{0.4} = 2.5$

L/D ratio $= 1.25$

Diagram factor, D.F. $= 0.75$

Condition of steam at inlet to the engine = Dry saturated.

(*i*) **Mean effective pressure, p_m :**

$$p_m = \text{D.F.} \left[\frac{p_1}{r}(1 + \log_e r) - p_b\right]$$

$$= 0.75 \left[\frac{10}{2.5}(1 + \log_e 2.5) - 1\right] \simeq \textbf{5.0 bar.}$$ **(Ans.)**

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OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. In a steam engine D-slide valve controls the flow of steam into and out of the steam engine cylinder
   (*a*) through one admission port and two exhaust ports
   (*b*) through one admission port and one exhaust port
   (*c*) through one exhaust port and two admission ports
   (*d*) through two exhaust ports and two admission ports.
2. In a steam engine
   (*a*) piston rod is connected to cross-head by gudgeon pin
   (*b*) connecting rod is connected to crankshaft by gudgeon pin
   (*c*) connecting rod is connected to piston by gudgeon pin
   (*d*) connecting rod is connected to cross-head by gudgeon pin.
3. For a steam engine the 'diagram factor' is defined as
   (*a*) area of hypothetical indicator diagram × area of actual indicator diagram
   (*b*) $\dfrac{\text{area of actual indicator diagram}}{\text{area of hypothetical indicator diagram}}$
   (*c*) $\dfrac{\text{area of hypothetical indicator diagram}}{\text{area of actual indicator diagram}}$
   (*d*) none of the above.
4. The function of D-slide valve in steam engine is
   (*a*) only to exhaust steam from the cylinder
   (*b*) only to admit steam in the cylinder
   (*c*) to admit steam and also exhaust steam from the cylinder
   (*d*) none of the above.
5. The component that imparts motion to the D-slide valve of steam engine is
   (*a*) flywheel (*b*) eccentric
   (*c*) crank (*d*) cam.
6. For throttle governing of steam engine
   (*a*) rate of steam consumption varies inversely as the speed of the engine
   (*b*) rate of steam consumption varies directly as the speed of the engine
   (*c*) rate of steam consumption is directly proportional to I.P. of the engine
   (*d*) rate of steam consumption is inversely proportional to I.P. of the engine.
7. Willan's line is a curve plotted of steam consumption rate in steam engines versus
   (*a*) speed (*b*) I.P.
   (*c*) F.P. (*d*) B.P.

ANSWERS

1. (*c*) 2. (*d*) 3. (*b*) 4. (*c*) 5. (*b*) 6. (*b*) 7. (*b*).

THEORETICAL QUESTIONS

1. Define an heat engine. How are heat engines classified ?
2. Define a reciprocating steam engine and state how the reciprocating steam engines are classified.
3. Using neat sketches enumerate and explain the various parts of a steam engine (reciprocating).
4. Explain the following terms as applied to steam engine :
   Cylinder bore, Piston stroke, Crank throw, Average piston speed, Dead centres, Swept volume and clearance.

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17 Compound Steam Engines

17.1. Introduction. 17.2. Advantages of compound steam engines. 17.3. Classification of compound steam engines. 17.4. Multi-cylinder engines. 17.5. Estimation of cylinder dimensions (compound steam engines). 17.6. Causes of loss of thermal efficiency in compound steam engines. 17.7. The governing of compound steam engines. 17.8. Uniflow steam engine—Worked Examples—Additional/Typical Examples—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

17.1. INTRODUCTION

A simple steam engine is one in which each of the engine cylinder receives steam direct from the boiler, and exhausts into the atmosphere or into a condenser. Compound steam engines are those which have two or more cylinders of successively increasing diameters so arranged that exhaust steam from the first and smallest cylinder is passed forward to do work in a second, and sometimes a third or fourth cylinder, before escaping to the condenser.

'Compounding' is primarily done to secure a *large total ratio of expansion* without the disadvantages attached to a very early cut-off in a single-cylinder. These ***disadvantages*** are :

(*i*) A large range of temperature, accompanied by excessive condensation and re-evaporation effects unless high superheats are used. The range of temperature is not merely the difference between the initial temperature and the exhaust temperature, but includes the fact that the cylinder is heating up during admission and cooling during expansion and exhaust so that with an early cut-off the ratio of the cooling period to the heating period is large.

(*ii*) A cut-off earlier than 30% of the stroke, with an ordinary slide valve, is accompanied by an early release and excessive compression.

(*iii*) When the cut-off is very early the evil effect of clearance is very much accentuated.

(*iv*) With an early cut-off the turning moment on the shaft is more uneven ; as a result a greater flywheel effect is required. This effect, however, is reduced if a number of cylinders and cranks are used.

17.2. ADVANTAGES OF COMPOUND STEAM ENGINES

The compound steam engines claim the following **advantages** over single stage steam engines :

1. Owing to reduction in temperature range per cylinder, the cylinder condensation is minimised.
2. As there is re-evaporation in the early stages of expansion the steam can be expanded still further in the later stages. Consequently, the loss due to condensation in compound steam engines is not cumulative and is generally restricted to low pressure cylinder.
3. The leakage past the valves and piston is minimised since the pressure difference across these components is less.

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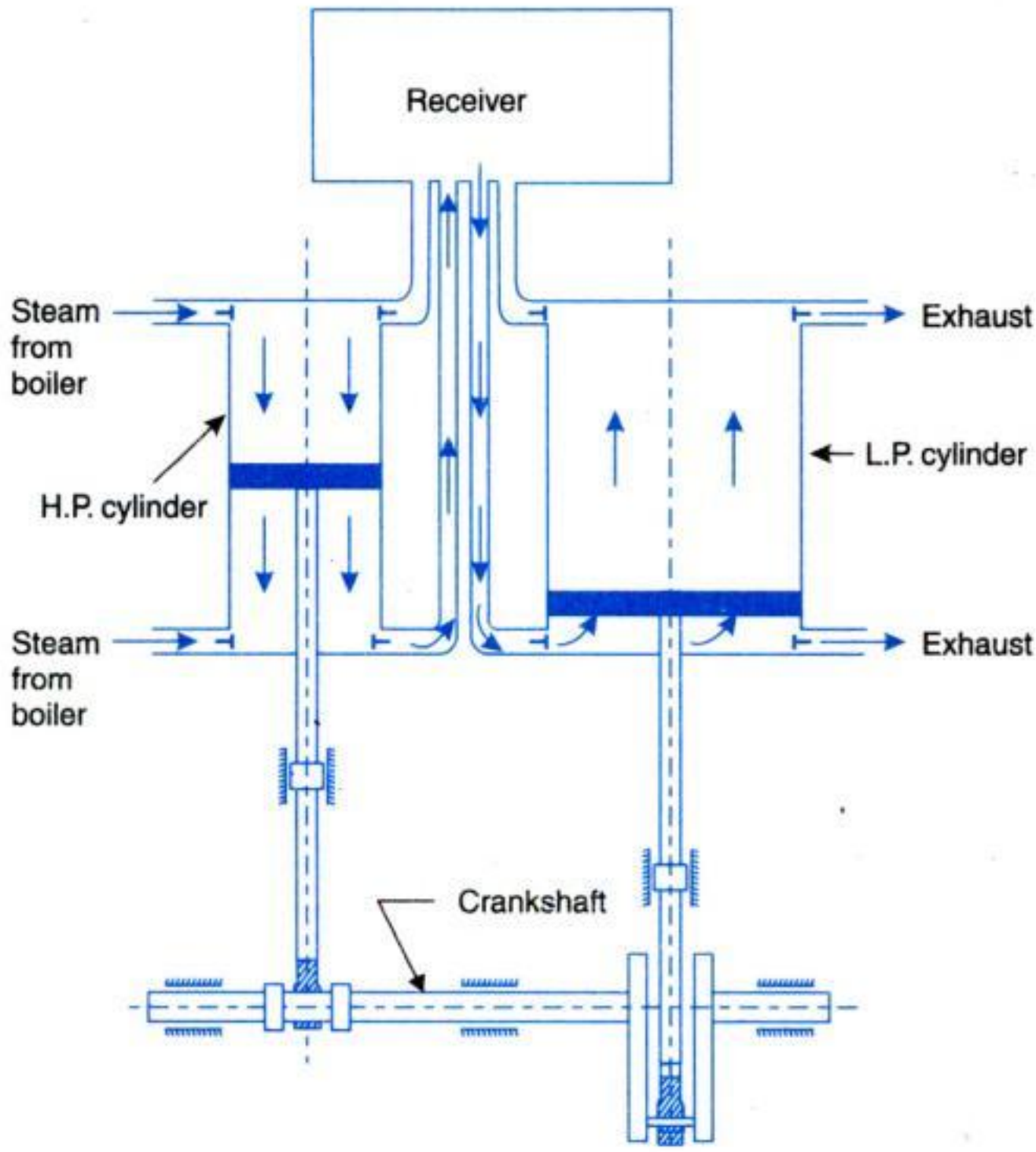

Fig. 17.4. Receiver type compound steam engine.

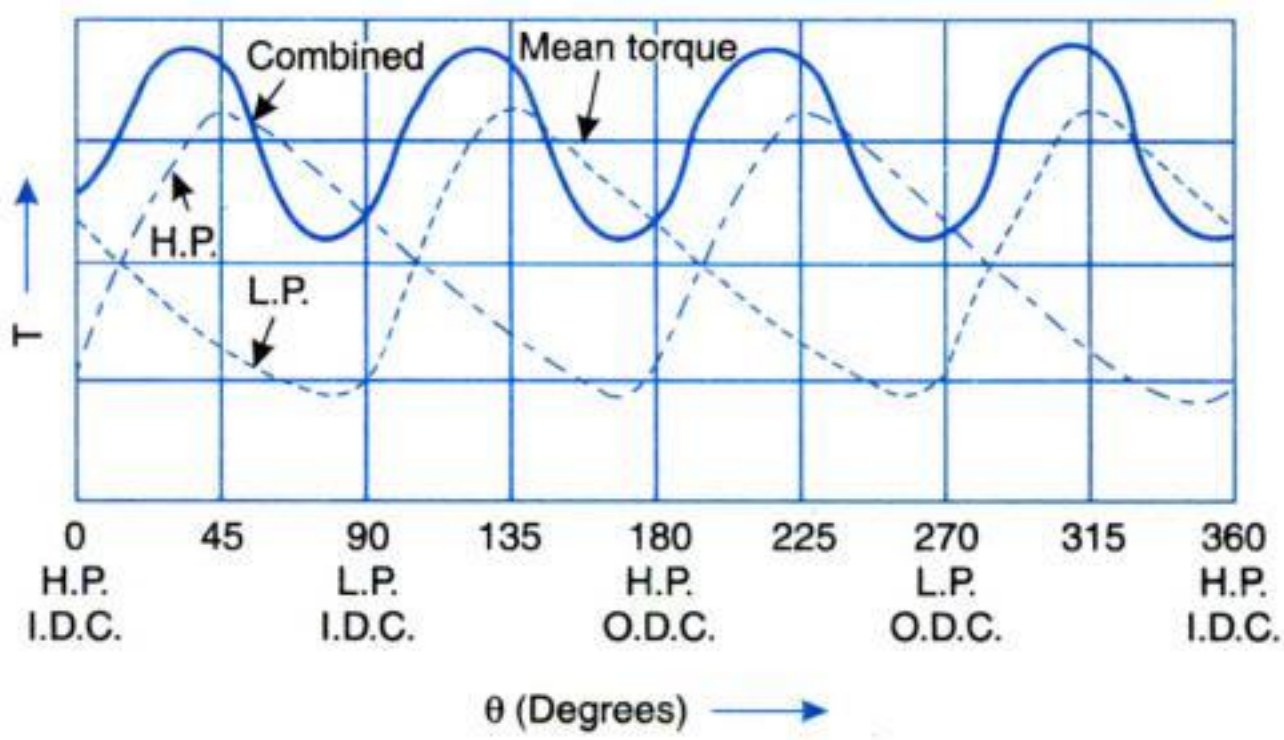

Fig. 17.5. T-θ curve.

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(*i*) ***If work done in each cylinder is same,*** then

$$p_1V_1\left(1+\log_e\frac{V_2}{V_1}\right)-p_3V_2=p_3V_3\left(1+\log_e\frac{V_4}{V_3}\right)-p_bV_4$$

$$=\frac{1}{2}\times\left[p_1V_1\left(1+\log_e\frac{V_4}{V_1}\right)-p_bV_4\right] \quad ...(17.4)$$

(*ii*) ***If the initial load/force on the pistons is same***, then

$$(p_1-p_3)\frac{\pi}{4}D^2_{\text{H.P.}}=(p_3-p_b)\frac{\pi}{4}D^2_{\text{L.P.}} \quad ...(17.5)$$

From the above equations, the required dimensions can be found as per data supplied and assumptions made.

17.6. CAUSES OF LOSS OF THERMAL EFFICIENCY IN COMPOUND STEAM ENGINES

Initial condensation becomes increasingly serious as the expansion ratio is increased and as the temperature range across the cylinder becomes greater. The most effective solution adopted in practice is that of compounding, but nevertheless the initial condensation and leakage occur on a reduced scale in each stage of the expansion. The pressure-drop in the receiver, which includes the wire-drawing at the exhaust and inlet ports and valves, is a further loss, resulting in an increase of entropy due to unresisted expansion, and overall available heat is reduced.

17.7. THE GOVERNING OF COMPOUND STEAM ENGINES

There are three methods of governing compound engines. These are :

1. **Throttle governing.** *Reducing the steam supply pressure in the H.P. cylinder* ;
2. **Cut-off governing on H.P. cylinder.** *Varying the point of cut-off in the H.P. cylinder*; and
3. **Cut-off governing on L.P. cylinder.** *Varying the point of cut-off in the L.P. cylinder.*

For simplicity, consider an engine with a cylinder volume ratio of 5 : 2, and with cut-off in the H.P. cylinder at full load of $\frac{1}{2}$. Let the back pressure in the L.P. cylinder be zero.

Fig. 17.9 illustrates the above three methods, the full load diagrams being 1-2-3-4-5-6-1, 1′-2′-3′-4′-5′-6′-1′, 1″-2″-3″- 4″-5″-6″-1″ respectively.

Throttle governing. Refer Fig. 17.9 (*a*). Suppose the full load pressure 6-1 is reduced to half by throttling, that is to 6-7, the cut-off in each cylinder remaining the same. The light load diagram will become 7-8-10-5-6-7. The high pressure work will be reduced from 1-2-3-7-1 to 7-8-9-11-7 and the low pressure work will be reduced from 7-3-4-5-6-7 to 11-9-10-5-6-11.

High pressure cut-off governing. Refer Fig. 17.9 (*b*). Suppose for the light load the cut-off in high pressure is reduced from $\frac{1}{2}$ to $\frac{1}{4}$, the initial pressure and the cut-off in low pressure remaining constant. The light load diagram will become 1′-8′-10′-5′-6′-1′, the high pressure work becoming 1′-8′-9′-11′-1′, which is not very different from 1′-2′-3′-7′-1′, while the low pressure work is reduced from 7′-3′-4′-5′-6′-7′ to 11′-9′-10′-5′-6′-11′. *Governing by varying the cut-off in the H.P. cylinder will therefore reduce the proportion of work done in the L.P. cylinder at light loads. With condensing engines at very light loads this may cause the average pressure to overcome the back pressure and frictional resistance, thus reducing the efficiency of the engine.*

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Solution. Power to be developed, I.P. = 90 kW

Engine speed, $N = 100$ r.p.m.

Admission steam pressure, $p_1 = 7.5$ bar

Condenser pressure, $p_b = 0.21$ bar

Expansion ratio, $R = 15$

Diagram factor, D.F. = 0.72.

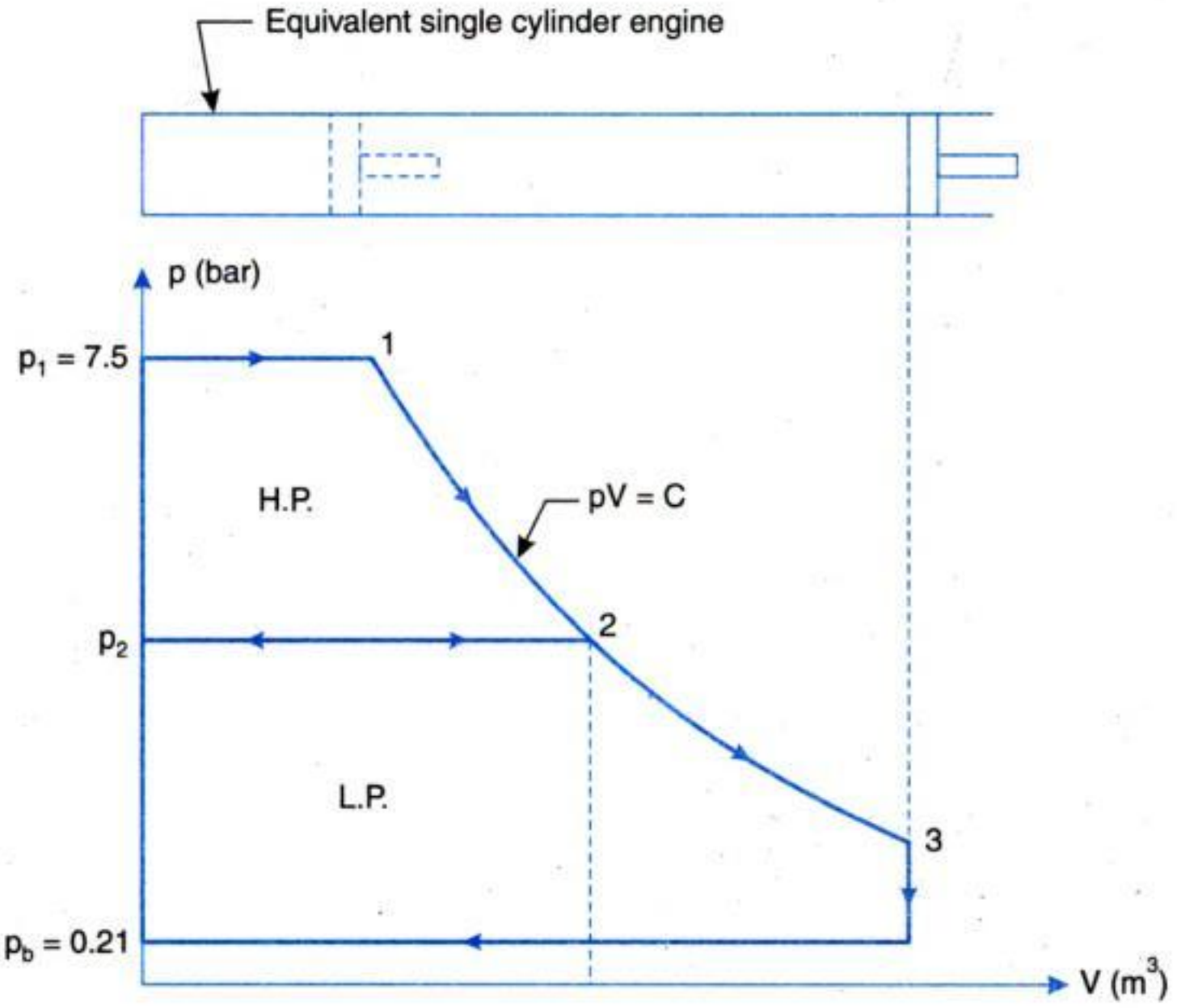

Fig. 17.12

Cylinder diameters, $D_{L.P.}$, $D_{H.P.}$:

Stroke of each piston = L.P. cylinder diameter

$p_{m(\text{actual})}$ referred to L.P. cylinder

$$= \text{D.F.} \left[\frac{p_1}{R}(1+\log_e R) - p_b\right]$$

$$= 0.72 \left[\frac{7.5}{15}(1+\log_e 15) - 0.21\right] = 1.18 \text{ bar}$$

Indicated power, $$\text{I.P.} = \frac{10\, p_{m(actual)}\, LAN}{3} = \frac{10 \times 1.18 \times D_{\text{L.P.}} \times \frac{\pi}{4} D^2_{\text{L.P.}} \times 100}{3}$$

$$= 90 = 308.92\ D^3_{\text{L.P.}}$$

$$\therefore \quad D_{\text{L.P.}} = \left(\frac{90}{308.92}\right)^{1/3} = 0.6629 \text{ m} \simeq 663 \text{ mm}$$

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☞**Example 17.3.** *A two-cylinder compound, double-acting steam engine is required to develop 62.5 kW (brake) at 350 r.p.m. when supplied with steam at 20 bar and exhausting to a condenser at 0.2 bar. Cut-off ratio in both cylinders to be 0.4. The stroke length for both cylinders is 250 mm. Estimate the cylinder diameters to develop equal work.*

Neglect clearance and assume hyperbolic expansion.

Take, mechanical efficiency = 85% and diagram factor for each cylinder = 0.8.

Solution. Power to be developed, B.P. = 62.5 kW

Engine speed, N = 350 r.p.m.

Steam supply pressure, p_1 = 20 bar

Condenser/back pressure, p_2 = 0.2 bar

Cut-off ratio in both cylinders = 0.4

Stroke length (for both cylinders), L = 250 mm

Mechanical efficiency = 85%, Diagram factor = 0.8] for each cylinder

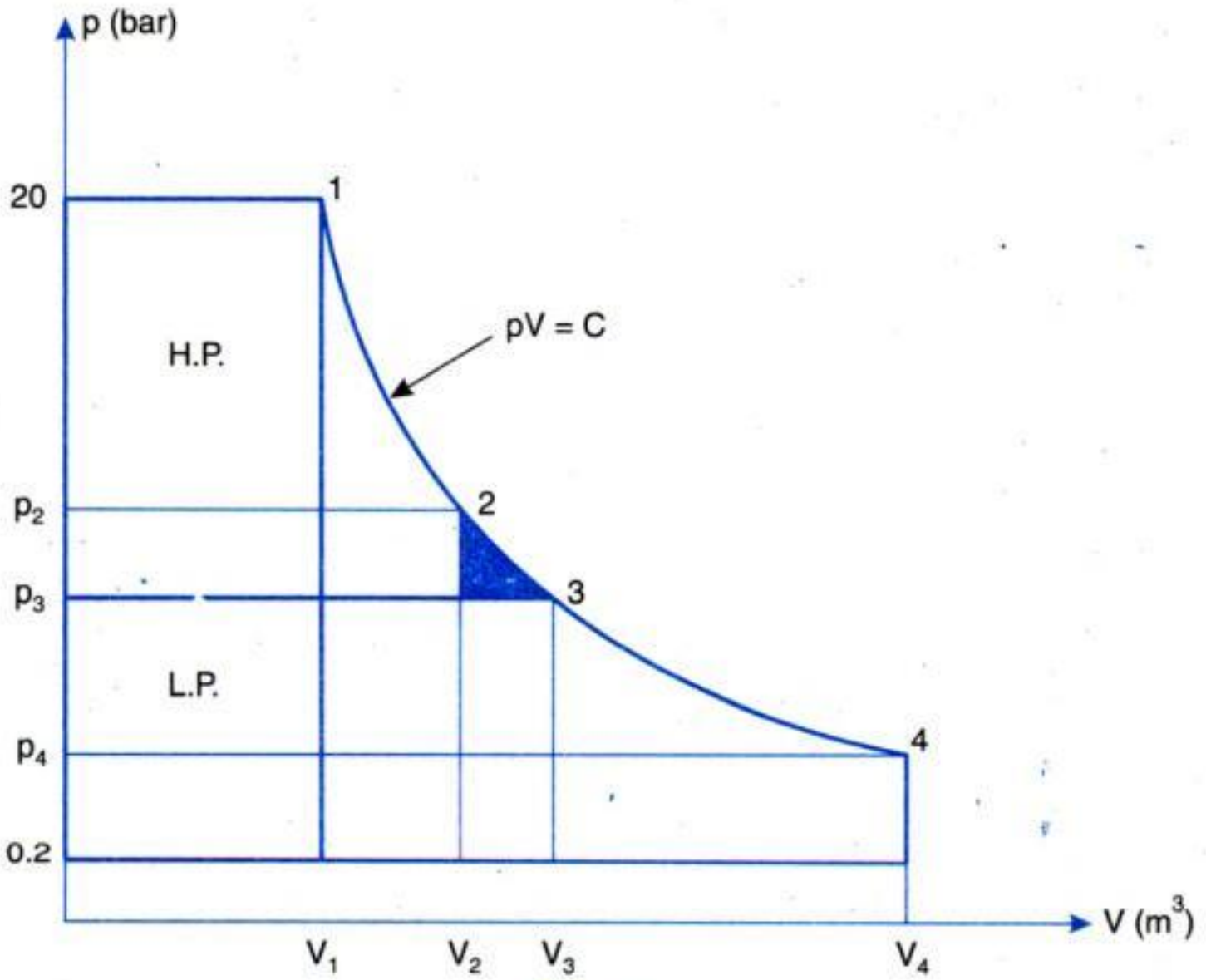

Fig. 17.14

Cylinder diameters :

Refer Fig. 17.14.

For equal work done in both cylinders,

$$\text{D.F.}\left[p_1V_1\left\{1+\log_e\frac{V_2}{V_1}\right\}-p_3V_2\right]=\text{D.F.}\left[p_3V_3\left\{1+\log_e\left(\frac{V_4}{V_3}\right)\right\}-p_bV_4\right]$$

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where 0.777 m^3/kg is the specific volume at 2.3 bar.

Total volume at cut-off $= \frac{V_2'}{10} + 0.5\ V_2' = 0.6\ V_2'$

But $0.6\ V_2' = 0.17\left(1 + \frac{V_2'}{7.77}\right) = 0.17 + \frac{0.17\ V_2'}{7.77}$

or $0.6\ V_2' = 0.17 + 0.0218\ V_2'$

$\therefore$ $V_2' = \frac{0.17}{(0.6 - 0.0218)} = 0.294\ m^3$

and $V_4' = 5V_2' = 5 \times 0.294 = 1.47\ m^3$

Work done/kg of steam $= p_{m(H.P.)}\ V_2' + p_{m(L.P.)} \times V_4'$

$= 10^5\ (7.63 \times 0.294 + 1.846 \times 1.47)$ N-m

$= 4.957 \times 10^5$ N-m

$\therefore$ Ideal indicated power,

$$\text{I.P.} = \frac{4.957 \times 10^5 \times 1090}{3600 \times 1000} = 150\ \text{kW}$$

Actual I.P. allowing for a diagram factor of 0.75

$= 0.75 \times 150 =$ **112.5 kW.** (Ans.)

☞**Example 17.5.** *A triple-expansion engine is required to develop 2940 kW under the following conditions :*

Pressure in the steam chest = *15 bar*
Piston speed = *210 m/min*
Exhaust pressure = *0.15 bar*
Cylinder volume ratios = *1 : 2.4 : 7.2*
Total ratio of expansion = *18*
Overall diagram factor = *0.62.*

Assuming equal initial loading on each piston, determine :

(*i*) *Cylinder diameters*
(*ii*) *Mean receiver pressures*
(*iii*) *Cut-off points in each cylinder.*

Assume hyperbolic expansion and neglect clearance.

Solution. Refer Fig. 17.16.

Let $V_2 = 1$ unit.

Since $V_2 : V_4 : V_6 = 1 : 2.4 : 7.2$ (given)

$\therefore$ $V_4 = 2.4$ units and $V_6 = 7.2$ units.

Also, total ratio of expansion, $R = V_6/V_1 = 18$

$\therefore$ $V_1 = \frac{V_6}{18} = \frac{7.2}{18} = 0.4$ units.

(*i*) **Cylinder diameters :**

Mean effective pressure referred to L.P. cylinder,

$$p_m = \text{D.F.} \left[\frac{p_1}{R}(1 + \log_e R) - p_b\right]$$

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ADDITIONAL/TYPICAL EXAMPLES

Example 17.7. *The following data refer to a double-acting compound steam engine :*

I.P.	*= 365 kW*
R.P.M.	*= 420*
Stroke	*= 60 cm*
Admission pressure	*= 10 bar*
Back pressure	*= 0.3 bar*
Expansion ratio	*= 10*
Diagram factor	*= 0.8*

Assuming complete expansion in H.P. cylinder and equal initial load, expansion follows the law pV = constant, and neglecting clearance, determine :

(i) The admission pressure for the low pressure cylinder.

(ii) The diameter of each cylinder.

Solution. I.P. = 365 kW, N = 420 r.p.m., L = 60 cm = 0.6 m,

$$p_1 = 10 \text{ bar}, p_b = 0.3 \text{ bar}, R = \frac{V_3}{V_1} = 10, \text{D.F.} = 0.8$$

(i), (ii) : $p_2 = ?, D_{\text{L.P.}} = ?, D_{\text{H.P.}} = ?$

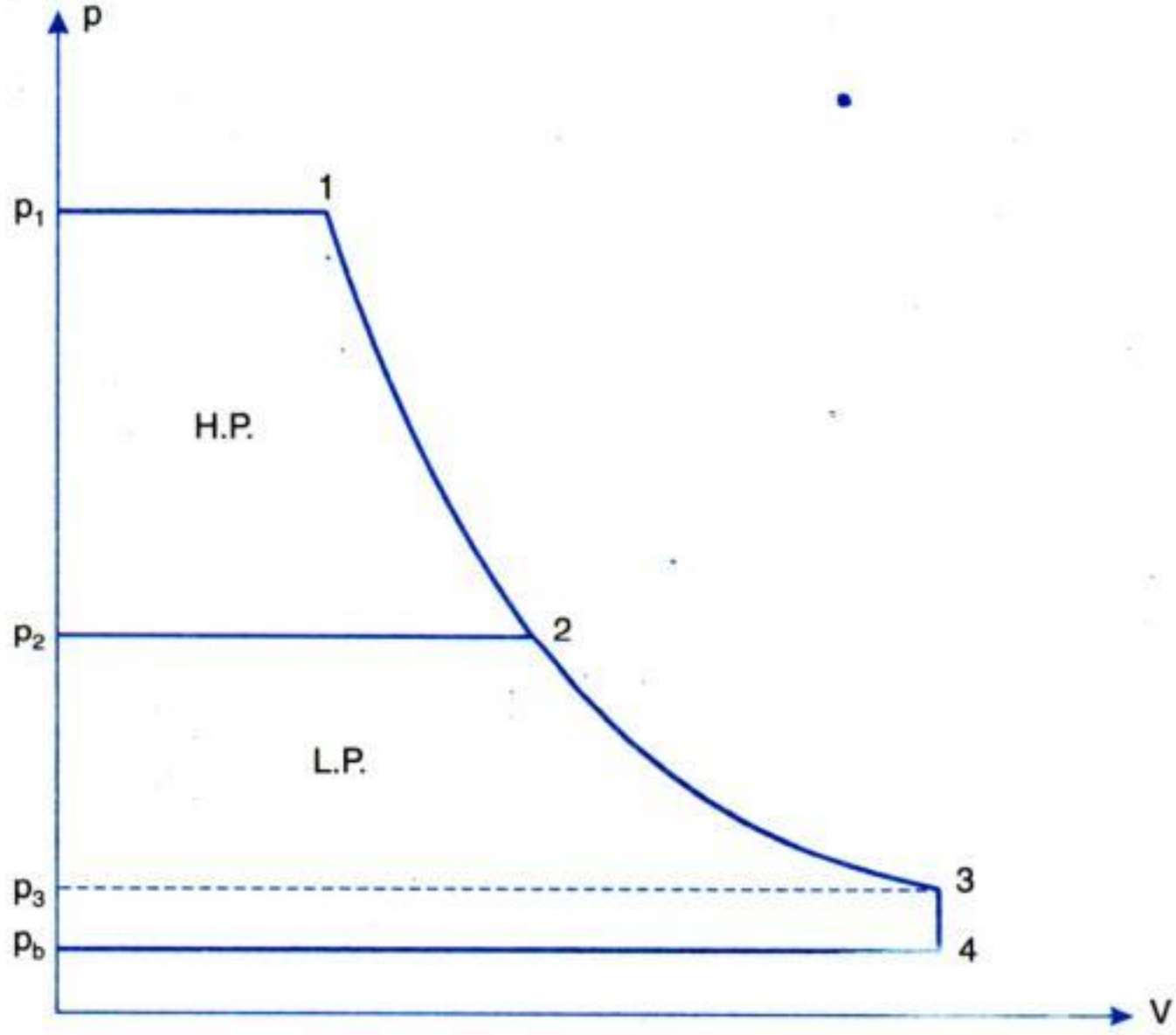

Fig. 17.17

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Since power developed in each cylinder is equal, work done in H.P. cylinder will be equal to the work done in L.P. cylinder.

$$\therefore \quad p_1V_1\left(1+\log_e\frac{V_2}{V_1}\right)-p_2V_2=\frac{1}{2}\left[p_1V_1\left(1+\log_e\frac{V_3}{V_1}\right)-p_bV_3\right]$$

Dividing both sides by p_1V_1 and considering $p_1V_1 = p_2V_2$, we have

$$\text{or} \qquad \log_e\frac{V_2}{V_1}=\frac{1}{2}\left[1+\log_e 15-\frac{0.21}{7.35}\times 5\right]$$

$$\therefore \qquad \frac{V_2}{V_1}=5.154$$

$$\therefore \text{ Cylinder volume,} \qquad V_2=5.154\times\frac{\left(\frac{\pi}{4}\times 0.657^2\times 0.657\right)}{15}=\frac{\pi}{4}\,D^2_{\text{H.P.}}\times 0.657$$

$$\text{or} \qquad D_{\text{H.P.}}=0.385 \text{ m or } 385 \text{ mm}$$

i.e., *Diameter of H.P. cylinder* **= 385 mm.** **(Ans.)**

HIGHLIGHTS

1. **'Compounding'** is primarily done to secure a large total ratio of expansion.
2. The compound steam engines are usually classified as follows :
   - (*i*) Tandem compound steam engines
   - (*ii*) Cross compound steam engines
     - (*a*) Woolfe type
     - (*b*) Receiver type.
3. In a 'double expansion' engine the steam expands in two cylinders.
   In a 'triple expansion' engine the expansion of steam is completed in three cylinders.
4. The following are the methods of governing compound steam engines :
   - (*i*) **Throttle governing.** Reducing the steam supply pressure in the H.P. cylinder.
   - (*ii*) **Cut-off governing on H.P. cylinder.** Varying the point of cut-off in the H.P. cylinder.
   - (*iii*) **Cut-off governing on L.P. cylinder.** Varying the point of cut-off in the L.P. cylinder.
5. Uniflow engines are fairly simple in design and of robust construction.

OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. The simple steam engine as compared to compound steam engine for the same output has
   (*a*) smaller flywheel (*b*) large flywheel
   (*c*) same size flywheel as output is same (*d*) same size flywheel as speed is same.
2. The leakage past the piston and initial condensation by compounding the steam engine is
   (*a*) increased (*b*) decreased
   (*c*) unaffected (*d*) depends on methods of compounding.
3. Woolfe type compound steam engines have
   (*a*) two cranks at 180° phase difference (*b*) two cranks at 90° phase difference
   (*c*) only one crank with no phase difference (*d*) two cranks with no phase difference.

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18 Steam Nozzles

18.1. Introduction. 18.2. Steam flow through nozzles—Velocity of steam—Discharge through the nozzle and conditions for its maximum value. 18.3. Nozzle efficiency. 18.4. Supersaturated or metastable expansion of steam in a nozzle. 18.5. General relationship between area, velocity and pressure in nozzle flow. 18.6. Steam injector—Worked examples—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

18.1. INTRODUCTION

*A **steam nozzle** may be defined as a passage of varying cross-section, through which heat energy of steam is converted to kinetic energy. Its major function is to produce steam jet with high velocity to drive steam turbines.* A turbine nozzle performs *two functions* :

(*i*) It transforms a portion of energy of steam (obtained from steam generating unit) into kinetic energy.

(*ii*) In the impulse turbine it directs the steam jet of high velocity against blades, which are free to move in order to convert kinetic energy into shaft work. In reaction turbines the nozzles which are free to move, discharge high velocity steam. The reactive force of the steam against the nozzle produces motion and work is obtained.

The cross-section of a nozzle at first tapers to a smaller section (*to allow for changes which occur due to changes in velocity, specific volume and dryness fraction as the steam expands*) ; the smallest section being known as ***throat,*** and then it diverges to a large diameter. The nozzle which converges to throat and diverges afterwards is known as ***convergent-divergent*** nozzle (Fig. 18.1). In convergent nozzle there is no divergence after the throat as shown in Fig. 18.2.

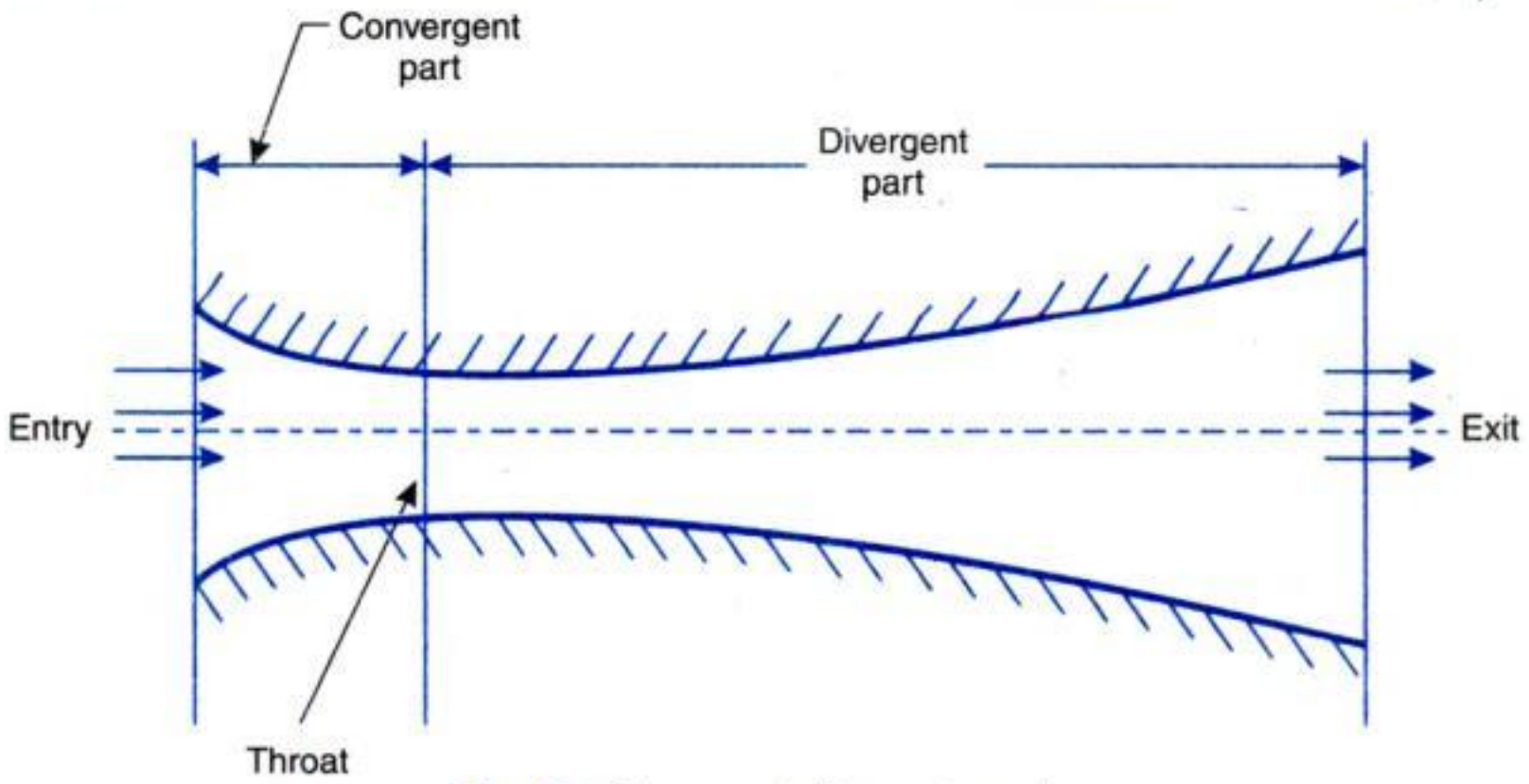

Fig. 18.1. Convergent-divergent nozzle.

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or $$\left(\frac{p_2}{p_1}\right)^{2-n-1} = \left(\frac{n+1}{2}\right)^n$$

or $$\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \qquad \text{...(18.9)}$$

Hence the discharge through the nozzle will be the *maximum* when critical pressure ratio, *i.e.*,

$$\frac{\text{Throat pressure}}{\text{Inlet pressure}} = \frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

For saturated steam : $n = 1.135$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}} = 0.58$$

For superheated steam : $n = 1.3$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} = \left(\frac{2}{2.3}\right)^{\frac{1.3}{0.3}} = 0.546$$

Substituting the value of $\frac{p_2}{p_1}$ from eqn. (18.9) into eqn. (18.8), we get the maximum discharge,

$$m_{max} = \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left[\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{2}{n}} - \left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{n+1}{n}}\right]}$$

$$= \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}-\frac{n+1}{n-1}} - 1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}} - 1\right]}$$

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condense until it density is about 8 times that of the saturated vapour of the same pressure. This effect is discussed below :

Refer Fig. 18.4. The point 1 represents initial state of the steam. The steam expands isentropically without any condensation to point 2, 2 being on the superheat constant pressure curve *AB* produced. At point 2 the limit of supersaturation is reached and steam reverts to its normal condition at 3 at the same enthalpy value as 2, and at the same pressure. The steam continues expanding isentropically to a lower pressure to point 4 instead of 4′ which would have been reached if thermal equilibrium had been maintained. Consequently, enthalpy drop is reduced and the condition of the final steam is improved. *The limiting condition of under-cooling at which condensation commences and is assumed to restore conditions of normal thermal equilibrium is called the* ***"Wilson Line"***.

It may be noted that when metastable conditions prevail the h-s chart/diagram should not be used and the expansion must be considered to follow the law $pv^{1.3} = C$, *i.e., with the index of expansion for superheated steam.* Thus,

$$\text{Enthalpy drop} = \frac{n}{(n-1)} p_1 v_1 \left[1 - \left(\frac{p_2}{p_2}\right)^{\frac{n-1}{n}}\right]$$

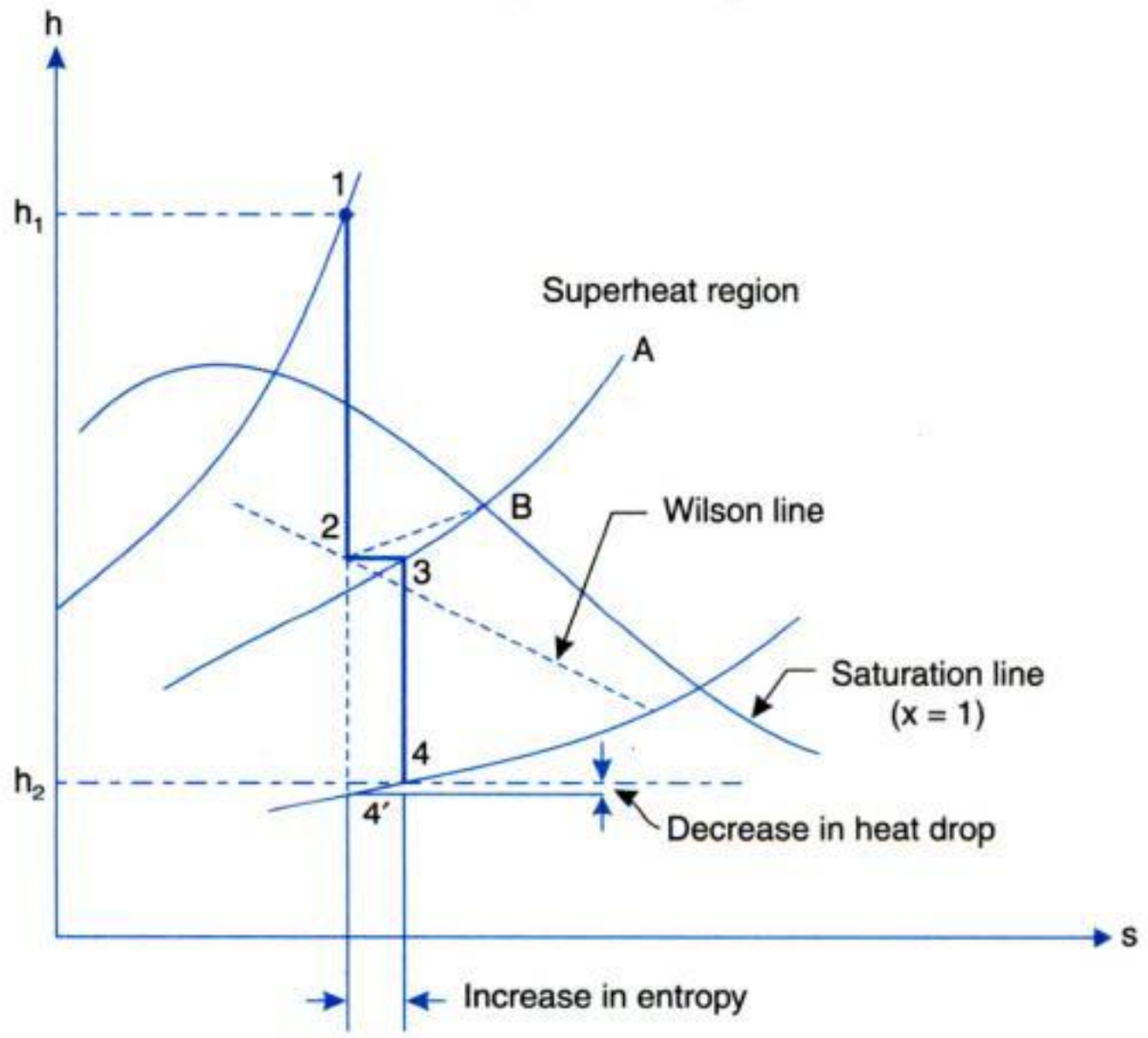

Fig. 18.4. Supersaturated flow of steam.

The relationship, $T_2/T_1 = \left(p_2/p_1\right)^{\frac{n-1}{n}}$ may be used to calculate *supercooled temperature.* The *'degree of undercooling' is then the difference between the saturation temperature and the supercooled temperature.*

Effects of supersaturation. In a nozzle in which supersaturation occurs the **effects** may be summarised as follows :

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18.6. STEAM INJECTOR

A steam injector is employed to force water into the boiler under pressure. It makes use of the principle of a steam nozzle by which it utilises the kinetic energy of a steam jet for increasing the pressure and velocity of a corresponding quantity of water.

The arrangement of injector for feeding the water from a tank to a boiler is shown in Fig. 18.7. The steam from the boiler is expanded to a high velocity be passing 'it through a convergent nozzle'. The steam jet enters the mixing cone and imparts its momentum to the incoming water supply from the feed water tank (the tank may be above or below the level of the steam injector). The cold water causes the steam to condense. The resulting jet at 2, formed by the steam and water is at atmospheric pressure and has a large velocity. The mixture then enters the delivery pipe at 3 through a diverging cone or diffuser, in which the kinetic energy is reduced and converted into pressure energy. This pressure energy is sufficient to overcome the boiler pressure and lifts the water through H_b. The pressure of water on leaving the delivery pipe must be about *20% higher than the boiler pressure in order to overcome all resistances.* The gap between the mixing cone and diverging cone is provided with an outlet through which any excess water may overflow during starting of steam injector.

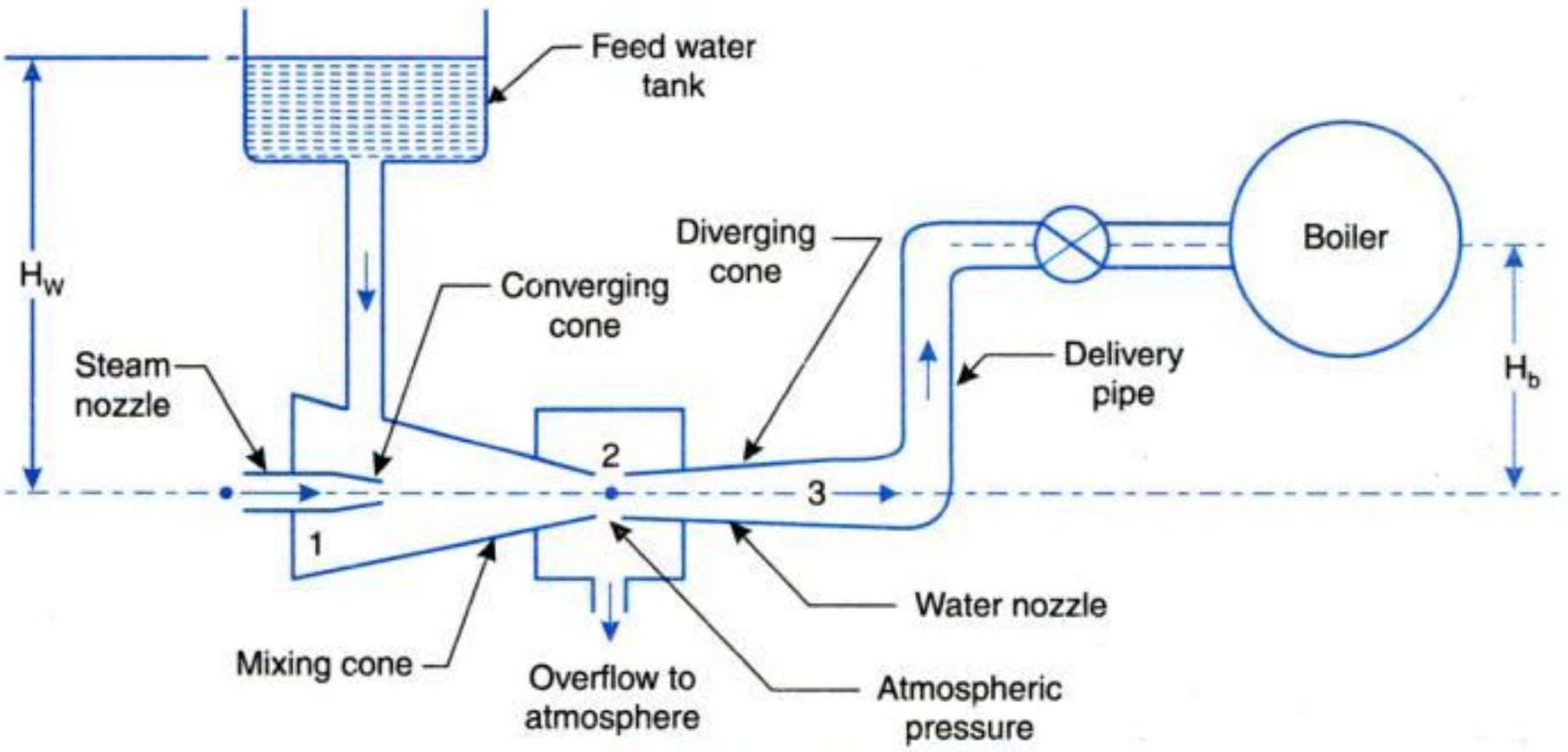

Fig. 18.7. Steam injector.

Let, C_s = Velocity of steam leaving the nozzle,

C_w = Velocity of water at entry to the steam nozzle,

C_m = Velocity of mixture leaving the combined nozzle or entering the water nozzle,

C_{wd} = Velocity of water in the delivery pipe,

m_w = Mass of water drawn from feed water tank per kg of steam supplied to the steam nozzle,

m_s = Mass of steam supplied per sec,

H_w = Water level over the steam injector,

H_b = Boiler inlet level over the steam injector, and

p_b = Absolute pressure inside the boiler.

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Also, $\dot{m} = \dfrac{A_2 C_2}{v_2} = \dfrac{A_2 C_2}{x_2 v_{g_2}}$ (where $\dot{m}$ = Mass flow rate in kg/s)

or Throat area, $A_2 = \dfrac{\dot{m}\, x_2\, v_{g_2}}{C_2} = \dfrac{\dot{m} \times 0.96 \times 0.297}{449.4} = 0.000634\ \dot{m}$

From Mollier chart, $x_3 = 0.9$

From steam tables, $v_{g_3} = 0.885\ \text{m}^3/\text{kg}$ (at 2 bar)

h_d' = Adiabatic heat drop between inlet and exit
$= h_1 - h_3 = 2780 - 2480 = 300\ \text{kJ/kg}$

Exit velocity, $C_3 = 44.72\ \sqrt{h_d'} = 44.72\ \sqrt{300} =$ **774.6 m/s.** **(Ans.)**

Exit area, $A_3 = \dfrac{\dot{m} x_3\, v_{g_3}}{C_3} = \dfrac{\dot{m} \times 0.9 \times 0.885}{774.6} = 0.001028\ \dot{m}$

$\therefore$ Ratio of $\dfrac{\text{exit area}}{\text{throat area}} = \dfrac{0.001028\,\dot{m}}{0.000638\,\dot{m}} =$ **1.62.** **(Ans.)**

Example 18.9. *The nozzles of a Delaval steam turbine are supplied with dry saturated steam at a pressure of 9 bar. The pressure at the outlet is 1 bar. The turbine has two nozzles with a throat diameter of 2.5 mm. Assuming nozzle efficiency as 90% and that of turbine rotor 35%, find the quality of steam used per hour and the power developed.*

Solution. $p_1 = 9$ bar, $p_3 = 1$ bar, p_2 = throat pressure, Number of nozzles = 2

We know that, $\dfrac{p_2}{p_1} = \left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$

Since the steam is dry and saturated, $n = 1.135$

$\therefore$ $\dfrac{p_2}{p_1} = \left(\dfrac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = 0.58$

or $p_2 = 0.58\ p_1 = 0.58 \times 9 = 5.22$ bar.

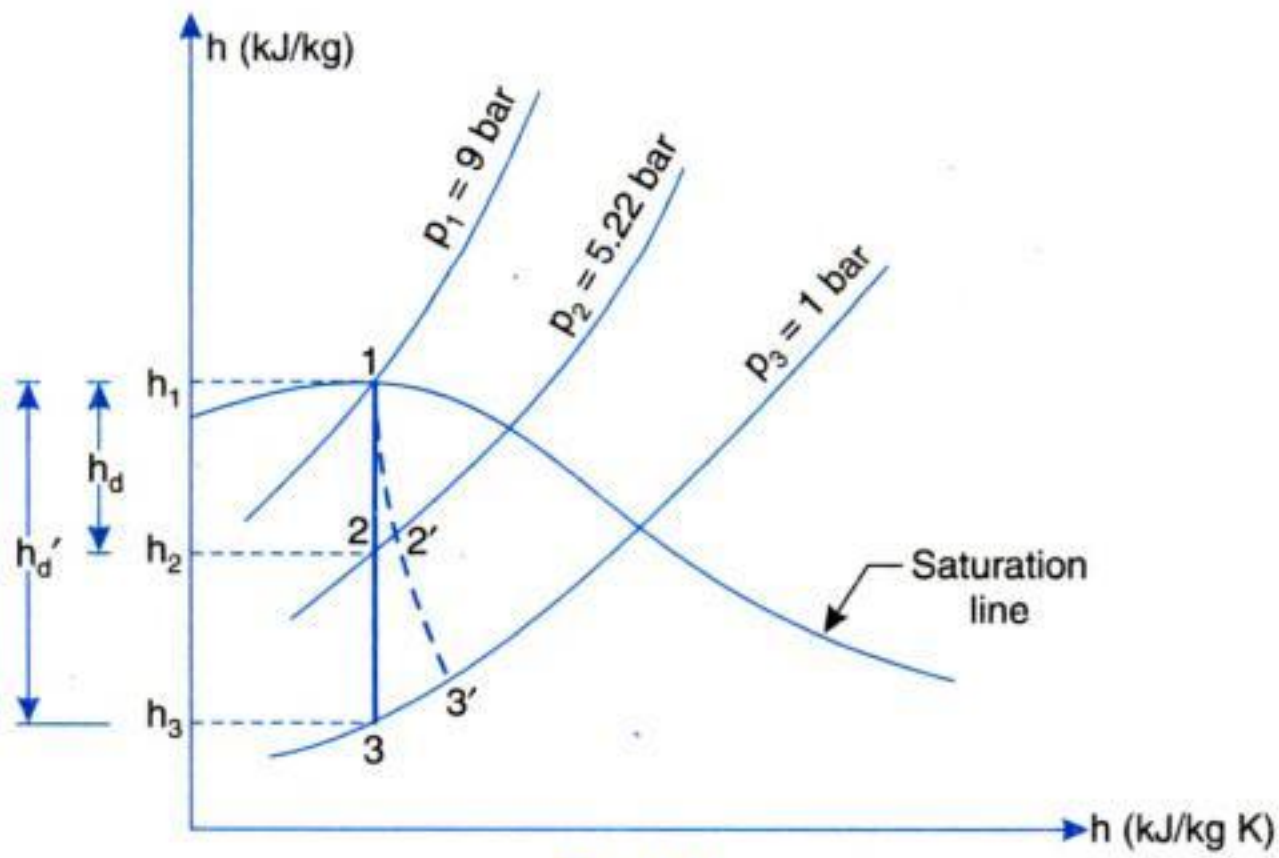

Fig. 18.12

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Now, $\dot{m} = \dfrac{A_2 C_2}{v_2} = \dfrac{A_2 \times 509.8}{0.287}$

$\therefore \quad A_2 = \dfrac{\dot{m} \times 0.287}{509.8} = \dfrac{500 \times 0.287}{3600 \times 509.8} = 7.82 \times 10^{-5}\ \text{m}^2$

i.e., $\dfrac{\pi}{4} D_2^2 = 7.82 \times 10^{-5}$

or $D_2 = \left(\dfrac{7.82 \times 10^{-5} \times 4}{\pi}\right)^{1/2} = 0.009978\ \text{m or } 9.9\ \text{mm}$

i.e., *Throat diameter* = **9.9 mm.** **(Ans.)**

At exit :

$C_3' = 44.72 \sqrt{k h_d'} = 44.72 \sqrt{(1 - 0.12) \times 490} = 928.6\ \text{m/s}$

$v_3' = x_3'\ v_{g_3} = 0.921 \times 1.69 = 1.556\ \text{m}^3/\text{kg}$

$\therefore \quad A_3 = \dfrac{\dot{m}\, v_3'}{C_3'} = \dfrac{500 \times 1.556}{3600 \times 928.6} = 0.0002327\ \text{m}^2$

i.e., $\dfrac{\pi}{4} D_3^2 = 0.0002327$

or $D_3 = \left(\dfrac{0.0002327 \times 4}{\pi}\right)^{1/2} = 0.0172\ \text{m or } 17.2\ \text{mm}$

i.e., *Exit diameter* = **17.2 mm.** **(Ans.)**

☞**Example 18.12.** *Steam at a pressure of 15 bar and dryness fraction 0.97 is discharged through a convergent-divergent nozzle to a back pressure of 0.2 bar. The mass flow rate is 9 kg/kWh. If the power developed is 220 kW, determine :*

(i) Throat pressure.

(ii) Number of nozzles required if each nozzle has a throat of rectangular cross-section of 4 mm × 8 mm.

(iii) If 12% of the overall isentropic enthalpy drop reheats by friction the steam in divergent portion find the cross-section of the exit rectangle.

Solution. $p_1 = 15$ bar ; $x_1 = 0.97$; $p_3 = 0.2$ bar

Mass flow rate = 9 kg/kWh ; Power developed = 220 kW

Throat dimensions = 4 mm × 8 mm

The isentropic index expansion for wet steam

$n = 1.035 + 0.1\, x_1 = 1.035 + 0.1 \times 0.97 = 1.132$

(i) **Throat pressure, p_2 :**

$$\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = \left(\frac{2}{1.132+1}\right)^{\frac{1.132}{1.132-1}} = (0.938)^{8.575} = 0.577$$

$\therefore \quad p_2 = 0.577 \times p_1 = 0.577 \times 15 =$ **8.65 bar.** **(Ans.)**

(ii) **Number of nozzles**

$h_1 = 2738$ kJ/kg ; $h_2 = 2630$ kJ/kg, $h_3 = 2080$ kJ/kg

Frictional heating = $0.12\ (h_1 - h_3) = 0.12\ (2738 - 2080) = 78.96$ kJ/kg

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$v_2 = 0.24$ m³/kg, $h_3 = 2580$ kJ/kg

$v_3' = 1.75$ m³/kg, $t_2 = 270°C$

i.e., **Temperature of the steam at throat = 270°C.** **(Ans.)**

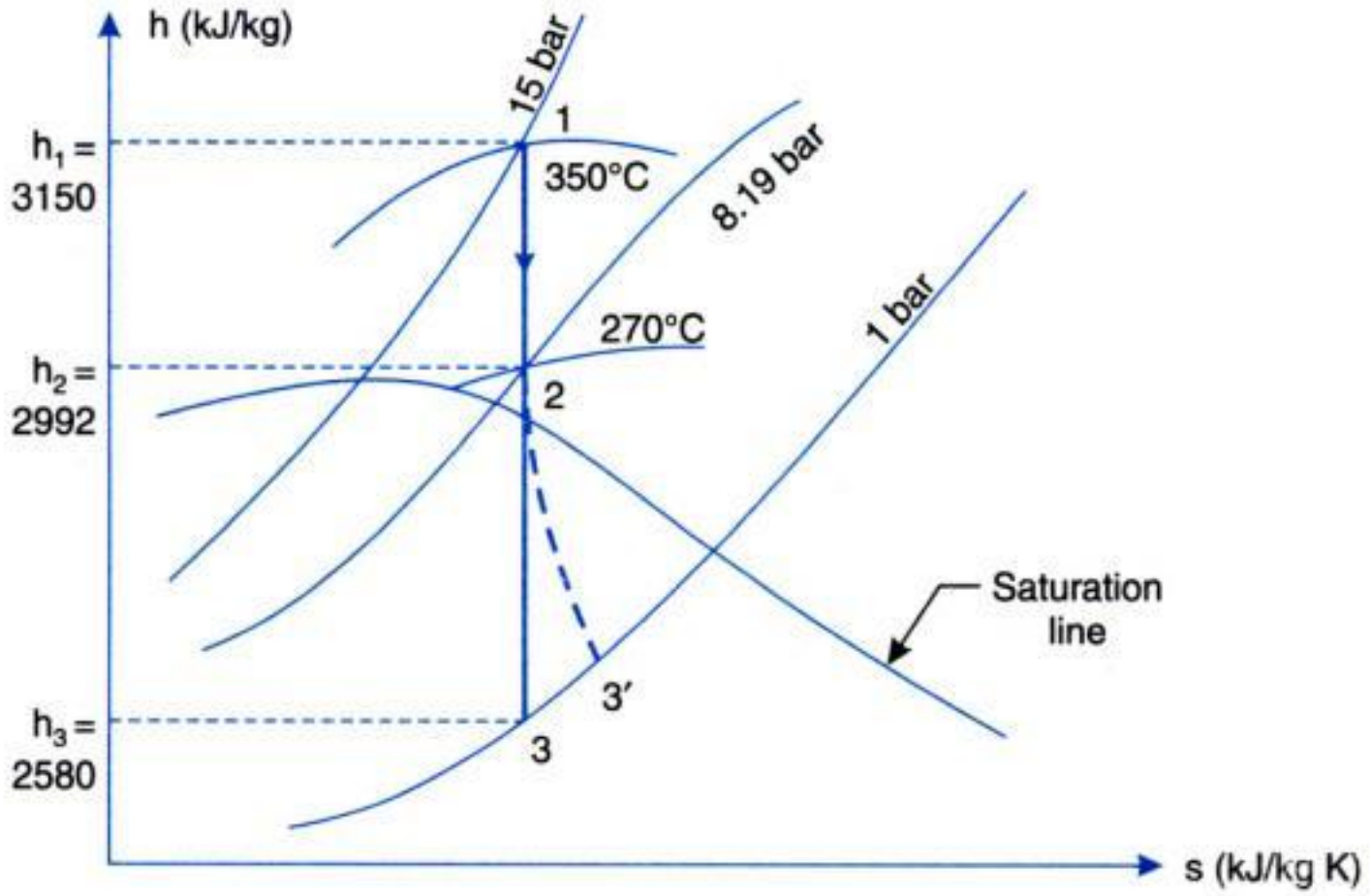

Fig. 18.17

Velocity of steam at throat,

$$C_2 = 44.72\ \sqrt{h_d} = 44.72\ \sqrt{(h_1 - h_2)}$$

$$= 44.72\ \sqrt{(3150 - 2992)} = \mathbf{562.12\ m/s.}\ \textbf{(Ans.)}$$

From the conditions at nozzle throat, mass flow rate,

$$\dot{m} = \frac{A_2 C_2}{v_2} = \frac{\pi / 4\,(6 / 1000)^2 \times 562.12}{0.24} = 0.0662\ \text{kg/s}$$

At exit :

$$C_3' = 44.72\ \sqrt{k h_d'} = 44.72\ \sqrt{0.88 \times (h_1 - h_3)}$$

$$= 44.72\ \sqrt{0.88 \times (3150 - 2580)} = 1001.5\ \text{m/s}$$

Exit area of the nozzle, $A_3 = \dfrac{\dot{m} \times v_3'}{C_3'} = \dfrac{0.0662 \times 1.75}{1001.5} = 0.0001156\ \text{m}^2$

i.e., $$\frac{\pi}{4} D_3^{\,2} = 0.0001156$$

$\therefore$ $$D_3 = \left(\frac{4 \times 0.0001156}{\pi}\right)^{1/2} = 0.012\ \text{m or } 12.1\ \text{mm}$$

If θ be the cone angle of nozzle,

$$\tan\theta = \frac{(12.1 - 6)}{2 \times 80} = 0.03812 \text{ or } \theta = 2°\ 11'$$

Thus, **cone angle = 2 × 2° 11′ = 4° 22′.** **(Ans.)**

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Example 18.17. *(a) Show that, when critical pressure ratio occurs, the velocity of a compressible fluid at the exit of a convergent nozzle is given by*

$$C_2 = \sqrt{\frac{2}{\gamma+1}} . a_1$$

where a_1 is the sonic velocity corresponding to the intial conditions. Assume critical pressure ratio $= \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma+1}}$, *where γ is the adiabatic index.*

(b) State the factors on which nozzle efficiency depends.

(c) Determine the throat and exit height of a Delaval nozzle to discharge 27 kg of a perfect gas per minute. The inlet and exit pressures are 480 kPa and 138 kPa respectively. Initial temperature of the gas is 535°C. Nozzle efficiency is 90% and frictional losses occur only after the throat. The molecular weight of the gas is 29 and its adiabatic index = 1.4. Assume square cross-section of the nozzle. **(AMIE Summer, 1998)**

Solution. (a) $\frac{p_2}{p_1} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ ...Given

The exit Mach. number is 1, $\therefore \quad T_2 = T^*$

$$\frac{T^*}{T_1} = \left(\frac{p^*}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left[\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right]^{\frac{\gamma-1}{\gamma}} = \frac{2}{\gamma+1}$$

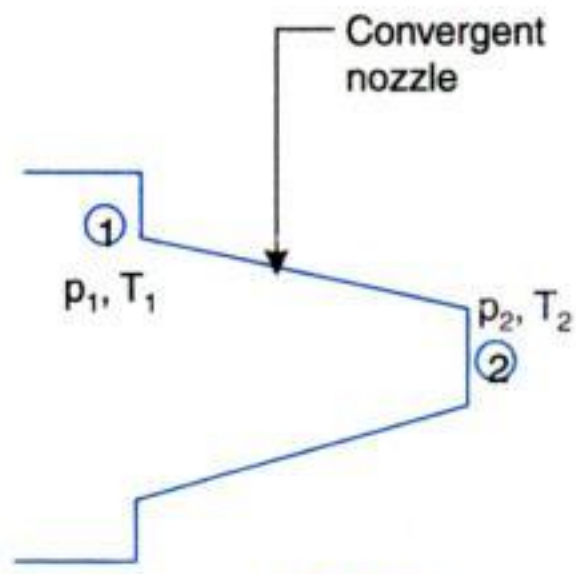

Fig. 18.20

The exit velocity is sonic velocity,

$$= \sqrt{\gamma R T_2} = \sqrt{\gamma R T^*}$$

$$= \sqrt{\gamma R \frac{2T_1}{\gamma+1}} = \sqrt{\frac{2}{\gamma+1}} \times \sqrt{\gamma R T_1}$$

$$= \sqrt{\frac{2}{\gamma+1}}\, a_1 . \quad \textbf{Proved.}$$

(*b*) **The factors on which nozzle efficiency depends are :**

1. Material of the nozzle.
2. Workmanship of the manufacture of nozzle.
3. Size and shape of the nozzle.
4. Reynolds number of flow.
5. Angle of divergence of divergent portion.
6. Nature of fluid flowing and its state.
7. Turbulence in fluid and its state.

(*c*) *Given :* $m_g = 27$ kg/min ; $p_1 = 480$ kPa ; $p_3 = 138$ kPa ; $T_1 = 535 + 273 = 808$ K ; $\eta_{nozzle} = 90\%$,

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Inlet : *From Mollier chart :*

$$h_1 = 3343 \text{ kJ/kg}, \; v_1 = 0.1078 \text{ m}^3/\text{kg}$$

$$h_{01} = h_1 + \frac{C_1^2}{2} = 3343 + \frac{60^2}{2 \times 1000}$$

$$= 3344.8 \text{ kJ/kg} \simeq h_1 \qquad [p_{01} \simeq p_1 = 30 \text{ bar}]$$

Also, $$\dot{m} = \frac{A_1 C_1}{v_1}$$

or $$A_1 = \frac{\dot{m} v_1}{C_1} = \frac{2 \times 0.1078}{60} = 0.003593 \text{ m}^2$$

or $$\frac{\pi}{4} D_1^2 = 0.003593$$

$\therefore$ $$\mathbf{D_1} = \left(\frac{0.003593 \times 4}{\pi} \right)^{1/2} = 0.0676 \text{ m} = \mathbf{67.6 \text{ mm}}.$$ **(Ans.)**

Throat **:** Assume that frictional losses occur in diverging part only ; *i.e.*, flow upto throat is frictionless.

$$\frac{p_2}{p_1} = 0.546$$

$\therefore$ $$p_2 = 30 \times 0.546 = 16.38 \text{ bar}$$

$$\left[\text{For superheat steam } n = 1.3 \quad \frac{p_2}{p_1} = \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}} = \left(\frac{2}{1.3+1} \right)^{\frac{1.3}{1.3-1}} = 0.546 \right]$$

From Mollier chart :

$$h_2 = 3155 \text{ kJ/kg}, \; v_2 = 0.19 \text{ m}^3/\text{kg}$$

$$C_2 = 44.72 \sqrt{(3344.8 - 3155)} = 616 \text{ m/s}$$

$\therefore$ $$A_2 = \frac{\dot{m} v_2}{C_2} = \frac{2 \times 0.19}{616} = 0.000617 \text{ m}^2$$

or $$\frac{\pi}{4} D_2^2 = 0.000617$$

$\therefore$ $$\mathbf{D_2} = \left(\frac{0.000617 \times 4}{\pi} \right)^{1/2} = 0.028 \text{ m} \quad \text{or} \quad \mathbf{28 \text{ mm}}.$$ **(Ans.)**

Exit **:** *From Mollier chart :* $h_3 = 2905$ kJ/kg, $v_3' = 0.4$ m^3/kg

$$h_3' = h_{01} - \eta_{nozzle} (h_{01} - h_3)$$

$$= 3344.8 - 0.9(3344.8 - 2905) = 2949 \text{ kJ/kg}$$

$\therefore$ $$C_3' = 44.72 \sqrt{(3344.8 - 2949)} \simeq 890 \text{ m/s}$$

$$A_3 = \frac{\dot{m} v_3'}{C_3'} = \frac{2 \times 0.4}{890} = 0.000899 \text{ m}^2$$

or $$\frac{\pi}{4} D_3^2 = 0.000899.$$

or $$\mathbf{D_3} = \left(\frac{0.000899 \times 4}{\pi} \right)^{1/2} = 0.0338 \text{ m} \quad \text{or} \quad \mathbf{33.8 \text{ mm}}.$$ **(Ans.)**

Example 18.19. *Determine the throat area, exit area and exit velocity for a steam nozzle to pass a mass flow of 0.2 kg/s when inlet conditions are 10 bar and 250°C and the final pressure is 2 bar. Assume expansion is isentropic and that the inlet velocity is negligible. Use $pv^{1.3}$ constant. Do not calculate from h-s chart.* **(N.U.)**

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$$= \frac{1.3}{1.3-1} \times \frac{30 \times 0.09 \times 10^5}{10^3} \left[1 - \left(\frac{4}{30}\right)^{\frac{1.3-1}{1.3}}\right]$$

$$= 1170\ (1 - 0.6282) = 435\ \text{kJ/kg}$$

$\therefore$ Velocity at point 2,

$$C_2 = 44.72\ \sqrt{(h_1 - h_2)}\ = 44.72\ \sqrt{435}\ = 932.7\ \text{m/s}$$

Also $$v_2 = v_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} = 0.09 \times \left(\frac{30}{4}\right)^{\frac{1}{0.3}} = 0.4239\ \text{m}^3/\text{kg}$$

$\therefore$ Mass flow rate, $$\dot{m} = \frac{A_2 \times C_2}{v_2}$$

$$5.2 = \frac{A_2 \times 932.7}{0.4239}$$

$\therefore$ $$A_2 = \frac{5.2 \times 0.4239}{932.7} = 0.002363\ \text{m}^2$$

Since aspect ratio is 3 : 1, if we assume breadth as x the length will be $3x$ and area of six nozzles will be

$$A_2 = 6 \times 3x \times x$$

$$0.002363 = 18x^2$$

$\therefore$ $$x = \left(\frac{0.002363}{18}\right)^{1/2} = \textbf{0.0114 m or 11.4 mm.}$$ (Ans.)

and *length* $= 3 \times 11.4 =$ **34.2 mm.** (Ans.)

(*ii*) **Degree of undercooling and supersaturation :**

Temperature at point 2 is found as follows :

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

$\therefore$ $$T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (273 + 350) \times \left(\frac{4}{30}\right)^{\frac{1.3-1}{1.3}}$$

$$= 623 \times 0.628 = 391.2\ \text{K or } 118.2°\text{C}$$

From steam tables saturation temperature at 4 bar

$$= 143.6°\text{C}$$

$\therefore$ *Degree of* ***undercooling*** $= 143.6 - 118.2 =$ **25.4°C.** (Ans.)

Saturation pressure corresponding to 118.2°C $\simeq$ 1.9 bar

$\therefore$ *Degree of* ***super saturation*** $= \frac{4}{1.9} =$ **2.1.** (Ans.)

(*iii*) **Loss in available heat drop** :

Isentropic enthalpy drop for expansion under thermal equilibrium conditions as read out from Mollier chart

$$h_1 - h_3 = 3115 - 2675 = 440\ \text{kJ/kg}$$

$\therefore$ Loss of available heat drop $= 440 - 435 = 5$ kJ/kg.

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OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. For a steam nozzle, if p_1 = inlet pressure, p_2 = exit pressure and n is the index of isentropic expansion, the mass flow rate per unit area is maximum if

(a) $\frac{p_2}{p_1} \le \left(\frac{2}{n+1}\right)^{\frac{n-1}{n}}$ (b) $\frac{p_2}{p_1} \le \left(\frac{1}{n+1}\right)^{\frac{n}{n+1}}$

(c) $\frac{p_2}{p_1} \le \left(\frac{2}{n+1}\right)^{\frac{n}{n+1}}$ (d) $\frac{p_2}{p_1} \le \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$.

2. The isentropic expansion of steam through nozzle for the steam initially superheated at inlet is approximated by equation

(a) $pv^{1.3} = C$ (b) $pv^{1.125} = C$

(c) $pv^{1.4} = C$ (d) $pv = C$.

3. The ratio of exit pressure to inlet pressure for maximum mass flow rate per unit area of steam through a nozzle when steam is initially dry saturated is

(a) 0.6 (b) 0.578

(c) 0.555 (d) 0.5457.

4. The ratio of exit pressure to inlet pressure for maximum mass flow rate per unit area of steam through nozzle when steam is initially superheated is

(a) 0.555 (b) 0.578

(c) 0.5457 (d) 0.6.

5. The critical pressure ratio of a convergent nozzle is defined as

(a) the ratio of outlet pressure to inlet pressure of nozzle

(b) the ratio of inlet pressure to outlet pressure of nozzle

(c) the ratio of outlet pressure to inlet pressure only when mass flow rate per unit area is minimum

(d) the ratio of outlet pressure to inlet pressure only when mass flow rate per unit area is maximum.

6. The isentropic expansion of steam through nozzle for the steam initially dry saturated at inlet is approximated by equation

(a) $pv = C$ (b) $pv^{1.4} = C$

(c) $pv^{1.3} = C$ (d) $pv^{1.135} = C$.

7. The effect of considering friction losses in steam nozzle for the same pressure ratio leads to

(a) increase in exit velocity from the nozzle

(b) decrease in exit velocity from nozzle

(c) no change in exit velocity from nozzle

(d) increase or decrease depending upon the exit quality of steam.

8. The effect of considering friction in steam nozzles for the same pressure ratio leads to

(a) increase in dryness fraction of exit steam

(b) decrease in dryness fraction of exit steam

(c) no change in the quality of exit steam

(d) decrease or increase of dryness fraction of exit steam depending upon inlet quality.

ANSWERS

1. (d) 2. (a) 3. (b) 4. (c) 5. (d) 6. (d) 7. (b)
8. (a).

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(*ii*) *Radial turbines* in which steam flows in a direction perpendicular to the axis of the turbine ; one or more low-pressure stages in such turbines are made axial.

3. According to the number of cylinders :

(*i*) Single cylinder turbines.

(*ii*) Double cylinder turbines.

(*iii*) Three cylinder turbines.

(*iv*) Four cylinder turbines.

Multi-cylinder turbines which have their rotors mounted on one and the same shaft and coupled to a single generator are known as *single shaft turbines* ; turbines with separate rotor shafts for each cylinder placed parallel to each other are known as **multiaxial turbines.**

4. According to the method of governing :

(*i*) *Turbines with throttle governing* in which fresh steam enters through one or more (depending on the power developed) simultaneously operated throttle valves.

(*ii*) *Turbines with nozzle governing* in which fresh steam enters through two or more consecutively opening regulators.

(*iii*) *Turbines with by pass governing* in which steam turbines besides being fed to the first stage is also directly fed to one, two or even three intermediate stages of the turbine.

5. According to heat drop process :

(*i*) *Condensing turbines with generators* ; in these turbines steam at a pressure less than atmospheric is directed to a condenser ; besides, steam is also extracted from intermediate stages for feed water heating, the number of such extractions usually being from 2–3 to as much 8–9. The latent heat of exhaust steam during the process of condensation is completely lost in these turbines.

(*ii*) *Condensing turbines with one or two intermediate stage extractions* at specific pressures for industrial and heating purposes.

(*iii*) *Back pressure turbines*, the exhaust steam from which is utilised for industrial or heating purposes ; to this type of turbines can also be added (in a relative sense) turbines with deteriorated vacuum, the exhaust steam of which may be used for heating and process purposes.

(*iv*) *Topping turbines* ; these turbines are also of the back pressure type with the difference that the exhaust steam from these turbines is further utilised in medium and low pressure condensing turbines. These turbines, in general, operate at high initial conditions of steam pressure and temperature, and are mostly used during extension of power station capacities, with a view to obtain better efficiencies.

(*v*) *Back pressure turbines with steam extraction from intermediate stages at specific pressure* ; turbines of this type are meant for supplying the consumer with steam of various pressures and temperature conditions.

(*vi*) *Low pressure turbines* in which the exhaust steam from reciprocating steam engines, power hammers, presses, etc., is utilised for power generation purposes.

(*vii*) *Mixed pressure turbines* with two or three pressure stages, with supply of exhaust steam to its intermediate stages.

6. According to steam conditions at inlet to turbine :

(*i*) *Low pressure turbines*, using steam at a pressure of 1.2 to 2 ata.

(*ii*) *Medium pressure turbines*, using steam at pressures of upto 40 ata.

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19.5. METHODS OF REDUCING WHEEL OR ROTOR SPEED

As already discussed under the heading 'simple impulse turbine' that if the steam is expanded from the boiler pressure to condenser pressure in one stage the speed of the rotor becomes *tremendously high which crops up practical complicacies*. There are several methods of reducing this speed to lower value ; all these methods utilise a multiple system of rotor in series, keyed on a common shaft and the steam pressure or jet velocity is absorbed in stages as the steam flows over the blades. This is known as '**compounding**'. The different methods of compounding are :

1. Velocity compounding.
2. Pressure compounding.
3. Pressure velocity compounding.
4. Reaction turbine.

1. Velocity compounding

Steam is expanded through a stationary nozzle from the boiler or inlet pressure to condenser pressure. So the pressure in the nozzle drops, the kinetic energy of the steam increases due to increase in velocity. A portion of this available energy is absorbed by a row of moving blades. The

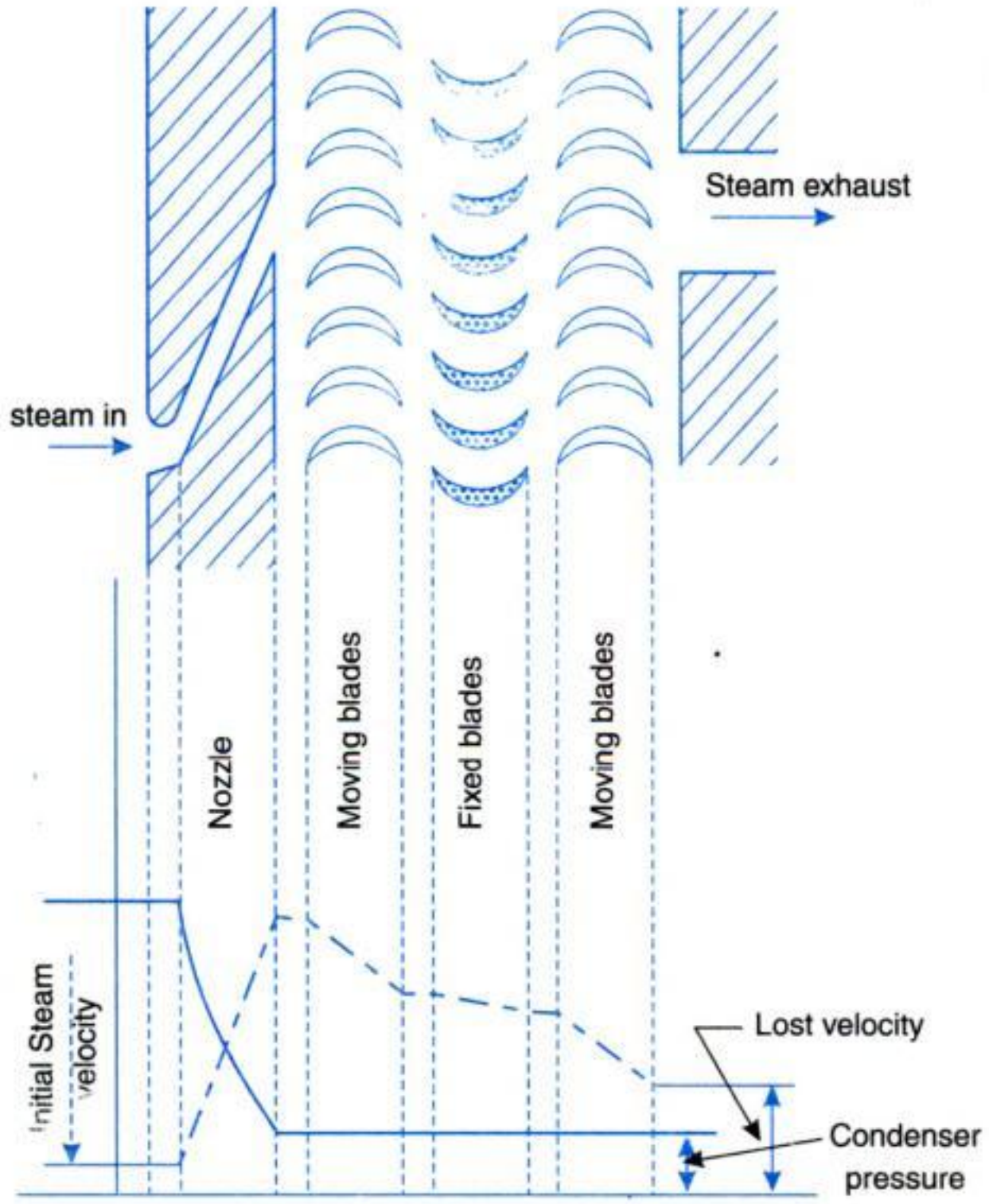

Fig. 19.3. Velocity compounding.

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β = Angle which the discharging steam makes with the tangent of the wheel at the exit of moving blade.

θ = Entrance angle of moving blade.

ϕ = Exit angle of moving blade.

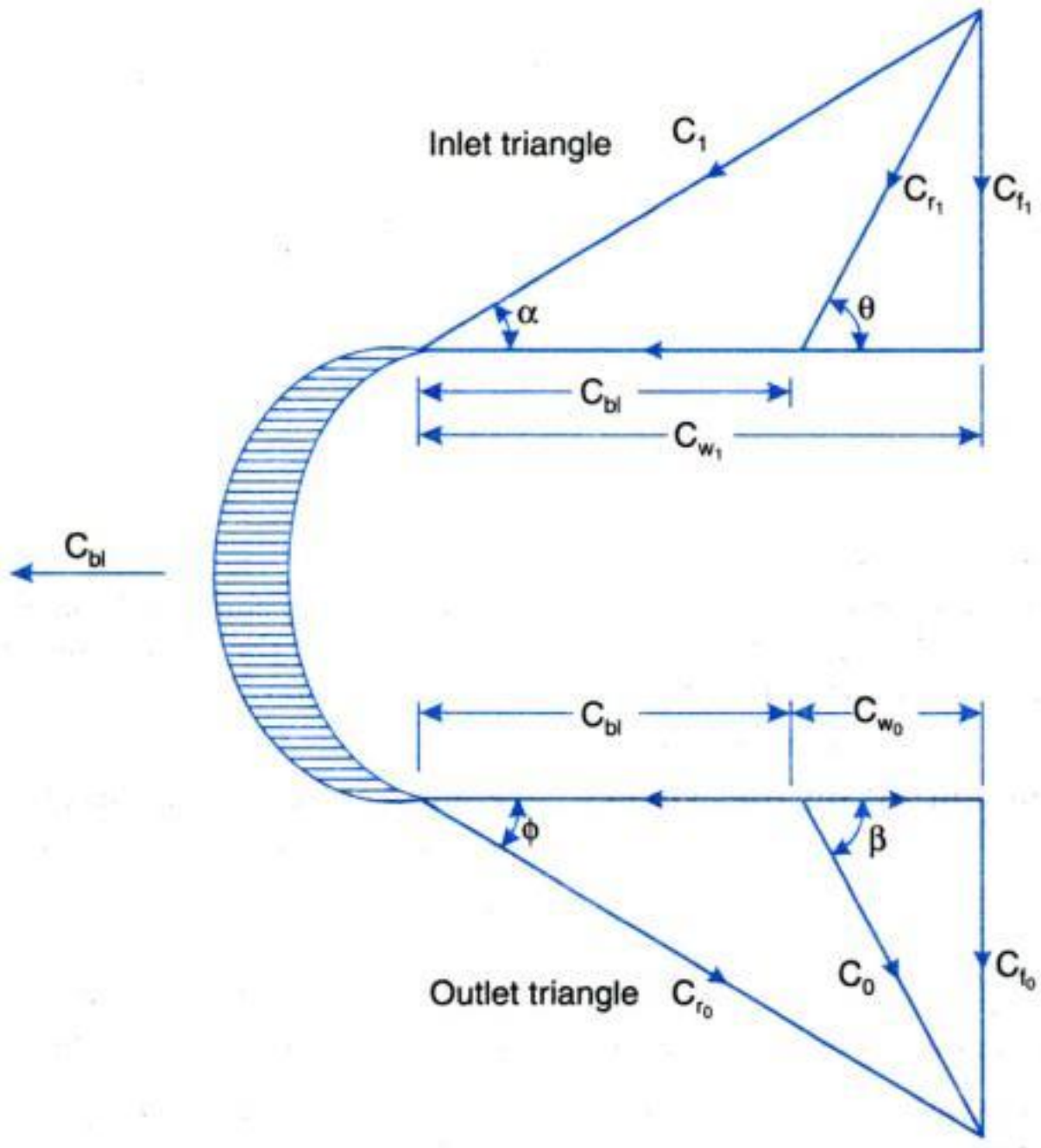

Fig. 19.6. Velocity diagram for moving blade.

The steam jet issuing from the nozzle at a velocity of C_1 impinges on the blade at an angle α. The tangential component of this jet (C_{w_1}) *performs work on the blade*, the axial component (C_{f_1}) however *does no work but causes the steam to flow through the turbine*. As the blades move with a tangential velocity of C_{bl}, the entering steam jet has a relative velocity C_{r_1} (with respect to blade) which makes an angle θ with the wheel tangent. The steam then glides over the blade without any shock and discharges at a relative velocity of C_0 at an angle ϕ with the tangent of the blades. The relative velocity at the inlet (C_{r_1}) is the same as the relative velocity at the outlet (C_{r_0}) if *there is no frictional loss at the blade*. The absolute velocity (C_0) of leaving steam make an angle β to the tangent at the wheel.

To have convenience in solving the problems on turbines it is a common practice to combine the two vector velocity diagrams on a common base which represents the blade velocity (C_{bl}) as shown in Fig. 19.7. This diagram has been obtained by superimposing the inlet velocity diagram on the outlet diagram in order that the blade velocity lines C_{bl} coincide.

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— From S draw a line *parallel to* LP ($\because$ $C_{f_1} = C_{f_0}$) and from point M draw an arc equal to $C_{r_0} (= 0.9 C_{r_1})$ to get the point of intersection N. Complete the triangle LMN. From N draw perpendicular NQ on PL produced to get C_{f_0}.

Measure θ and ϕ (the blade angles) from the velocity diagram.

$\theta = 37°$ and $\phi = 42°$. **(Ans.)**

Power developed, P :

$$P = \frac{\dot{m}_s (C_{w_1} + C_{w_0}) \times C_{bl}}{1000} = \frac{10 \times 306 \times 105}{1000} = \mathbf{321.3\ kW.}$$ **(Ans.)**

☞ **Example 19.9.** *In an impulse turbine (with a single row wheel) the mean diameter of the blades is 1.05 m and the speed is 3000 r.p.m. The nozzle angle is 18°, the ratio of blade speed to steam speed is 0.42 and the ratio of the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of the blade is to be made 3° less than the inlet angle. The steam flow is 10 kg/s. Draw the velocity diagram for the blades and derive the following :*

(i) Tangential thrust on the blades *(ii) Axial thrust on the blades*

(iii) Resultant thrust on the blades *(iv) Power developed in the blades*

(v) Blading efficiency. **(P.U.)**

Solution. Mean diameter of the blades, D = 1.05 m

Speed of the turbine, N = 3000 r.p.m.

Nozzle angle, $\alpha = 18°$

Ratio of blade speed to steam speed, $\rho = 0.42$

Ratio, $\frac{C_{r_0}}{C_{r_1}} = 0.84$

Outlet blade angle, $\phi = \theta - 3°$

Steam flow rate $\dot{m}_s = 10$ kg/s

Blade speed, $C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1.05 \times 3000}{60} = 164.5$ m/s

But $\rho = \frac{C_{bl}}{C_1} = 0.42$ (given)

$\therefore$ $C_1 = \frac{C_{bl}}{0.42} = \frac{164.5}{0.42} = 392$ m/s

With the data, $C_1 = 392$ m/s ;

$\alpha = 18°$, complete ΔLMS

$\theta = 30°$ (on measurement)

$\therefore$ $\phi = 30° - 3 = 27°$.

Now complete the ΔLMN by taking $\phi = 27°$ and $C_{r_0} = 0.84\ C_{r_1}$.

Finally complete the whole diagram as shown in Fig. 19.20.

(i) Tangential thrust on the blades :

Tangential thrust $= \dot{m}_s (C_{w_1} + C_{w_0}) = 10 \times 390 = \mathbf{3900\ N.}$ **(Ans.)**

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In a simple impulse turbine the nozzles are inclined at 20° to the direction of motion of the moving blades. The steam leaves the nozzle at 375 m/s. The blade velocity is 165 m/s. Calculate suitable inlet and outlet angles for the blades in order that the axial thrust is zero. The relative velocity of steam as it flows over the blades is reduced by 15% by friction. Also, determine the power developed for a flow rate of 10 kg/s.

Solution. Nozzles, $\alpha = 20°$

Velocity of steam issuing from the nozzles, $C_1 = 375$ m/s

Blade speed $C_{bl} = 165$ m/s

Axial thrust = zero *i.e.*, $C_{f_1} = C_{f_0}$

$\frac{C_{r_0}}{C_{r_1}} = (1 - 0.15) = 0.85$, *i.e.*, 15% loss due to friction, steam flow rate, $\dot{m}_s = 10$ kg/s.

Inlet and outlet angles :

With the above given data, draw velocity diagram to a suitable scale as shown in Fig. 19.25.

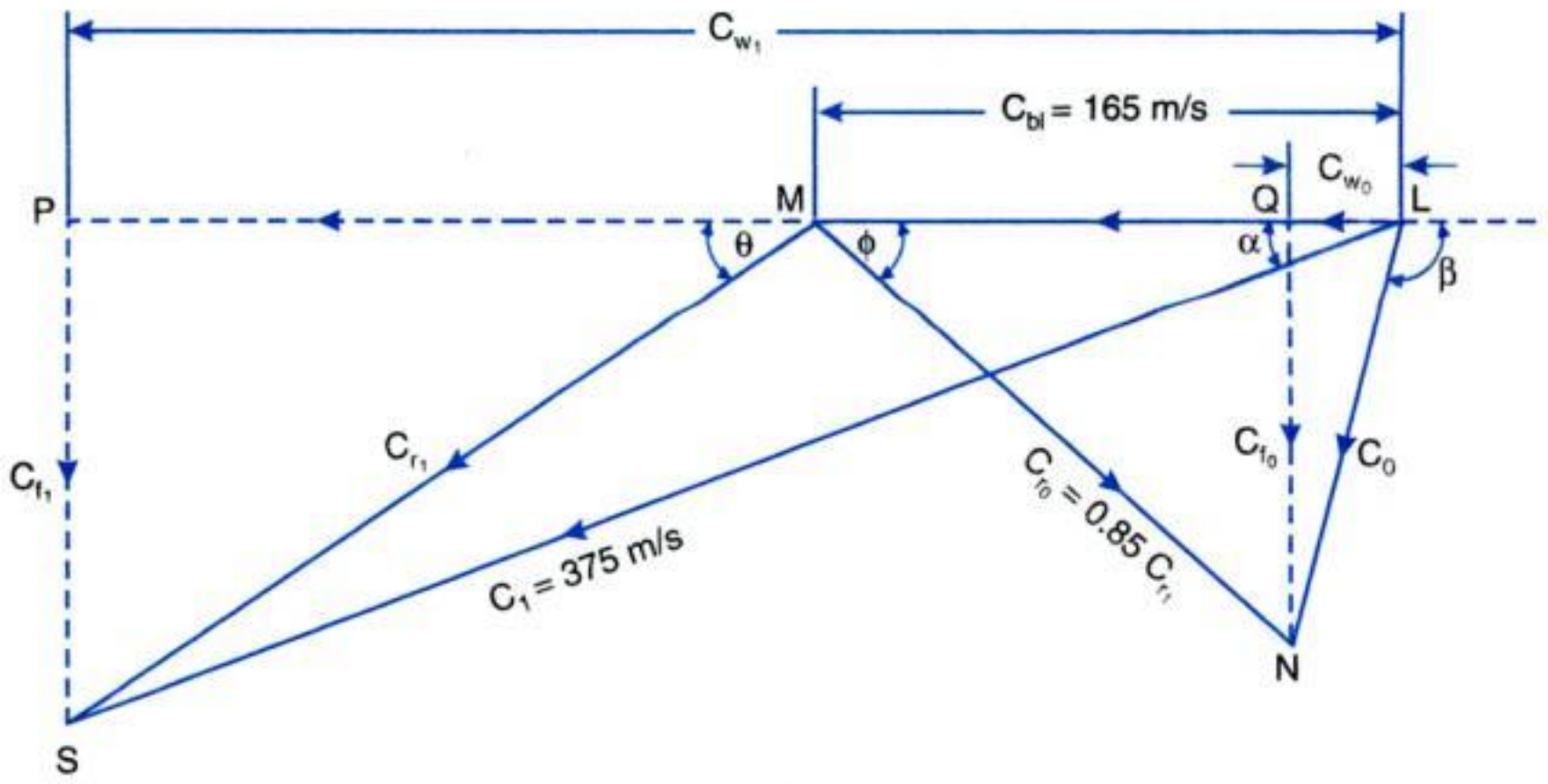

Fig. 19.25

By measurement (from velocity diagram),

$$\left.\begin{aligned} \theta &= \mathbf{35°} \\ \phi &= \mathbf{42°} \\ \beta &= \mathbf{100°} \end{aligned}\right\} . \quad \textbf{(Ans.)}$$

Power developed, P :

Also, $C_{w_1} = 354$ m/s ; $C_{w_0} = 24$ m/s (By measurement)

$\therefore$ *Power developed,* $P = \dfrac{\dot{m}_s [C_{w_1} + C_{w_0}] \times C_{bl}}{1000}$

$$= \frac{\dot{m}_s [C_{w_1} + (-C_{w_0})] \times C_{bl}}{1000} = \frac{10 [354 + (-24)] \times 165}{1000} = \mathbf{544.5\ kW.} \quad \textbf{(Ans.)}$$

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(*i*) **Axial thrust on the blades**

$$= \dot{m}_s[(C_{f_1} - C_{f_0}) + (C_{f_1}' - C_{f_0}')]$$

$$= 2.5\ [(180 - 138) + (122 - 107)] = \mathbf{142.5\ N.}\quad \textbf{(Ans.)}$$

(*ii*) **Power developed**

$$= \frac{\dot{m}_s[(C_{w_1} + C_{w_0}') + (C_{w_1}' + C_{w_0}')]C_{bl}}{1000}$$

$$= \frac{2.5(924 + 324) \times 125}{1000} = \mathbf{390\ kW.}\quad \textbf{(Ans.)}$$

(*iii*) **Efficiency**

$$= \frac{(924 + 324) \times 125}{C_1^{\,2}/2} = \frac{(924 + 324) \times 125}{650^2/2}$$

$$= \mathbf{0.738\ or\ 73.8\%.}\quad \textbf{(Ans.)}$$

Example 19.21. *The first stage of an impulse turbine is compounded for velocity and has two rings of moving blades and one ring of fixed blades. The nozzle angle is 20° and the leaving angles of the blades are respectively as follows :*

First moving 20°, fixed 25° and second moving 30°. Velocity of steam leaving the nozzles is 600 m/sec and the steam velocity relative to the blade is reduced by 10% during the passage through each ring. Find the diagram efficiency and power developed for a steam flow of 4 kg per second. Blade speed may be taken as 125 m/sec. **(M.U.)**

Solution.

$$C_{bl} = 125\ \text{m/s},\ C_1 = 600\ \text{m/s}$$

$$\alpha = 20°,\ \phi = 20°$$

$$\alpha' = 25°,\ \phi' = 30°$$

$$K = \left(1 - \frac{10}{100}\right) = 0.9$$

$$\dot{m}_s = 4\ \text{kg/s}$$

With these values velocity triangles can be drawn (Fig. 19.31).

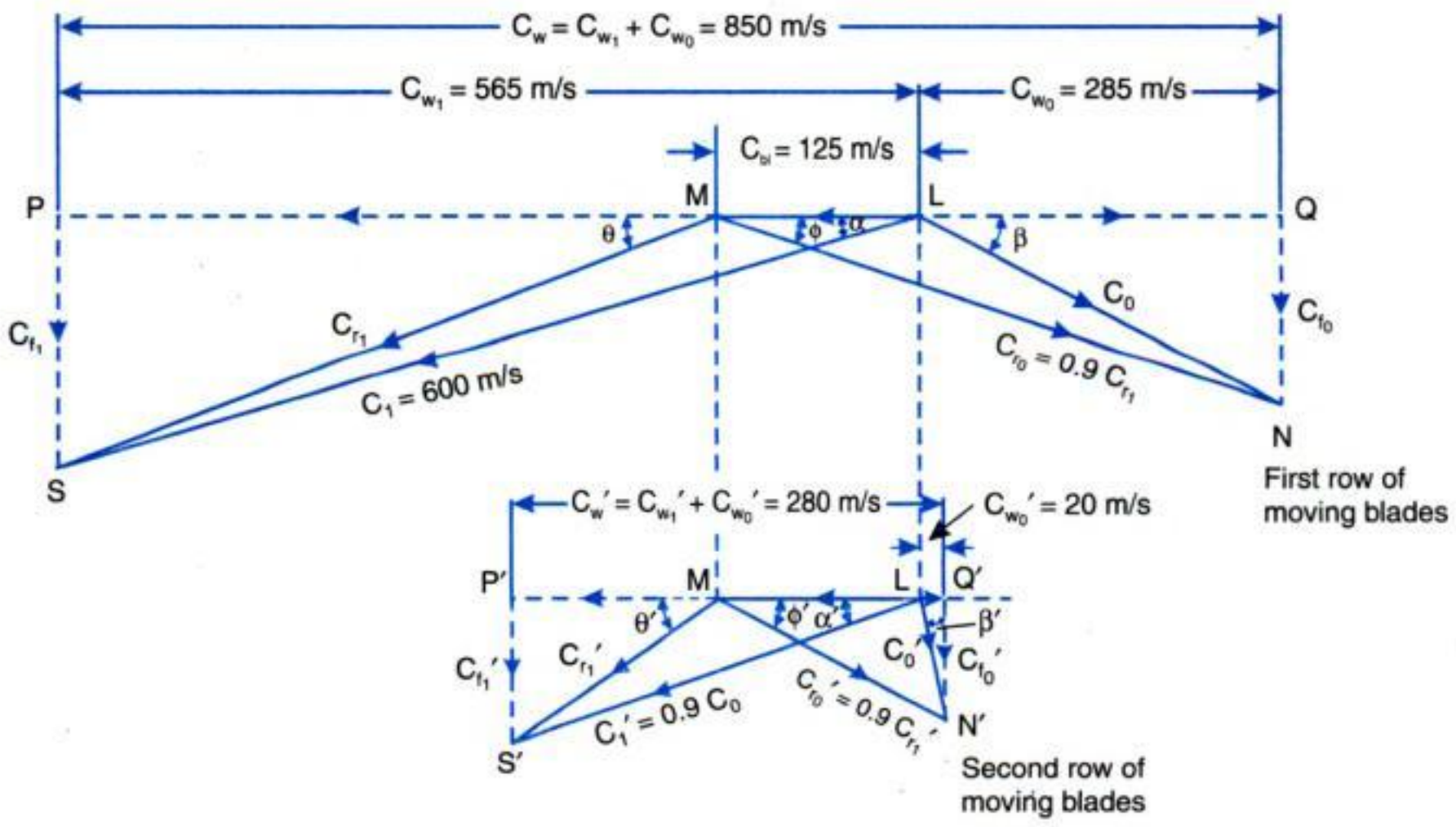

Fig. 19.31

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The pressure drop in reaction turbines takes place in both fixed and moving blades. The division generally is given in terms of enthalpy drops. The criterion used is the *degree of reaction*. It is defined as

$$\frac{\text{Enthalpy drop in rotor blades}}{\text{Total enthalpy drop in stage}} = \frac{\Delta h_m}{\Delta h_f + \Delta h_m}$$ (Refer Fig. 19.36)

A special case is when the degree of reaction is zero ; it means no heat drop in the moving blades. This becomes a case of impulse stage. Other common case is of Parson's turbine which has the same reason for both the fixed and moving blades. The blades are *symmetrical*, *i.e.*, the exit angle of moving blade is equal to the exit angle of the fixed blade and the inlet angle of the moving blade is equal to the inlet angle of the fixed blade. Since the blades are symmetrical the velocity diagram is also symmetrical. In such a case the degree of reaction is 50%.

Applying the steady flow energy equation to the fixed blades and assuming that the velocity of steam entering the fixed blade is equal to the absolute velocity of steam leaving the previous moving row, we have

$$\Delta h_f = \frac{C_1^{\,2} - C_0^{\,2}}{2}$$

Similarly, for the moving blades

$$\Delta h_m = \frac{C_{r_0}^{\,2} - C_{r_1}^{\,2}}{2}$$

But $$C_1 = C_{r_0} \text{ and } C_0 = C_{r_1}$$

$\therefore$ $$\Delta h_f = \Delta h_m$$

Hence degree of reaction $$= \frac{\Delta h_m}{\Delta h_m + \Delta h_m} = \frac{1}{2}$$

This is a proof that **Parson's reaction turbine is a 50% reaction turbine.**

Example 19.26. (*a*) *Explain the functions of the blading of a reaction turbine.*

(*b*) *A certain stage of a Parson's turbine consists of one row of fixed blades and one row of moving blades. The details of the turbine are as below :*

The mean diameter of the blades = *68 cm*

R.P.M. of the turbine = *3,000*

The mass of steam passing per sec = *13.5 kg*

Steam velocity at exit from fixed blades = *143.7 m/s*

The blade outlet angle = *20°*

Calculate the power developed in the stage and gross efficiency, assuming carry over co-efficient as 0.74 and the efficiency of conversion of heat energy into kinetic energy in the blade channel as 0.92. **(M.U.)**

Solution. (*a*) The blades of reaction turbine has to perform two functions :

1. They change the direction of motion of steam causing change of momentum, responsible for motive force.

2. The blades also act as nozzles causing pressure drop as steam moves in the blade passage.

(*b*) $$D = 0.68 \text{ m}, N = 3000 \text{ r.p.m.}, \dot{m} = 13.5 \text{ kg/s}$$

$$C_{r_0} = 143.7 \text{ m/s}, \phi = 20°, \psi = 0.74, \eta = 0.92$$

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Specific volume of steam at nozzle outlet or blade inlet, $v_1 = 3.45$ m^3/kg

Specific volume of steam at blade outlet, $v_0 = 3.95$ m^3/kg

Power developed by the turbine = 287 kW

Efficiency of nozzle and blades combinedly = 90%

Carry over co-efficient, $\psi = 0.82$

Blade speed, $$C_{bl} = \frac{\pi DN}{60}$$

$$\therefore \quad D = \frac{60\, C_{bl}}{\pi N} = \frac{60 \times 145}{\pi \times 3000} = 0.923 \text{ m}$$

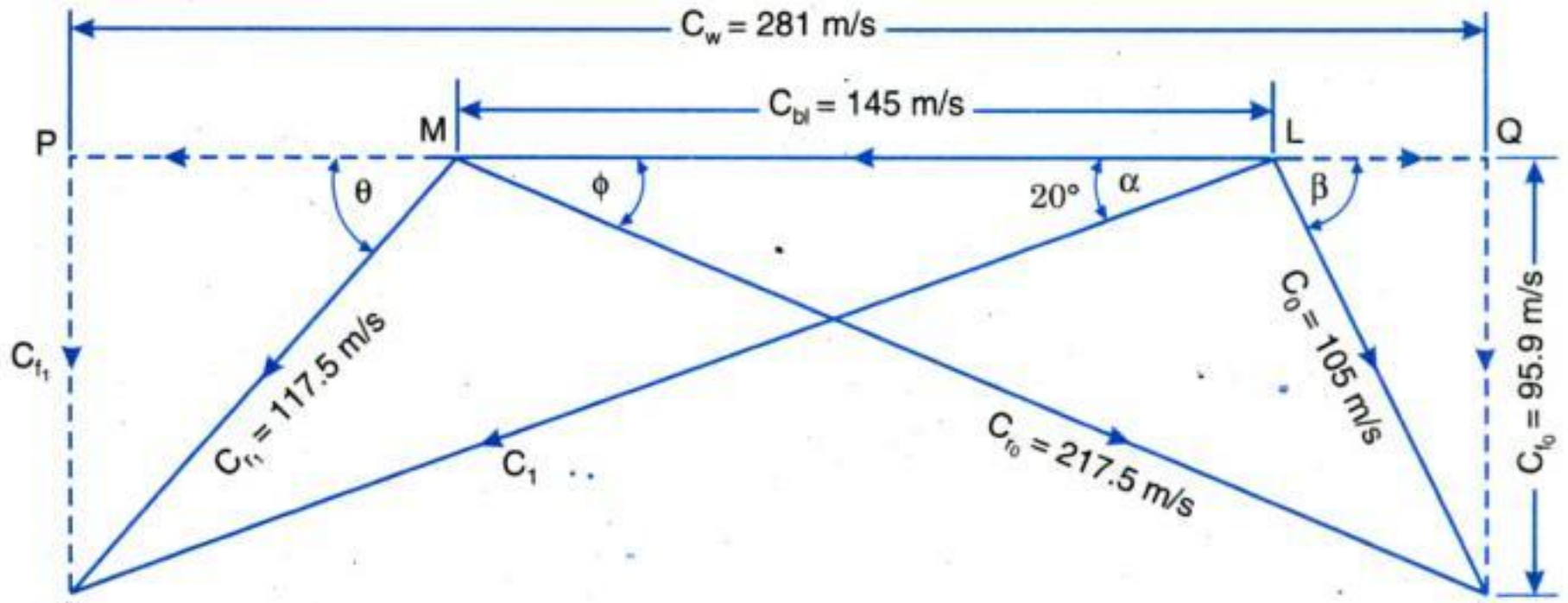

Fig. 19.53

Mass flow rate, $$\dot{m}_s = \frac{C_{f_1} \times \pi Dh}{v_1} = \frac{C_1 \sin \alpha \times \pi Dh}{v_1}$$

$$= \frac{245 \times \sin 20° \times \pi \times 0.923 \times 0.1}{3.45} = 7.04 \text{ kg/s}$$

Also $$\dot{m}_s = \frac{C_{f_0}\, \pi Dh}{v_0}$$

$$\therefore \quad C_{f_0} = \frac{\dot{m}_s v_0}{\pi Dh} = \frac{7.04 \times 3.95}{\pi \times 0.923 \times 0.1} = 95.9 \text{ m/s}$$

The power is given by, $$P = \frac{\dot{m}_s \times C_{bl} \times C_w}{1000}$$

$$287 = \frac{7.04 \times 145 \times C_w}{1000}$$

$$\therefore \quad C_w = \frac{287 \times 1000}{7.04 \times 145} = 281 \text{ m/s}$$

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2. Extraction pass out turbine

It is found that in several cases the power available from a back pressure turbine (through which the whole of the steam flows) is appreciably less than that required in the factory and this may be due to the following reasons :

(*i*) Small heating or process requirements ;

(*ii*) A relatively high exhaust pressure ; and

(*iii*) A combination of the both.

In such a case it would be possible to install a back-pressure turbine to provide the heating steam and a condensing turbine to generate extra power, but it is possible, and useful, to combine functions of both machines in a single turbine. Such a machine is called **extraction or pass out turbine** and here at some point intermediate between inlet and exhaust some steam is extracted or passed out for process or heating purposes. In this type of turbine a sensitive governor is used which controls the admission of steam to the high pressure section so that regardless of power or process requirements, constant speed is maintained.

Exhaust or low pressure turbine

If an uninterrupted supply of low pressure steam is available (such as from reciprocating steam engines exhaust) it is possible to improve the efficiency of the whole plant by fitting an exhaust or low pressure turbine. The use of exhaust turbine is chiefly made where there are several reciprocating steam engines which work intermittently ; and are non-condensing (*e.g.*, rolling mill and colliery engines). The exhaust steam from these engines is expanded in an exhaust turbine and then condensed. In this turbine some form of heat accumulator is needed to collect the more or less irregular supply of low pressure steam from the non-condensing steam engines and deliver it to the turbine at the rate required. In some cases when the supply of low pressure steam falls below the demand, live steam from the boiler, with its pressure and temperature reduced ; is used to make up the deficiency.

The necessary drop in pressure may be obtained by the use of a reducing valve, or for large flows, more economically by expansion through another turbine. The high pressure and low pressure turbines are sometimes combined on a common spindle and because of two supply pressures this combined unit is known as '**mixed pressure turbine**'.

HIGHLIGHTS

1. The **steam turbine** is a prime mover in which the potential energy of the steam is transformed into kinetic energy, and latter in its turn is transformed into the mechanical energy of rotation of the turbine shaft.
2. The most important **classification** of steam turbines is as follows :
   (*i*) Impulse turbines (*ii*) Reaction turbines
   (*iii*) Combination of impulse and reaction turbines.
3. The main difference between **Impulse** and **Reaction turbines** lies in the way in which steam in expanded while its moves through them. In the former type, steam expands in the nozzle and its pressure does not change as it moves over the blades while in the latter type the steam expands continuously as it passes over the blades and thus there is a gradual fall in pressure during expansion.
4. The different methods of compounding are :
   (*i*) Velocity compounding (*ii*) Pressure compounding
   (*iii*) Pressure velocity compounding (*iv*) Reaction turbine.
5. Force (tangential) on the wheel = $\dot{m}_s(C_{w_1} + C_{w_0})$ Nm

   Power per wheel $= \dfrac{\dot{m}_s(C_{w_1} + C_{w_0}) \times C_{bl}}{1000}$ kW

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2. It supplies to the boiler pure and hot feed water as the condensed steam which is discharged from the condenser and collected in a hot well, can be used as feed water for the boiler.

20.2. VACUUM

Vacuum is *sub-atmospheric pressure*. It is measured as the *pressure depression below atmospheric*. The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in the volume of the low pressure steam or vapour. The back pressure in steam engine or steam turbine can lowered from 1.013 to 0.2 bar abs. or even less. Since the steam engines are intermittent flow machines and as such cannot take the advantage of a very low vacuum, therefore, for most steam engines the exhaust pressure is about 0.2 to 0.28 bar abs. On the other hand, in steam turbines, which are continuous flow machines, the back pressure may be about 0.025 bar abs.

20.3. ORGANS OF A STEAM CONDENSING PLANT

A steam condensing plant mainly consists of the following *organs/elements* :

1. Condenser (To condense the steam).
2. Supply of cooling (or injection) water.
3. Wet air pump (To remove the condensed steam, the air and uncondensed water vapour and gases from the condenser ; separate pumps may be used to deal with air and condensate).
4. Hot well (where the condensate can be discharged and from which the boiler feed water is taken).
5. Arrangement for recooling the cooling water in case surface condenser is employed.

20.4. CLASSIFICATION OF CONDENSERS

Mainly, condensers are of two types : (1) Jet condensers, (2) Surface condenser.

In ***jet condensers***, the *exhaust steam and water come in direct contact with each other and temperature of the condensate is the same as that of cooling water leaving the condenser.* The cooling water is usually sprayed into the exhaust steam to cause, rapid condensation.

In ***surface condensers***, *the exhaust steam and water do not come into direct contact.* The steam passes over the outer surface of tubes through which a supply of cooling water is maintained. There may be single-pass or double-pass. In single-pass condensers, the water flows in one direction only through all the tubes, while in two-pass condenser the water flows in one direction through the tubes and returns through the remainder.

A jet condenser is simpler and cheaper than a surface condenser. It should be installed when the cooling water is cheaply and easily made suitable for boiler feed or when a cheap source of boiler and feed water is available. A surface condenser is most commonly used because the condensate obtained is not thrown as a waste but returned to the boiler.

20.4.1. Jet Condensers

These condensers may be classified as :

(*a*) Parallel flow type

(*b*) Counter flow type

(*c*) Ejector type.

Parallel flow and counter flow condensers are further sub-divided into two types : (*i*) Low level type (*ii*) High level type.

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3. In jet condensers, a little quantity of air accompanies the injection water (in which it is dissolved).

Note. (*i*) In jet condensers, the quantity of air dissolved in injection water is about 0.5 kg/10000 kg of water.

(*ii*) In surface condensers of reciprocating steam engines, the air leakage is about 15 kg/10000 kg of steam whereas in surface condensers of well designed and properly maintained steam turbine plants the air leakage is about 5 kg/10000 kg of steam.

In order to check whether there is air leakage in the condenser, the following procedure is adopted :

1. Keep the plant running until the temperature and pressure conditions are steady in the condenser.
2. The steam condenser be isolated by shutting off steam supply and simultaneously closing the condensate and air extraction pumps.

In case there is a leakage the readings of vacuum gauge and thermometer will record a fall.

The following methods are used to check the source of air leakage.

1. Put the steam condenser under air pressure and note its effect on soap water at the points where infiltration is likely to occur.
2. Put the peppramen oil on the suspected joint (when the condenser is operating) and make a check on the peppramen odour in the discharge of air ejector.
3. Large leakages in steam condenser under vacuum can be detected by moving/passing candle flame over possible openings.

20.6. EFFECTS OF AIR LEAKAGE IN A CONDENSER

The following are the *effects of air leakage in a condenser* :

1. **Lowered thermal efficiency.** The leaked air in the condenser results in increased back-pressure on the primemover which means there is loss of heat drop and consequently thermal efficiency of steam power plant is lowered.

2. **Increased requirement of cooling water**. The leaked air in the condenser lowers the partial pressure of steam which means a lowered saturation temperature of steam. As the saturation temperature of steam lowers, its latent heat increases. So it will require increased amount of cooling water for increased latent heat.

3. **Reduced heat transfer**. Air has poor thermal conductivity. Hence leaked air reduces the rate of heat transfer from the vapour, and consequently it requires surface of the tubes of a surface condenser to be increased for a given condenser capacity.

4. **Corrosion**. The presence of air in the condenser increases the corrosive action.

20.7. METHODS FOR OBTAINING MAXIMUM VACUUM IN CONDENSERS

Following are some of the methods used to obtain *maximum possible vacuum in condensers used in modern steam power plants*.

1. **Air pump.** Air pumps are provided to maintain a desired vacuum in the condenser by extracting the air and other non-condensable gases. They are usually classified as : (*a*) ***Wet air pumps*** which remove a *mixture of condensate and non-condensable gases.* (*b*) ***Dry air pump*** which removes the *air only*.

2. **Steam air ejector.** When a wet air pump (also called extraction pump) is employed then use is made of steam air ejectors to remove air from the mixture. The operation of the ejector consists in utilising the viscous drag of a high velocity steam jet for the ejection of air and other non-condensable gases from a chamber ; it is chiefly used for exhausting the air from steam condensers. In the case of ejectors used for steam plants where a high vacuum pressure is maintained

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WORKED EXAMPLES

Example 20.1. *If a barometer stands at 760 mm and condenser vacuum is at 710 mm and temperature is 30°C, calculate the mass of air per kg of uncondensed steam.*

Solution. Barometric reading = 760 mm of Hg

Condenser vacuum = 710 mm of Hg

Total absolute pressure in the condenser = 760 – 710 = 50 mm of Hg

= 50 × 0.001333 = 0.06665 bar

(∵ 1 mm Hg = 0.001333 bar)

Temperature in the condenser = 30°C

Corresponding to this temperature pressure of steam from steam tables

= 0.0425 bar

∴ Pressure exerted by air = 0.06665 – 0.0425 = 0.024 bar

And specific volume of steam = 32.89 m^3/kg (From steam tables)

Now $pV = m_a RT$

$$m_a = \frac{pV}{RT}$$

$$= \frac{0.024 \times 10^5 \times 32.89}{287 \times (30 + 273)} = \mathbf{0.908\ kg.}\ \textbf{(Ans.)}$$

Example 20.2. *Steam enters a condenser at 36°C and with barometer reading 760 mm. If the vacuum of 695 mm is produced find the vacuum efficiency.*

Solution. From steam tables corresponding to 36°C :

Partial pressure of steam, p_s = 0.0595 bar

$$= \frac{0.0595}{0.001333} \text{ mm of Hg} = 44.64 \text{ mm of Hg}$$

∴ Maximum obtainable vacuum = 760 – 44.64 = 715.36 mm of Hg

$$\therefore \textit{ Vacuum efficiency} = \frac{\text{Actual vacuum}}{\text{Max. obtainable vacuum}} = \frac{695}{715.36} \times 100 = \mathbf{97\%.}\ \textbf{(Ans.)}$$

Example 20.3. *The outlet and inlet temperatures of cooling water to a condenser are 37.5°C and 30°C respectively. If the vacuum in the barometer is 706 mm of mercury with barometer reading 760 mm determine efficiency.*

Solution. $t_{w_1} = 30°C,\ t_{w_2} = 37.5°C$

Absolute pressure in the condenser = 760 – 706 = 54 mm of Hg

= 54 × 0.001333 = 0.072 bar

From steam tables corresponding to 0.072 bar, $t_s \simeq 40°C$

∴ *Condenser efficiency*

$$= \frac{\text{Rise in temp. of cooling water}}{\text{(Temperature corresponding to vacuum in condenser – Inlet temperature of cooling water)}}$$

$$= \frac{(t_{w_2} - t_{w_1})}{(t_s - t_{w_1})} = \frac{37.5 - 30}{40 - 30} = \frac{7.5}{10} = \mathbf{0.75\ or\ 75\%.}\ \textbf{(Ans.)}$$

Example 20.4. *The surface condenser is designed to handle 16000 kg of steam per hour. The steam enters the condenser at 0.09 bar abs. and 0.88 dryness fraction and the condensate leaves the condenser at the corresponding saturation temperature. Determine the rise in cooling*

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Example 20.17. *A primemover uses 15000 kg of steam per hour and develops 2450 kW. The steam is supplied at 30 bar and 350°C. The exhaust from the primemover is condensed at 725 mm Hg when barometer records 755 mm Hg. The condensate temperature from the condenser is 31°C and the rise of temperature of circulating water is from 8°C to 18°C. Determine : (i) The quality of steam entering the condenser, (ii) The quantity of circulating cooling water and the ratio of cooling.*

Assume that no air is present in the condenser and all mechanical drive losses are negligible.

Solution. Quantity of steam used, m_s = 15000 kg/h

Pressure of steam = 30 bar, 350°C

Vacuum reading = 725 mm Hg

Barometer reading = 755 mm Hg

Condensate temperature = 31°C

Rise of temperature of cooling water $t_{w_2} - t_{w_1} = 18 - 8 = 10°C$.

(*i*) **Quality of steam entering the condenser, x :**

Condenser pressure = 755 – 725 = 30 mm Hg

$= 30 \times 1.333 \times 10^{-3} = 0.0399$ bar

From steam table :

At 0.0399 bar : $h_f = 121.5$ kJ/kg, $h_{fg} = 2432.9$ kJ/kg

At 30 bar, 350°C : $h_g = 3115.3$ kJ/kg

The work done in the turbine = 2450 kW or kJ/s ...(*i*)

Enthalpy drop in the turbine/sec.

$$= \frac{15000}{3600} [3115.3 - (121.5 + x \times 2432.9)] \text{ kJ/s} \quad ...(ii)$$

Since mechanical drive losses are negligible, the expressions (*i*) and (*ii*) are equal.

$$\therefore \quad 2450 = \frac{15000}{3600} [3115.3 - (121.5 + x \times 2432.9)]$$

or $$\frac{2450 \times 3600}{15000} = 3115.3 - 121.5 - x \times 2432.9$$

$$588 = 3115.3 - 121.5 - 2432.9x$$

$$\therefore \quad x = \frac{3115.3 - 121.5 - 588}{2432.9} = \mathbf{0.988.} \quad \textbf{(Ans.)}$$

(*ii*) **Quantity of circulating water, m_w :**

Ratio of cooling :

Heat lost by condensing steam = Heat gained by cooling water

$$m_s [h_f + x\, h_{fg} - h_c] = m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

or $$m_w = \frac{m_s [h_f + x\, h_{fg} - h_c]}{c_{pw} \times (t_{w_2} - t_{w_1})}$$

$$= \frac{15000 [121.5 + 0.988 \times 2432.9 - 4.186 \times 31]}{4.186 \times 10}$$

$$= \mathbf{858375 \text{ kg/h.}} \quad \textbf{(Ans.)}$$

[where c_{pw} = specific heat of water = 4.186 kJ/kg K]

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5. The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.

21.3. THE CARNOT CYCLE

This cycle has the *highest possible efficiency* and consists of four simple operations namely,

(*a*) Isothermal expansion

(*b*) Adiabatic expansion

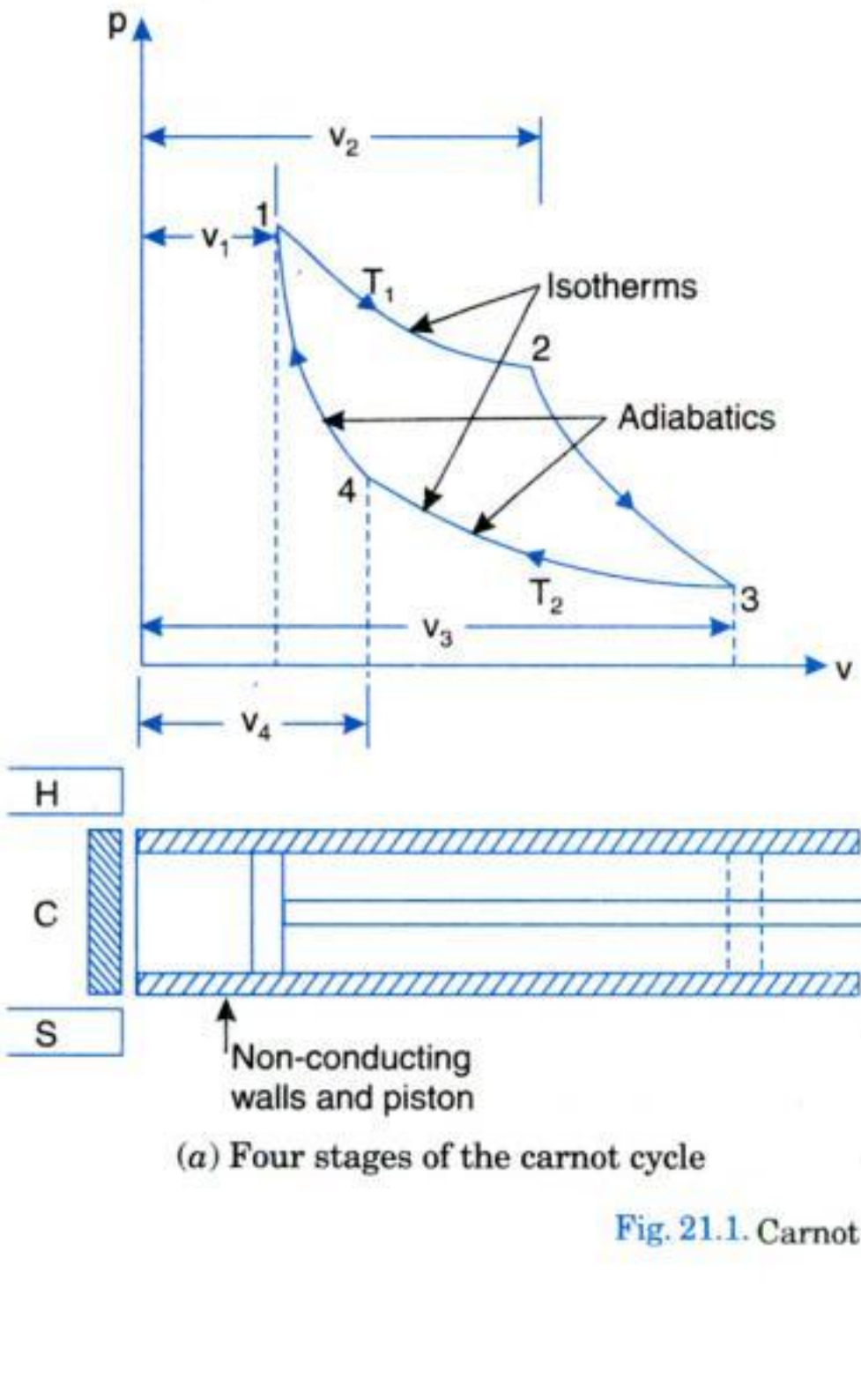
(*c*) Isothermal compression

(*d*) Adiabatic compression.

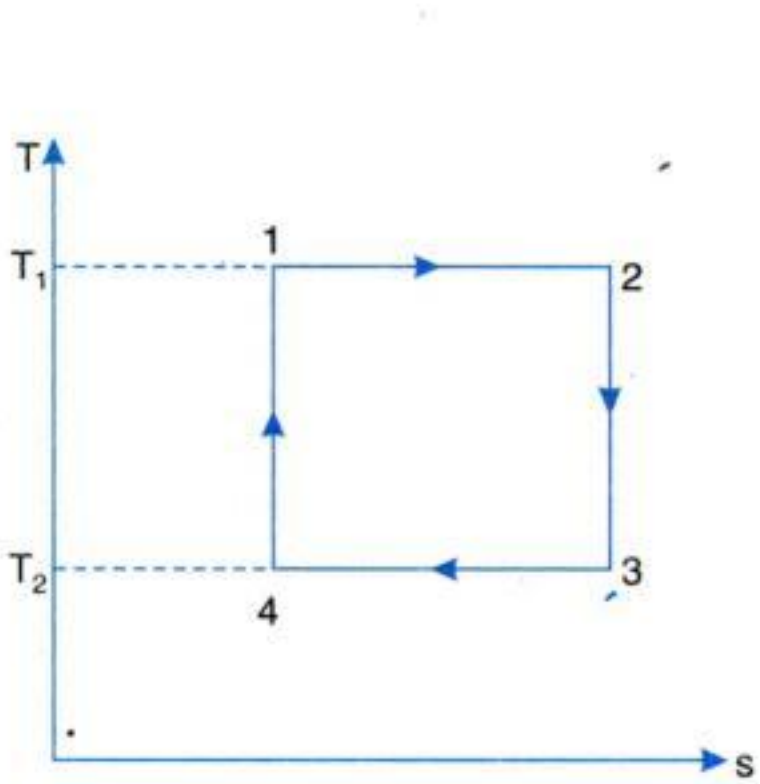
The condition of the Carnot cycle may be imagined to occur in the following way :

One kg of a air is enclosed in the cylinder which (except at the end) is made of perfect non-conducting material. A source of heat '*H*' is supposed to provide unlimited quantity of heat, non-conducting cover '*C*' and a sump '*S*' which is of infinite capacity so that its temperature remains unchanged irrespective of the fact how much heat is supplied to it. The temperature of source *H* is T_1 and the same is of the working substance. The working substance while rejecting heat to sump '*S*' has the temperature. T_2 *i.e.*, the same as that of sump *S*.

Following are the *four stages* of the Carnot cycle. Refer Fig. 21.1 (*a*).



(*a*) Four stages of the carnot cycle



(*b*) *T-s* diagram

Fig. 21.1. Carnot cycle.

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Solution. Refer Fig. 21.4.

Maximum pressure, $p_1 = 18$ bar

Maximum temperature, $T_1 = (T_2) = 410 + 273 = 683$ K

Ratio of isentropic (or adiabatic) compression, $\frac{V_4}{V_1} = 6$

Ratio of isothermal expansion, $\frac{V_2}{V_1} = 1.5$.

Volume of the air at the beginning of isothermal expansion, $V_1 = 0.18$ m^3.

(*i*) **Temperatures and pressures at the main points in the cycle :**

For the *isentropic process* 4–1 :

$$\frac{T_1}{T_4} = \left(\frac{V_4}{V_1}\right)^{\gamma - 1} = (6)^{1.4 - 1} = (6)^{0.4} = 2.05$$

$$\therefore \quad T_4 = \frac{T_1}{2.05} = \frac{683}{2.05} = 333.2 \text{ K} = T_3$$

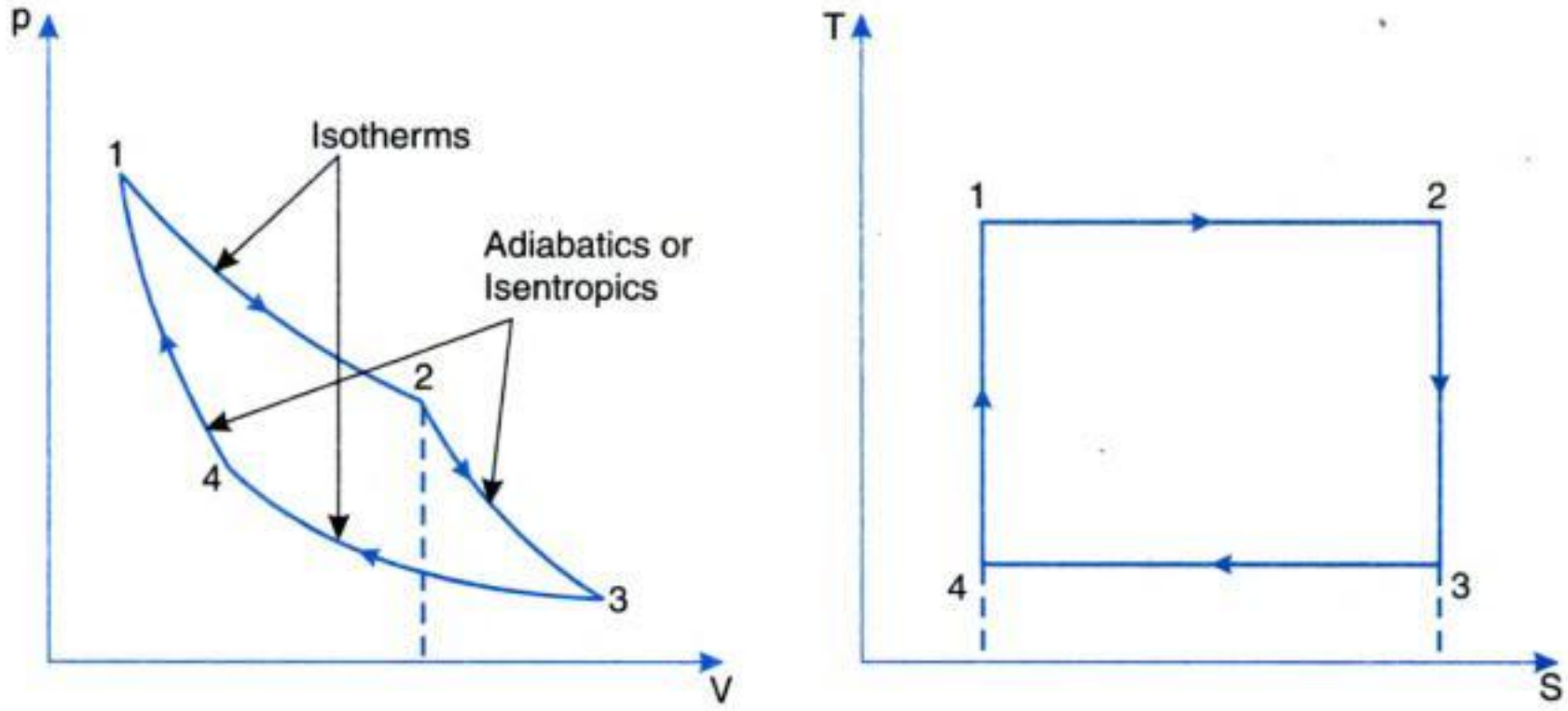

Fig. 21.4

Also, $$\frac{p_1}{p_4} = \left(\frac{V_4}{V_1}\right)^{\gamma} = (6)^{1.4} = 12.29$$

$$\therefore \quad p_4 = \frac{p_1}{12.29} = \frac{18}{12.29} = 1.46 \text{ bar}$$

For the *isothermal process* 1–2 :

$$p_1 V_1 = p_2 V_2$$

$$p_2 = \frac{p_1 V_1}{V_2} = \frac{18}{1.5} = 12 \text{ bar}$$

For *isentropic process* 2–3, we have :

$$p_2 V_2^{\gamma} = p_3 V_3^{\gamma}$$

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$$= \frac{v_1}{\gamma - 1}\left[(r^{\gamma-1} - 1)(p_4 - p_1)\right]$$

$$= \frac{p_1 v_1}{\gamma - 1}\left[(r^{\gamma-1} - 1)(r_p - 1)\right] \quad \text{...[21.4 (a)]}$$

Mean effective pressure ($\mathbf{p_m}$) is given by :

$$p_m = \left[\left(\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}\right) \div (v_1 - v_2)\right] \text{ bar} \quad \text{...(21.5)}$$

Also

$$p_m = \frac{\left[\frac{p_1 v_1}{\gamma - 1}(r^{\gamma-1} - 1)(r_p - 1)\right]}{(v_1 - v_2)}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma-1} - 1)(r_p - 1)}{v_1 - \frac{v_1}{r}}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma-1} - 1)(r_p - 1)]}{v_1\left(\frac{r-1}{r}\right)}$$

i.e.,

$$\mathbf{p_m} = \frac{p_1 r[(r^{\gamma-1} - 1)(r_p - 1)]}{(\gamma - 1)(r - 1)} \quad \text{...(21.6)}$$

Example 21.7. *The efficiency of an Otto cycle is 60% and $\gamma = 1.5$. What is the compression ratio ?*

Solution. Efficiency of Otto cycle, $\eta = 60\%$

Ratio of specific heats, $\gamma = 1.5$

Compression ratio, $r = ?$

Efficiency of Otto cycle is given by,

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

$$0.6 = 1 - \frac{1}{(r)^{1.5-1}}$$

or

$$\frac{1}{(r)^{0.5}} = 0.4 \quad \text{or} \quad (r)^{0.5} = \frac{1}{0.4} = 2.5 \quad \text{or} \quad r = 6.25$$

Hence, *compression ratio* = **6.25. (Ans.)**

Example 21.8. *An engine of 250 mm bore and 375 mm stroke works on Otto cycle. The clearance volume is 0.00263 m^3. The initial pressure and temperature are 1 bar and 50°C. If the maximum pressure is limited to 25 bar, find the following :*

(i) The air standard efficiency of the cycle..

(ii) The mean effective pressure for the cycle.

Assume the ideal conditions.

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(*iii*) **Work done :**

Again, for *adiabatic compression 1–2,*

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} = (6.9)^{1.4 - 1} = (6.9)^{0.4} = 2.16$$

or $$T_2 = T_1 \times 2.16 = 311 \times 2.16 = 671.7 \text{ K or } 398.7°C$$

For *adiabatic expansion process 3–4*

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma - 1} = (r)^{\gamma - 1} = (6.9)^{0.4} = 2.16$$

or $$T_4 = \frac{T_3}{2.16} = \frac{2223}{2.16} = 1029 \text{ K or } 756°C$$

Heat supplied *per kg of air*

$$= c_v(T_3 - T_2) = 0.717(2223 - 671.7)$$
$$= 1112.3 \text{ kJ/kg or air}$$

$$\left[c_v = \frac{R}{\gamma - 1} = \frac{0.287}{1.4 - 1} = 0.717 \text{ kJ/ kg K} \right]$$

Heat rejected per kg of air

$$= c_v(T_4 - T_1) = 0.717(1029 - 311)$$
$$= 514.8 \text{ kJ/kg of air}$$

∴ *Work done per kg of air* = Heat supplied – heat rejected
$$= 1112.3 - 514.8$$
$$= \mathbf{597.5 \text{ kJ}} \text{ or } \mathbf{597500 \text{ Nm.}} \quad \textbf{(Ans.)}$$

☞ **Example 21.13.** *An engine working on Otto cycle has a volume of 0.45 m^3, pressure 1 bar and temperature 30°C at the beginning of compression stroke. At the end of compression stroke, the pressure is 11 bar. 210 kJ of heat is added at constant volume. Determine :*

(*i*) *Pressures, temperatures and volumes at salient points in the cycle.*

(*ii*) *Percentage clearance.*

(*iii*) *Efficiency.*

(*iv*) *Mean effective pressure.*

(*v*) *Ideal power developed by the engine if the number of working cycles per minute is 210. Assume the cycle is reversible.*

Solution. Refer Fig. 21.12

Volume,	$V_1 = 0.45 \text{ m}^3$
Initial pressure,	$p_1 = 1$ bar
Initial temperature,	$T_1 = 30 + 273 = 303$ K
Pressure at the end of compression stroke,	$p_2 = 11$ bar
Heat added at constant volume	= 210 kJ
Number of working cycles/min.	= 210.

(*i*) **Pressures, temperatures and volumes at salient points :**

For *adiabatic compression 1–2*

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

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$$= \left(\frac{r}{\rho}\right)^{\gamma-1} \qquad \left(\because \ \frac{v_4}{v_3} = \frac{v_1}{v_3} = \frac{v_1}{v_2} \times \frac{v_2}{v_3} = \frac{r}{\rho}\right)$$

$$\therefore \qquad T_4 = \frac{T_3}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = \frac{\rho \,.\, T_1 (r)^{\gamma-1}}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = T_1 \,.\, \rho^{\gamma}$$

By inserting values of T_2, T_3 and T_4 in eqn. (*i*), we get

$$\eta_{\text{diesel}} = 1 - \frac{(T_1 \,.\, \rho^{\gamma} - T_1)}{\gamma\,(\rho \,.\, T_1 \,.\, (r)^{\gamma-1} - T_1 \,.\, (r)^{\gamma-1})} = 1 - \frac{(\rho^{\gamma} - 1)}{\gamma (r)^{\gamma-1} (\rho - 1)}$$

or $$\eta_{\text{diesel}} = 1 - \frac{1}{\gamma (r)^{\gamma-1}} \left[\frac{\rho^{\gamma} - 1}{\rho - 1}\right] \qquad \text{...(21.7)}$$

It may be observed that eqn. (21.7) for efficiency of diesel cycle is different from that of the Otto cycle only in bracketed factor. This factor is always greater than unity, because $\rho > 1$. Hence *for a given compression ratio, the Otto cycle is more efficient.*

The *net work* for diesel cycle can be expressed in terms of *pv* as follows :

$$W = p_2(v_3 - v_2) + \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}$$

$$= p_2\,(\rho v_2 - v_2) + \frac{p_3 \rho v_2 - p_4 r v_2}{\gamma - 1} - \frac{p_2 v_2 - p_1 r v_2}{\gamma - 1}$$

$$\left[\because \ \frac{v_3}{v_2} = \rho \quad \therefore \ v_3 = \rho v_2 \ \text{ and } \frac{v_1}{v_2} = r \quad \therefore \ v_1 = r v_2 \right.$$
$$\left. \text{But } v_4 = v_1 \quad \therefore \quad v_4 = r v_2 \right]$$

$$= p_2 v_2\,(\rho - 1) + \frac{p_3 \rho v_2 - p_4 r v_2}{\gamma - 1} - \frac{p_2 v_2 - p_1 r v_2}{\gamma - 1}$$

$$= \frac{v_2 [p_2 (\rho - 1)(\gamma - 1) + p_3 \rho - p_4 r - (p_2 - p_1 r)]}{\gamma - 1}$$

$$= \frac{v_2 \left[p_2 (\rho - 1)(\gamma - 1) + p_3 \left(\rho - \frac{p_4 r}{p_3}\right) - p_2 \left(1 - \frac{p_1 r}{p_2}\right)\right]}{\gamma - 1}$$

$$= \frac{p_2 v_2 [(\rho - 1)(\gamma - 1) + \rho - \rho^{\gamma} \,.\, r^{1-\gamma} - (1 - r^{1-\gamma})}{\gamma - 1}$$

$$\left[\because \ \frac{p_4}{p_3} = \left(\frac{v_3}{v_4}\right)^{\gamma} = \left(\frac{\rho}{r}\right)^{\gamma} = \rho^{\gamma} r^{-\gamma}\right]$$

$$= \frac{p_1 v_1 r^{\gamma-1} [(\rho - 1)(\gamma - 1) + \rho - \rho^{\gamma} r^{1-\gamma} - (1 - r^{1-\gamma})}{\gamma - 1}$$

$$\left[\because \ \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} \ \text{ or } \ p_2 = p_1 \,.\, r^{\gamma} \text{ and } \frac{v_1}{v_2} = r \ \text{ or } \ v_2 = v_1\, r^{-1}\right]$$

$$= \frac{p_1 v_1 r^{\gamma-1} [\gamma(\rho - 1) - r^{1-\gamma} (\rho^{\gamma} - 1)]}{(\gamma - 1)} \qquad \text{...(21.8)}$$

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For the *adiabatic (or isentropic) process 1–2*

$$p_1V_1^{\gamma} = p_2V_2^{\gamma} \quad \text{or} \quad \frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = (r)^{\gamma}$$

$\therefore$ $$\boldsymbol{p_2} = p_1 \cdot (r)^{\gamma} = 1 \times (15)^{1.4} = \mathbf{44.31\ bar.}$$ **(Ans.)**

Also, $$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (15)^{1.4-1} = 2.954$$

$\therefore$ $$\boldsymbol{T_2} = T_1 \times 2.954 = 300 \times 2.954 = \mathbf{886.2\ K.}$$ **(Ans.)**

$$\boldsymbol{V_2} = V_c = \frac{V_s}{r-1} = \frac{0.00942}{15-1} = \mathbf{0.0006728\ m^3.}$$ **(Ans.)**

$$\boldsymbol{p_2} = p_3 = \mathbf{44.31\ bar.}$$ **(Ans.)**

% cut-off ratio $$= \frac{\rho - 1}{r-1}$$

$$\frac{8}{100} = \frac{\rho - 1}{15-1}$$

i.e., $$\rho = 0.08 \times 14 + 1 = 2.12$$

$\therefore$ $$V_3 = \rho\, V_2 = 2.12 \times 0.0006728 = \mathbf{0.001426\ m^3.}$$ **(Ans.)**

[V_3 can also be calculated as follows :

$V_3 = 0.08V_s + V_c = 0.08 \times 0.00942 + 0.0006728 = 0.001426\ m^3$]

For the *constant pressure process 2–3,*

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

$\therefore$ $$\boldsymbol{T_3} = T_2 \times \frac{V_3}{V_2} = 886.2 \times \frac{0.001426}{0.0006728} = \mathbf{1878.3\ K.}$$ **(Ans.)**

For the *isentropic process 3–4,*

$$p_3V_3^{\gamma} = p_4V_4^{\gamma}$$

$$p_4 = p_3 \times \left(\frac{V_3}{V_4}\right)^{\gamma} = p_3 \times \frac{1}{(7.07)^{1.4}}$$

$$= \frac{44.31}{(7.07)^{1.4}} = \mathbf{2.866\ bar.}$$ **(Ans.)**

[$\because$ $\frac{V_4}{V_3} = \frac{V_4}{V_2} \times \frac{V_2}{V_3} = \frac{V_1}{V_2} \times \frac{V_2}{V_3}$

$= \frac{r}{\rho}$, $\because$ $V_4 = V_1 = \frac{15}{2.12} = 7.07$]

Also, $$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{1}{7.07}\right)^{1.4-1} = 0.457$$

$\therefore$ $$\boldsymbol{T_4} = T_3 \times 0.457 = 1878.3 \times 0.457 = \mathbf{858.38\ K.}$$ **(Ans.)**

$$\boldsymbol{V_4} = V_1 = \mathbf{0.0101\ m^3.}$$ **(Ans.)**

(*ii*) **Theoretical air standard efficiency :**

$$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right] = 1 - \frac{1}{1.4\,(15)^{1.4-1}}\left[\frac{(2.12)^{1.4}-1}{2.12-1}\right]$$

$$= 1 - 0.2418 \times 1.663 = \mathbf{0.598}\ \text{or}\ \mathbf{59.8\%.}$$ **(Ans.)**

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$$\therefore \qquad T_2 = T_1 \times 2.408 = 303 \times 2.408 = 729.6 \text{ K}$$

For the *constant volume process 2–3,*

$$\frac{T_3}{p_3} = \frac{T_2}{p_2}$$

$$\therefore \qquad T_3 = T_2 \cdot \frac{p_3}{p_2} = 729.6 \times \frac{60}{21.67} = 2020 \text{ K}$$

Also, $$\frac{\rho - 1}{r - 1} = \frac{4}{100} \quad \text{or} \quad 0.04$$

$$\therefore \qquad \frac{\rho - 1}{9 - 1} = 0.04 \quad \text{or} \quad \rho = 1.32$$

For the *constant pressure process 3–4,*

$$\frac{V_4}{T_4} = \frac{V_3}{T_3} \quad \text{or} \quad \frac{T_4}{T_3} = \frac{V_4}{V_3} = \rho$$

$$\therefore \qquad T_4 = T_3 \times \rho = 2020 \times 1.32 = 2666.4 \text{ K}$$

Also expansion ratio, $$\frac{V_5}{V_4} = \frac{V_5}{V_2} \times \frac{V_2}{V_4} = \frac{V_1}{V_2} \times \frac{V_3}{V_4} = \frac{r}{\rho} \qquad [\because V_5 = V_1 \text{ and } V_2 = V_3]$$

For *adiabatic process 4–5,*

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

$$\therefore \qquad T_5 = T_4 \times \left(\frac{\rho}{r}\right)^{\gamma - 1} = 2666.4 \times \left(\frac{1.32}{9}\right)^{1.4 - 1} = 1237 \text{ K}$$

Also $$p_4 V_4^{\gamma} = p_5 V_5^{\gamma}$$

$$p_5 = p_4 \cdot \left(\frac{V_4}{V_5}\right)^{\gamma} = 60 \times \left(\frac{r}{\rho}\right)^{\gamma} = 60 \times \left(\frac{1.32}{9}\right)^{1.4} = 4.08 \text{ bar}$$

Heat supplied, $$Q_s = c_v(T_3 - T_2) + c_p(T_4 - T_3)$$
$$= 0.71\ (2020 - 729.6) + 1.0\ (2666.4 - 2020) = 1562.58 \text{ kJ/kg}$$

Heat rejected, $$Q_r = c_v\ (T_5 - T_1)$$
$$= 0.71\ (1237 - 303) = 663.14 \text{ kJ/kg}$$

$$\eta_{\text{air-standard}} = \frac{Q_s - Q_r}{Q_s} = \frac{1562.85 - 663.14}{1562.58} = \mathbf{0.5756} \text{ or } \mathbf{57.56\%}. \quad \textbf{(Ans.)}$$

(*ii*) **Power developed by the engine, P :**

Mass of air in the cycle is given by

$$m = \frac{p_1 V_1}{R T_1} = \frac{1 \times 10^5 \times 0.0165}{287 \times 303} = 0.0189 \text{ kg}$$

$\therefore$ Work done per cycle $= m(Q_s - Q_r)$
$= 0.0189\ (1562.58 - 663.14) = 16.999 \text{ kJ}$

Power developed = Work done per cycle × no. of cycles per second
$= 16.999 \times 3 =$ **50.99 say 51 kW.** **(Ans.)**

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or $$\frac{T_2}{T_1} = \frac{v_2}{v_1} = \frac{\alpha}{r} \qquad ...(iii) \quad \left(\frac{v_2}{v_1} = \frac{v_2}{v_3} \times \frac{v_3}{v_1} = \frac{v_2}{v_3} \times \frac{v_4}{v_1} = \frac{\alpha}{r}\right)$$

During *adiabatic expansion 4–1,*

$$\frac{T_4}{T_1} = \left(\frac{v_1}{v_4}\right)^{\gamma - 1} = (r)^{\gamma - 1}$$

$$T_1 = \frac{T_4}{(r)^{\gamma - 1}} \qquad ...(iv)$$

Putting the value of T_1 in eqn. (*iii*), we get

$$T_2 = \frac{T_4}{(r)^{\gamma - 1}} \cdot \frac{\alpha}{r}$$

$$= \frac{\alpha T_4}{r^{\gamma}} \qquad ...(v)$$

Substituting the value of T_2 in eqn. (*ii*), we get

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

Finally putting the values of T_1, T_2 and T_3 in eqn. (*i*), we get

$$\eta = 1 - \gamma \left(\frac{\frac{T_4}{r^{\gamma - 1}} - \frac{\alpha T_4}{(r)^{\gamma}}}{T_4 - \left(\frac{\alpha}{r}\right)^{\gamma} . T_4} \right) = 1 - \gamma \left(\frac{r - \alpha}{r^{\gamma} - \alpha^{\gamma}} \right)$$

Hence, air standard efficiency $= 1 - \gamma \cdot \left(\frac{r - \alpha}{r^{\gamma} - \alpha^{\gamma}} \right)$ ...(21.14)

Example 21.31. *A perfect gas undergoes a cycle which consists of the following processes taken in order :*

(a) Heat rejection at constant pressure.

(b) Adiabatic compression from 1 bar and 27°C to 4 bar.

(c) Heat addition at constant volume to a final pressure of 16 bar.

(d) Adiabatic expansion to 1 bar.

Calculate : (i) Work done/kg of gas.

(ii) Efficiency of the cycle.

Take : $c_p = 0.92$, $c_v = 0.75$.

Solution. Refer Fig. 21.31.

Pressure, $p_2 = p_1 = 1$ bar

Temperature, $T_2 = 27 + 273 = 300$ K

Pressure after adiabatic compression, $p_3 = 4$ bar

Final pressure after heat addition, $p_4 = 16$ bar

For *adiabatic compression 2–3,*

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.22 - 1}{1.22}} = 1.284 \qquad \left[\gamma = \frac{c_p}{c_v} = \frac{0.92}{0.75} = 1.22\right]$$

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or $$T_2' = \frac{445.8 - 300}{0.8} + 300 = 482.2 \text{ K}$$

Now, heat supplied by the fuel = heat taken by the burning gases

$$0.9 \times m_f \times C = (m_a + m_f) \times c_p \times (T_3 - T_2')$$

$$\therefore \quad C = \left(\frac{m_a + m_f}{m_f}\right) \times \frac{c_p(T_3 - T_2')}{0.9} = \left(\frac{m_a}{m_f} + 1\right) \times \frac{c_p(T_3 - T_2')}{0.9}$$

or $$42000 = \left(\frac{m_a}{m_f} + 1\right) \times \frac{1.00(1148 - 482.27)}{0.9} = 739.78 \left(\frac{m_a}{m_f} + 1\right)$$

$$\therefore \quad \frac{m_a}{m_f} = \frac{42000}{739.78} - 1 = 55.77 \text{ say } 56$$

$\therefore$ **A/F ratio = 56 : 1.** **(Ans.)**

ADDITIONAL/TYPICAL EXAMPLES

Example 21.40. *In an engine working on the Otto cycle between given lower and upper limits of absolute temperature, T_1 and T_2 respectively, show that for maximum work to be done per kg, the ratio of compression is given by :*

$$r = \left(\frac{T_3}{T_1}\right)^{1.25}$$

where, γ = Ratio of specific heats = 1.4

Solution. Fig. 21.42, shows the cycle for the Otto cycle.

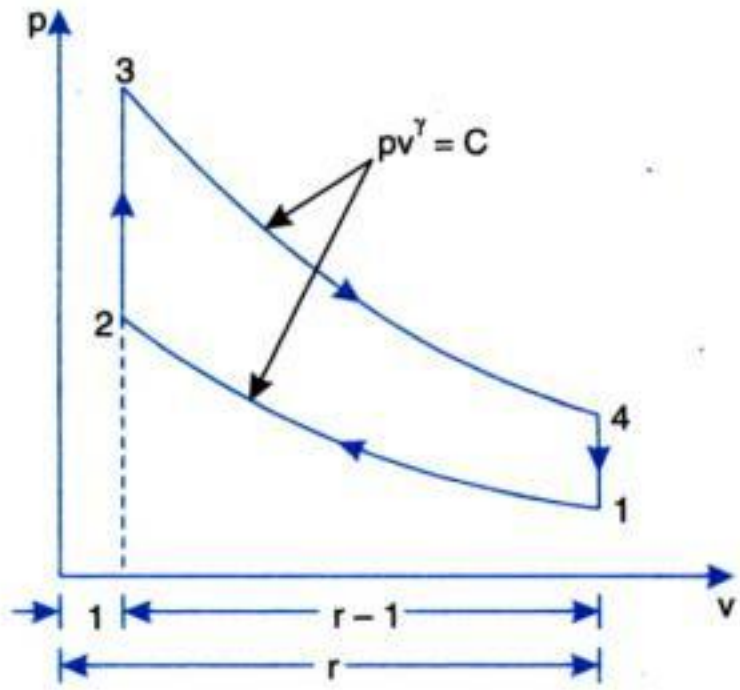

Fig. 21.42

$$T_2 = T_1 \times (r)^{\gamma-1} \; ; \; T_4 = T_3 \times \left(\frac{1}{r}\right)^{\gamma-1}$$

Work done, $W = c_v(T_3 - T_2) - c_v(T_4 - T_1)$

$$= c_v(T_3 - T_1 \times r^{\gamma-1}) - c_v \left\{T_3\left(\frac{1}{r}\right)^{\gamma-1} - T_1\right\}$$

For maximum work, W is differentiated with the variable r and equated to 0.

i.e., $$\frac{dW}{dr} = [0 - (\gamma - 1)T_1(r)^{\gamma-2}]$$

$$-[-T_3 \times (\gamma - 1)r^{-\gamma} - 0] = 0$$

or, $$(\gamma - 1)\, T_1\, (r)^{\gamma-2} = (\gamma - 1)\, T_3\, r^{-\gamma}$$

or, $$\frac{T_3}{T_1} = \frac{(r)^{\gamma-2}}{(r)^{-\gamma}} = (r)^{2(\gamma-1)}$$

$\therefore$ $$\mathbf{r} = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}} = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(1.4-1)}} = \left(\frac{T_3}{T_1}\right)^{1.25}$$ **...Proved.**

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ANSWERS

1. (*b*) 2. (*b*) 3. (*d*) 4. (*d*) 5. (*a*) 6. (*b*).

THEORETICAL QUESTIONS

1. What is a cycle ? What is the difference between an ideal and actual cycle ?
2. What is an air-standard efficiency ?
3. What is relative efficiency ?
4. Derive expressions of efficiency in the following cases :
   (*i*) Carnot cycle (*ii*) Diesel cycle (*iii*) Dual combustion cycle.
5. Explain "Air standard analysis" which has been adopted for I.C. engine cycles. State the assumptions made for air standard cycles.
6. Derive an expression for 'Atkinson cycle'.

UNSOLVED EXAMPLES

1. A Carnot engine working between 377°C and 37°C produces 120 kJ of work. Determine :
   (*i*) The heat added in kJ. (*ii*) The entropy change during heat rejection process.
   (*iii*) The engine thermal efficiency. [**Ans.** (*i*) 229.5 kJ ; (*ii*) 0.353 kJ/K ; (*iii*) 52.3%]
2. Find the thermal efficiency of a Carnot engine whose hot and cold bodies have temperatures of 154°C and 15°C respectively. [**Ans.** 32.55%]
3. Derive an expression for change in efficiency for a change in compression ratio. If the compression ratio is increased from 6 to 8, what will be the percentage increase in efficiency ? [**Ans.** 8%]
4. The efficiency of an Otto cycle is 50% and γ is 1.5. What is the compression ratio ? [**Ans.** 4]
5. An engine working on Otto cycle has a volume of 0.5 m^3, pressure 1 bar and temperature 27°C at the commencement of compression stroke. At the end of compression stroke, the pressure is 10 bar. Heat added during the constant volume process is 200 kJ. Determine :
   (*i*) Percentage clearance (*ii*) Air standard efficiency
   (*iii*) Mean effective pressure
   (*iv*) Ideal power developed by the engine if the engine runs at 400 r.p.m. so that there are 200 complete cycles per minutes. [**Ans.** (*i*) 23.76% ; (*ii*) 47.2% ; (*iii*) 2.37 bar ; (*iv*) 321 kW]
6. The compression ratio in an air-standard Otto cycle is 8. At the beginning of compression process, the pressure is 1 bar and the temperature is 300 K. The heat transfer to the air per cycle is 1900 kJ/kg of air. Calculate :
   (*i*) Thermal efficiency (*ii*) The mean effective pressure.
   [**Ans.** (*i*) 56.47% ; (*ii*) 14.24 bar]
7. An engine 200 mm bore and 300 mm stroke works on Otto cycle. The clearance volume is 0.0016 m^3. The initial pressure and temperature are 1 bar and 60°C. If the maximum pressure is limited to 24 bar, find :
   (*i*) The air-standard efficiency of the cycle (*ii*) The mean effective pressure for the cycle.
   Assume ideal conditions. [**Ans.** (*i*) 54.08% ; (*ii*) 1.972 bar]
8. Calculate the air standard efficiency of a four stroke Otto cycle engine with the following data :
   Piston diameter (bore) = 137 mm ; Length of stroke = 130 mm ;
   Clearance volume 0.00028 m^3.
   Express clearance as a percentage of swept volume. [**Ans.** 56.1% ; 14.6%]
9. In an ideal Diesel cycle, the temperatures at the beginning of compression, at the end of compression and at the end of the heat addition are 97°C, 789°C and 1839°C. Find the efficiency of the cycle.
   [**Ans.** 59.6%]

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—Fig. 22.2 shows the effect of humidity on properties of air. R is the characteristic gas constant ; c_p and c_v are the specific heats at constant pressure and constant volume respectively, γ = ratio of specific heats.

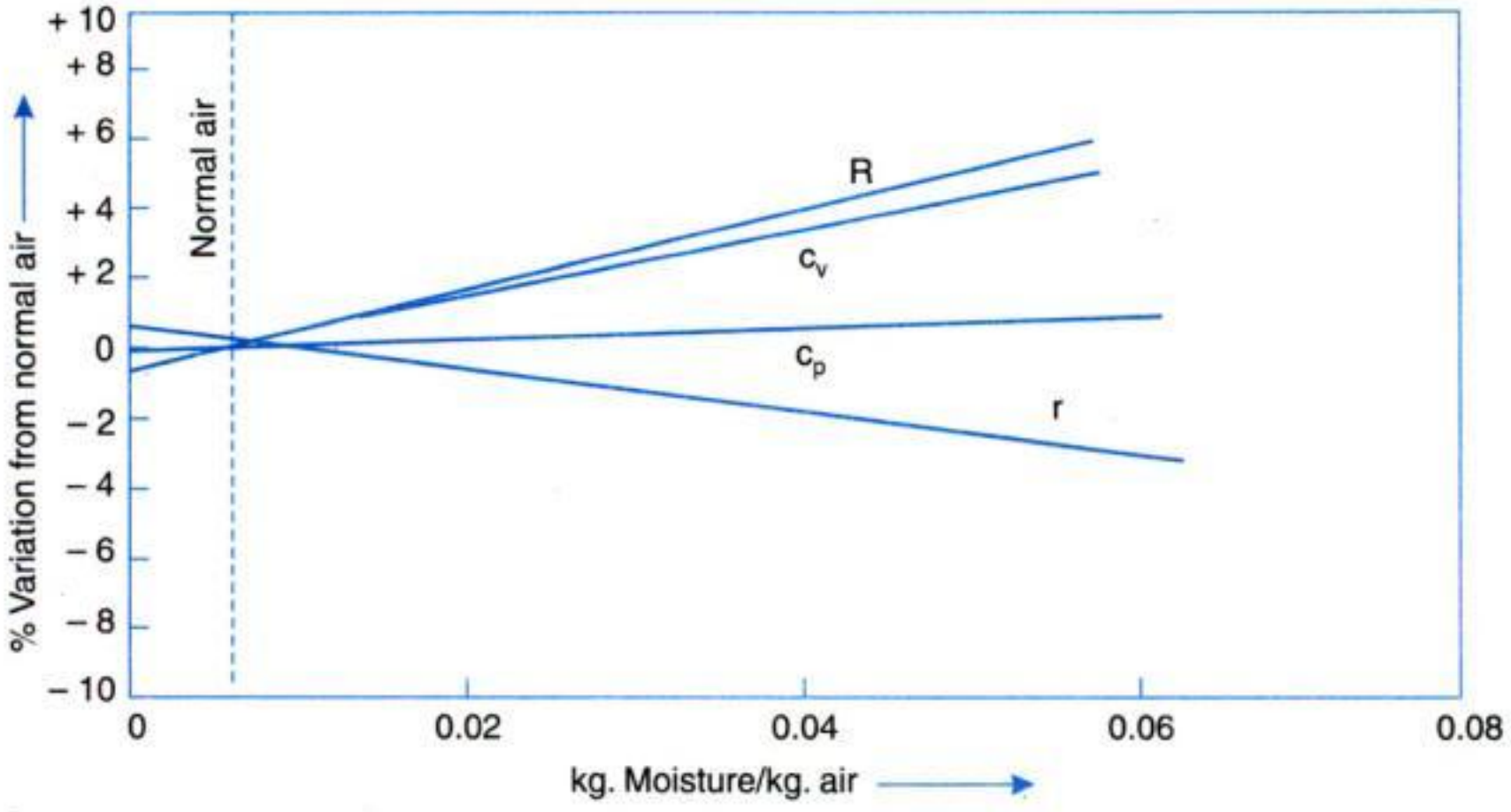

Fig. 22.2. Effect of humidity on properties of air.

22.2. CHANGE OF INTERNAL ENERGY AND ENTHALPY DURING A PROCESS WITH VARIABLE SPECIFIC HEATS

The small change in internal energy (du) of a *unit mass* of a gas for small change in temperature (dT) is given by :

$$du = c_v\, dT$$

$$\therefore \quad u_2 - u_1 = \int_{T_1}^{T_2} c_v\, dT = \int_{T_1}^{T_2} (b + KT)\, dT$$

$$= b(T_2 - T_1) + \frac{K}{2}\,(T_2^2 - T_1^2)$$

$$= (T_2 - T_1)\left[b + K\left(\frac{T_1 + T_2}{2}\right)\right] = (T_2 - T_1)\,(b + KT_m)$$

where, T_m (mean temperature) $= \dfrac{T_1 + T_2}{2}$

$c_{vm} = b + KT_m$ (c_{vm} means mean specific heat at constant volume)

$$\therefore \quad u_2 - u_1 = c_{vm}\,(T_2 - T_1) \qquad \ldots(22.6)$$

Similarly, change in enthalpy is given by :

$$dh = c_p\, dT$$

$$\therefore \quad h_2 - h_1 = c_{pm}\,(T_2 - T_1) \qquad \ldots(22.7)$$

where, c_{pm} mean specific heat at constant pressure.

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HIGHLIGHTS

1. The specific heat varies largely with temperature but not very significantly with pressure except at very high pressure.
2. The specific heats of gases increase with the rise in temperature since the vibrational energy of molecules increases with temperature.
3. Change of *internal energy* (per kg) during a process with variable specific heats
$$u_2 - u_1 = c_{vm}\,(T_2 - T_1)$$
Change of *enthalpy*, $h_2 - h_1 = c_{pm}\,(T_2 - T_1)$.
4. Heat transfer during a process with variable specific heats,
$$dQ = \left[\left(\frac{\gamma' - n}{\gamma' - 1}\right) - \left(\frac{n-1}{a-b}\right) KT\right] dW.$$
5. In case of isentropic expansion with variable specific heats
$$p(v)^{a/b} \,.\, e^{KT/b} = \text{constant}.$$
6. 'Entropy change' during a process with variable specific heats,
$$s_2 - s_1 = a \log_e \left(\frac{T_2}{T_1}\right) - (a-b) \log_e \left(\frac{p_2}{p_1}\right) + K(T_2 - T_1).$$
7. Percentage variation in air standard efficiency on account of percentage variation in c_v in case of :

(*i*) **Otto cycle :**
$$\frac{d\eta}{\eta} = -\frac{1-\eta}{\eta}\,(\gamma - 1) \log_e r \times \frac{dc_v}{c_v}.$$

(*ii*) **Diesel cycle :**
$$\frac{d\eta}{\eta} = -\frac{1-\eta}{\eta}\,(\gamma - 1)\left[\log_e r - \frac{\rho^\gamma \,.\, \log_e \rho}{\rho^\gamma - 1} + \frac{1}{\gamma}\right] \frac{dc_v}{c_v}.$$

OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer :

1. The value of c_v of a real gas
(*a*) increases with increase in temperature
(*b*) decreases with increase in temperature
(*c*) remains constant with increase in temperature
(*d*) first increases with increase in temperature and then decreases with further increase in temperature.
2. The value c_p of a real gas
(*a* increases with increase in tempcrature
(*b*) decreases with increase in temperature
(*c*) remains constant with increase in temperature
(*d*) first increases with increase in temperature and then decreases with further increase in temperature.
3. The value of c_p
(*a*) decreases with increase in moisture content in air
(*b*) increases with increase in moisture content in air
(*c*) remain same irrespective of the moisture content in air
(*d*) increases with increase in moisture content and then decreases with further increase in moisture content in air.
4. The value of $\gamma\,(= c_p/c_v)$ for air
(*a*) decreases with increase in moisture content in air
(*b*) increase with increases in moisture content in air

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6. **According to the speed of the engine :**

(*i*) Low speed engine (*ii*) Medium speed engine

(*iii*) High speed engine.

7. **According to method of ignition :**

(*i*) Spark ignition (S.I.) engine (*ii*) Compression ignition (C.I.) engine.

8. **According to method of cooling the cylinder :**

(*i*) Air-cooled engine (*ii*) Water-cooled engine.

9. **According to method of Governing :**

(*i*) Hit and miss governed engine (*ii*) Quality governed engine

(*iii*) Quantity governed engine.

10. **According to valve arrangement :**

(*i*) Overhead valve engine (*ii*) *L*-head type engine

(*iii*) *T*-head type engine (*iv*) *F*-head type engine.

11. **According to number of cylinders :**

(*i*) Singlecylinder engine (*ii*) Multi-cylinder engine.

23.4. APPLICATIONS OF I.C. ENGINES

The I.C. engines are generally used for :

(*i*) Road vehicles (*e.g.*, scooter, motorcycle, buses etc.)

(*ii*) Air craft

(*iii*) Locomotives

(*iv*) Construction in civil engineering equipment such as bull-dozer, scraper, power shovels etc.

(*v*) Pumping sets

(*vi*) Cinemas

(*vii*) Hospital

(*viii*) Several industrial applications.

Note. Prime movers in all *construction equipment* are invariable I.C. engines, unless of course, when drive is electric. Use of steam source for this equipment is almost absolete.

23.5. BASIC IDEA OF I.C. ENGINES

The basic idea of internal combustion engine is shown in Fig. 23.1. The cylinder which is closed at one end is filled with a mixture of fuel and air. As the crankshaft turns it pushes cylinder. The piston is forced up and compresses the mixture in the top of the cylinder. The mixture is set alight and, as it burns, it creates a gas pressure on the piston, forcing it down the cylinder. This motion is shown by arrow '1'. The piston pushes on the rod which pushes on the crank. The crank is given rotary (turning) motion as shown by the arrow '2'. The flywheel fitted on the end of the crankshaft stores energy and keeps the crank turning steadily.

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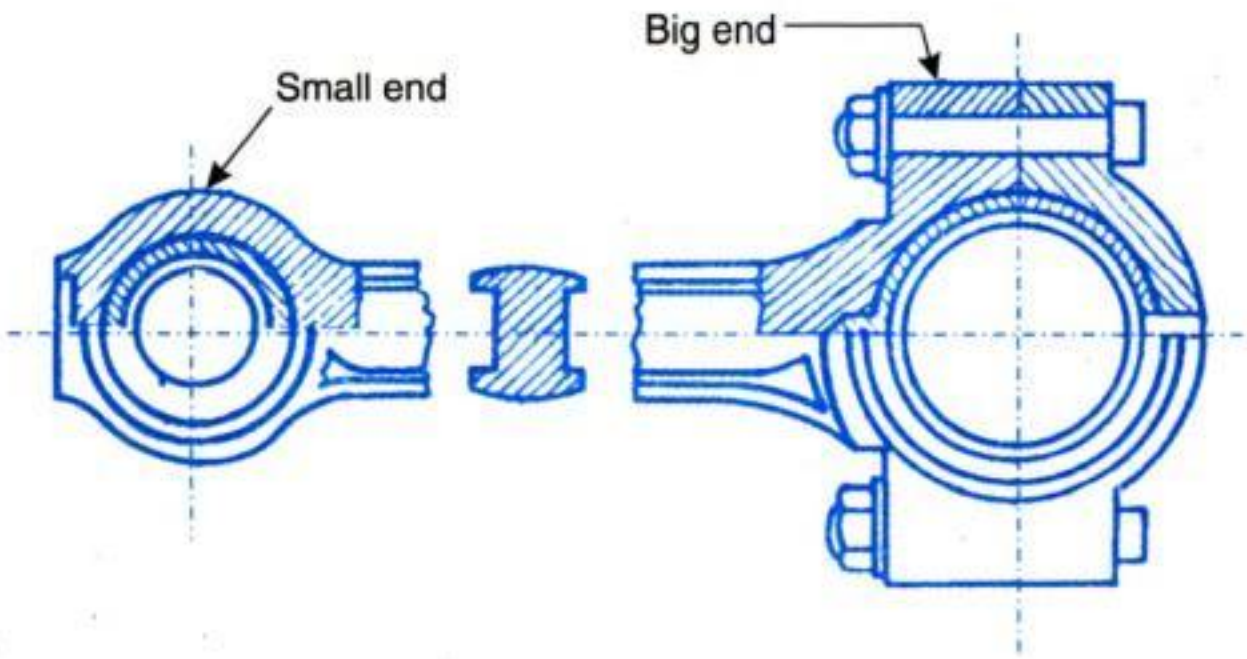

Fig. 23.11. Connecting rod.

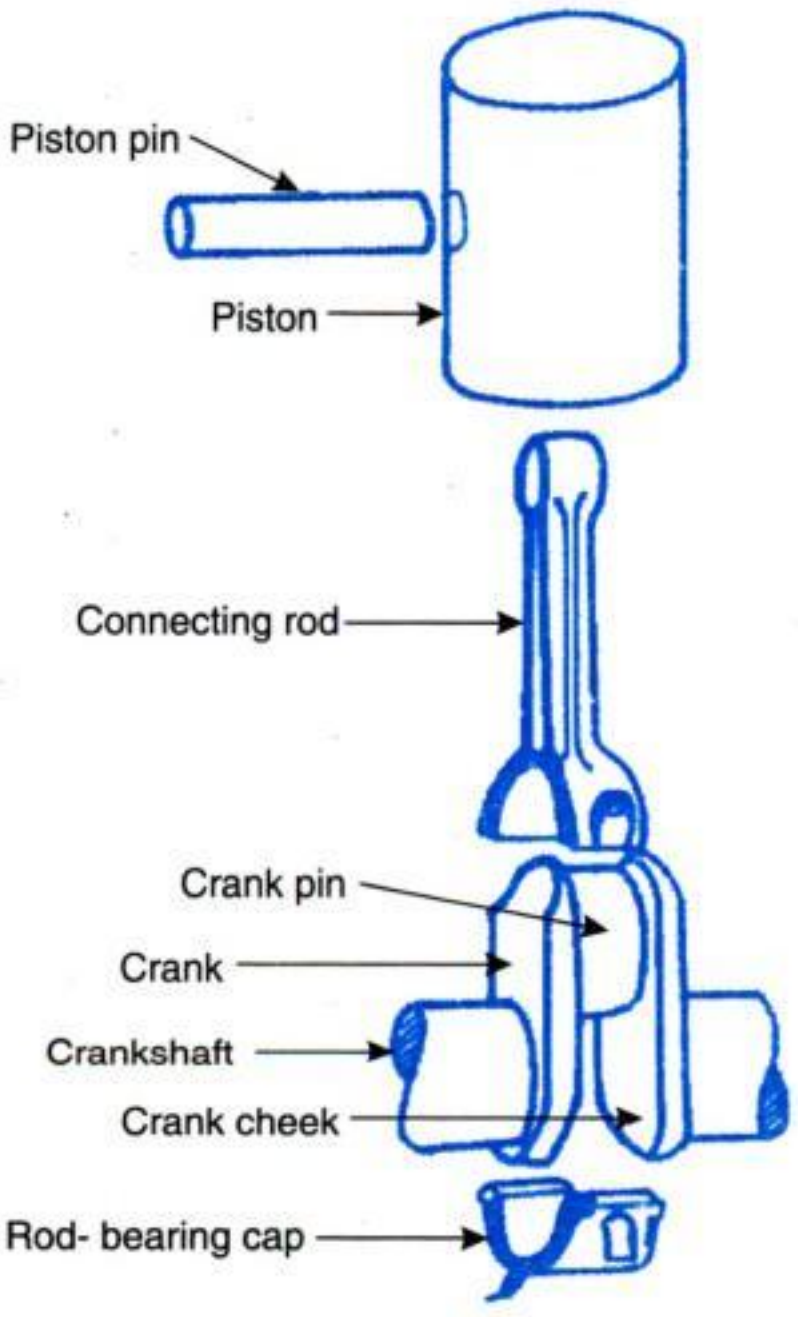

Fig. 23.12

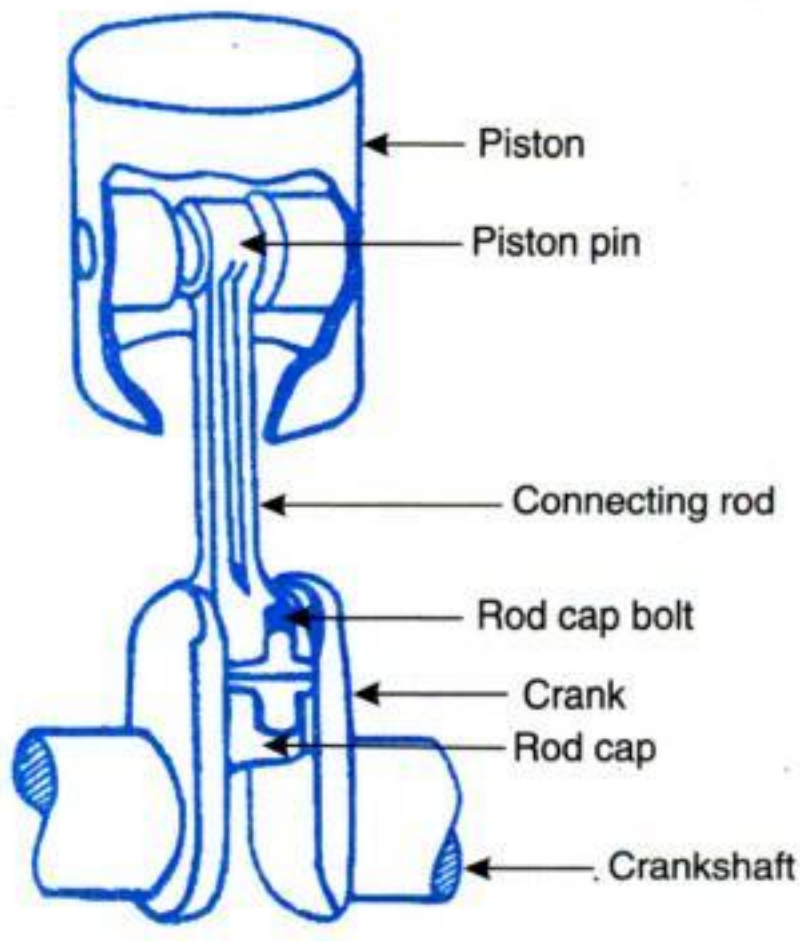

Fig. 23.13

8. **Crankshaft**

The crank is part of the crankshaft. The crankshaft of an internal combustion engine receives via its cranks the efforts supplied by the pistons to the connecting rods. All the engines auxiliary mechanisms with mechanical transmission are geared in one way or the another to the crankshaft. *It is usually a steel forging, but some makers use special types of cast iron such as*

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extension of the lower arms. For given values of weight of the ball, weight of the sleeve and height of the governor, a Proell governor runs at a *lower speed* than a Porter governor. *In order to give the same equilibrium speed a ball of smaller mass may be used in Proell governor.*

4. Hartnell governor

The Hartnell governor is a spring loaded governor in which the controlling force, to a great extent, is provided by the spring thrust.

Fig. 23.22 shows one of the types of Hartnell governors. It consists of casing fixed to the spindle. A compressed spring is placed inside the casing which presses against the top of the casing and on adjustable collars. The sleeve can move up and down on the vertical spindle depending upon the speed of the governor. Governor balls are carried on bell crank lever which are pivoted on the lower end of the casing. The balls will fly outwards or inwards as the speed of the governor shaft increases or decreases respectively.

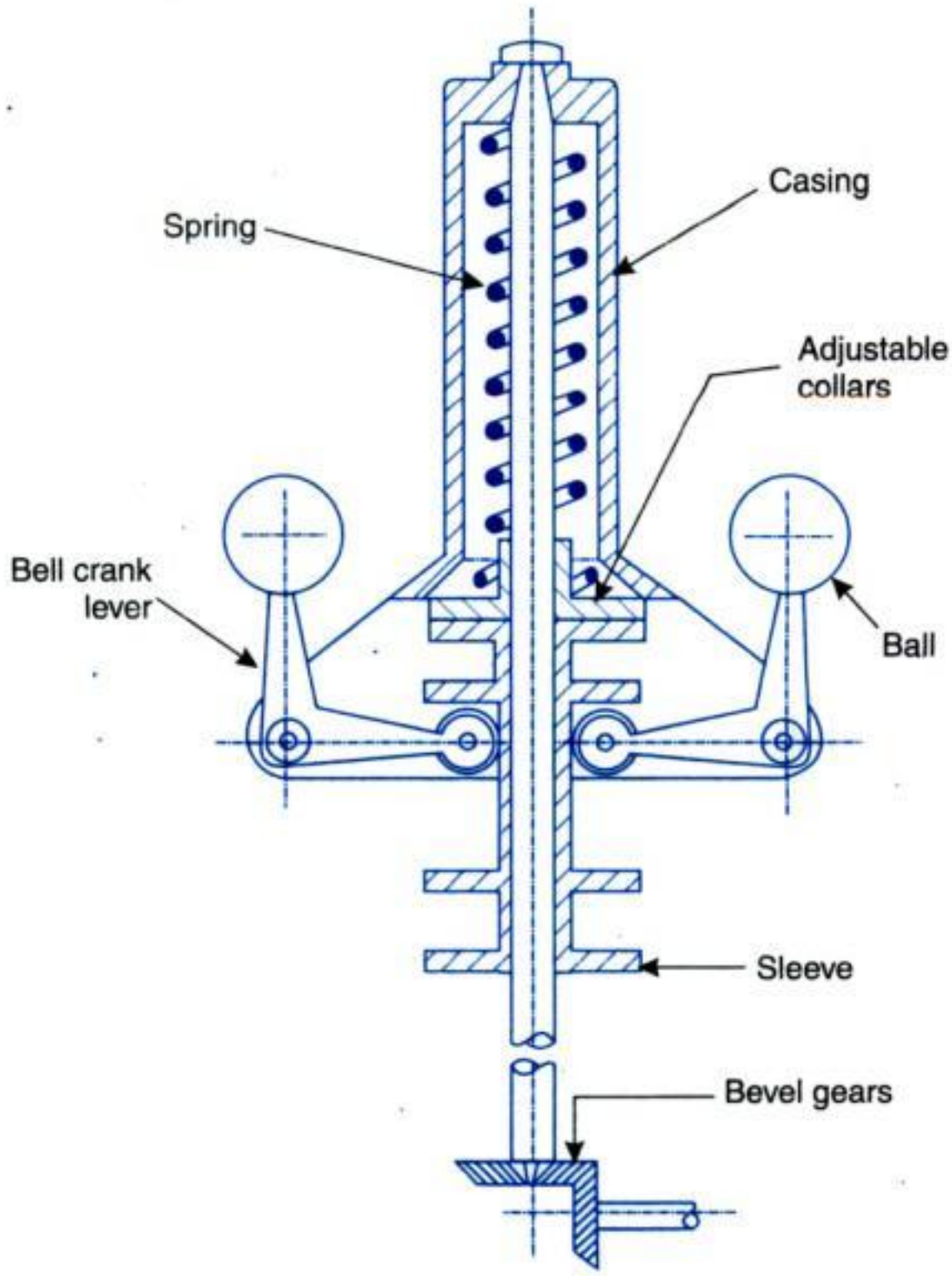

Fig. 23.22. Hartnell governor.

13. Valves and valve operating mechanism.

With few exceptions the inlet and exhaust of internal combustion engines are controlled by poppet valves. These valves are held to their seating by strong springs, and as the valves usually open inwards, the pressure in the cylinder helps to keep them closed. The valves are lifted from

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The loop which has area 4-5-1 is called *negative loop* ; it gives the pumping loss due to admission of fuel air mixture and removal of exhaust gases. The area 1-2-3-4 is the total or gross work obtained from the piston and network can be obtained by subtracting area 4-5-1 from area 1-2-3-4.

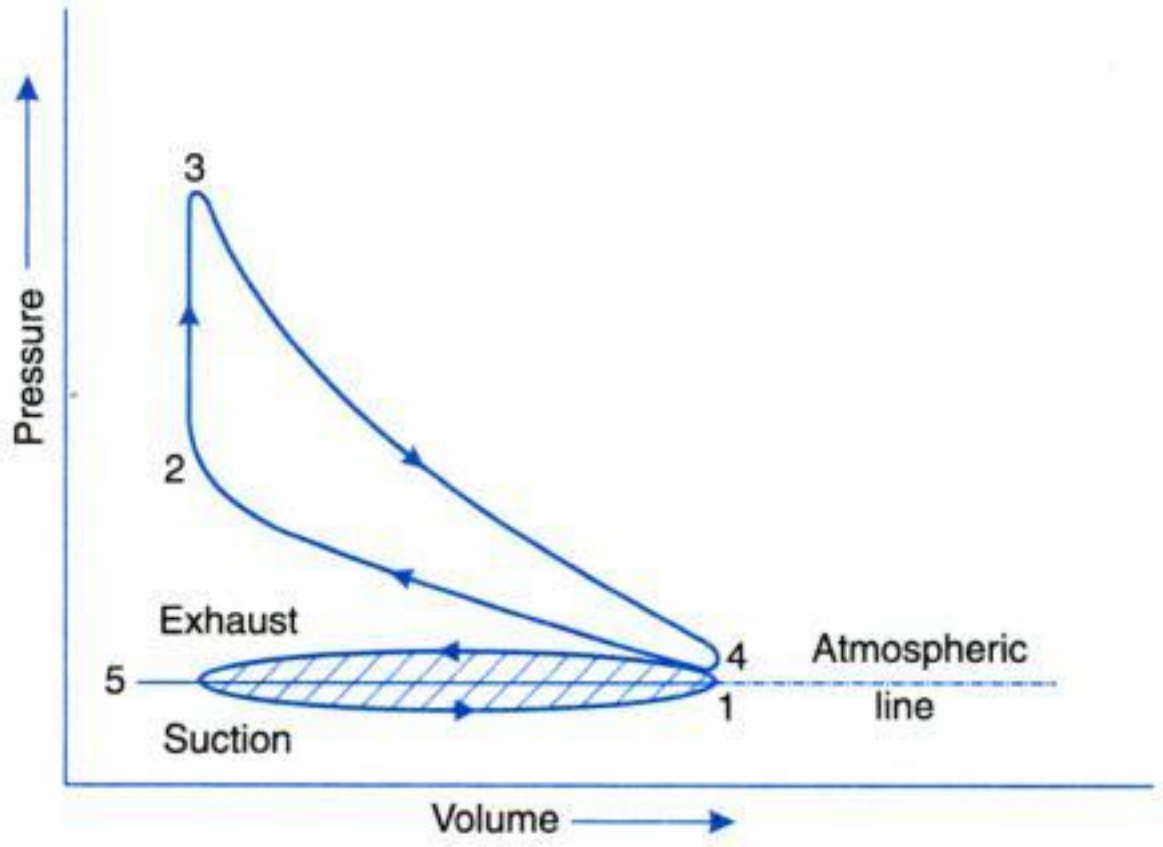

Fig. 23.33. Actual p-V diagram of a four-stroke Otto cycle engine.

Diesel engines (four-stroke cycle). As is the case of Otto four-stroke ; this cycle too is completed in *four-strokes* as follows. (Refer Fig. 23.34).

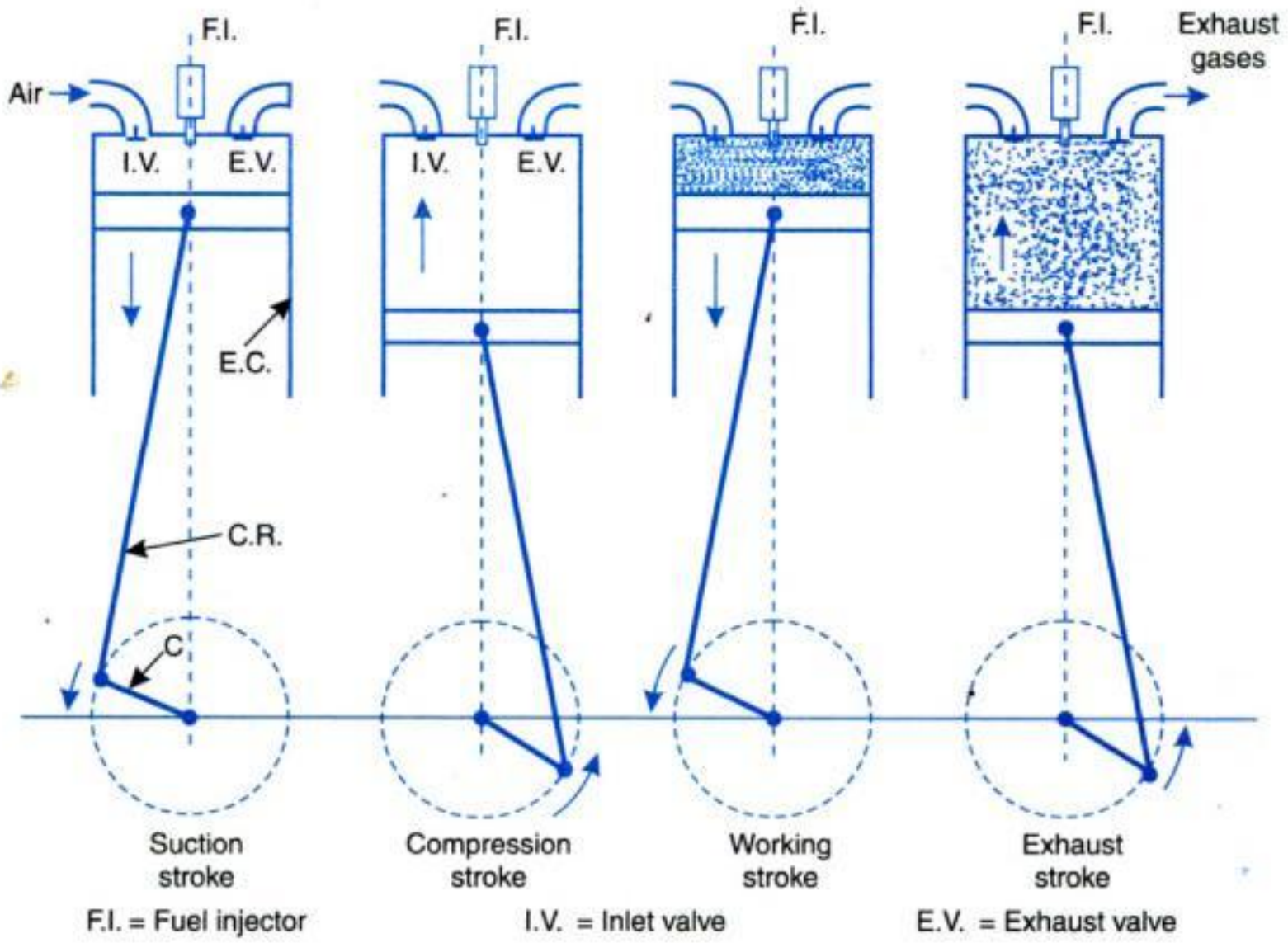

Fig. 23.34. Four-stroke Diesel cycle engine.

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1. **Ignition delay period.** The delay period is counted from the start of injection to the point where the p-θ combustion curve departs from air compression (or no ignition or motoring) curve. The delay period can be roughly sub-divided into ***physical delay*** and ***chemical delay***. *The period of physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions*. In the physical delay period, the *fuel is atomized, vaporized, mixed with air, and raised in temperature*. In the *chemical delay period reaction starts slowly and then accelerates until inflammation or ignition takes place* (it may be noted that the ignition delay in the S.I. engine is essentially equivalent to the chemical delay in the C.I. engine).

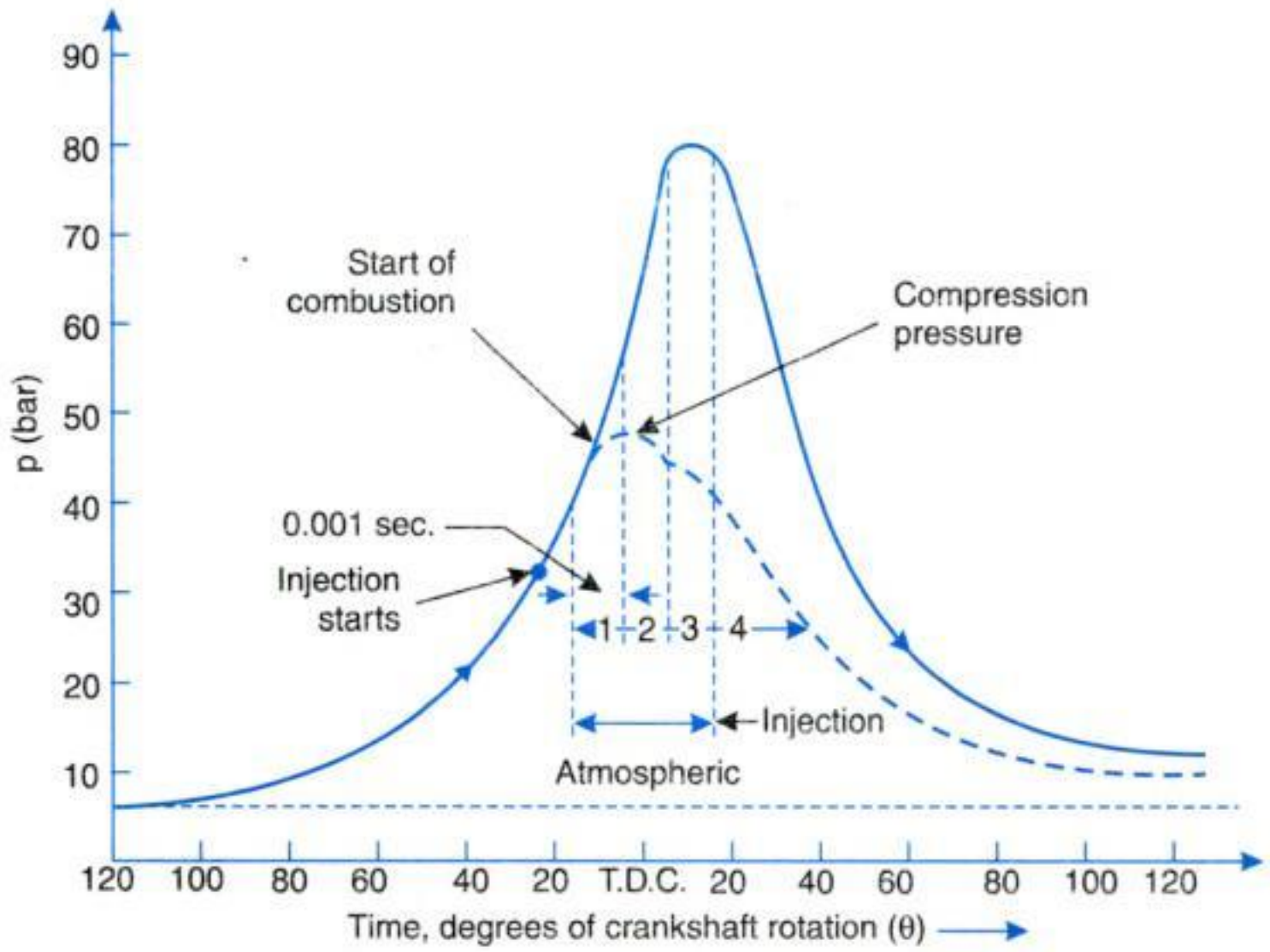

Fig. 23.63. Combustion phenomenon of C.I. engine.

The delay period exerts a great influence in the C.I. engine combustion phenomenon. It is clear that the pressure reached during the second stage will depend upon the duration of the delay period ; the longer the delay, the more rapid and higher the pressure rise, since more fuel will be present in the cylinder before the rate of burning comes under control. This causes rough running and may cause *diesel knock*. Therefore we must *aim to keep the delay period as short as possible, both for the sake of smooth running and in order to maintain control over the pressure changes. But some delay period is necessary otherwise the droplets would not be dispersed in the air for complete combustion*. However, the delay period imposed upon is greater than what is needed and the designer's efforts are to shorten it as much as possible.

2. **Period of rapid or uncontrolled combustion.** The second stage of combustion in C.I. engines, after the delay period, is the period of rapid or uncontrolled combustion. This period is counted from the end of the delay period to the point of maximum pressure on the indicator diagram. In this second stage of combustion, the rise of pressure is rapid because during the delay period the droplets of fuel have had time to spread themselves out over a wide area and they have fresh air all around them. *About one-third of heat is evolved during this process.*

The rate of pressure rise depends on the amount of fuel present at the end of delay period, degree of turbulence, fineness of atomization and spray pattern.

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23.37. SUPERCHARGING

The purpose of supercharging is to raise the volumetric efficiency above that value which can be obtained by normal aspiration.

The engine is an air pump. Increasing the air consumption permits greater quantities of fuel to be added, and results in a greater potential output. The indicated power produced is almost directly proportional to the engine air consumption. While brake power is not so closely related to air consumption, it is nevertheless, dependent upon the mass of air consumed. It is desirable, then, that the engine takes in greatest possible mass of air.

Three possible methods which might be utilized to increase the air consumption of an engine are :

1. *Increasing the piston displacement*, but this increases the size and weight of the engine, and *introduces additional cooling problems.*
2. *Running the engine at higher speeds,* which results in *increased fluid and mechanical friction losses, and imposes greater inertia stresses on engine parts.*
3. *Increasing the density of the charge,* such that a greater mass of charge is introduced into the same volume or same total piston displacement.

The last method of increasing the air capacity of an engine is widely used, and is termed *supercharging*.

The apparatus used to increase the air density is known as a *supercharger*. It is merely a *compressor* which provides a denser charge to the engine, thereby enabling the consumption of a greater mass of charge with the same total piston displacement. During the process of compressing the charge, the supercharger produces the following effects :

(*i*) *Provides better mixing of the air-fuel mixture*. The turbulent effect created by the supercharger assists in additional mixing of the fuel and air particles. The arrangement of certain types of superchargers, particularly the *centrifugal type*, also encourages more even distribution of the charge to the cylinders.

(*ii*) *The temperature of the charge is raised as it is compressed, resulting in a higher temperature within the cylinders*. This is partially beneficial in that it helps to produce better vapourisation of fuel (in case of S.I. engines) but deterimental in that it tends to lessen the density of the charge. The increase in temperature of the charge also effects the detonation of the fuel.

Supercharging tends to increase the possibility of detonation in a S.I. engine and lessen the possibility in a C.I. engine.

(*iii*) Power is required to drive the supercharger. This is usually taken from the engine and thereby removes, from over-all engine output, some of the gain in power obtained through supercharging.

Compressors used are of the following three types :

(*i*) **Positive displacement type** used with many reciprocating engines in stationary plants, vehicles and marine installations.

(*ii*) **Axial flow type** seldom used to supercharge reciprocating engines, it is widely used as the compressor unit of the *gas turbines.*

(*iii*) **Centrifugal type** *widely used* as the *supercharger for reciprocating engines, as well as compressor for gas turbines.* It is almost exclusively used as the supercharger with reciprocating power plants for *aircraft because it is relatively light and compact, and produces continuous flow rather than pulsating flow as in some positive displacement types.*

Supercharging of S.I. engines

The schematic arrangement for supercharging S.I. engine and an ideal p-V diagram for a supercharged constant volume cycle are shown in Fig. 23.66 (*a*), (*b*).

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If, W = weight at the end of the rope, N,
S = spring balance reading, N,
N = engine speed, r.p.m.,
D_b = diameter of the brake wheel, m,
d = diameter of the rope, m., and
$(D_b + d)$ = effective diameter of the brake wheel,

Then work/revolution = Torque × angle turned per revolution

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

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

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

$$\therefore \quad \text{B.P.} = \frac{(W - S)\, \pi\, (D_b + d)\, N}{60 \times 1000} \text{ kW} \qquad \text{...(23.15)}$$

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

or

$$\left(= \frac{T \times 2\pi N}{60 \times 1000} \text{ kW}\right) \qquad \text{...(23.16)}$$

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

Measurement of Indicated power (I.P.)

The power developed in the engine cylinder or at the piston is necessarily greater than that at the crankshaft due to engine losses. Thus,

I.P. = B.P. + engine losses.

Indicated power is usually determined with the help of a p-V diagram taken with the help of an indicator. In case indicated power cannot be measured directly, it is made possible by measuring the brake power and also the engine losses. If the *indicator diagram* is available, the indicated power may be *computed by measuring the area of* diagram, either with a *planimeter* or by *ordinate method*, and dividing by the stroke measurement in order to obtain the mean effective pressure (m.e.p.).

i.e., $$p_{mi} = \frac{\text{Net area of diagram in mm}^2}{\text{Length of diagram in mm}} \times \text{Spring constant}$$

where p_{mi} is in bar. ...(23.17)

(The spring constant is given in bar per mm of vertical movement of the indicator stylus.)

Engine indicators

The main types of engine indicators are :

1. *Piston indicator*
2. *Balanced diaphragm type indicator*

(*i*) The Farnborough balanced engine indicator
(*ii*) Dickinson-Newell indicator
(*iii*) MIT balanced pressure indicator
(*iv*) Capacitance-type balance pressure indicator.

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23.41. THE WANKEL ROTARY COMBUSTION (RC) ENGINE

Introduction

- Scheffel in 1952, to get a patent for *rotary engine*, utilised the *principle that oval or elliptical rotors can be designed to maintain contact ; while turning about fixed centres, and that three or more rotors can be run to enclose between them a continuously varying volume*. The four volumes between the rotors with the suitable arrangement of ports, ignition system and adequate compression ratio, could be made to execute a *four-phase Otto cycle*. However, this design failed due to its complexity and great difficulties/problems involved in its manufacture.
- Felix Wankel (German inventor), in 1954, got a patent for design of four-phase rotary engine working on the Otto cycle principle.
- Later Dr. Frocede made certain modifications and an engine was developed, called as KKM (Kreiskolbenmotor) : now popularly known as Wankel Rotating Combustion (RC) engine.

Construction and Working

Refer Fig. 23.75

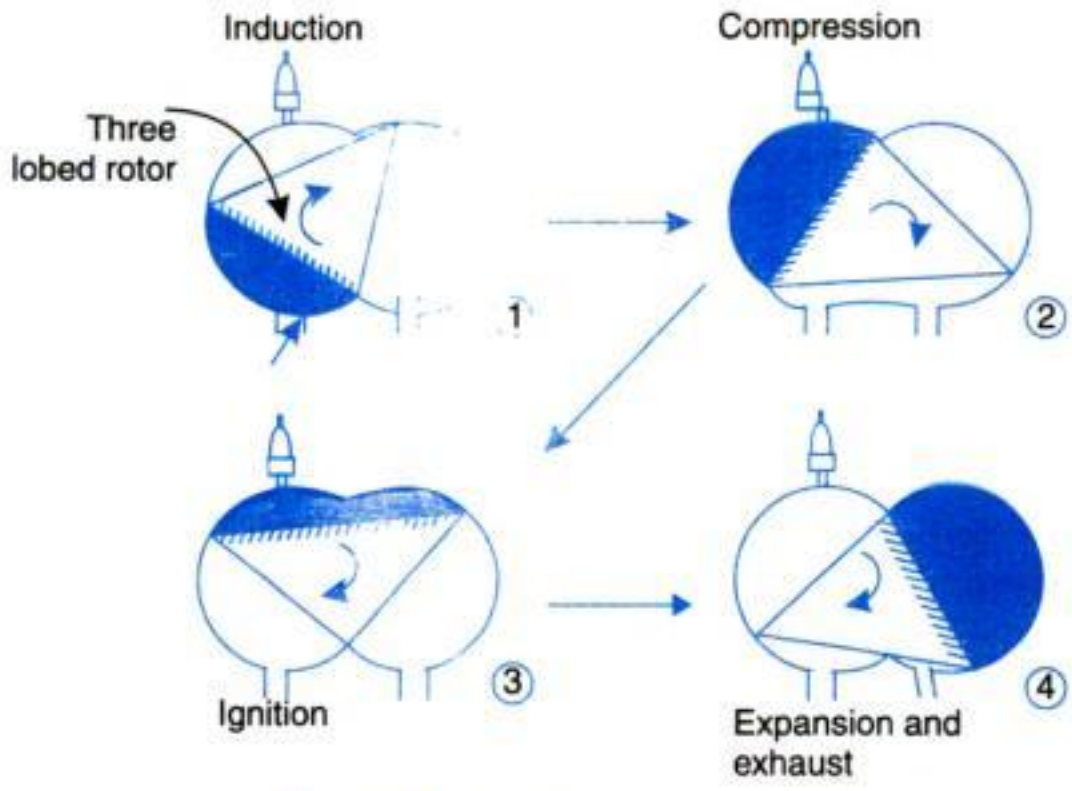

Fig. 23.75. The Wankel engine.

Construction : It consists of the following parts :

(*i*) ***Rotor*** *(*three lobed*)*.

(*ii*) ***Eccentric or output shaft*** with its integral eccentric. No connecting rod is required as the rotor rotates directly on the eccentric shaft. The *output torque is transmitted to the shaft through eccentric.*

(*iii*) ***Internal and external timing gears.*** They maintain the phase relationship between the rotation of the rotor and the eccentric shaft and eventually *control the orbital motion of the motor.*

Working :

- The Wankel engine works on the *four-phase principle.* (The word phase corresponds to stroke of the reciprocating engine).
- The engine having three lobed rotor is driven eccentrically in a casing in such a way that there are *three separate volumes trapped between the rotor and the casing.*

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Solution. Number of cylinders, $n = 4$
Power developed, $P = 14.7$ kW
Engine speed, $N = 1000$ r.p.m.
Indicated mean effective pressure, $p_{mi} = 5.5$ bar
Length of stroke, $L = 1.5\ D$ (bore)

For four stroke cycle, $k = \frac{1}{2}$

$\boldsymbol{L} = ?, \boldsymbol{D} = ?$

Indicated power developed, $\text{I.P.} = \frac{np_{mi}\,LANk \times 10}{6}$ kW

$$14.7 = \frac{4 \times 5.5 \times 1.5D \times \pi / 4D^2 \times 1000 \times \frac{1}{2} \times 10}{6}$$

$$\therefore \quad D^3 = \frac{14.7 \times 6 \times 4 \times 2}{4 \times 5.5 \times 1.5 \times \pi \times 1000 \times 10} = 0.0006806$$

$\therefore$ $D =$ **0.0879 or 87.9 mm.** **(Ans.)**

and $L =$ **1.5 × 87.9 = 131.8 mm.** **(Ans.)**

Example 23.7. *A single-cylinder, four-stroke cycle oil engine is fitted with a rope brake. The diameter of the brake wheel is 600 mm and the rope diameter is 26 mm. The dead load on the brake is 200 N and the spring balance reads 30 N. If the engine runs at 450 r.p.m., what will be the brake power of the engine ?*

Solution. Diameter of the brake wheel, $D_b = 600$ mm $= 0.6$ m
Rope diameter, $d = 26$ mm $= 0.026$ m
Dead load on the brake, $W = 200$ N
Spring balance reading, $S = 30$ N
Engine speed, $N = 450$ r.p.m.

Brake power, B.P. :

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

$$= \frac{(200 - 30)\pi(0.6 + 0.026) \times 450}{60 \times 1000} = \textbf{2.5 kW.} \quad \textbf{(Ans.)}$$

Example 23.8. *A four-cylinder four-stroke, spark-ignition engine develops a maximum brake torque of 160 Nm at 3000 r.p.m. Calculate the engine displacement, bore and stroke. The brake mean effective pressure at the maximum engine torque point is 960 kPa. Assume bore is equal to stroke.* **(GATE)**

Solution. *Given* $n = 4$; $k = \frac{1}{2}$ (engine being 4-stroke cycle), $T_b = 160$ Nm,

$N = 3000$ r.p.m., $p_{mb} = 960$ kPa $= 960 \times 10^3$ N/m^2 $= 9.6$ bar ; D = L

D, L, displacement :

$$\text{Power developed} = \frac{2\pi NT_b}{60 \times 1000} \text{ kW} = \frac{p_m LANk \times 10}{6} \text{ kW}$$

i.e., $$\frac{2\pi NT_b}{60 \times 1000} = \frac{p_{mb} LANk \times 10}{6}$$

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Indicated power. I.P. :

Speed of the car $= 48 \text{ km/h} = \dfrac{48 \times 1000}{60} = 800 \text{ m/min.}$

In N_t are the revolutions made by the tyre per minute, then $\pi D N_t = 800$

$$\therefore \quad N_t = \frac{800}{\pi \times 0.650} = 392 \text{ r.p.m.}$$

As the rear axle ratio is 39 : 8,

$$\therefore N_e \text{ (speed of the engine shaft)} = \frac{392 \times 39}{8} = 1911 \text{ r.p.m.}$$

$$\text{I.P.} = \frac{n p_{mi} LANk \times 10}{6} \text{ kW}$$

$$= \frac{4 \times 5.625 \times 0.09 \times \pi / 4 \times 0.075^2 \times 1911 \times \frac{1}{2} \times 10}{6}$$

$$= \mathbf{14.25 \text{ kW.}} \quad \text{(Ans.)}$$

Indicated thermal efficiency :

To find indicated thermal efficiency, let us find $\dot{m}_f$ first :

Speed of the car $= \dfrac{48}{60} = 0.8 \text{ km/min.}$

Time for covering 3.2 km $= \dfrac{3.2}{0.8} = 4 \text{ min.}$

Amount of fuel consumed in 4 min. = 0.227 kg

$$\therefore \text{ Fuel consumed/sec} = \frac{0.227}{4 \times 60} = 0.000946 \text{ kg/s}$$

Now, *Indicated thermal efficiency*,

$$\eta_{th(I)} = \frac{\text{I.P.}}{\dot{m}_f \times C} = \frac{14.25}{0.000946 \times 43470} = \mathbf{0.346} \text{ or } \mathbf{34.6\%.} \quad \text{(Ans.)}$$

Example 23.26. *The following readings were taken during the test of a single-cylinder four stroke oil engine :*

Cylinder diameter	*= 250 mm*
Stroke length	*= 400 mm*
Gross m.e.p.	*= 7 bar*
Pumping m.e.p.	*= 0.5 bar*
Engine speed	*= 250 r.p.m.*
Net load on the brake	*= 1080 N*
Effective diameter of the brake	*= 1.5 metres*
Fuel used per hour	*= 10 kg*
Calorific value of fuel	*= 44300 kJ/kg*

Calculate : (i) Indicated power ; (ii) Brake power ;
(iii) Mechanical efficiency ; (iv) Indicated thermal efficiency.

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$$\eta_{\text{air-standard}} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(6.5)^{1.4-1}} = 0.527 \text{ or } 52.7\%$$

$$\therefore \quad \eta_{\text{relative}} = \frac{0.23}{0.527} = \mathbf{0.436} \text{ or } \mathbf{43.6\%}. \quad \textbf{(Ans.)}$$

Example 23.30. *The compression curve on the indicator diagram for a gas engine follows the law $pV^{1.3}$ = constant. At two points on the curve at $\frac{1}{4}$ stroke and $\frac{3}{4}$ stroke the pressures are 1.4 bar and 3.6 bar respectively. Determine the compression ratio of the engine. Calculate the thermal efficiency and the gas consumption per I.P. hour, if the relative efficiency is 0.4 and the gas has the calorific value of 18800 kJ/m³.*

Solution. Refer Fig. 23.79.

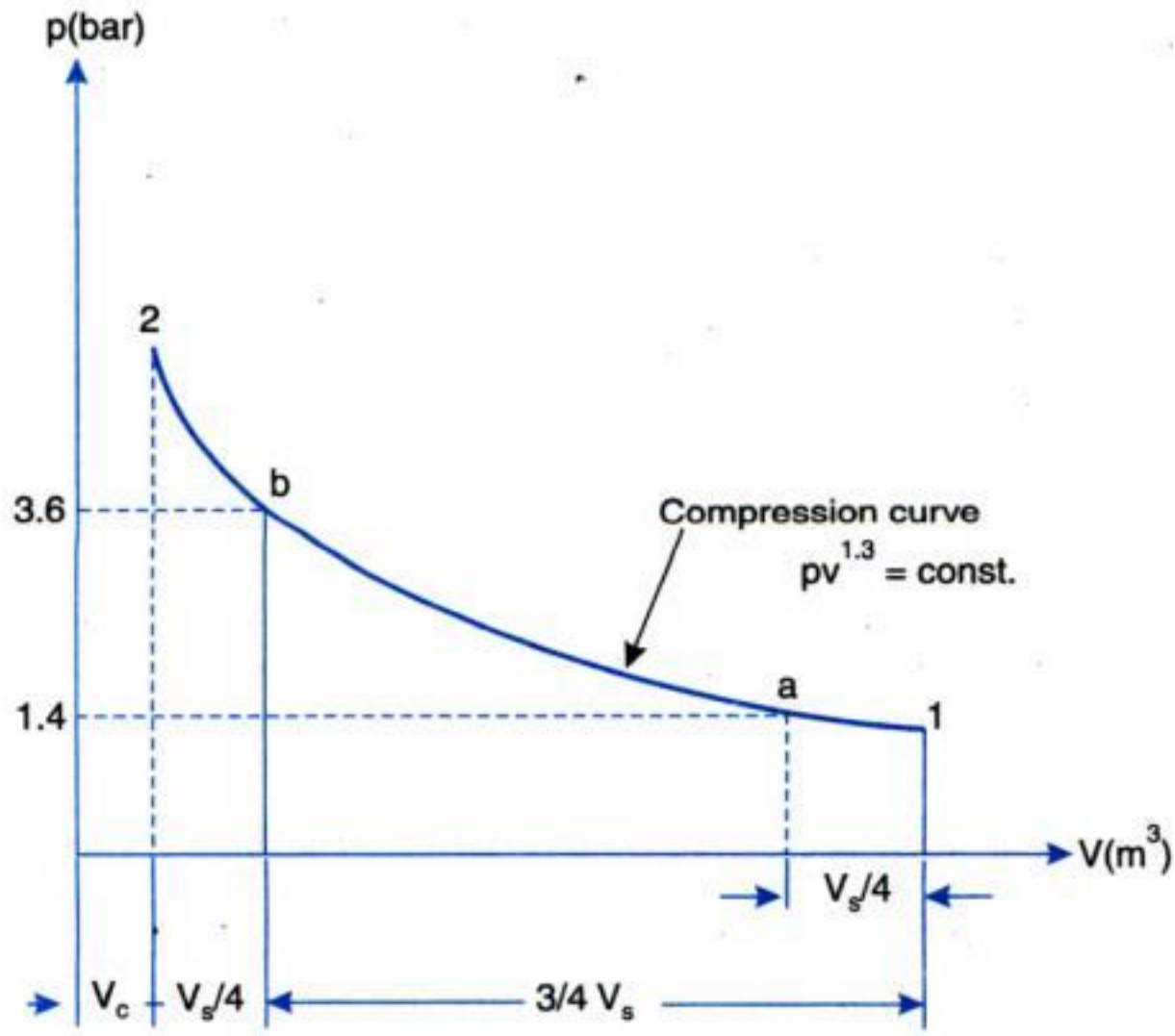

Fig. 23.79

Compression law, $pV^{1.3} = \text{constant}$

Pressure at '*a*', $p_a = 1.4$ bar

Pressure at '*b*', $p_b = 3.6$ bar

Volume at '*a*', $V_a = V_c + 0.75V_s$

Volume at '*b*', $V_b = V_c + 0.25V_s$

Also, $p_a V_a^{1.3} = p_b V_b^{1.3}$

or $$\frac{V_a}{V_b} = \left(\frac{p_b}{p_a}\right)^{1/1.3} = \left(\frac{3.6}{1.4}\right)^{1/1.3} = 2.067$$

Also, $$\frac{V_a}{V_b} = \frac{V_c + 0.75V_s}{V_c + 0.25V_s} = 2.067$$

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∴ Volume of air-fuel mixture supplied to the engine per sec.

$$\frac{mRT}{P} = \frac{0.0906 \times 287 \times (25 + 273)}{1 \times 10^5} = 0.07748 \text{ m}^3/\text{s}$$

$$\eta_{vol.} = \frac{\text{Mass of mixture supplied /. sec.}}{\text{Swept volume}}$$

$$0.78 = \frac{0.07748}{\text{Swept volume}}$$

∴ Swept volume $= \frac{0.07748}{0.78} = 0.0993 \text{ m}^3/\text{s}$

But swept volume/sec. $= \left(\frac{\pi}{4} D^2 \times L\right) \times \text{no. of cylinders} \times \frac{\text{r.p.m.}}{2 \times 60}$

$$= \frac{\pi}{4} \times D^2 \times 1.25D \times 4 \times \frac{2400}{2 \times 60} = 0.0993$$

$$\therefore \quad D^3 = \frac{0.0993 \times 4 \times 2 \times 60}{\pi \times 1.25 \times 4 \times 2400} = 0.001264$$

∴ $D = \mathbf{0.108\ m}$ or **108 mm.** (Ans.)

and $L = 108 \times 1.25 = \mathbf{135\ mm.}$ (Ans.)

☞ **Example 23.26.** *A single-cylinder four-stroke gas engine has a bore of 180 mm and stroke of 340 mm and is governed on hit-and-miss principle. When running at 400 r.p.m. at full load, indicator cards are taken which give a working loop mean effective pressure of 6.4 bar, and a pumping loop mean effective pressure of 0.36 bar. Diagrams from the dead cycle give a mean effective pressure of 0.64 bar. The engine was run light at the same speed (i.e., with no load), and a mechanical counter recorded 46 firing strokes per minute.*

Calculate : (i) Full load brake power.

(ii) Mechanical efficiency of the engine. **(P.U.)**

Solution. Number of cylinders, $n = 1$

Bore, $D = 180 \text{ mm} = 0.18 \text{ m}$

Stroke, $L = 340 \text{ mm} = 0.34 \text{ m}$

Engine speed, $N = 400$ r.p.m.

Working loop mean effective pressure $= 6.4$ bar

Pumping loop mean effective pressure $= 0.36$ bar

Mean effective pressure (dead cycle) $= 0.64$ bar

Firing strokes/min. $= 46$

Refer Fig. 23.84.

(*i*) **Full load brake power, B.P. :**

Net indicated mean effective pressure,

$p_{mi\ (net)}$ = Working (or power) loop mean effective pressure − pumping loop mean effective pressure

$= 6.4 - 0.36 = 6.04$ bar

Also, working cycles/min $= 46$

and, dead cycles/min $= \left(\frac{400}{2} - 46\right) = 154$

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$$28 = \frac{1 \times p_{mb} \times 0.00178 \times 2500 \times \frac{1}{2} \times 10}{6} \qquad [\because \quad LA = 0.00178 \text{ m}^3]$$

$$\therefore \qquad p_{mb} = \frac{28 \times 6 \times 2}{0.00178 \times 2500 \times 10} = \textbf{7.55 bar.} \quad \textbf{(Ans.)}$$

(*iv*) **Relative efficiency :**

$$\eta_{\text{air-standard}} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(6.5)^{1.4-1}} = 0.527 \text{ or } 52.7\%$$

Brake thermal efficiency,

$$\eta_{\text{th.(B)}} = \frac{\text{B.P.}}{\dot{m}_f \times C} = \frac{28}{\frac{0.135}{60} \times 43900} = 0.2835 \text{ or } 28.35\%$$

$$\therefore \qquad \eta_{\text{relative}} = \frac{\eta_{\text{thermal (B)}}}{\eta_{\text{air-standard}}} = \frac{0.2835}{0.527} = \textbf{0.5379 or 53.79\%.} \quad \textbf{(Ans.)}$$

Example 23.38. *A six-cylinder, four-stroke CI engine is tested against a water brake dynamometer for which B.P. = $WN/17 \times 10^3$ in kW, where W is the brake load in newton and N is the speed of the engine in the r.p.m. The air consumption was measured by means of a sharp edged orifice. During the test following observations were taken :*

Bore	*= 10 cm*
Stroke	*= 14 cm*
Speed	*= 2500 r.p.m.*
Brake load	*= 480 N*
Barometer reading	*= 76 cm of Hg*
Orifice diameter	*= 3.3 cm*
Co-efficient of discharge of orifice	*= 0.62*
Pressure drop across orifice	*= 14 cm of Hg*
Room temperature	*= 25°C*
Fuel consumption	*= 0.32 kg/min.*

Calculate the following :

(i) The volumetric efficiency ; (ii) The brake mean effective pressure (b.m.e.p.) ; (iii) The engine torque ; (iv) The brake specific fuel consumption (b.s.f.c.).

Solution. (*i*) **Volumetric efficiency,** η_{vol} :

V_s = Swept volume,

$$= \frac{\pi}{4} D^2 L \times \frac{N}{60 \times 2} \times \text{No. of cylinders, for 4-stroke.} \quad (\text{where } N = \text{r.p.m.})$$

$$= \frac{\pi}{4} (0.1)^2 \times 0.14 \times \frac{2500}{60 \times 2} \times 6 = 0.137 \text{ m}^3/\text{s}.$$

Barometer $= 76 \text{ cm Hg} = \left[\frac{76}{100} \times 13.6 \times 10^3 \times 9.81\right] \times 10^{-3} = 101.4 \text{ kN/m}^2$

$$\rho_a = \frac{p}{R_a T} = \frac{101.3}{0.287\,(273 + 25)} = 1.1844 \text{ kg/m}^3$$

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Solution. Number of cylinders, $n = 4$

Bore, $D = 60 \text{ mm} = 0.06 \text{ m}$

Stroke, $L = 90 \text{ mm} = 0.09 \text{ m}$

Speed, $N = 2800 \text{ r.p.m.}$

Torque arm $= 0.37 \text{ m}$

Not brake lead $= 160 \text{ N}$

Specific gravity of petrol $= 0.74$

Fuel consumption $= 8.986 \text{ litres/h}$

$= 8.986 \times 1 \times 0.74 \text{ kg/h}$

Calorific value $= 44100 \text{ kJ/kg}$

(*i*) **Engine torque, T :**

Engine torque, $T = \text{Net brake load} \times \text{Torque arm}$

$= 160 \times 0.37 = \mathbf{59.2 \text{ Nm.}}$ **(Ans.)**

Brake power, $\text{B.P.} = \dfrac{2\pi NT}{60 \times 1000} = \dfrac{2\pi \times 2800 \times 59.2}{60 \times 1000} = 17.36 \text{ kW}$

(*ii*) **Brake mean effective pressure, p_{mb} :**

$$\text{B.P.} = \frac{np_{mb}LANk \times 10}{6}$$

$$17.36 = \frac{4 \times p_{mb} \times 0.09 \times \dfrac{\pi}{4} \times (0.06)^2 \times 2800 \times \dfrac{1}{2} \times 10}{6}$$

$$\therefore \quad p_{mb} = \frac{17.36 \times 6 \times 4 \times 2}{4 \times 0.09 \times \pi \times (0.06)^2 \times 2800 \times 10} = \mathbf{7.31 \text{ bar.}} \text{ (Ans.)}$$

(*iii*) **Brake thermal efficiency, $\eta_{th\ (B)}$:**

$$\eta_{th(B).} = \frac{\text{B.P.}}{\dot{m}_f \times C} = \frac{17.36}{\dfrac{(8.986 \times 1 \times 0.74)}{3600} \times 44100} = \mathbf{0.213} \text{ or } \mathbf{21.3\%.} \text{ (Ans.)}$$

(*iv*) **Specific fuel consumption, s.f.c. :**

$$\text{s.f.c.} = \frac{m_f}{\text{B.P.}} = \frac{6.65}{17.36} = \mathbf{0.383 \text{ kg/kWh.}} \text{ (Ans.)}$$

(*v*) **Mechanical efficiency, $\eta_{mech.}$:**

Since the speed is constant, substituting the brake loads instead of the values of B.P. as follows :

$$IP_1 = BP - BP_1 = 160 - 110 = 50 \text{ N}$$
$$IP_2 = BP - BP_2 = 160 - 107 = 53 \text{ N}$$
$$IP_3 = BP - BP_3 = 160 - 104 = 56 \text{ N}$$
$$IP_4 = BP - BP_4 = 160 - 110 = 50 \text{ N}$$

Hence for the engine, the indicated load is given by

$$IP = IP_1 + IP_2 + IP_3 + IP_4 = 50 + 53 + 56 + 50 = 209 \text{ N}$$

$$\therefore \quad \eta_{mech.} = \frac{BP}{IP} = \frac{160}{209} = \mathbf{0.765} \text{ or } \mathbf{76.5\%.} \text{ (Ans.)}$$

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37. Octane number of the fuel used commercially for diesel engine in India is in the range
(*a*) 80 to 90 (*b*) 60 to 80
(*c*) 60 to 70 (*d*) 40 to 45.

38. The knocking tendency in C.I. engines increases with
(*a*) decrease of compression ratio (*b*) increase of compression ratio
(*c*) increasing the temperature of inlet air (*d*) increasing cooling water temperature.

39. Desirable characteristic of combustion chamber for S.I. engines to avoid knock is
(*a*) small bore (*b*) short ratio of flame path to bore
(*c*) absence of hot surfaces in the end region of gas (*d*) all of the above.

ANSWERS

1. (*c*)	**2.** (*d*)	**3.** (*b*)	**4.** (*b*)	**5.** (*a*)	**6.** (*c*)	**7.** (*a*)
8. (*c*)	**9.** (*b*)	**10.** (*a*)	**11.** (*a*)	**12.** (*a*)	**13.** (*b*)	**14.** (*c*)
15. (*c*)	**16.** (*a*)	**17.** (*b*)	**18.** (*a*)	**19.** (*a*)	**20.** (*c*)	**21.** (*b*)
22. (*d*)	**23.** (*b*)	**24.** (*a*)	**25.** (*b*)	**26.** (*c*)	**27.** (*d*)	**28.** (*c*)
29. (*a*)	**30.** (*b*)	**31.** (*d*)	**32.** (*b*)	**33.** (*a*)	**34.** (*a*)	**35.** (*b*)
36. (*c*)	**37.** (*d*)	**38.** (*a*)	**39.** (*d*).			

THEORETICAL QUESTIONS

1. Name the two general classes of combustion engines and state how do they basically differ in principle ?

2. Discuss the relative advantages and disadvantages of internal combustion and external combustion engines.

3. What are the two basic types of internal combustion engines ? What are the fundamental differences between the two ?

4. What is the function of a governor ? Enumerate the types of governors and discuss with a neat sketch the Porter governor.

5. Differentiate between a flywheel and a governor.

6. (*a*) State the function of a carburettor in a petrol engine.
(*b*) Describe a simple carburettor with a neat sketch and also state its limitations.

7. Explain with neat sketches the construction and working of the following :
(*i*) Fuel pump (*ii*) Injector.

8. Explain the following terms as applied to I.C. engines :
Bore, stroke, T.D.C., B.D.C., clearance volume, swept volume, compression ratio and piston speed.

9. Explain with suitable sketches the working of a four-stroke otto engine.

10. Discuss the difference between ideal and actual valve timing diagrams of a petrol engine.

11. In what respects four-stroke diesel cycle (compression ignition) engine differs from four-stroke cycle spark ignition engine ?

12. Discuss the difference between theoretical and actual valve timing diagrams of a diesel engine.

13. What promotes the development of two-stroke engines ? What are the two main types of two-stroke engines.

14. Describe with a suitable sketch the two-stroke cycle spark ignition (SI) engine. How its indicator diagram differs from that of four-stroke cycle engine ?

15. Compare the relative advantages and disadvantages of four-stroke and two-stroke cycle engines.

16. Discuss with suitable sketches the following ignition systems used in petrol engines :
(*a*) Coil or battery-ignition system
(*b*) Magneto-ignition system.

17. State the relative advantages and disadvantages of battery and magneto-ignition systems.

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24 Air Compressors

24.1. General aspects. 24.2. Classification of air compressors. 24.3. **Reciprocating compressors**—Construction and working of a reciprocating compressor (single stage)—Single-stage compressor—equation for work (neglecting clearance)—Equation for work (with clearance volume)—Volumetric efficiency—Actual p-V diagram for single-stage compressor— Multistage compression—Efficiency of compressor—How to increase isothermal efficiency ?—Clearance in compressors—Effect of clearance volume—Free air delivered and displacement—Compressor performance—Effect of atmospheric conditions on the output of a compressor—Control of compressors—Arrangement of reciprocating compressors—Intercooler—Compressed air motors—Reciprocating air motor—Rotary type air motor. 24.4. **Rotary compressors**—Classification—Displacement compressors—Steady flow compressors. 24.5. Comparison between reciprocating and centrifugal compressors. 24.6. Comparison between reciprocating and rotary air compressors. 24.7. Comparison between centrifugal and axial flow compressors—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

24.1. GENERAL ASPECTS

The compressed air finds application in the following *fields* :

1. It is widely employed for powering small engines, generally those of portable nature. Compressed air is used in such diversified fields as :

(*i*) operating tools in factories ;

(*ii*) operating drills and hammers in road building ;

(*iii*) excavating ;

(*iv*) tunneling and mining ;

(*v*) starting diesel engines ; and

(*vi*) operating brakes on buses, trucks and trains.

2. A large quantity of air at moderate pressure is used in smelting of various metals such as melting iron, in blowing converters, and cupola work.

3. Large quantities of air are used in the air-conditioning, drying, and ventilation fields. In many of these cases, there is little resistance to the flow of air ; and hence it does not have to be compressed (*i.e.,* measurably decreased in volume). For such cases fans serve the purpose of moving the air to the desired location. In other cases, particularly in drying work, there is appreciable resistance to the flow of air and a compressor of some sort is required to build up sufficient pressure to overcome the resistance to flow.

The function of a compressor is to take a definite quantity of fluid (usually gas, and most often air) *and deliver it at a required pressure.*

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$$W = \frac{n}{n-1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} \quad ...(24.4)$$

$$= \frac{n}{n-1} mRT_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} \quad ...(24.5)$$

If the air delivery temperature T_2 is required then this can be obtained by using this equation :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \quad \text{or} \quad T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \quad ...(24.6)$$

24.3.3. Equation for Work (with clearance volume)

In practice, all reciprocating compressors will have a *clearance volume.* The clearance volume is *that volume which remains in the cylinder after the piston has reached the end of its inward stroke.*

Refer Fig. 24.4. At point 1, the cylinder is full of intake air, volume V_1 and the piston is about to commence its compression stroke. The air is compressed polytropically according to some law $pV^n = C$ to delivery pressure p_2 and volume V_2. At 2 the delivery valve theoretically opens and for the remainder of the stroke, 2 to 3, the compressed air is delivered from the cylinder. At 3 the piston has reached the end of its inward stroke and so on, delivery of compressed air ceases at 3. V_3

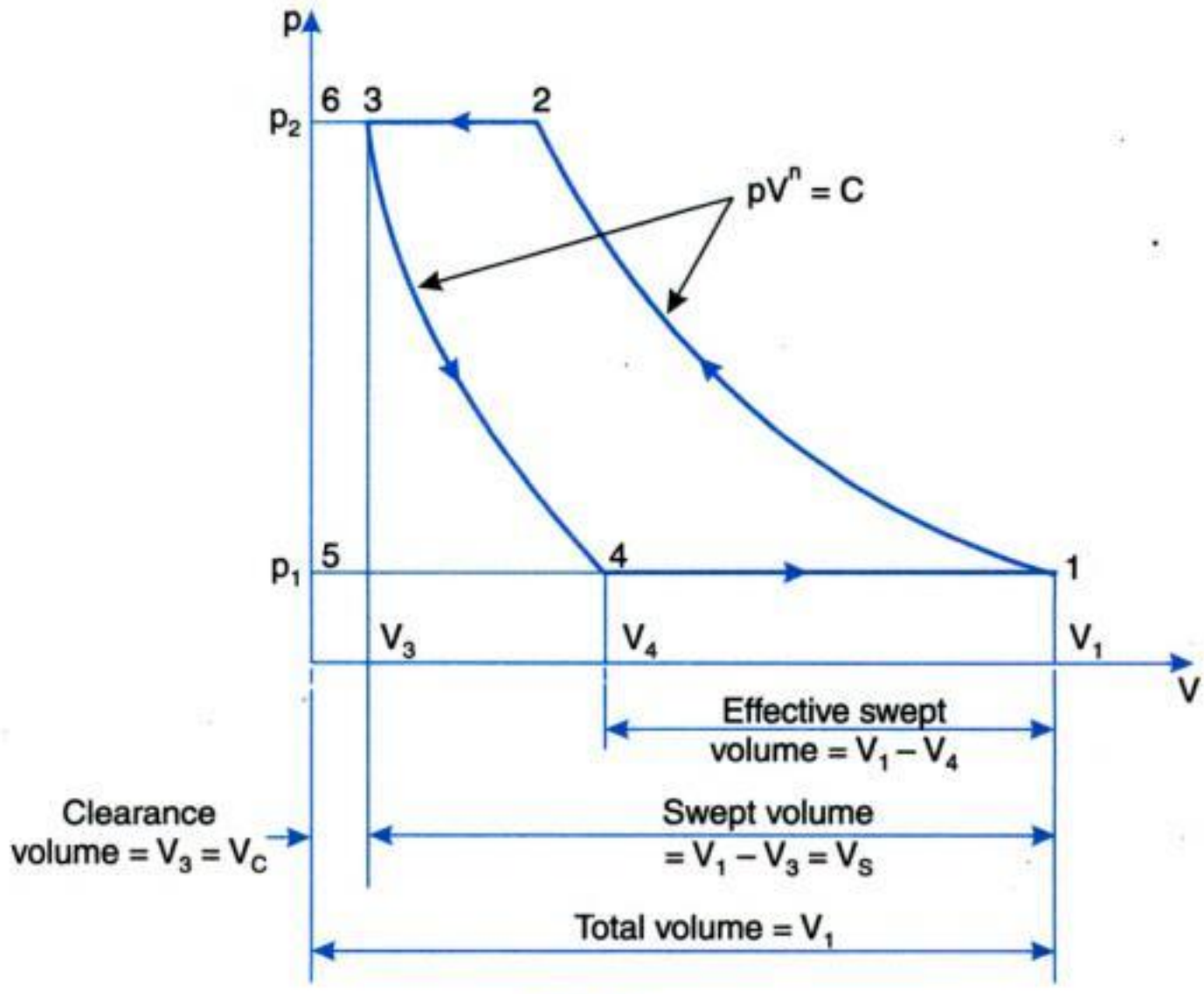

Fig. 24.4

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The direct effect of clearance is to make the volume taken in per stroke less than the swept volume, and because of the necessary increase in size of the compressor (to maintain the output) the power to drive the compressor is slightly increased. *The maximum compression pressure is also controlled by the clearance volume.*

The *value of clearance* may be expressed as follows :

(*i*) Since less precision is required in machining and erection, a large clearance cheapens a compressor and tends to increase its reliability.

(*ii*) A variable clearance is a convenient and safe way of controlling the output of a constant speed compressor.

(*iii*) Increasing the clearance in one stage throws more work on the stage below. In this way the temperature rise in the higher stages, consequent on controlling the output by throttling the L.P. suction, may be limited.

24.3.10. Effect of Clearance Volume

The ***clearance volume*** is *the volume within the cylinder when the piston it at the end of its inward travel plus the volume within the passages leading to the valves. The effect of clearance volume is to reduce the volume actually aspirated.* Therefore clearance volume should be as small as possible, but it cannot be reduced to zero since, for mechanical reasons, the piston face cannot be allowed to come into contact with cylinder head. The clearance volume of the compressor is given as *a percentage of stroke volume*.

The *p-V* diagram for a single-stage and a single-acting air-compressor with clearance volume is shown in Fig. 24.11. At the end of delivery stroke, the high pressure air is left in the

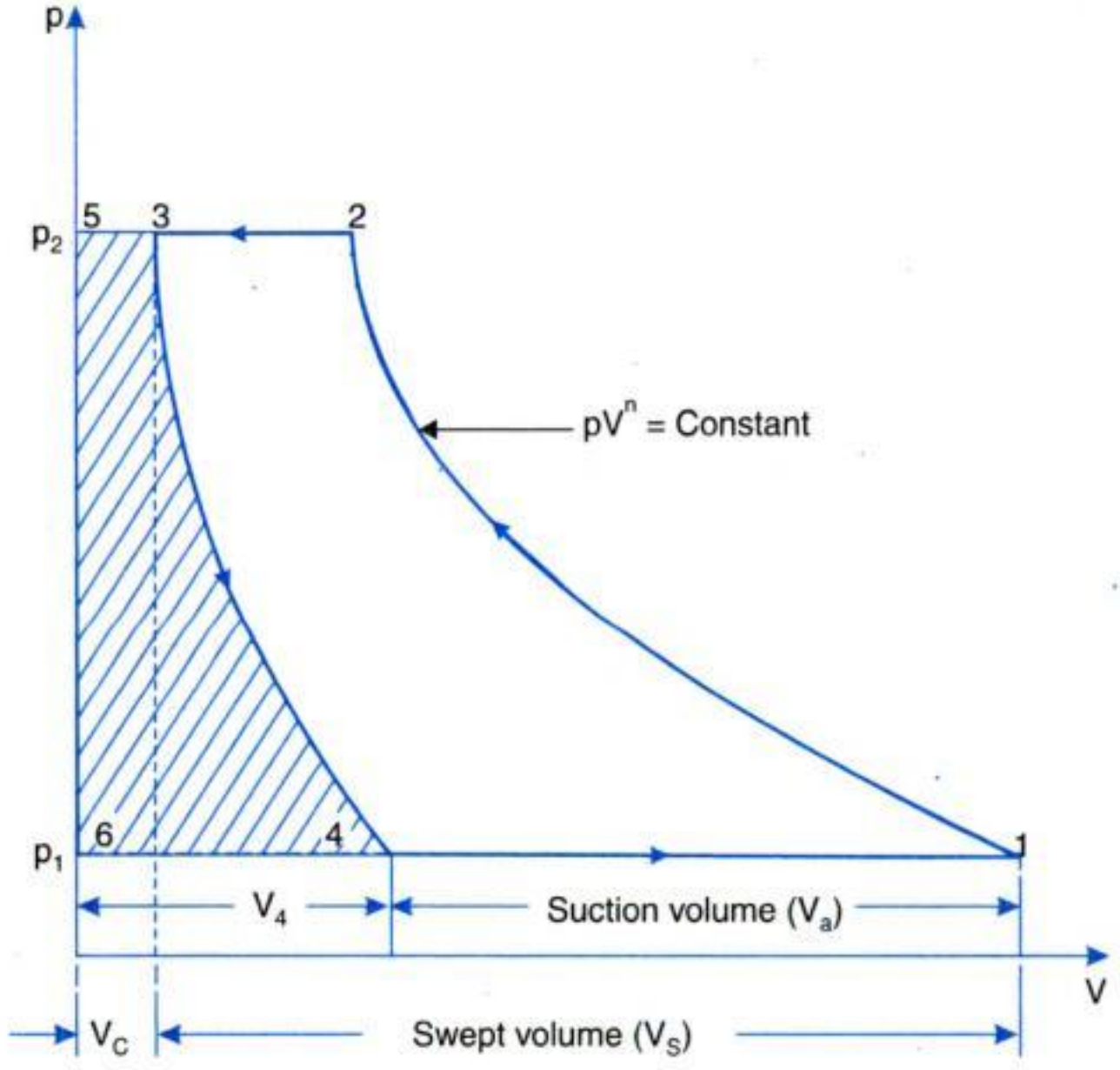

Fig. 24.11

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i.e., *Cylinder bore, D* = **0.1414 m** or **141.4 mm.** **(Ans.)**

Power input to the compressor = $\frac{4.23}{0.85}$ = 4.98 kW

$\therefore$ *Motor power* = $\frac{4.98}{0.9}$ = **5.53 kW.** **(Ans.)**

Example 24.3. *An air compressor takes in air at 1 bar and 20°C and compresses it according to law* $pv^{1.2}$ = *constant. It is then delivered to a receiver at a constant pressure of 10 bar.* R = *0.287 kJ/kg K. Determine :*

(i) Temperature at the end of compression ;

(ii) Work done and heat transferred during compression per kg of air.

Solution. Refer Fig. 24.15.

$T_1 = 20 + 273 = 293$ K ; p_1 = 1 bar ; p_2 = 10 bar

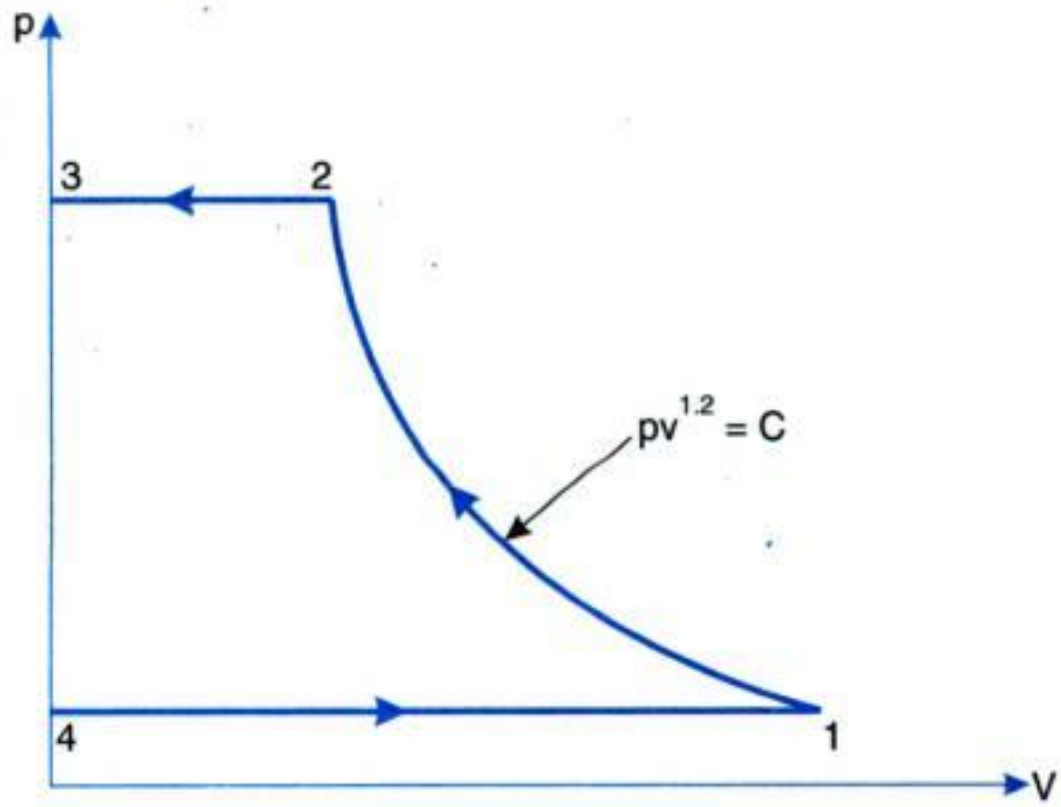

Fig. 24.15

Law of compression : $pv^{1.2} = C$; R = 0.287 J kJ/kg K

(*i*) Temperature at the end of compression, T_2 :

For *compression process 1–2*, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\frac{10}{1}\right)^{\frac{1.2-1}{1.2}} = 1.468$$

or $$T_2 = T_1 \times 1.468 = 293 \times 1.468 = 430 \text{ K or } \mathbf{157°C.} \quad \textbf{(Ans.)}$$

(*ii*) Work done and heat transferred during compression per kg of air :

Work done, $W = mRT_1 \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$ ...[Eqn. (24.5)]

$$= 1 \times 0.287 \times 293 \times \left(\frac{1.2}{1.2-1}\right)\left[\left(\frac{10}{1}\right)^{\frac{1.2-1}{1.2}} - 1\right] = \mathbf{236.13 \text{ kJ/kg of air.}} \quad \textbf{(Ans.)}$$

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i.e., $V_4 = 4.423\ V_3 = 4.423 \times 0.05\ V_s = 0.221\ V_s$

$\therefore$ $(V_1 - V_4) = 1.05\ V_s - 0.221\ V_s = 0.0233$

$\therefore$ $$V_s = \frac{0.0233}{(1.05 - 0.221)} = 0.0281\ \text{m}^3$$

i.e., *Swept volume of the cylinder* = **0.0281 m³.** **(Ans.)**

(*ii*) **The delivery temperature, T_2 :**

Using the relation, $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

$\therefore$ $$T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 288 \times \left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} = 450\ \text{K}$$

$\therefore$ *Delivery temperature* = 450 − 273 = **177°C.** **(Ans.)**

(*iii*) **Indicated power :**

Indicated power $$= \frac{n}{n-1}\ p_1(V_1 - V_4)\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\}$$

$$= \frac{1.3-1}{1.3} \times \frac{1.013 \times 10^5 \times 14}{10^3 \times 60}\left\{\left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} - 1\right\}\ \text{kW}$$

$$= 57.56\ \text{kW}.$$

i.e., *Indicated power* = **57.56 kW.** **(Ans.)**

☞**Example 24.7.** *A single-stage, double-acting compressor has a free air delivery (F.A.D.) of 14 m^3/min. measured at 1.013 bar and 15°C. The pressure and temperature in the cylinder during induction are 0.95 bar 32°C. The delivery pressure is 7 bar and index of compression and expansion, n = 1.3. The clearance volume is 5% of the swept volume. Calculate :*

(i) Indicated power required ; *(ii) Volumetric efficiency.*

Solution.

Free air delivery, F.A.D. = 14 m³/min. (measured at 1.013 bar and 15°C)

Induction pressure, $p_1 = 0.95$ bar

Induction temperature, $T_1 = 32 + 273 = 305$ K

Delivery pressure, $p_2 = 7$ bar

Index of compression and expansion, $n = 1.3$

Clearance volume, $V_3 = V_c = 0.05\ V_s$

(*i*) **Indicated power :**

Mass delivered per minute, $$m = \frac{pV}{RT} = \frac{1.013 \times 10^3 \times 14}{0.287 \times 288 \times 10^3} = 17.16\ \text{kg/min.}$$

[where F.A.D. per minute is V at p(= 1.013 bar) and T(15 + 273 = 288 K)]

To find T_2, using the equation,

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

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Bore, $D = 100 \text{ mm} = 0.1 \text{ m}$

Stroke length, $L = 150 \text{ mm} = 0.15 \text{ m}$

Clearance volume, $V_c = 0.03\ V_s$

Mechanical efficiency, $\eta_{mech.} = 85\%$

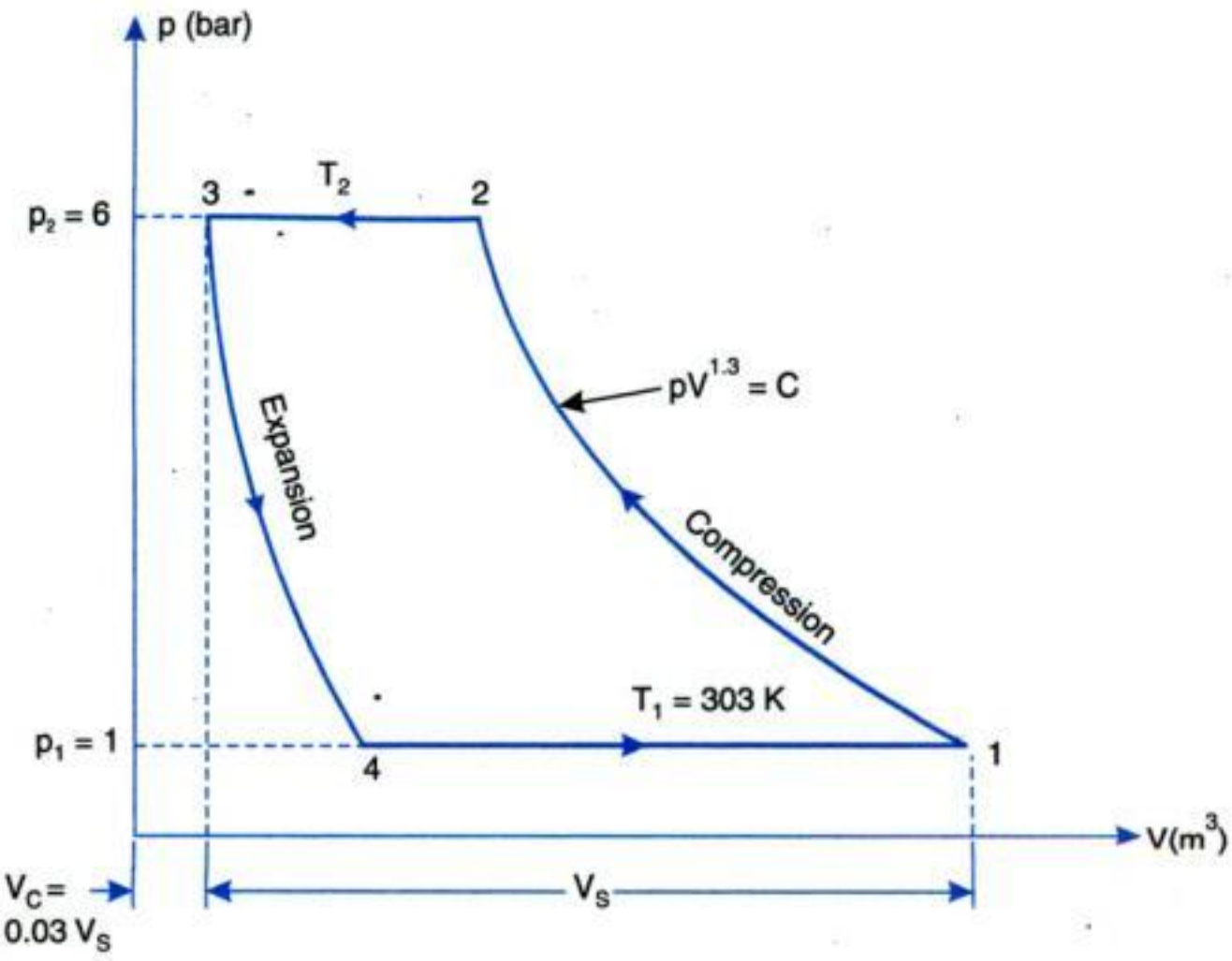

Fig. 24.21

(*i*) Volumetric efficiency of compressor, $\eta_{vol.}$:

$$\eta_{vol.} = 1 + k - k\left(\frac{p_2}{p_1}\right)^{1/n} \qquad \text{...[Eqn. (24.12)]}$$

where $k = \dfrac{V_c}{V_s} = 0.03$

$$\therefore \quad \eta_{vol.} = 1 + 0.03 - 0.03\left(\frac{6}{1}\right)^{\frac{1}{1.3}} = \mathbf{0.91096} \text{ or } \mathbf{91.096\%}. \quad \textbf{(Ans.)}$$

(*ii*) Power required to drive the compressor :

$$\text{Indicated power} = \frac{n}{n-1}\, mRT_1\left[\left(\frac{p_2}{p_1}\right)^{1/n} - 1\right]$$

$$= \frac{1.3}{1.3-1} \times \frac{0.6}{60} \times 0.287 \times 303\left[\left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right] = 1.93 \text{ kW}$$

$$\therefore \text{ Power required to drive the compressor} = \frac{1.93}{\eta_{mech.}} = \frac{1.93}{0.85} = \mathbf{2.27 \text{ kW}}. \quad \textbf{(Ans.)}$$

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Similarly for the *compression process 2–3*, we have

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4000}{700}\right)^{\frac{1.4-1}{1.4}} = 1.6454 \text{ or } T_3 = 310 \times 1.645 = 510.1 \text{ K}$$

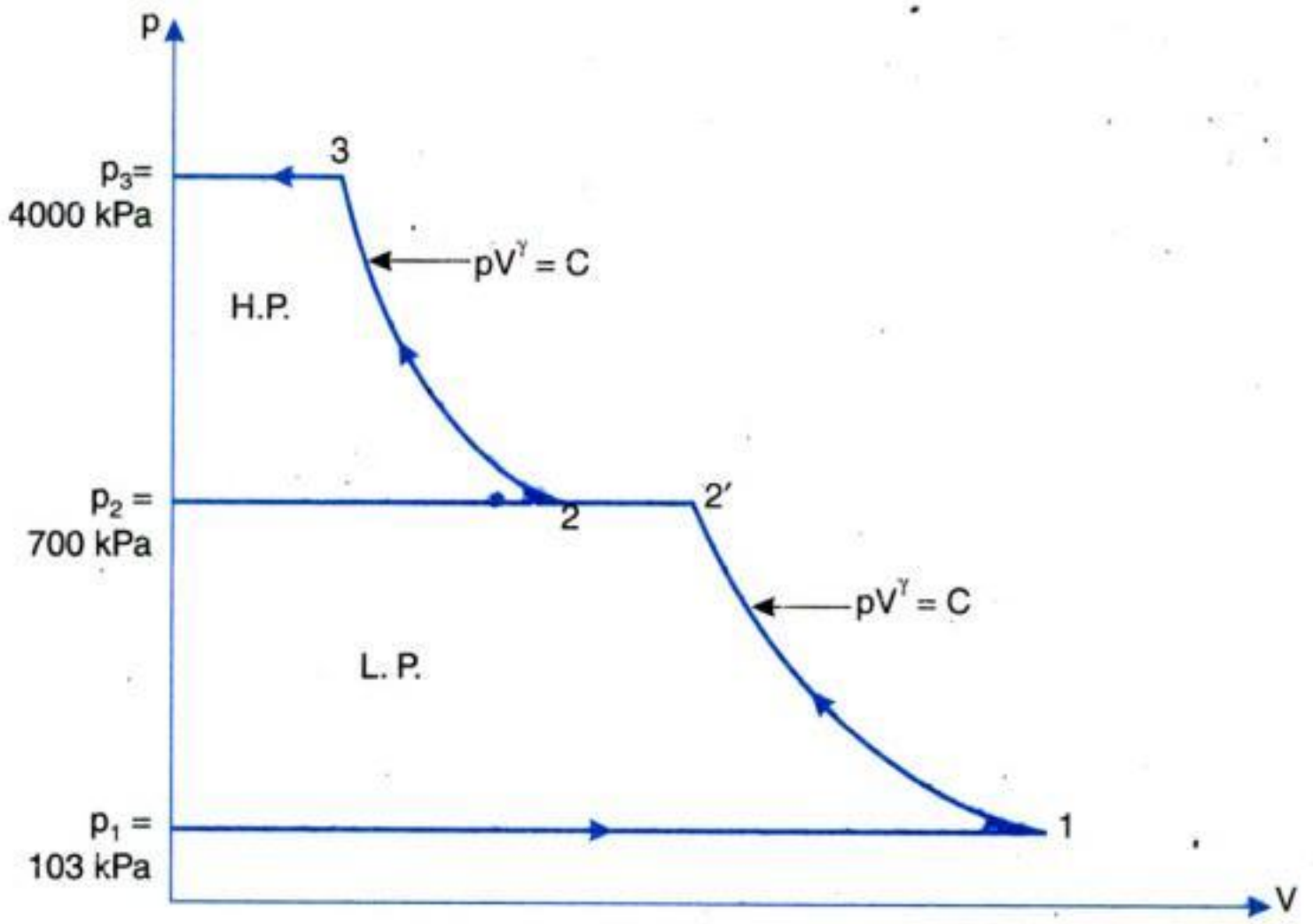

Fig. 24.26

∴ Work required to run the compressor,

$$W = \frac{\gamma}{\gamma-1}\,[mR\,(T_2' - T_1) + mR\,(T_3 - T_2)]$$

$$= \frac{\gamma}{\gamma-1} \times mR\,[(T_2' - T_1) + (T_3 - T_2)]$$

$$= \frac{1.4}{1.4-1} \times \frac{35.89}{3600} \times 0.287\,[(518.7 - 300) + (510.1 - 310)] = 4.194 \text{ kN m/s}$$

Hence power required to run the compressor = **4.194 kW.** **(Ans.)**

Example 24.17. *A trial on a two-stage single-acting reciprocating air compressor gave the following data :*

Free air delivered	$= 6\ m^3/min$
Atmospheric pressure and temperature	$= 1$ *bar and* $27°C$
Delivery pressure	$= 40$ *bar*
Speed	$= 400$ *r.p.m.*
Intermediate pressure	$= 6$ *bar*
Temperature at the inlet to the second stage	$= 27°C$
Law of compression	$= pV^{1.3} = constant$
Mechanical efficiency	$= 80\%$
Stroke of L.P.	$=$ *diameter of L.P.* $=$ *stroke of H.P.*

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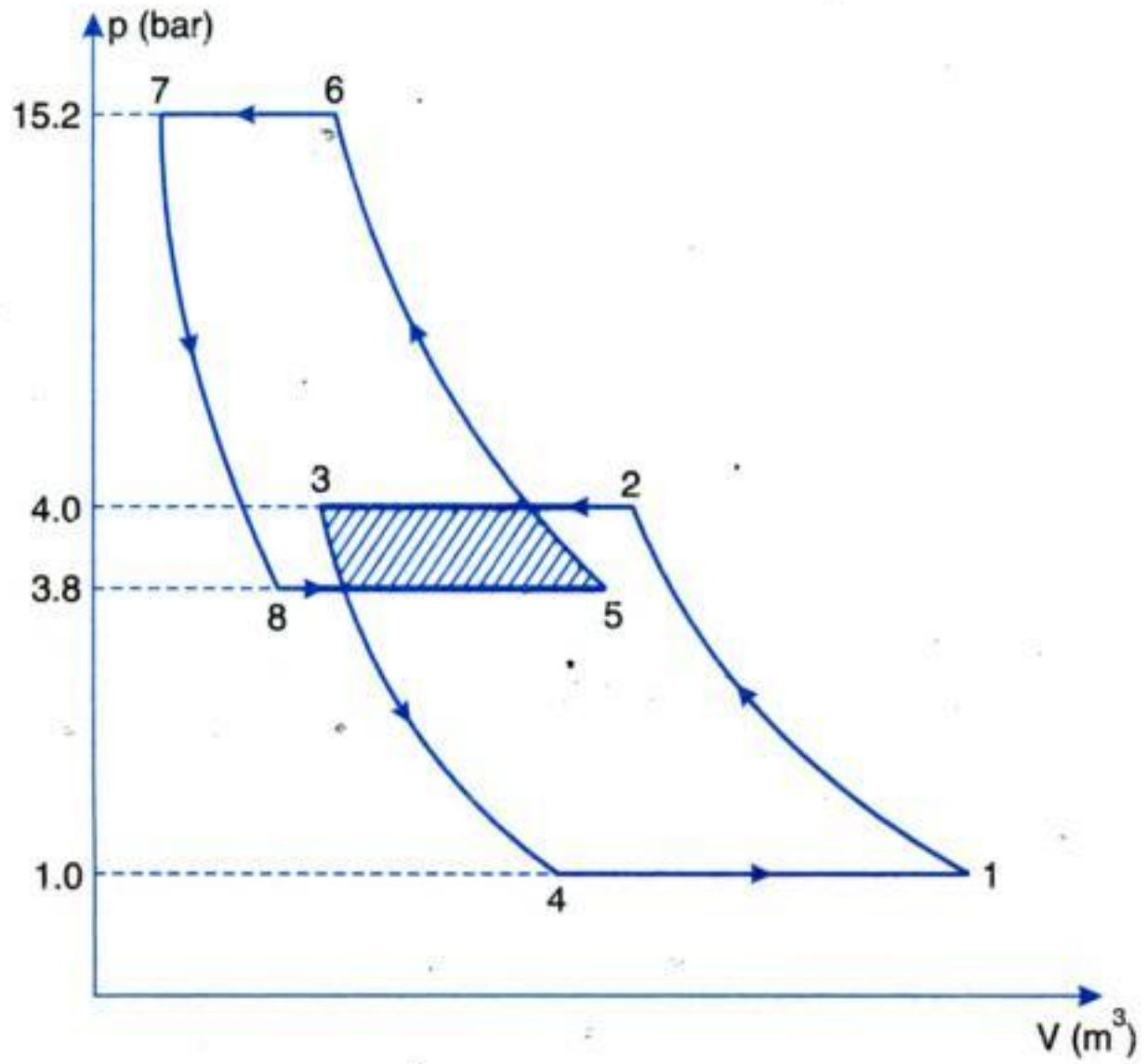

Fig. 24.32

$$= 0.9238 \text{ (or } 92.38\%) \qquad \left[\because \; k = \frac{V_c}{V_s} = \frac{4}{100} = 0.04\right]$$

$\therefore$ Volume of air drawn, referred to condition at 1

$$= 0.9238 \times 17.91 = 16.54 \text{ m}^3/\text{min}$$

Thus, mass of air/min, $m = \dfrac{p_1 V_1}{RT_1} = \dfrac{1.0 \times 10^5 \times 16.54}{0.287 \times 300 \times 10^3} = 19.21$ kg/min

Also, $$T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{p_n}} = 300 \left(\frac{4}{1}\right)^{\frac{1.3-1}{1.3}} = 413 \text{ K}$$

(*i*) Heat rejected in the intercooler :

Heat rejected in the intercooler

$$= mc_p(T_2 - T_5) = 19.21 \times 1.0035 \; (413 - 300)$$

$$= \mathbf{2178.3 \text{ kJ/min.}} \quad \textbf{(Ans.)}$$

(*ii*) Diameter of H.P. cylinder :

Volume of air drawn in H.P. cylinder per minute

$$V_s = \frac{mRT_5}{p_5} = \frac{19.21 \times 0.287 \times 300 \times 10^3}{3.8 \times 10^5} = 4.352 \text{ m}^3/\text{min.}$$

Since the pressure ratio and clearance percentage in H.P. and L.P. cylinders are the same, therefore, the volumetric efficiency of both cylinders is same referred to condition at the start of compression.

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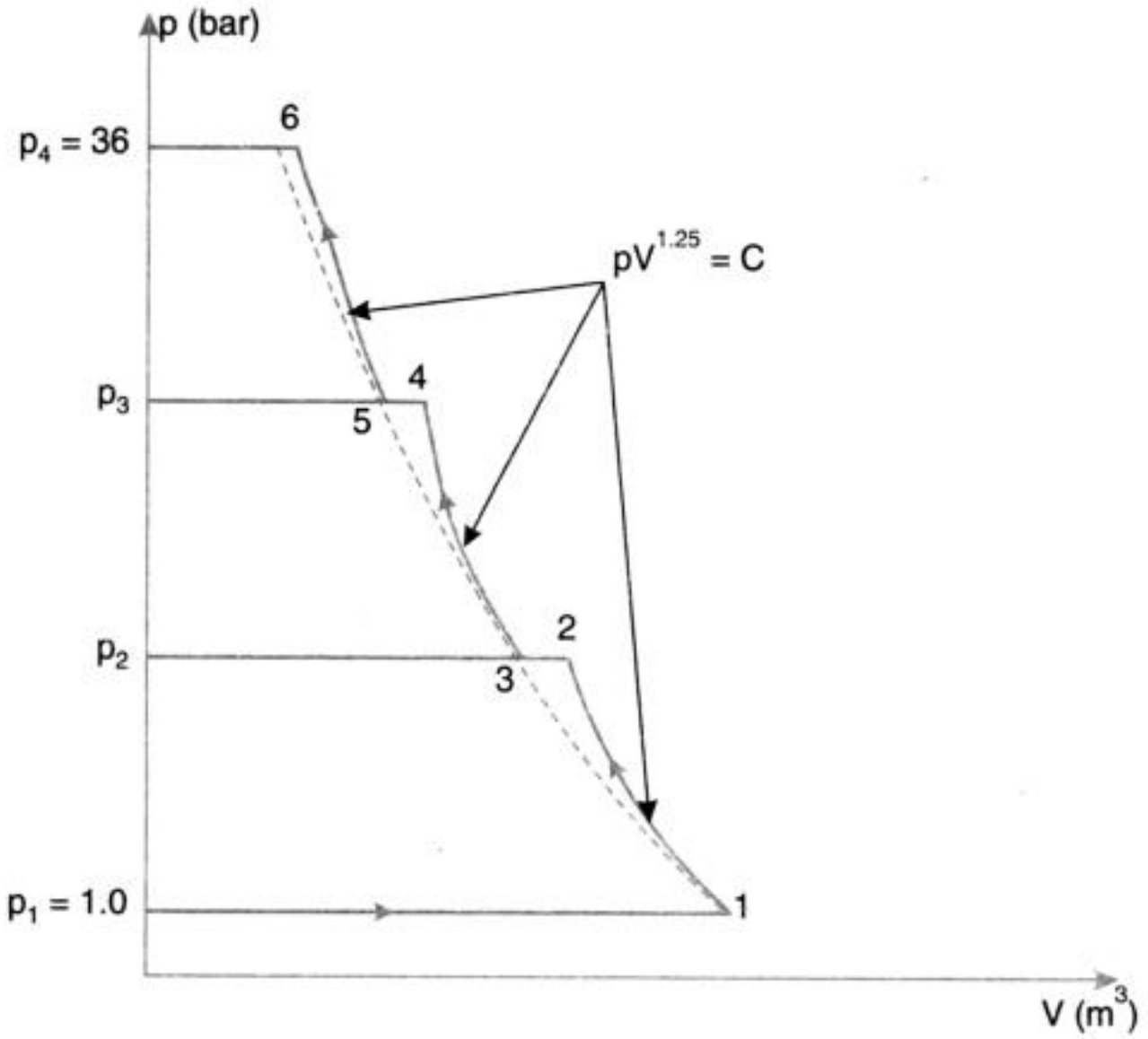

Fig. 24.35

Now $$T_2 = T_1 \left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n} \times \frac{1}{x}}$$ [where x = no. of stages = 3]

$$\therefore \quad T_2 = 300 \left(\frac{36}{1}\right)^{\left(\frac{1.25-1}{1.25}\right) \times \frac{1}{3}} = 380.9 \text{ K}$$

Mass of air handled per minute

$$= \frac{p_1 V_1}{RT_1} = \frac{1.0 \times 10^5 \times 15}{0.287 \times 1000 \times 300} = 17.42 \text{ kg/min.}$$

Total work done in three stages (in kJ/s)

$$= \left[\frac{n}{n-1} . mR (T_2 - T_1)\right] \times x$$

$$= \left[\frac{1.25}{(1.25-1)} \times \frac{17.42}{60} \times 0.287 \times (380.9 - 300)\right] \times 3$$

$$= 101.11 \text{ kJ/s or } 101.11 \text{ kW}$$

i.e., **Indicated power required = 101.11 kW.** (Ans.)

☞**Example 24.29.** *A 3-stage double-acting compressor, operating at 200 r.p.m. takes in air at 1.0 bar and 20°C. The low pressure cylinder size is 350 mm × 400 mm. The intermediate pressure cylinder and the high pressure cylinder have the same stroke as the low pressure cylinder. The discharge pressures from the first stage and second stage are 4.0 bar and 16.0 bar and the air is finally delivered at 64.0 bar. The air is cooled to initial temperature in the intercooler after each stage and there is a drop of pressure of 0.2 bar in each of the intercoolers. The clearance volume in each cylinder is 4% of the stroke volume, but the compression indices are 1.2, 1.25 and 1.3 for*

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$$\therefore \qquad x = \frac{4.828}{1.386} = 3.48 \text{ say } 4$$

Hence the number of stages, x = 4. (Ans.)

(*ii*) **Exact stage-pressure ratios :**

Again using the relation

$$\frac{(p)_{x+1}}{(p)_x} = \left[\frac{(p)_{x+1}}{(p)_1}\right]^{1/x} = \left(\frac{125}{1}\right)^{1/4} = \mathbf{3.343.}$$ (Ans.)

(*iii*) **Intermediate pressures :**

Extreme pressures are already fixed

i.e., $p_1 = 1$ bar, $p_{4+1} = p_5 = 125$ bar

Also $$\frac{(p)_{4+1}}{(p)_4} = \frac{p_5}{p_4} = 3.343$$

$$\therefore \qquad p_4 = \frac{p_5}{3.343} = \frac{1.25}{3.343} = \mathbf{37.39 \text{ bar.}}$$ (Ans.)

Similarly $$\frac{p_4}{p_3} = 3.343$$

$$\therefore \qquad p_3 = \frac{p_4}{3.343} = \frac{37.39}{3.343} = \mathbf{11.18 \text{ bar.}}$$ (Ans.)

and $$p_2 = \frac{p_3}{3.343} = \frac{11.18}{3.343} = \mathbf{3.344 \text{ bar.}}$$ (Ans.)

Example 24.31. *The requirement is to compress air at 1 bar and 25°C and deliver it at 160 bar using multi-stage compression and intercoolers. The maximum temperature during compression must not exceed 125°C and also cooling in the intercooler is done so as not to drop the temperature below 40°C. The law of compression followed is $pV^{1.25}$ = constant for all stages.*

Calculate : (*i*) *Number of stages required,*

(*ii*) *Work input per kg of air, and*

(*iii*) *Heat rejected in the intercoolers.*

Take R = 0.287 kJ/kg K, c_v = 0.71 kJ/kg K.

Solution. Let, p_s = Suction pressure = 1 bar,

T_s = Suction temperature = 25 + 273 = 298 K,

p_1 = Delivery pressure from 1st stage = entry pressure to 2nd stage,

T_1 = Delivery temperature after every stage,

p_d = Delivery pressure from (N + 1)th stage,

x = Number of stages after the 1st stage, and

T_{ratio} = Temperature ratio for 2nd stage, 3rd stage ..... nth stage.

(*i*) **Number of stages :**

Thus, $$\frac{p_1}{p_s} = \left(\frac{T_1}{T_s}\right)^{\frac{n}{n-1}}$$

$$= \left(\frac{125+273}{25+273}\right)^{\frac{1.25}{1.25-1}} = \left(\frac{398}{298}\right)^5 = 4.25$$

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Solution. Volume of air to be compressed, $V = 0.08\ m^3$

Intake pressure, $p_1 = 1.0$ bar

Pressure after compression, $p_2 = 1.5$ bar

Actual work done, $W_{actual} = (p_2 - p_1)\ V = 10^5(1.5 - 1.0) \times 0.08 = 4000$ Nm.

Also ideal work done per revolution is given by,

$$W_{ideal} = \frac{\gamma}{\gamma - 1}\ p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} - 1\right]$$

$$= \frac{1.4}{1.4 - 1} \times 1.0 \times 10^5 \times 0.08 \left[\left(\frac{1.5}{1.0}\right)^{\frac{1.4-1}{1.4}} - 1\right] = 3438.89 \text{ Nm.}$$

$$\therefore \quad \eta_{compressor} = \frac{W_{ideal}}{W_{actual}} = \frac{3438.89}{4000} = \mathbf{0.8597} \text{ or } \mathbf{85.97\%}. \quad \textbf{(Ans.)}$$

24.4.3. Steady-flow Compressors

The compressors in which compression occurs by transfer of kinetic energy from a rotor are called ***Steady-flow compressors.***

The centrifugal type of compressor was used in the earliest gas turbine units for aircraft.

- For low pressure ratios (no greater than about 4 : 1) the centrifugal compressor is *lighter* and is able to *operate effectively over a wide range of mass flows at any one speed,* than its axial-flow counterpart.
- For *larger units with higher pressure ratios the axial flow compressor is more efficient and is usually preferred.* For industrial and large marine gas turbine plants axial flow compressors are usually used, although some units *may employ two or more centrifugal compressors.*
- For *aircraft* the trend has been to higher pressure ratios, and the compressor is usually of the *axial flow.* In aircraft units the advantage of the smaller diameter axial flow compressor can offset the disadvantage of the increased length and weight compared with an equivalent centrifugal compressor.

Advantages *of centrifugal compressors over axial flow compressors :*

1. Smaller length.
2. Contaminated atmosphere does not deteriorate the performance.
3. Can perform efficiently over wide range of mass flows at any speed.
4. Cheaper to produce.
5. More robust.
6. Less prone to icing troubles at high altitudes.

Disadvantages :

1. Large frontal area.
2. Lower maximum efficiency.

Uses : The centrifugal compressors are used in :

(*i*) Superchargers (*ii*) Turbo-prop.

- **Centrifugal compressors** *are preferred where simplicity, light weight, ruggedness are more important than maximum efficiency and smaller diameter.*

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24.4.3.2.2. Width of Blades of Impeller and Diffuser

If the mass of the air flowing per second is constant and is known, then the width of blades of impeller and diffuser can be calculated as follows :

Let, $\dot{m}$ = Mass of air flowing per second,

b_1 = Width (or height) of impeller at inlet,

C_{f_1} = Velocity of flow at inlet of the impeller,

v_1 = Volume of 1 kg of air at the inlet,

r_1 = Radius of impeller at the inlet,

Then, $$\dot{m} = \frac{\text{Volume of air flowing per second}}{\text{Volume of 1 kg of air}} = \frac{2\pi r_1 b_1 \times C_{f_1}}{v_1}$$

But as the air is trapped radially,

$$C_{f_1} = C_1$$

$$\therefore \quad \dot{m} = \frac{2\pi r_1 b_1 \times C_1}{v_1} \qquad \text{...(24.71)}$$

i.e., $$b_1 = \frac{\dot{m} v_1}{2\pi_1 r_1 C_1} \qquad \text{...[24.71 (a)]}$$

Similarly the width of impeller blade at the outlet can be found by using suffix 2 in eqn. (24.71)

$$\dot{m} = \frac{2\pi r_2 b_2 \times C_{f_2}}{v_2} \qquad \text{...(24.72)}$$

The width or height of the impeller blades at the outlet and height of diffuser blade at the inlet should be same theoretically.

The width or height of the diffuser blades at the outlet, is given by

$$\dot{m} = \frac{2\pi r_d b_d \times C_{fd}}{v_d} \qquad \text{...(24.73)}$$

where suffix '*d*' represents the quantities at the *outlet of the diffuser*.

If, n = Number of blades on the impeller, and

t = Thickness of the blade,

then eqns. (24.71), (24.72) and (24.73) are expressed as follows :

$$\dot{m} = \frac{(2\pi r_1 - nt) b_1 C_{f_1}}{v_1} \qquad \text{...(24.74)}$$

$$\dot{m} = \frac{(2\pi r_2 - nt) b_2 C_{f_2}}{v_2} \qquad \text{...(24.75)}$$

$$\dot{m} = \frac{(2\pi r_d - nt) b_d C_{f_d}}{v_d} \qquad \text{...(24.76)}$$

24.4.3.2.3. Isentropic Efficiency of the Compressor

The following *losses* occur when air flows through the impeller :

(*i*) Friction between the air layers moving with relative velocities and friction between the air and flow passages

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- The number of vanes may vary from 10 to 30 but *should not coincide with number of vanes in impeller* to avoid resistance.
- $\frac{D_4}{D_3}$ may vary from 1.25 to 1.6 and the maximum diffusion angle is around 10°.

- *Isentropic efficiency of vaned diffuser at design condition is higher than that of vaneless but its off-design performance is poor than that of vaneless.*

24.4.3.2.9. Compressor Characteristics. When flow is taking place in an impeller channel, there are certain inlet losses, friction and separation losses and discharge losses in the diffuser. If these losses and the effect of slip and non-uniform distribution of radial velocity around the periphery of the impeller are taken into account, the head-capacity characteristic for the backward-curved vanes would take the form *LM* as shown in Fig. 24.57.

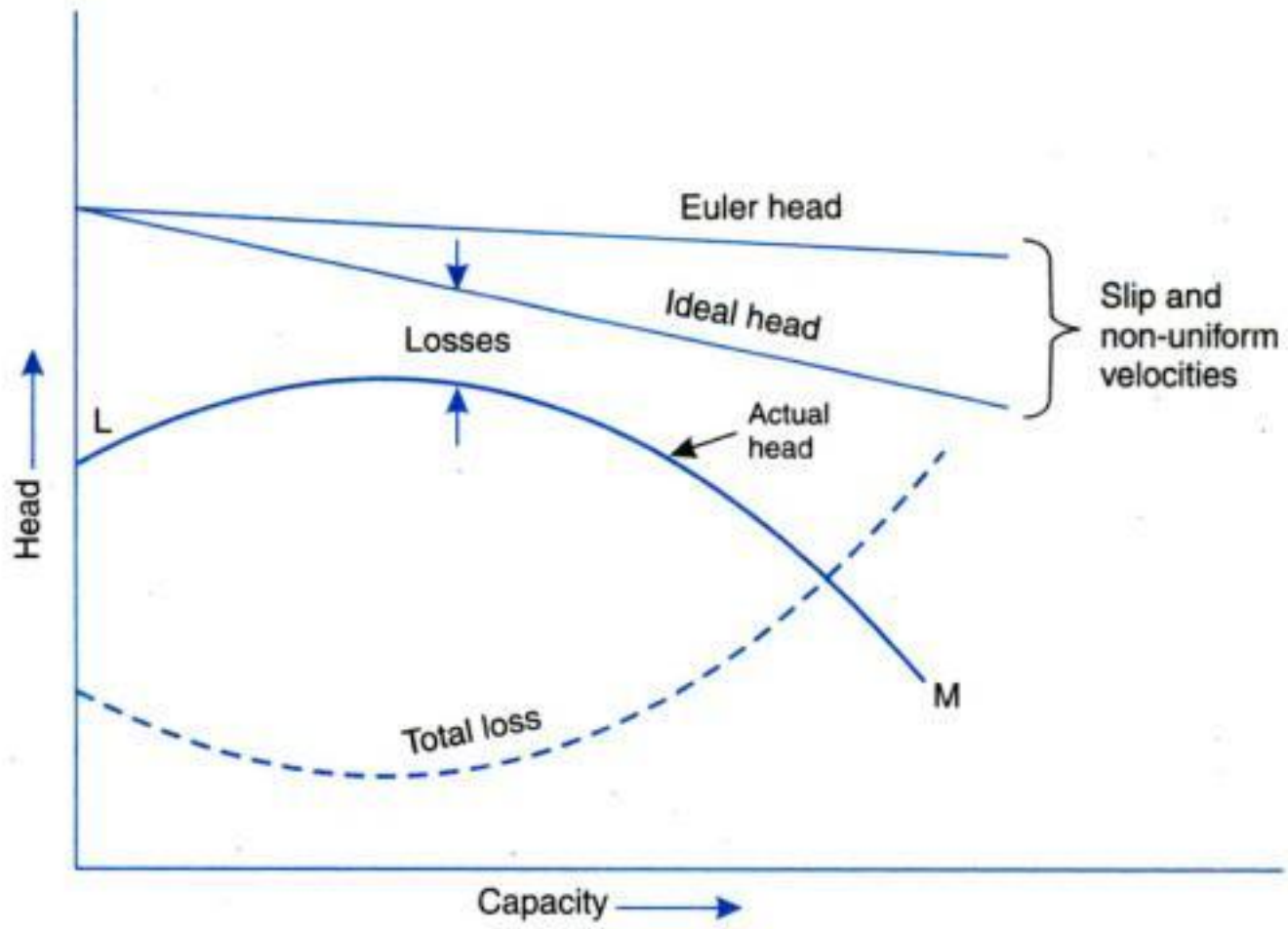

Fig. 24.57. Actual characteristics of a centrifugal compressor.

24.4.3.2.10. Surging and Choking. Fig. 24.58 shows a typical characteristics of a centrifugal compressor at one particular sped. *Consider that the compressor is running at point N* :

- If now the *resistance to flow is increased* (say, by, closing the valve provided at the delivery line of the compressor), the equilibrium point moves to *M*.
- Any further restriction to the flow will cause the operating point to shift to the left, ultimately arriving at point *L*. At this point *maximum pressure ratio is obtained*. If the flow is still

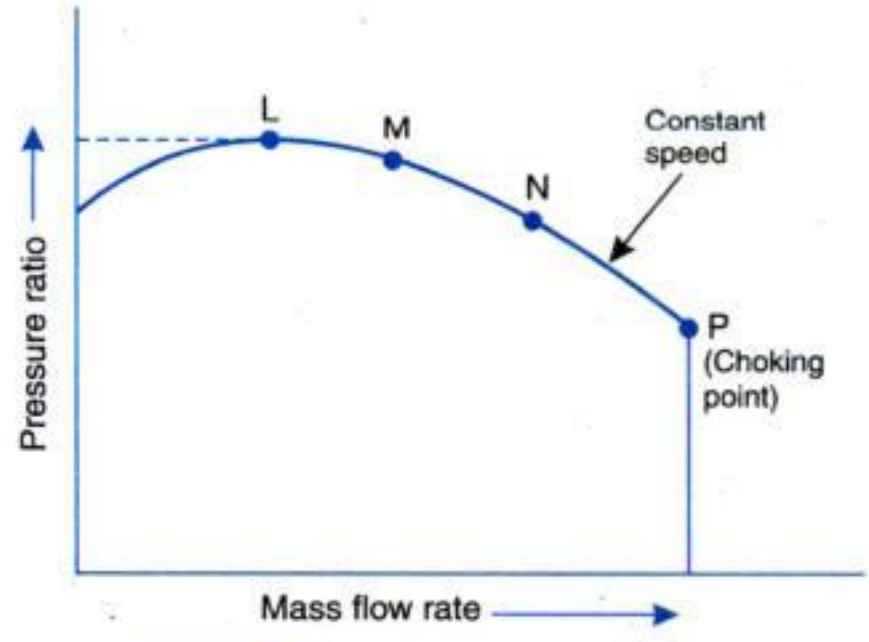

Fig. 24.58. A typical characteristic at one particular speed.

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Work consumed by the compressor is also given by Euler's equation without prewhirl as :

$$W = \frac{C_{w_2} \times C_{bl_2}}{1000} \text{ kJ/kg} = 165.8 \text{ kJ/kg}$$

But $$\frac{C_{w_2}}{C_{bl_2}} = 0.9 \quad \therefore \quad C_{w_2} = 0.9 C_{bl_2}$$

$$\therefore \qquad 165.8 = \frac{C_{bl_2}^2 \times 0.9}{1000}$$

or $$C_{bl_2} = \left(\frac{165.8 \times 1000}{0.9}\right)^{1/2} = 429.2 \text{ m/s}$$

But $$C_{bl_2} = 429.2 = \frac{\pi d_2 N}{60} = \frac{\pi d_2 \times 20000}{60}$$

$$\therefore \qquad d_2 = \frac{429.2 \times 60}{\pi \times 20000} = 0.4098 \text{ m} \quad \text{or} \quad 40.98 \text{ cm say } \mathbf{41 \text{ cm.}} \quad \textbf{(Ans.)}$$

(*iii*) **Power required, P :**

$$P = \dot{m} \times 165.8 = 8.8 \times 165.8 = \mathbf{1459 \text{ kW.}} \quad \textbf{(Ans.)}$$

(*iv*) **Eye diameter if hub diameter is 12 cm, d_1 :**

From continuity equation, we have

$$\dot{m} = \frac{\pi}{4} (d_1^2 - d_h^2) \times C_1 \times \rho_1$$

But density at entry is given by,

$$\rho_1 = \frac{p_1}{RT_1} = \frac{1 \times 10^5}{287 \times 293} = 1.189 \text{ kg/m}^3$$

$$\therefore \qquad 8.8 = \frac{\pi}{4} (d_1^2 - 0.12^2) \times 145 \times 1.189$$

or $$d_1^2 = \frac{8.8 \times 4}{\pi \times 145 \times 1.189} + 0.12^2 = 0.07939 \text{ m}^2$$

$$\therefore \qquad d_1 = \mathbf{0.2818 \text{ m}} \text{ or } \mathbf{28.2 \text{ cm.}} \quad \textbf{(Ans.)}$$

Example 24.39. *A centrifugal compressor running at 10000 r.p.m. delivers 660 m^3/min. of free air. The air is compressed from 1 bar and 20°C to a pressure ratio of 4 with an isentropic efficiency of 82%. Blades are radial at outlet of impeller and flow velocity of 62 m/s may be assumed throughout constant. The outer radius of impeller is twice the inner and the slip factor may be assumed as 0.9. The blade area co-efficient may be assumed 0.9 at inlet. Calculate :*

(*i*) *Final temperature of air.* (*ii*) *Theoretical power.*

(*iii*) *Impeller diameters at inlet and outlet.*

(*iv*) *Breadth of impeller at inlet.*

(*v*) *Impeller blade angle at inlet.*

(*vi*) *Diffuser blade angle at inlet.*

Solution. *Given* : $N = 10000$ r.p.m. ; Volume of air delivered, $V = 660$ m^3/min. ;

$p_1 = 1$ bar, $T_1 = 20 + 273 = 293$ K ; $r_p = 4$, $\eta_{isen} = 0.82$; $C_{f_2} = 62$ m/s ;

$r_2 = 2r_1$; $\phi_s = 0.9$; Blade area co-efficient, $k_a = 0.9$.

(*i*) **Final temperature of air, T_2 :**

$$\frac{T_2'}{T_1} = (r_p)^{\gamma - 1/\gamma} = (4)^{\frac{1.4-1}{1.4}} = 1.486$$

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(iii) Work of compressor per kg of air.

(iv) The theoretical power required.

Solution. *Given :* r_p (pressure ratio) = 3.6

$$T_{01} = 15 + 273 = 288 \text{ K} ; \eta_{isen} = 0.8$$

$$\dot{m} = 12 \text{ kg/s} ; \phi_p \text{ (Pressure co-efficient)} = 0.73 ;$$

$$C_{bl_2} = 120 \text{ m/s}$$

(i) The static pressure and temperature at inlet and outlet of compressor :

Total head temperature rise

$$\Delta T_0 = \frac{T_{01}\,(r_p^{\frac{\gamma-1}{\gamma}} - 1)}{\eta_{isen}} = \frac{288\,[(3.6)^{\frac{1.4-1}{1.4}} - 1]}{0.8} = 159.1°\text{C}$$

$$\therefore \quad \mathbf{T_{02}} = T_{01} + \Delta T_0 = 288 + 159.1 = \mathbf{447.1\ K.} \quad \textbf{(Ans.)}$$

The static temperature at exist is

$$\mathbf{T_2} = T_{02} - \frac{C_{bl_2}^2}{2c_p} = 447.1 - \frac{120^2}{2 \times 1.005 \times 10^3}$$

$$= 447.1 - 7.1 = \mathbf{440\ K.} \quad \textbf{(Ans.)}$$

The *static pressure at exit is*

$$p_2 = p_{02} - \frac{\rho_2 C_{bl_2}^2}{2} \qquad \left(\text{where } \rho_2 = \frac{p_2}{RT_2}\right)$$

i.e.,

$$p_2 = p_{02} - \frac{p_2 \times 120^2}{2 \times 0.287 \times 440 \times 10^3} = p_{02} - 0.057\, p_2$$

But $p_{02} = 1.013 \times 3.6 = 3.65$ bar

$\therefore$ $p_2 = 3.65 - 0.057\, p_2$ or $1.057\, p_2 = 3.65$

$\therefore$ $\mathbf{p_2 = 3.45\ bar.}$ **(Ans.)**

To calculate *static conditions at inlet :*

$$T_{01} = T_1 + \frac{140^2}{2c_p} = T_1 + \frac{140^2}{2 \times 1.005 \times 10^3} = T_1 + 9.75$$

$\therefore$ $\mathbf{T_1 = 288 - 9.75 = 278.25\ K.}$ **(Ans.)**

$$p_1 = p_{01} - \frac{p_1 \times 140^2}{2 \times 0.287 \times 278.25 \times 10^3} = 1.013 - 0.123\, p_1$$

$\therefore$ $\mathbf{p_1 = 0.9\ bar.}$ **(Ans.)**

(ii) Static pressure ratio $= \dfrac{p_2}{p_1} = \dfrac{3.45}{0.9} = \mathbf{3.83.}$ **(Ans.)**

(iii) Work done on air $= c_p \times \Delta T_0$

$= 1.005 \times 159.1 = \mathbf{159.89\ kJ/kg\ of\ air.}$ **(Ans.)**

(iv) Theoretical power required to drive compressor

$= \dot{m}\, c_p\, \Delta T_0 = 12 \times 1.005 \times 159.1 = \mathbf{1918.7\ kW.}$ **(Ans.)**

Example 24.43. *Air at a temperature of 300 K flows in a centrifugal compressor running at 18000 r.p.m. The other data given is as follows :*

Isentropic total head efficiency = *0.76*

Outer diameter of blade tip = *550 mm*

Slip factor = *0.82*

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- Due to diffusion in the diverging passages formed by rotor blades, there is some pressure rise. This is at the expense of relative velocity and so the relative velocity decreases from C_{r1} to C_{r2}. Since work is being done on the air by rotor blades, the air would ultimately leave the rotor with increased absolute velocity C_2.
- The air then enters the stator blades and the diffusion and deceleration takes place in the diverging passage of stator blades. Finally the air leaves the stator blades with velocity C_3 at an angle α_3 and is redirected to the *next stage*. Generally it is *assumed that absolute velocity C_3 leaving the compressor stage equals the approach velocity C_1*.

From the velocity triangles, we have

$$\frac{C_{bl}}{C_f} = \tan \alpha_1 + \tan \beta_1 \qquad ...(24.85)$$

and,

$$\frac{C_{bl}}{C_f} = \tan \beta_2 + \tan \alpha_2 \qquad ...(24.86)$$

Assume *1 kg of flow of air* through the *compressor stage*.

From Newton's second law of motion,

Tangential force per kg = $C_{w2} - C_{w1}$

Work absorbed by the stage per kg of air,

$$W_{st} = C_{bl}(C_{w2} - C_{w1}) = c_p(T_{02} - T_{01})_{act} \qquad ...(24.87)$$

where C_{w1} and C_{w2} are the whirl components of absolute velocities at inlet and exit of rotating blades.

(It may be noted that here whirl component at the entrance of the compressor is not *zero* because of the fact that air flows *axially and not radially*.)

The expression for work done may be put in terms of flow/axial velocity and air angles.

$$W_{st} = C_{bl}\, C_f \quad (\tan \alpha_2 - \tan \alpha_1) \qquad ...(24.88)$$

From eqns. (24.85) and (24.86), we have

$$W_{st} = C_{bl}\, C_f \quad (\tan \beta_1 - \tan \beta_2) \qquad ...(24.89)$$

$$\left[\because \quad \frac{C_{w1}}{C_f} = \tan \alpha_1, \text{ and } \frac{C_{w2}}{C_f} = \tan \alpha_2\right]$$

The work input to the axial flow compressor may also be obtained by the Euler's equation. For each kg of air delivered, we have

$$W_{st} = C_{bl_2}\, C_{w2} - C_{bl_1}\, C_{w1}$$

Since $$C_{w2} - C_{w1} = [(C_{bl} - C_{rw2}) - (C_{bl} - C_{rw1})] = C_{rw1} - C_{rw2}$$

Further $C_{bl_1} = C_{bl_2} = C_{bl}$, the above equation is modified as,

$$W_{st} = C_{bl}(C_{rw1} - C_{rw2}) = C_{bl}\, \Delta C_{rw} = c_p\, (\Delta T_0)_{act}$$

By use of velocity triangle and cosine theorem,

$$W_{st} = \underbrace{\frac{C_{r1}^2 - C_{r2}^2}{2}}_{\text{(First term)}} + \underbrace{\frac{C_2^2 - C_3^2}{2}}_{\text{(Second term)}} \qquad ...(24.90)$$

Here $$C_3 = C_1$$

— The *first term* on the right side of the above equation introduces the part of the work supplied by a rotating cascade, which is converted into pressure due to *diffusion action* in rotating cascade itself.

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11.	*Adaptability*	Adaptability to low speed drive	Adaptability to high speed, low maintenance cost drivers such as turbines
12.	*Operating attention*	More	Less
13.	*Mixing of working fluid with lubricating oil*	Always a chance	No chance
14.	*Suitability*	For low, medium and high pressures and low and medium gas volumes.	For low and medium pressures and large gas volumes.

24.6. COMPARISON BETWEEN RECIPROCATING AND ROTARY AIR COMPRESSORS

S. No.	*Aspects*	*Reciprocating air compressors*	*Rotary air compressors*
1.	*Suitability*	Suitable for low discharge of air at high pressure	Suitable for handling large volumes of air at low pressures.
2.	*Operational speed*	Low	Usually high
3.	*Air supply*	Pulsating	Continuous
4.	*Balancing*	Cyclic vibrations occur	Less vibrations
5.	*Lubricating system*	Generally complicated	Generally simple lubrication systems are required
6.	*Quality of air delivered*	Generally contaminated with oil	Air delivered is relatively more clean.
7.	*Air compressor size*	Large for the given discharge	Small for same discharge
8.	*Free air handled*	250–300 m^3/min	2000–3000 m^3/min
9.	*Delivery pressure*	High	Low
10.	*Usual standard of compression*	Isothermal compression	Isentropic compression

24.7. COMPARISON BETWEEN CENTRIFUGAL AND AXIAL FLOW COMPRESSORS

S. No.	*Aspects*	*Centrifugal compressors*	*Axial flow compressors*
1.	*Type of flow*	Axial (Parallel to the direction of axis of the machine)	Radial
2.	*Pressure ratio per stage*	High, about 4.5 : 1. Thus unit is *compact* — In supersonic compressors, the pressure ratio is about 10 but at the cost of efficiency. Operation is *not* so difficult and risky.	Low, about 1.2 : 1. This is due to absence of centrifugal action. To achieve the pressure ratio equal to that per stage in centrifugal compressor 10 to 20 stages are required. Thus, the unit is *less compact* and *less rugged.*
3.	*Isothermal efficiency*	About 80 to 82%	About 86 to 88% (with modern aerofoil blades)
4.	*Frontal area*	Larger	Smaller (This makes the axial flow compressors *more suitable for jet engines due to less drag*).

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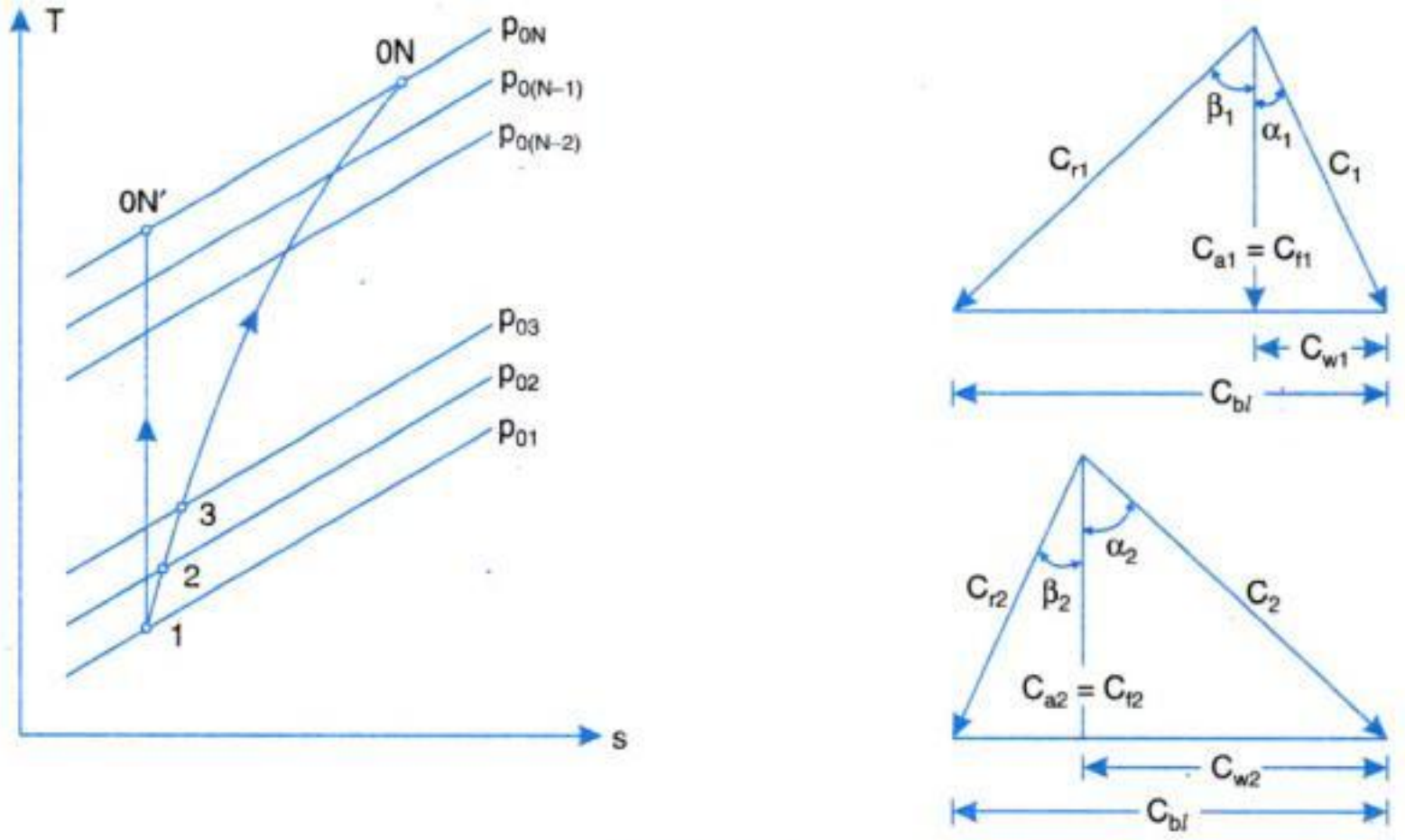

Fig. 24.69

Let suffix N denotes the *number of stages.*

With isentropic compression the temperature of air leaving the compressor stage is

$$T_{0N}' = T_{01}\left(\frac{p_{0N}}{p_{01}}\right)^{(\gamma-1/\gamma)} = 290 \times (5)^{\frac{1.4-1}{1.4}} = 459.3 \text{ K}$$

But $$\eta_{isen} = \frac{T_{0N}' - T_{01}}{T_{0N} - T_{01}}$$

$$0.92 = \frac{459.3 - 290}{T_{oN} - 290}$$

$\therefore$ $$T_{0N} = \frac{459.3 - 290}{0.92} + 290 = 474 \text{ K}$$

The work consumed by the compressor

$$= c_p(T_{0N} - T_{01}) = (C_{w2} - C_{w1})\, C_{bl} \times \text{N}$$

or $$c_p(T_{0N} - T_{01}) = C_f\,(\tan\alpha_2 - \tan\alpha_1)\, C_{bl}\,.\,N$$

$\therefore$ $$\tan\alpha_2 - \tan\alpha_1 = \frac{c_p(T_{0N} - T_{01})}{C_f.C_{bl}.N} = \frac{1.005\,(474 - 290) \times 10^3}{90 \times 160 \times 8} = 1.605 \qquad ...(i)$$

From velocity triangles, we have

$$\frac{C_{bl}}{C_f} = \tan\alpha_1 + \tan\beta_1 = \frac{160}{90} = 1.778 \qquad ...(ii)$$

Adding (i) and (ii), we get

$$\tan\beta_1 = \frac{1.605 + 1.778}{2} = 1.6915 \qquad (\because \quad \alpha_1 = \beta_1)$$

or $$\beta_1 = \tan^{-1}(1.6915) = 59.4°$$

$\therefore$ $$\boldsymbol{\beta_1 = \alpha_2 = 59.4°.}$$ **(Ans.)**

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(*v*) Impeller blade angle at inlet.
(*vi*) Diffuse blade angle at inlet. [Ans. (*i*) 466.85 K ; (*ii*) 2077.7 kW ; (*iii*) 46.745 cm, 94.9 cm ; (*iv*) 12.2 cm; (*v*) 15.7°; (*vi*) 8.9°]

26. A single inlet type centrifugal compressor handles 8 kg/s of air. The ambient air conditions are 1 bar and 20°C. The compressor runs at 22000 r.p.m. with isentropic efficiency of 82%. The air is compressed in the compressor from 1 bar static pressure to 4.2 bar total pressure. The air enters the impeller eye with a velocity of 150 m/s with no prewhirl. Assuming that the ratio of whirl speed to tip speed is 0.9, calculate :
(*i*) Rise in total temperature during compression if the change in K.E. is negligible.
(*ii*) The tip diameter of the impeller.
(*iii*) Power required.
(*iv*) Eye diameter if the hub diameter is 10 cm.
[Ans. (*i*) 167.67°C ; (*ii*) 37.56 cm ; (*iii*) 1348 kW ; (*iv*) 25.9 cm]

27. In an axial flow compressor, the overall stagnation pressure ratio achieved is 4 with overall stagnation isentropic efficiency 86 per cent. The inlet stagnation pressure and temperature are 1 bar and 320 K. The mean blade speed is 190 m/s. The degree of reaction is 0.5 at the mean radius with relative air angles of 10° and 30° at rotor inlet and outlet respectively. The work done factor is 0.9. Calculate :
(*i*) Stagnation polytropic efficiency.
(*ii*) Number of stages.
(*iii*) Inlet temperature and pressure.
(*iv*) Blade height in the first stage if the hub-tip ratio is 0.4, mass flow rate is 20 kg/s.
[Ans. (*i*) 88.4% ; (*ii*) 11 ; (*iii*) 287.39 K, 0.6864 bar; (*iv*) 11.4 cm]

28. A multi-stage axial flow compressor delivers 18 kg/s of air. The inlet stagnation condition is 1 bar and 20°C. The power consumed by the compressor is 4260 kW. Calculate :
(*i*) Delivery pressure.
(*ii*) Number of stages.
(*iii*) Overall isentropic efficiency of the compressor.
Assume temperature rise in the first-stage is 18°C. The polytropic efficiency of compression is 0.9 and the stage stagnation pressure ratio is constant. [Ans. (*i*) 6.41 bar ; (*ii*) 10 ; (*iii*) 87.24%]

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- 1–2 represents : *Ideal isentropic compression.*
- 3–4 represents : *Ideal isentropic expansion.*

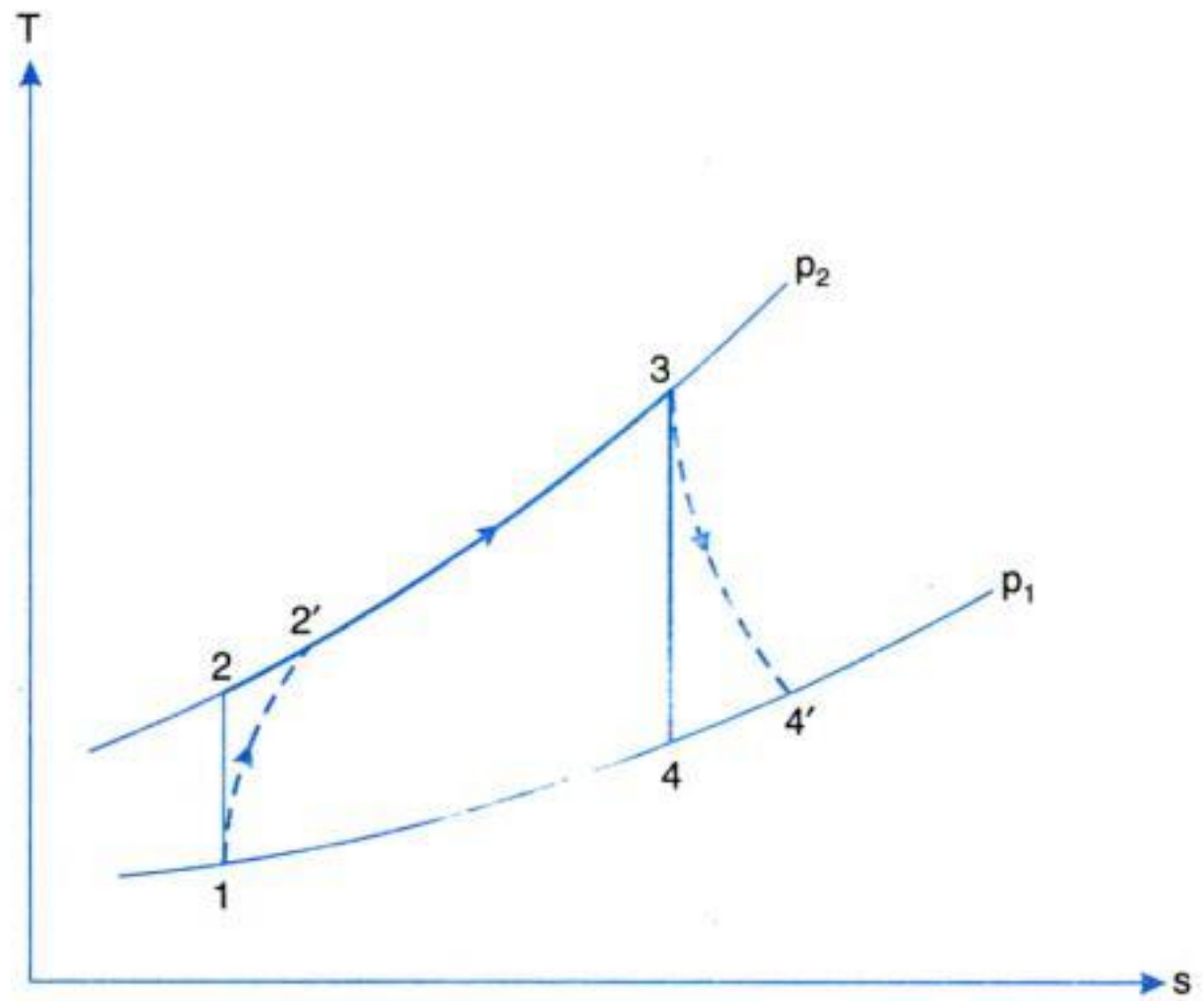

Fig. 25.2

Assuming change in kinetic energy between the various points in the cycle to be negligibly small compared with enthalpy changes and then applying the flow equation to each part of cycle, for unit mass, we have

Work input (compressor) $= c_p (T_2' - T_1)$

Heat supplied (combustion chamber) $= c_p (T_3 - T_2')$

Work output (turbine) $= c_p (T_3 - T_4')$

$\therefore$ Network output = Work output – Work input

$$= c_p (T_3 - T_4') - c_p(T_2' - T_1)$$

and
$$\eta_{thermal} = \frac{\text{Network output}}{\text{Heat supplied}}$$

$$= \frac{c_p (T_3 - T_4') - c_p (T_2' - T_1)}{c_p (T_3 - T_2')}$$

Compressor isentropic efficiency, η_{comp}

$$= \frac{\text{Work input required in isentropic compression}}{\text{Actual work required}}$$

$$= \frac{c_p (T_2 - T_1)}{c_p (T_2' - T_1)} = \frac{T_2 - T_1}{T_2' - T_1} \qquad ...(25.1)$$

Turbine isentropic efficiency, $\eta_{turbine}$

$$= \frac{\text{Actual work output}}{\text{Isentropic work output}}$$

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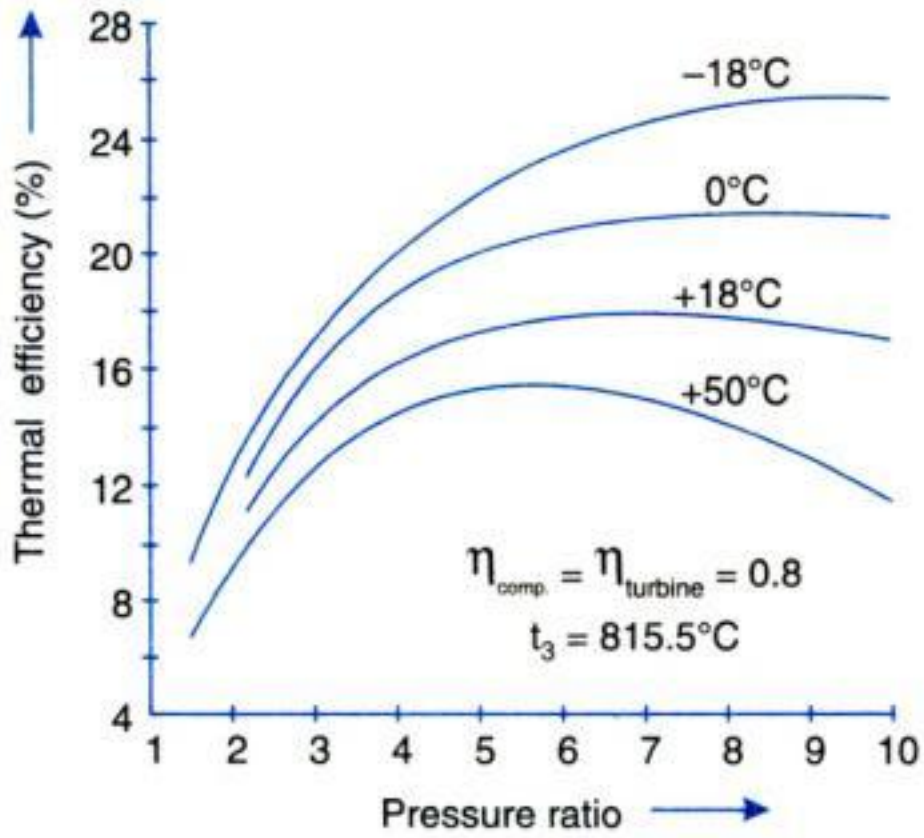

Fig. 25.12

25.4.4. Closed Cycle Gas Turbine (Constant pressure or joule cycle).

Fig. 25.13 shows a gas turbine operating on a constant pressure cycle in which the closed system consists of air behaving as an ideal gas. The various operations are as follows : Refer Figs. 25.14 and 25.15.

Operation 1–2 : The air is compressed isentropically from the lower pressure p_1 to the upper pressure p_2, the temperature rising from T_1 to T_2. No heat flow occurs.

Operation 2–3 : Heat flow into the system increasing the volume from V_2 to V_3 and temperature from T_2 to T_3 whilst the pressure remains constant at p_2. Heat received = $mc_p\,(T_3 - T_2)$.

Operation 3–4 : The air is expanded isentropically from p_2 to p_1, the temperature falling from T_3 to T_4. No heat flow occurs.

Operation 4–1 : Heat is rejected from the system as the volume decreases from V_4 to V_1 and the temperature from T_4 to T_1 whilst the pressure remains constant at p_1. Heat rejected = $mc_p\,(T_4 - T_1)$

$$\eta_{air\text{-}standard} = \frac{\text{Work done}}{\text{Heat received}}$$

$$= \frac{\text{Heat received/cycle} - \text{Heat rejected/cycle}}{\text{Heat received/cycle}}$$

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

Now, from isentropic expansion

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

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In a closed cycle plant, in practice, the control of power output is achieved by varying the mass flow by the use of a reservoir in the circuit. The *reservoir maintains the design pressure and temperature and therefore achieves an approximately constant level of efficiency for varying* loads. In this cycle since it is closed, *gases other than air with favourable properties can be used* ; furthermore it is possible to burn solid fuels in the combustion heaters. *The major factor responsible for inefficiency in this cycle is the large irreversible temperature drop which occurs in the air heaters between the furnace and circulating gas.*

Note. 1. In a closed cycle gas turbines, although air has been extensively used, the use of 'helium' which though of a lower density, has been inviting the attention of manufacturers for its use, for large output gas turbine units. *The specific heat of helium at constant pressure is about 'five times' that of air, therefore for each kg mass flow the heat drop and hence energy dealt within helium machines is nearly five times of those in case of air.* The *surface area of the heat exchanger for helium can be kept as low as 1/3* of that *required for gas turbine plant using air as working medium.* For the same temperature ratio and for the plants of the same output the *cross-sectional area required for helium is much less than that for air.* It may therefore be concluded that the *size of helium unit is considerably small comparatively.*

2. Some gas turbine plants work on a combination of two cycles the open cycle and the closed cycle. Such a combination is called the *semi-closed cycle. Here a part of the working fluid is confined within the plant and another part flows from and to atmosphere.*

25.4.5. Merits and Demerits of Closed Cycle Gas Turbine Over Open Cycle Gas Turbine

Merits of closed cycle :

1. Higher thermal efficiency
2. Reduced size
3. No contamination
4. Improved heat transmission
5. Improved part load efficiency
6. Lesser fluid friction
7. No loss of working medium
8. Greater output
9. Inexpensive fuel.

Demerits of closed cycle :

1. Complexity
2. Large amount of cooling water is required. This limits its use to stationary installation or marine use where water is available in abundance.
3. Dependent system.
4. The weight of the system per H.P. developed is high comparatively, therefore not economical for moving vehicles.
5. Requires the use of a very large air heater.

25.5. CONSTANT VOLUME COMBUSTION TURBINES

Refer Fig. 25.18. In a constant volume combustion turbine, the compressed air from an air compressor C is admitted into the combustion chamber D through the valve A. When the valve A is closed, the fuel is admitted into the chamber by means of a fuel pump P. Then the mixture is ignited by means of a spark plug S. The combustion takes place at constant volume with increase of pressure. The valve B opens and the hot gases flow to the turbine T, and finally, they are discharged, into atmosphere. The energy of the hot gases is thereby converted into mechanical energy. For continuous running of the turbine these operations are repeated.

The main demerit associated with this type of turbine is that the *pressure difference and velocities of hot gases are not constant ; so the turbine speed fluctuates.*

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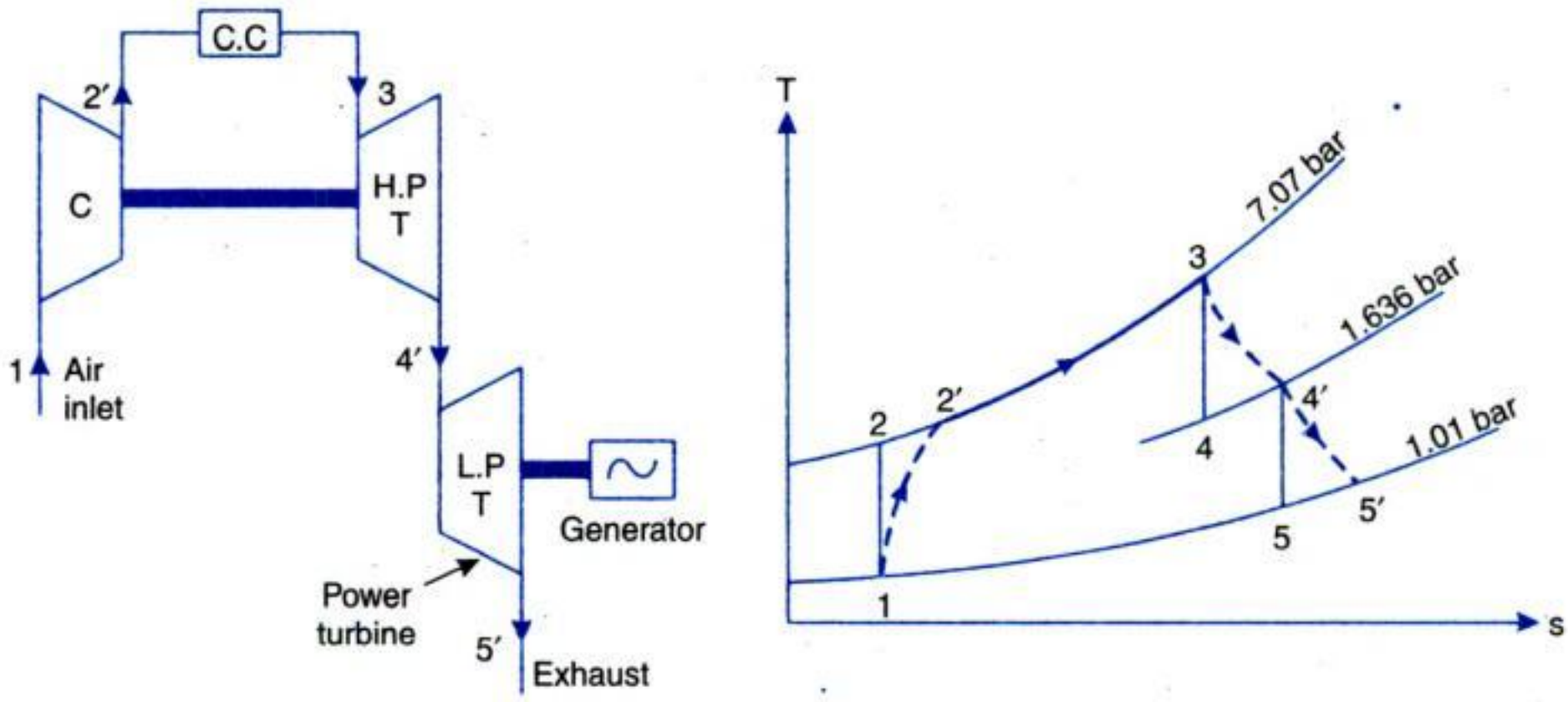

Fig. 25.27

$$\therefore \qquad T_4' = 883 - \frac{262.9}{1.15} = 654.4 \text{ K}$$

i.e., *Temperature of gases entering the power turbine* = **654.4 K.** **(Ans.)**

Again, for H.P. turbine :

$$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4} \quad i.e., \quad 0.85 = \frac{883 - 654.4}{883 - T_4}$$

$$\therefore \qquad T_4 = 883 - \left(\frac{883 - 654.4}{0.85}\right) = 614 \text{ K}$$

Now, considering *isentropic expansion process 3-4,* we have

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma - 1}{\gamma}}$$

or

$$\frac{p_3}{p_4} = \left(\frac{T_3}{T_4}\right)^{\frac{\gamma}{\gamma - 1}} = \left(\frac{883}{614}\right)^{\frac{1.33}{0.33}} = 4.32$$

i.e.,

$$p_4 = \frac{p_3}{4.32} = \frac{7.07}{4.32} = 1.636 \text{ bar}$$

i.e., *Pressure of gases entering the power turbine* = **1.636 bar.** **(Ans.)**

(*ii*) **Net power developed per kg/s mass flow, P :**

To find the power output it is now necessary to calculate T_5'.

The pressure ratio, $\frac{p_4}{p_5}$, is given by $\frac{p_4}{p_3} \times \frac{p_3}{p_5}$

i.e.,

$$\frac{p_4}{p_5} = \frac{p_4}{p_3} \times \frac{p_2}{p_1} = \frac{7}{4.32} = 1.62 \qquad (\because \quad p_2 = p_3 \text{ and } p_5 = p_1)$$

Then,

$$\frac{T_4'}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma - 1}{\gamma}} = (1.62)^{\frac{0.33}{1.33}} = 1.127$$

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$$= \frac{(919.5 - 677) - (471 - 293)}{(919.5 - 619)} = \frac{64.5}{300.5} = \mathbf{0.2146} \text{ or } \mathbf{21.46\%}$$

$\therefore$ **Increase in thermal efficiency** = 21.46 – 16.16 = **5.3%.** **(Ans.)**

Example 25.15. *A 4500 kW gas turbine generating set operates with two compressor stages ; the overall pressure ratio is 9 : 1. A high pressure turbine is used to drive the compressors, and a low-pressure turbine drives the generator. The temperature of the gases at entry to the high pressure turbine is 625°C and the gases are reheated to 625°C after expansion in the first turbine. The exhaust gases leaving the low-pressure turbine are passed through a heat exchanger to heat air leaving the high pressure stage compressor. The compressors have equal pressure ratios and intercooling is complete between the stages. The air inlet temperature to the unit is 20°C. The isentropic efficiency of each compressor stage is 0.8, and the isentropic efficiency of each turbine stage is 0.85, the heat exchanger thermal ratio is 0.8. A mechanical efficiency of 95% can be assumed for both the power shaft and compressor turbine shaft. Neglecting all pressure losses and changes in kinetic energy calculate :*

(i) The thermal efficiency ; *(ii) Work ratio of the plant ;*

(iii) The mass flow in kg/s.

Neglect the mass of the fuel and assume the following :

For air : c_{pa} = 1.005 kJ/kg K and γ = 1.4

For gases in the combustion chamber and in turbines and heat exchanger, c_{pg} = 1.15 kJ/kg K and γ = 1.333.

Solution. Refer Fig. 25.33

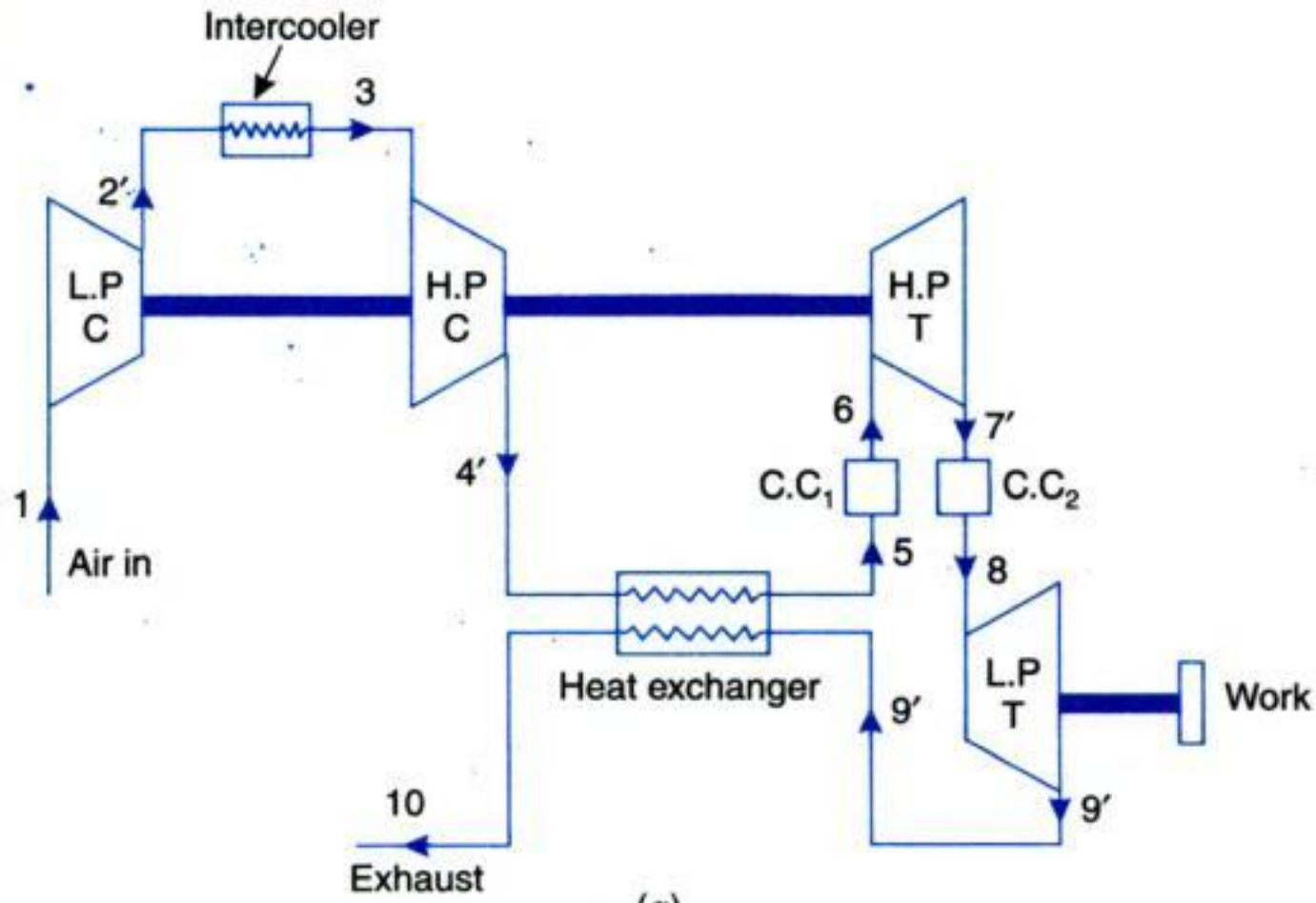
Given : $T_1 = 20 + 273 + 293$ K, $T_6 = T_8 = 625 + 273 = 898$ K

Efficiency of each compressor stage = 0.8

Efficiency of each turbine stage = 0.85, $\eta_{mech.} = 0.95$, $\varepsilon = 0.8$

(*i*) Thermal efficiency, $\eta_{thermal}$:

Since the pressure ratio and the isentropic efficiency of each compressor is the same then the work input required for each compressor is the same since both compressors have the same air inlet temperature *i.e.*, $T_1 = T_3$ and $T_2' = T_4'$.



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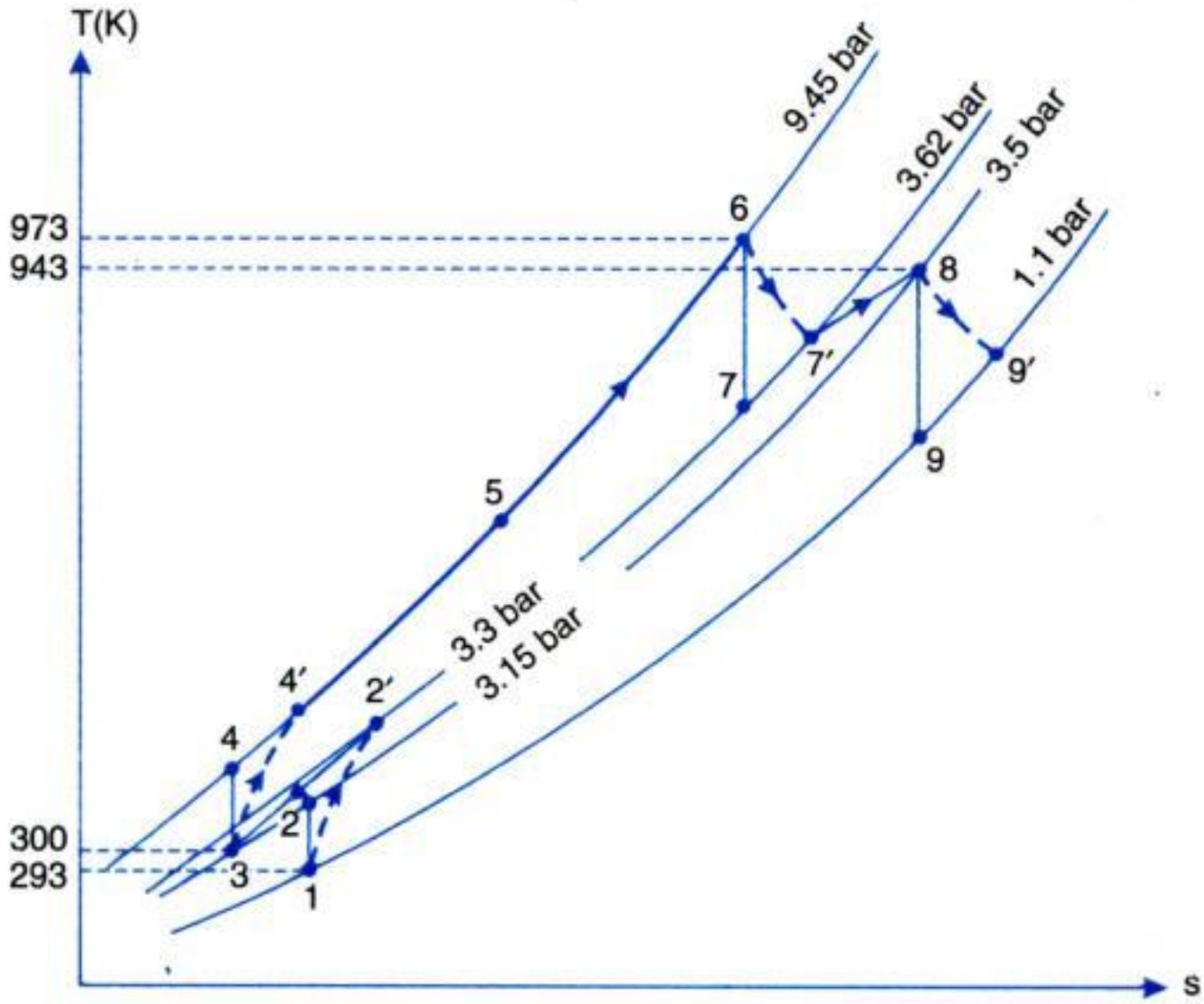

Fig. 25.36

Now, $$\eta_{compressor\ (H.P.)} = \frac{T_4 - T_3}{T_4' - T_3}$$

$$0.82 = \frac{411 - 300}{T_4' - 300}$$

$$\therefore \quad T_4' = \left(\frac{411 - 300}{0.82}\right) + 300 = 435 \text{ K}$$

Similarly, $$\frac{T_6}{T_7} = \left(\frac{p_6}{p_7}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{9.45}{3.62}\right)^{\frac{1.33 - 1}{1.33}} = 1.268$$

$$\therefore \quad T_7 = \frac{T_6}{1.268} = \frac{973}{1.268} = 767 \text{ K}$$

Also, $$\eta_{turbine\ (H.P.)} = \frac{T_6 - T_7'}{T_6 - T_7}$$

$$0.85 = \frac{973 - T_7'}{973 - 767}$$

$$\therefore \quad T_7' = 973 - 0.85\ (973 - 767) = 798 \text{ K}$$

Again, $$\frac{T_8}{T_9} = \left(\frac{p_8}{p_9}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{3.5}{1.1}\right)^{\frac{1.33 - 1}{1.33}} = 1.332$$

$$\therefore \quad T_9 = \frac{T_8}{1.332} = \frac{943}{1.332} = 708 \text{ K}$$

$$\eta_{turbine\ (L.P.)} = \frac{T_8 - T_9'}{T_8 - T_9}$$

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Thrust is the force *produced due to change of momentum.*

Now, absolute velocity of gases leaving aircraft = $(C_j - C_a)$

Absolute velocity of air entering the aircraft = 0

$$\therefore \quad \text{Change of momentum} = \left(1+\frac{m_f}{m_a}\right)(C_j - C_a)$$

Hence, $$\text{thrust, } T = \left(1+\frac{m_f}{m_a}\right)(C_j - C_a) \text{ N/kg of air/s} \qquad ...(25.11)$$

$$T = (C_j - C_a) \text{ N/kg of air/s, } \textit{neglecting mass of fuel} \qquad ...(25.12)$$

Thrust power (T.P.) :

It is defined as the rate at which work must be developed by the engine if the aircraft is to be kept moving at a constant velocity C_a against friction force or drag.

$\therefore$ Thrust power = Forward thrust × speed of aircraft

or $$\text{T.P.} = \left[\left(1+\frac{m_f}{m_a}\right)(C_j - C_a)\right] C_a \text{ W/kg of air} \qquad ...(25.13)$$

$$= (C_j - C_a)\, C_a \text{ W/kg of air if } \textit{mass of fuel is neglected}$$

$$= \frac{(C_j - C_a)\, C_a}{1000} \text{ kW/kg of air} \qquad ...(25.14)$$

Propulsive power (P.P.) :

The *energy required to change the momentum of the mass flow of gas represents the propulsive power.* It is expressed as the *difference between the rate of kinetic energies of the entering air and exit gases.*

Mathematically, $$\text{P.P.} = \Delta\text{K.E.} = \frac{\left(1+\frac{m_f}{m_a}\right) C_j^{\,2}}{2} - \frac{C_a^{\,2}}{2} \text{ W/kg} \qquad ...(25.15)$$

$$= \frac{C_j^{\,2} - C_a^{\,2}}{2} \text{ W/kg, } \textit{neglecting mass of fuel}$$

$$= \frac{C_j^{\,2} - C_a^{\,2}}{2 \times 1000} \text{ kW/kg of air} \qquad ...(25.16)$$

Propulsive efficiency ($\eta_{\text{prop.}}$) :

The *ratio of thrust power to propulsive power is called the propulsive efficiency of the propulsive unit.*

$$\eta_{prop.} = \frac{\text{Thrust power}}{\text{Propulsive power}} = \frac{\left[\left(1+\frac{m_f}{m_a}\right)(C_j - C_a)\right] C_a}{\left[\frac{\left(1+\frac{m_f}{m_a}\right) C_j^{\,2}}{2} - \frac{C_a^{\,2}}{2}\right]} = \frac{2\left[\left(1+\frac{m_f}{m_a}\right)(C_j - C_a)\right] C_a}{\left[\left(1+\frac{m_f}{m_a}\right) C_j^{\,2} - C_a^{\,2}\right]} \qquad ...(25.17)$$

Neglecting the mass of fuel,

$$\eta_{prop.} = \frac{2(C_j - C_a)\, C_a}{C_j^{\,2} - C_a^{\,2}} = \frac{2(C_j - C_a)\, C_a}{(C_j + C_a)(C_j - C_a)}$$

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or $1.005\,(392.2 - 231) = 1.087(1 + 0.0148)\,(963 - T_4')$

or $$T_4' = 963 - \frac{1.005(392.2 - 231)}{1.087(1 + 0.0148)} = 816.13 \text{ K}$$

Let r_{pt} = expansion pressure ratio in turbine *i.e.*, $r_{pt} = \dfrac{p_3}{p_4}$

r_{pj} = expansion pressure ratio in jet tube *i.e.*, $r_{pj} = \dfrac{p_4}{p_5}$

$\therefore$ $$r_{pt} \times r_{pj} = \frac{p_3}{p_4} \times \frac{p_4}{p_5} \simeq 5.2$$

Now, $$\eta_t = \frac{T_3 - T_4'}{T_3 - T_4} \quad \text{or} \quad T_4 = T_3 - \frac{T_3 - T_4'}{\eta_t}$$

$$= 963 - \frac{963 - 816.13}{0.86} = 792.2 \text{ K}$$

Also, $$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{p_3}{p_4}\right)^{\frac{1.33 - 1}{1.33}} \quad \text{or} \quad \frac{963}{792.2} = (r_{pt})^{0.248}$$

or $$r_{pt} = \left(\frac{963}{792.2}\right)^{\frac{1}{0.248}} = 2.197$$

$\therefore$ $$r_{pj} = \frac{p_4}{p_5} = \frac{5.2}{2.197} = 2.366$$

Thus, $$\frac{T_4'}{T_5} = (r_{pj})^{\frac{\gamma - 1}{\gamma}} = (2.366)^{\frac{1.33 - 1}{1.33}} = 1.238$$

or $$T_5 = \frac{T_4'}{1.238} = \frac{816.13}{1.238} = 659.23 \text{ K}$$

Substituting the values in eqn. (*ii*), we get

$$0.9 = \frac{C_j^2/2}{1.087 \times 1000(816.13 - 659.23) + 195^2/2} = \frac{C_j^2/2}{189562.8}$$

$\therefore$ $$C_j = \sqrt{0.9 \times 189562.8 \times 2} = 584.13 \text{ m/s}$$

(*i*) **Overall efficiency, η_0 :**

$$\eta_0 = \frac{\left[\left(1 + \dfrac{m_f}{m_a}\right) C_j - C_a\right] C_a}{\left(\dfrac{m_f}{m_a}\right) \times C} = \frac{[(1 + 0.0148) \times 584.13 - 215]215}{1000 \times 0.0148 \times 42500}$$

$$= \mathbf{0.1291} \text{ or } \mathbf{12.91\%}. \quad \text{(Ans.)}$$

(*ii*) **Rate of air consumption, $\dot{m}_a$:**

Thrust power = Thrust × Velocity of the unit

$$750 = \left[\frac{\left\{\left(1 + \dfrac{m_f}{m_a}\right) C_j - C_a\right\} \dot{m}_a}{1000}\right] C_a$$

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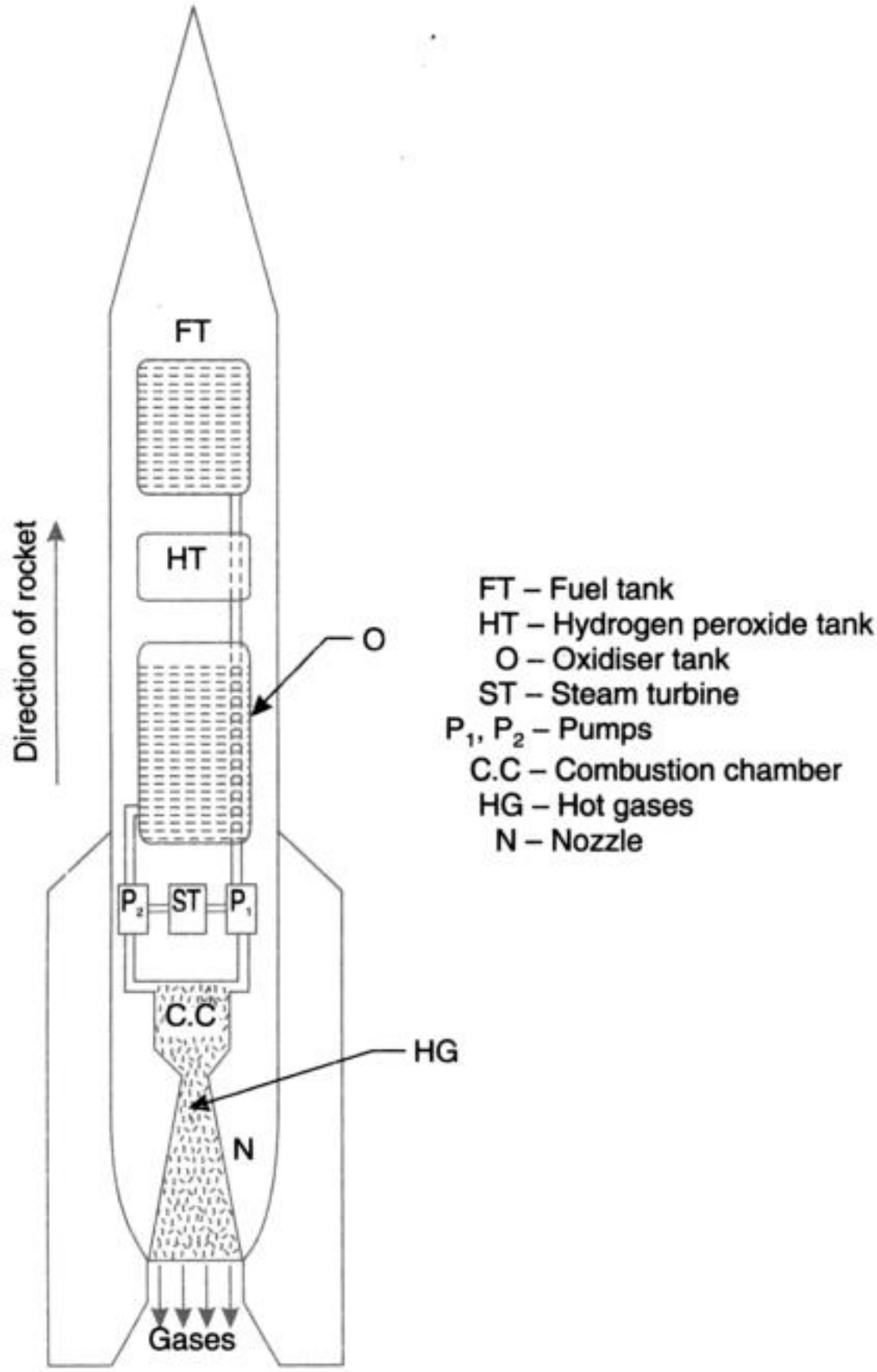

Fig. 25.46. Rocket.

25.8.5.1. Requirements of an ideal rocket propellent

An ideal rocket propellent should have the following *characteristics/properties :*

1. High heat value
2. Reliable smooth ignition
3. Stability and ease of handling and storing
4. Low toxicity and corrosiveness
5. Highest possible density so that it occupies less space.

25.8.5.2. Applications of rockets

The fields of application of rockets are as follows :

1. Long range artillary

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26 Refrigeration

26.1. Fundamentals of refrigeration—Introduction—Elements of refrigeration systems—Refrigeration systems—Co-efficient of performance (C.O.P.)—Standard rating of a refrigeration machine. 26.2. Air refrigeration system—Introduction—Reversed Carnot cycle—Reversed Brayton cycle—Merits and demerits of air refrigeration system. 26.3. Simple vapour compression system—Introduction—Simple vapour compression cycle—Functions of parts of a simple vapour compression system—Vapour compression cycle on temperature-entropy (T-s) diagram—Pressure-enthalpy (p-h) chart—Simple vapour compression cycle on p-h chart—Factors affecting the performance of a vapour compression system—Actual vapour compression cycle—Volumetric efficiency—Mathematical analysis of vapour compression refrigeration. 26.4. Vapour absorption system—Introduction—Simple vapour absorption system—Practical vapour absorption system—Comparison between vapour compression and vapour absorption systems. 26.5. Refrigerants—Classification of refrigerants—Desirable properties of an ideal refrigerant—Properties and uses of commonly used refrigerants—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

26.1. FUNDAMENTALS OF REFRIGERATION

26.1.1. Introduction

Refrigeration *is the science of producing and maintaining temperatures below that of the surrounding atmosphere.* This means the removing of heat from a substance to be cooled. Heat always passes downhill, from a warm body to a cooler one, until both bodies are at the same temperature. Maintaining perishables at their required temperatures is done by refrigeration. Not only perishables but today many human work spaces in offices and factory buildings are air-conditioned and a refrigeration unit is the heart of the system.

Before the advent of mechanical refrigeration water was kept cool by storing it in semi-porous jugs so that the water could seep through and evaporate. The evaporation carried away heat and cooled the water. This system was used by the Egyptians and by Indians in the South-west. Natural ice from lakes and rivers was often cut during winter and stored in caves, straw-lined pits, and later in sawdust-insulated buildings to be used as required. The Romans carried pack trains of snow from Alps to Rome for cooling the Emperor's drinks. Though these methods of cooling all make use of natural phenomena, they were used to maintain a lower temperature in a space or product and may properly be called refrigeration.

In simple, *refrigeration means the cooling of or removal of heat from a system.* The equipment employed to maintain the system at a low temperature is termed as *refrigerating system* and the system which is kept at lower temperature is called *refrigerated system.* Refrigeration is generally produced in one of the following *three ways* :

(*i*) By *melting of a solid.*

(*ii*) By *sublimation of a solid.*

(*iii*) By *evaporation of a liquid.*

Most of the commercial refrigeration is produced by the evaporation of a liquid called *refrigerant. Mechanical refrigeration* depends upon the evaporation of liquid refrigerant and its circuit

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Heat removed by the plant /min

$$= \frac{320.29 \times 20 \times 1000}{8} = 800725 \text{ kJ/h}$$

(*i*) **Capacity of the refrigerating plant** $= \frac{800725}{14000}$ **= 57.19 tonnes.** **(Ans.)**

(*ii*) $T_1 = 25 + 273 = 298$ K

$T_2 = -8 + 273 = 265$ K

∴ **C.O.P. of reversed Carnot cycle**

$$= \frac{T_2}{T_1 - T_2} = \frac{265}{298 - 265} = \mathbf{8.03.} \quad \textbf{(Ans.)}$$

(*iii*) **Power required :**

Actual C.O.P. $= \frac{1}{3} \times$ Carnot C.O.P. $= \frac{1}{3} \times 8.03 = 2.67$

But actual C.O.P. $= \frac{\text{Net refrigerating effect/min}}{\text{Work done /min}} = \frac{R_n}{W}$

$$2.67 = \frac{800725}{W} \text{ kJ/h}$$

∴ $$W = \frac{800725}{2.67} = 299897 \text{ kJ/h} = 83.3 \text{ kJ/s}$$

∴ *Power required to run the plant* **= 83.3 kW.** **(Ans.)**

Example 26.8. *A heat pump is used for heating the interior of a house in cold climate. The ambient temperature is – 5°C and the desired interior temperature is 25°C. The compressor of heat pump is to be driven by a heat engine working between 1000°C and 25°C. Treating both cycles as reversible, calculate the ratio in which the heat pump and heat engine share the heating load.* **(P.U.)**

Solution. Refer Fig. 26.3. *Given :* $T_1 = 1000 + 273 = 1273$ K ; $T_2 = 25 + 273 = 298$ K ;

$T_3 = -5 + 273 = 268$ K ; $T_4 = 25 + 273 = 298$ K

The ratio in which the heat pump and heat engine share the heating load, $\frac{Q_4}{Q_1}$ **:**

Since both the cycles are *reversible*, therefore,

$$\frac{Q_3}{Q_4} = \frac{T_3}{T_4} \text{ and } \frac{Q_2}{Q_1} = \frac{T_2}{T_1}$$

or $$\frac{Q_3}{Q_4} = \frac{268}{298} \text{ or } Q_3 = \frac{268}{298} Q_4 \text{ and } \frac{Q_2}{Q_1} = \frac{298}{1273}$$

Heat engine drives the heat pump,

∴ $$W = (Q_1 - Q_2) = Q_4 - Q_3$$

Dividing both sides by Q_1, we have

$$1 - \frac{Q_2}{Q_1} = \frac{Q_4 - Q_3}{Q_1}$$

$$1 - \frac{298}{1273} = \frac{Q_4 - \frac{268}{298} Q_4}{Q_1}$$

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(*i*) Mass of air circulated per minute, m :

For the *expansion process 1–2*, we have

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1000}{100}\right)^{\frac{1.4-1}{1.4}} = 1.9306$$

or $$T_2 = \frac{T_1}{1.9306} = \frac{303}{1.9306} = 156.9 \text{ K}$$

Refrigerating effect per kg = $1 \times c_p (T_3 - T_2) = 1.005 (268 - 156.9) = 111.66$ kJ/kg

$\therefore$ Mass of air circulated per minute = $\frac{\text{Refrigerating effect}}{\text{Refrigerating effect per kg}}$

$$= \frac{2000}{111.66} = \mathbf{17.91 \text{ kg/min.}} \quad \textbf{(Ans.)}$$

(*ii*) Compressor work ($W_{comp.}$), expander work ($W_{exp.}$) and cycle work (W_{cycle}) :

For *compression process 3–4*, we have

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1000}{10}\right)^{\frac{1.4-1}{1.4}} = 1.9306$$

or $$T_4 = 268 \times 1.9306 = \mathbf{517.4 \text{ K.}} \quad \textbf{(Ans.)}$$

Compressor work, $W_{comp.}$: $\frac{\gamma}{\gamma-1} mR (T_4 - T_3)$

$$= \frac{1.4}{1.4-1} \times 17.91 \times 0.287 (517.4 - 268)$$

$$= \mathbf{4486.85 \text{ kJ/min.}} \quad \textbf{(Ans.)}$$

Expander work, $W_{exp.}$: $\frac{\gamma}{\gamma-1} mR (T_1 - T_2)$

$$= \frac{1.4}{1.4-1} \times 17.91 \times 0.287 (303 - 156.9)$$

$$= \mathbf{2628.42 \text{ kJ/min.}} \quad \textbf{(Ans.)}$$

Cycle work, W_{cycle} : $W_{comp.} - W_{exp.}$

$$= 4486.85 - 2628.42 = \mathbf{1858.43 \text{ kJ/min.}} \quad \textbf{(Ans.)}$$

(*iii*) C.O.P. and power required (P) :

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work required}} = \frac{2000}{1858.43} = \mathbf{1.076} \quad \textbf{(Ans.)}$$

Power required, P = Work per second = $\frac{1858.43}{60}$ kJ/s or kW = **30.97 kW.** **(Ans.)**

☞**Example 26.11.** *A refrigerating machine of 6 tonnes capacity working on Bell-Coleman cycle has an upper limit of pressure of 5.2 bar. The pressure and temperature at the start of the compression are 1.0 bar and 16°C respectively. The compressed air cooled at constant pressure to a temperature of 41°C enters the expansion cylinder. Assuming both expansion and compression processes to be adiabatic with $\gamma = 1.4$, calculate :*

(i) Co-efficient of performance.

(ii) Quantity of air in circulation per minute.

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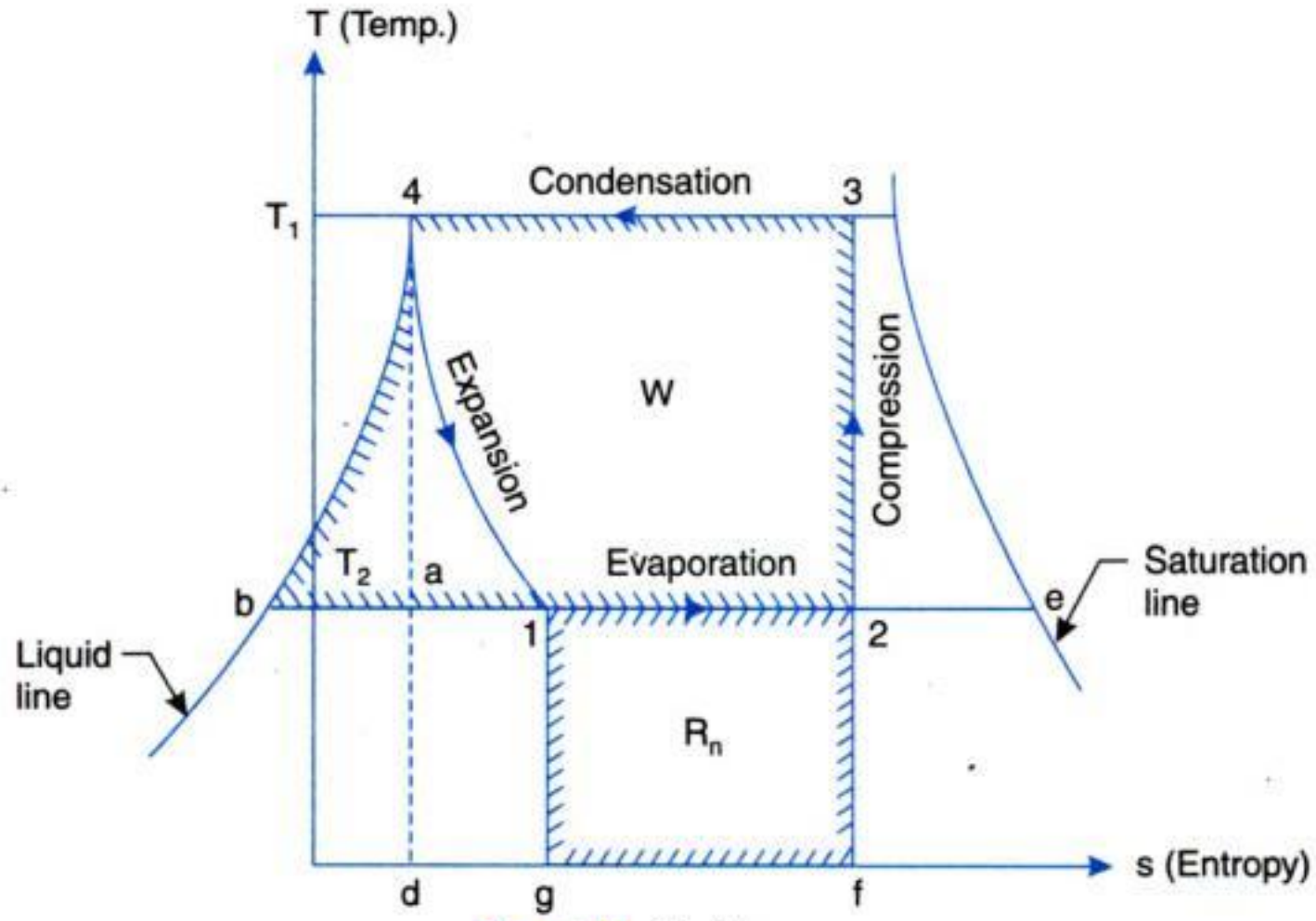

Fig. 26.12. *T-s* diagram.

26.3.5. Pressure-Enthalpy (p-h) Chart

The diagram commonly used in the analysis of the refrigeration cycle are :

(*i*) Pressure-enthalpy (*p-h*) chart (*ii*) Temperature-entropy (*T-s*) chart.

Of the two, the pressure-enthalpy diagram seems to be the more useful.

The condition of the refrigerant in any thermodynamic state can be represented as a point on the *p-h* chart. The point on the *p-h* chart that represents the condition of the refrigerant in any one particular thermodynamic state may be located if any two properties of the refrigerant for that state are known, the other properties of the refrigerant for that state can be determined directly from the chart for studying the performance of the machines.

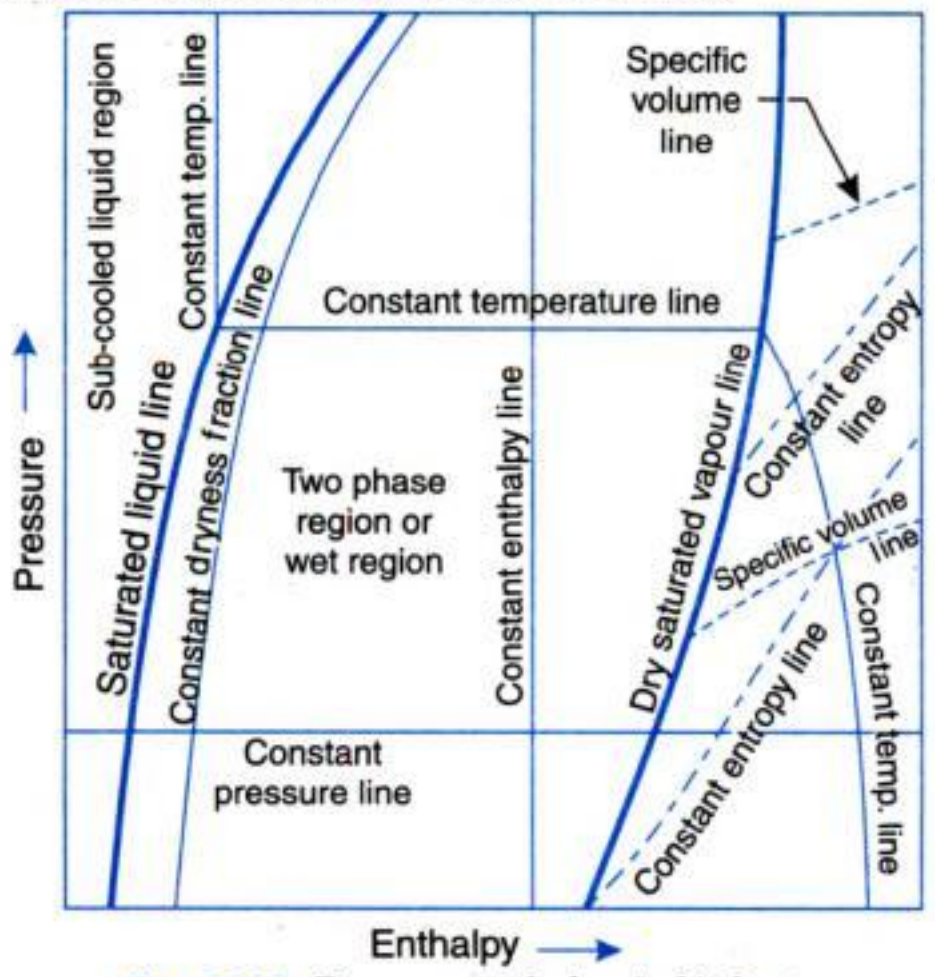

Fig. 26.13. Pressure enthalpy (*p-h*) chart.

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26.3.9. Volumetric Efficiency

A compressor which is theoretically perfect would have neither clearance nor losses of any type and would pump on each stroke a quantity of refrigerant equal to piston displacement. No actual compressor is able to do this, since it is impossible to construct a compressor without clearance or one that will have no wire drawing through the suction and discharge valves, no superheating of the suction gases upon contact with the cylinder walls, or no leakage of gas past the piston or the valves. All these factors effect the volume of gas pumped or the capacity of the compressor, some of them affect the H.P. requirements per tonne of refrigeration developed.

'Volumetric efficiency' is defined as the *ratio of actual volume of gas drawn into the compressor* (at evaporator temperature and pressure) on *each stroke to the piston displacement*. If the effect of *clearance alone* is considered, the resulting expression may be termed *clearance volumetric efficiency*. The expression used for grouping into one constant all the factors affecting efficiency may be termed *total volumetric efficiency*.

Clearance volumetric efficiency. *'Clearance volume' is the volume of space between the end of the cylinder and the piston when the latter is in dead centre position*. The clearance volume is expressed as a percentage of piston displacement. In Fig. 26.20 the piston displacement is shown as 4′-1.

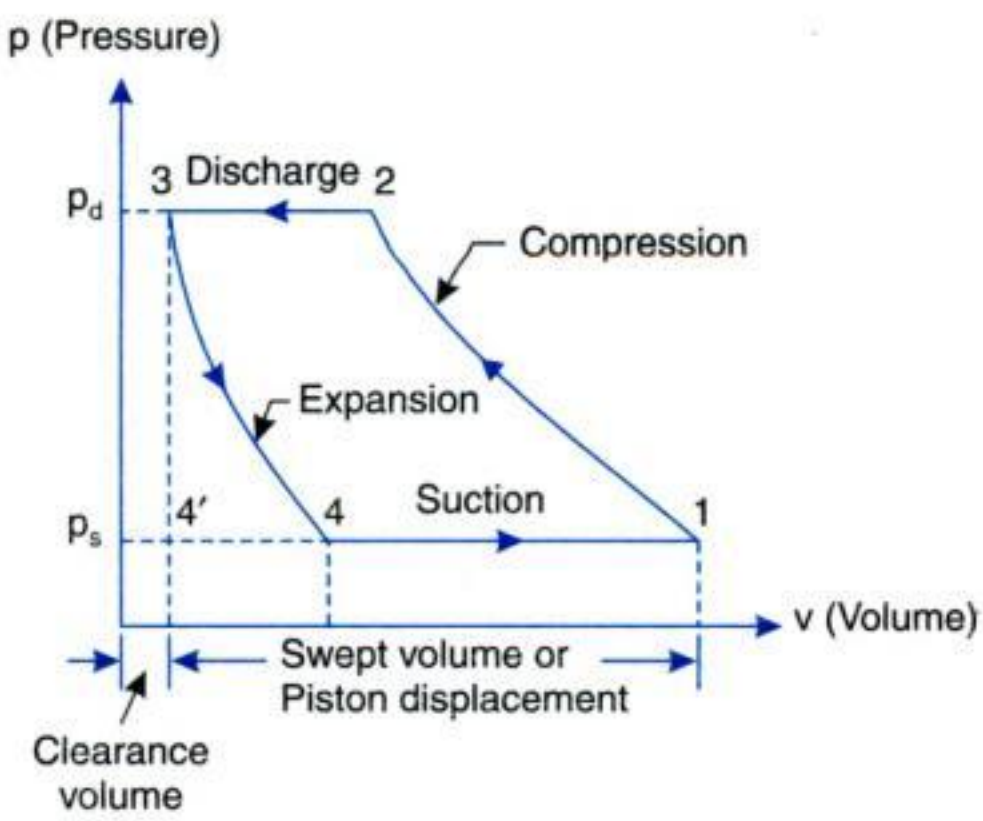

Fig. 26.20

During the suction stroke 4′-1, the vapour filled in clearance space at a discharge pressure p_d expands along 3-4 and the suction valve opens only when pressure has dropped to suction pressure p_s, therefore actual volume sucked will be $(v_1 - v_4)$ while the swept volume is $(v_1 - v_4')$. The *ratio of actual volume of vapour sucked to the piston displacement is defined as clearance volumetric efficiency*.

Thus,

$$\text{Clearance volumetric efficiency, } \eta_{cv} = \frac{v_1 - v_4}{v_1 - v_4'} = \frac{v_1 - v_4}{v_1 - v_3} \qquad (\because \quad v_4' = v_3)$$

Considering *polytropic expansion process 3-4*, we have

$$p_s v_4^{\,n} = p_d v_3^{\,n}$$

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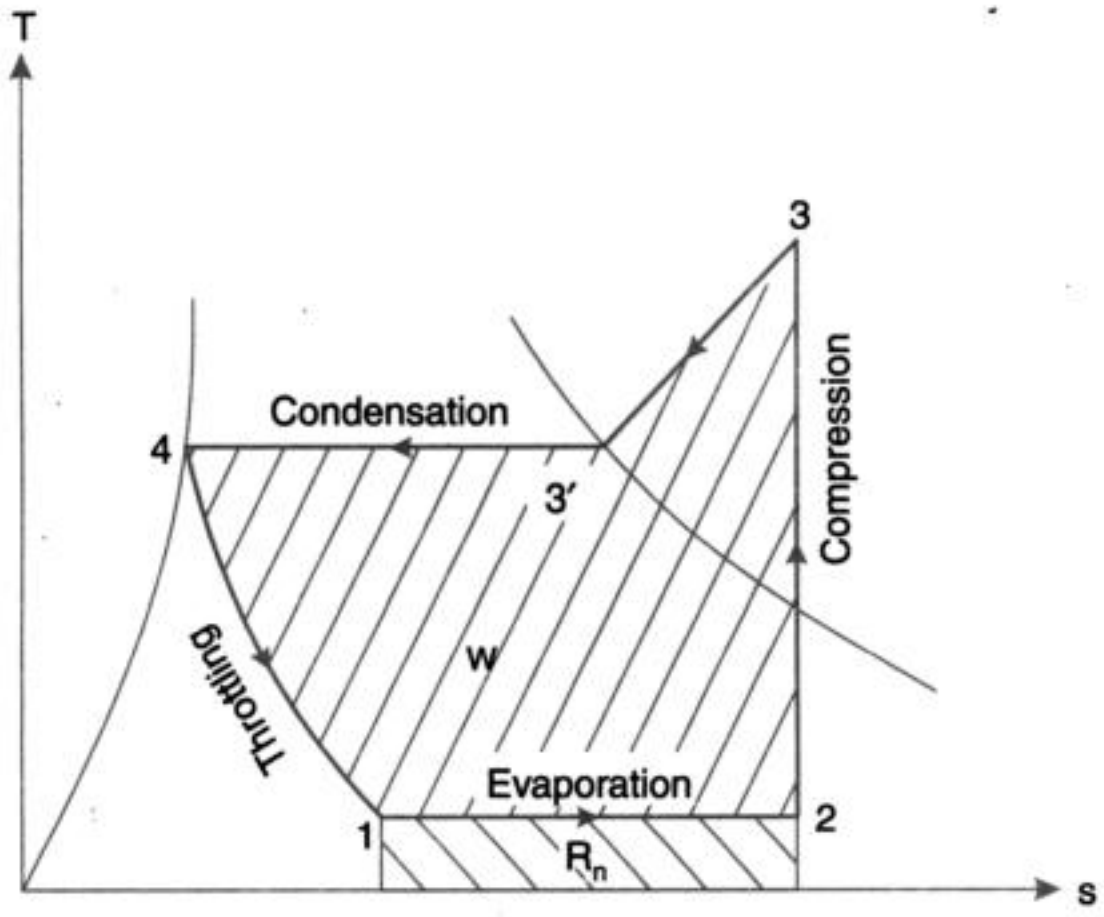

Fig. 26.26

(*ii*) **The C.O.P. :**

$$\text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{183.19 - 74.59}{209.41 - 183.19} = \mathbf{4.142.} \quad \text{(Ans.)}$$

(*iii*) **The power required to drive the compressor, P :**

$$P = \dot{m}\ (h_3 - h_2) = 0.179\ (209.41 - 183.19) = \mathbf{4.69\ kW.} \quad \text{(Ans.)}$$

(*iv*) **The rate of heat rejection to the condenser :**

The rate of heat rejection to the condenser

$$= \dot{m}\ (h_3 - h_4) = 0.179\ (209.41 - 74.59) = \mathbf{24.13\ kW.} \quad \text{(Ans.)}$$

Example 26.16. (*i*) *What are the advantages of using an expansion valve instead of an expander in a vapour compression refrigeration cycle ?*

(*ii*) *Give a comparison between centrifugal and reciprocating compressors.*

(*iii*) *An ice-making machine operates on ideal vapour compression refrigeration cycle using refrigerant R-12. The refrigerant enters the compressor as dry saturated vapour at – 15°C and leaves the condenser as saturated liquid at 30°C. Water enters the machine at 15°C and leaves as ice at – 5°C. For an ice production rate of 2400 kg in a day, determine the power required to run the unit. Find also the C.O.P. of the machine. Use refrigerant table only to solve the problem. Take the latent heat of fusion for water as 335 kJ/kg.*

Solution. (*i*) If an expansion cylinder is used in a vapour compression system, *the work recovered would be extremely small, in fact not even sufficient to overcome the mechanical friction.* It will not be possible to gain any work. Further, the *expansion cylinder is bulky.* On the other hand the *expansion valve is a very simple and handy device, much cheaper than the expansion cylinder.* It *does not need installation, lubrication or maintenance.*

The expansion valve *also controls the refrigerant flow rate according to the requirement, in addition to serving the function of reducting the pressure of the refrigerant.*

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Example 26.21. *A refrigeration cycle uses Freon-12 as the working fluid. The temperature of the refrigerant in the evaporator is – 10°C. The condensing temperature is 40°C. The cooling load is 150 W and the volumetric efficiency of the compressor is 80%. The speed of the compressor is 720 r.p.m. Calculate the mass flow rate of the refrigerant and the displacement volume of the compressor.*

Properties of Freon-12

Temperature (°C)	*Saturation pressure (MPa)*	*Enthalpy (kJ/kg)*		*Specific volume (m^3/kg) Saturated vapour*
		Liquid	*Vapour*	
– 10	*0.22*	*26.8*	*183.0*	*0.08*
40	*0.96*	*74.5*	*203.1*	*0.02*

(GATE, 1995)

Solution. *Given* : Cooling load = 150 W ; $\eta_{vol.} = 0.8$; N = 720 r.p.m.

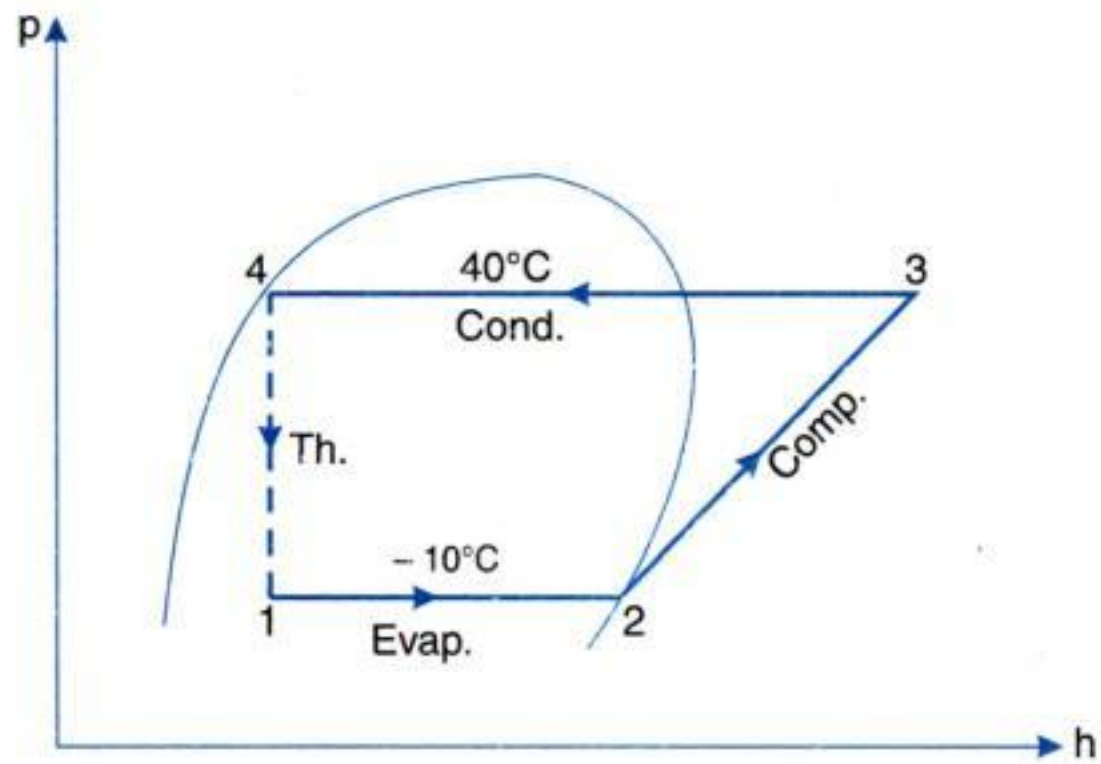

Fig. 26.32

Mass flow rate of the refrigerant $\dot{m}$:

Refrigerating effect = $h_2 - h_1$

$= 183 - 74.5 = 108.5$ kJ/kg

Cooling load $= \dot{m} \times (108.5 \times 1000) = 150$

or $$\dot{m} = \frac{150}{108.5 \times 1000} = \mathbf{0.001382\ kJ/s.}$$ **(Ans.)**

Displacement volume of the compressor :

Specific volume at entry to compressor,

$v_2 = 0.08\ m^3/kg$ (From table)

$\therefore$ Displacement volume of compressor $= \dfrac{\dot{m} v_2}{\eta_{vol.}} = \dfrac{0.001382 \times 0.08}{0.8}$

$= \mathbf{0.0001382\ m^3/s.}$ **(Ans.)**

Example 26.22. *In a simple vapour compression cycle, following are the properties of the refrigerant R-12 at various points :*

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(*v*) *C.O.P.*

(*vi*) *Heat removed through condenser.*

(*vii*) *Theoretical bore and stroke of the compressor.*

Neglect valve throttling and clearance effect.

Solution. Refer Fig. 26.37. Following the procedure as given in the previous example plot the points 1, 2, 3 and 4 on *p-h* chart for freon-22. The following values are obtained :

$$h_2 = 615 \text{ kJ/kg}$$

$$h_3 = 664 \text{ kJ/kg}$$

$$h_4 = h_1 = 446 \text{ kJ/kg}$$

$$v_2 = 0.14 \text{ m}^3\text{/kg}.$$

(*i*) **Refrigerating effect per kg** = $h_2 - h_1 = 615 - 446 =$ **169 kJ/kg.** **(Ans.)**

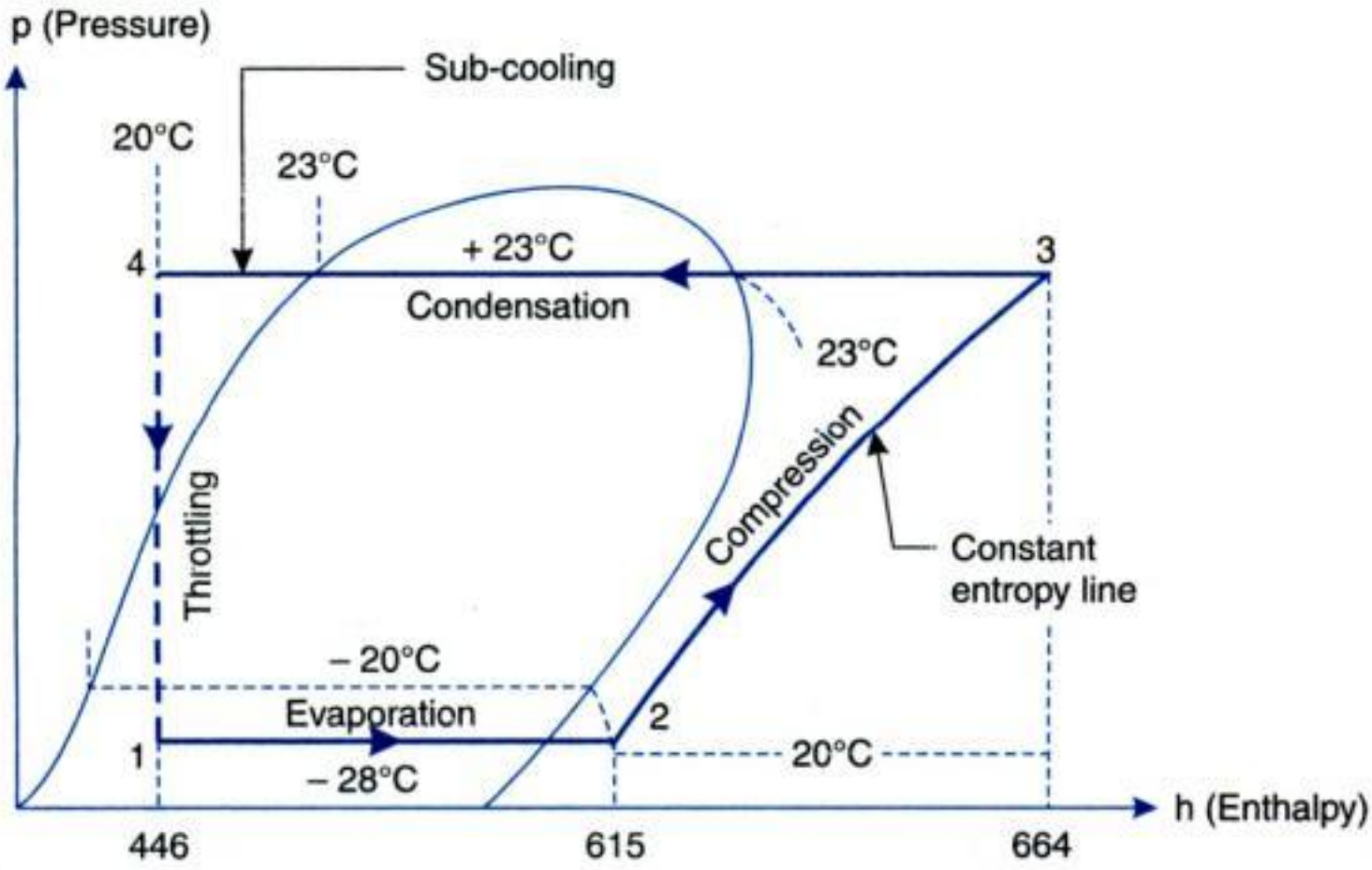

Fig. 26.37

(*ii*) **Mass of refrigerant to be circulated per minute,**

$$m = \frac{20 \times 14000}{169 \times 60} = \textbf{27.6 kg/min.} \quad \textbf{(Ans.)}$$

(*iii*) **Theoretical piston displacement**

= Specific volume at suction × Mass of refrigerant used/min

= $0.14 \times 27.6 = 3.864 \text{ m}^3\text{/min}$

(*iv*) **Theoretical power**

$$= m \times (h_3 - h_2) = \frac{27.6}{60} (664 - 615) = 22.54 \text{ kJ/s}$$

= **22.54 kW.** **(Ans.)**

(*v*) $$\textbf{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{615 - 446}{664 - 615} = \textbf{3.45.} \quad \textbf{(Ans.)}$$

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5. Methyl chloride (CH_3Cl)

Properties :

(*i*) It is a colourless liquid with a faint, sweet, non-irritating odour.

(*ii*) It has liquid specific gravity of 1.002 at atmospheric pressure.

(*iii*) It is neither flammable nor toxic.

Uses :

It has been used in the past in both domestic and commercial applications. It should never be used with aluminium.

6. R-11 (Trichloro monofluoro methane)

Properties :

(*i*) It is composed of one carbon, three chlorine and one fluorine atoms (or parts by weight) and is *non-corrosive, non-toxic* and *non-flammable.*

(*ii*) It dissolves natural rubber.

(*iii*) It has a boiling point of – 24°C.

(*iv*) It mixes completely with mineral lubricating oil under all conditions.

Uses :

It is employed for 50 tonnes capacity and over in small office buildings and factories. A centrifugal compressor is used in the plants employing this refrigerant.

• *Its leakage is detected by a halide torch.*

7. R-12 (Dichloro-difluoro methane) or Freon-12

Properties :

(*i*) It is *non-toxic, non-flammable*, and *non-explosive*, therefore it is *most suitable refrigerant.*

(*ii*) It is fully oil miscible therefore it simplifies the problem of oil return.

(*iii*) The operating pressures of R-12 in evaporator and condenser under *standard tonne of refrigeration* are 1.9 bar abs. and 7.6 bar abs. (app.).

(*iv*) Its latent heat at – 15°C is 161.6 kJ/kg.

(*v*) C.O.P. = 4.61.

(*vi*) It does not break even under the extreme operating conditions.

(*vii*) It condenses at moderate pressure and under atmospheric conditions.

Uses :

1. It is suitable for high, medium and low temperature applications.
2. It is used for domestic applications.
3. It is *excellent electric insulator therefore it is universally used in sealed type compressors.*

8. R-22 (Monochloro-difluro methane) or Freon-22

R-22 refrigerant is superior to R-12 in many respects. It has the following properties and uses :

Properties :

(*i*) The compressor displacement per tonne of refrigeration with R-22 is 60% less than the compressor displacement with R-12 as refrigerant.

(*ii*) R-22 is miscible with oil at condenser temperature but tries to separate at evaporator temperature when the system is used for very low temperature applications (– 90°C). Oil

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14. State merits and demerits of 'Vapour compression system' over 'Air refrigeration system'.
15. State the functions of the following parts of a simple vapour compression system :
    (*i*) Compressor, (*ii*) Condenser,
    (*iii*) Expansion valve, and (*iv*) Evaporator.
16. Show the vapour compression cycle on 'Temperature-Entropy' (*T*-*s*) diagram for the following cases :
    (*i*) When the vapour is dry and saturated at the end of compression.
    (*ii*) When the vapour is superheated after compression.
    (*iii*) When the vapour is wet after compression.
17. What is the difference between 'Wet compression' and 'Dry compression' ?
18. Write a short note on 'Pressure Enthalpy (*p*-*h*) chart'.
19. Show the simple vapour compression cycle on a *p*-*h* chart.
20. Discuss the effect of the following on the performance of a vapour compression system :
    (*i*) Effect of suction pressure (*ii*) Effect of delivery pressure
    (*iii*) Effect of superheating (*iv*) Effect of sub-cooling of liquid
    (*v*) Effect of suction temperature and condenser temperature.
21. Show with the help of diagrams, the difference between theoretical and actual vapour compression cycles.
22. Define the terms 'Volumetric efficiency' and 'Clearance volumetric efficiency'.
23. Derive an expression for 'Clearance volumetric efficiency'.
24. Explain briefly the term 'Total volumetric efficiency'.
25. Explain briefly simple vapour absorption system.
26. Give the comparison between a vapour compression system and a vapour absorption system.

UNSOLVED EXAMPLES

1. The co-efficient of performance of a Carnot refrigerator, when it extracts 8350 kJ/min from a heat source, is 5. Find power required to run the compressor. [**Ans.** 27.83 kW]
2. A reversed cycle has refrigerating C.O.P. of 4,
    (*i*) Determine the ratio T_1/T_2 ; and
    (*ii*) If this cycle is used as heat pump, determine the C.O.P. and heat delivered.
    [**Ans.** (*i*) 1.25 ; (*ii*) 50 kW, 5]
3. An ice plant produces 10 tonnes of ice per day at 0°C, using water at room temperature of 20°C. Estimate the power rating of the compressor motor if the C.O.P. of the plant is 2.5 and overall electromechanical efficiency is 0.9.
    Take latent heat of freezing for water = 335 kJ/kg
    Specific heat of water = 4.18 kJ/kg. [**Ans.** 21.44 kW]
4. An air refrigeration system operating on Bell Coleman cycle, takes in air from cold room at 268 K and compresses it from 1.0 bar to 5.5 bar. The index of compression being 1.25. The compressed air is cooled to 300 K. The ambient temperature is 20°C. Air expands in an expander where the index of expansion is 1.35. Calculate : (*i*) C.O.P. of the system (*ii*) Quantity of air circulated per minute for production of 1500 kg of ice per day at 0°C from water at 20°C. (*iii*) Capacity of the plant in terms of kJ/s.
    Take c_p = 4.18 kJ/kg K for water, c_p = 1.005 kJ/kg K for air
    Latent heat of ice = 335 kJ/kg. [**Ans.** (*i*) 1.974 ; (*ii*) 5.814 kg/min ; (*iii*) 7.27 kJ/s]
5. The temperature in a refrigerator coil is 267 K and that in the condenser coil is 295 K. Assuming that the machine operates on the reversed Carnot cycle, calculate :
    (*i*) $C.O.P._{(ref.)}$
    (*ii*) The refrigerating effect per kW of input work.
    (*iii*) The heat rejected to the condenser. [**Ans.** (*i*) 9.54 ; (*ii*) 9.54 kW ; (*iii*) 10.54 kW]

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Disadvantages of central system :

The disadvantage of the central system is that it results in *large size ducts which are costly and occupy large space.*

27.2.3.2. Zoned System

When in a building several rooms or floors are to be served, it is necessary to consider means by which the varying heat gains in the different departments can be dealt with. Some rooms may have some, and others not ; some may be crowded and others empty ; again some may contain heat-producing equipment. Variations in requirements of this kind are the most common case with which air-conditioning has to deal, and for this central system is unsuitable. *Zoned system is one approach.*

In zoned system the building is divided into zones such that as nearly as possible conditions may be expected to exist. Each zone is provided with its own local recirculating fan and booster cooler or heater, and this unit receives fresh air supply conditioned to some average temperature and corrected for humidity by means of what is in effect a central plant. Such an arrangement is shown in Fig. 27.3 (next page). In this case the central plant is on the roof. The circulating units are fixed overhead adjacent to the corridor, taking then return air there from, and distributing ducts are run above the corridor false ceiling delivering into the various rooms, floor by floor each floor, constituting a separate zone. The return air from the room passes through grilles into the corridor which acts as the returning air collecting duct. The cooling or heating booster coils could be served from circulating water mains, each coil being controlled locally according to the requirements of the zone served.

27.2.3.3. Unitary Systems

The components of unitary air-conditioned system are assembled in the factory itself. These assembled units are usually installed in or immediately adjacent to a zone or space to be conditioned. The package units are available in the size ranges of greatest usage to obtain economics of factory production.

Unitary system is commonly preferred for 15 tonnes capacity or above 200 m^3/min. of flow. The units of even 100 tonnes capacity have also been manufactured.

Various *factory assembled units* available are :

(*i*) Attic (or exhaust) fan.
(*ii*) Remote units.
(*iii*) Self contained units.
(*iv*) Room air-conditioners.
(*v*) Unit air coolers.

(*i*) **Attic fans.** An attic or exhaust fan is a cooling unit without any heat transfer element such as a cooling coil. When the sun sets the temperature of outdoor air reduces to cool levels whereas the indoor temperatures are high. To reduce the inside temperature an attic fan is placed in the attic. It *draws outdoor air into several rooms of the building through various doors and windows and finally discharges from attic to outdoors.* Consequently *a circulation of cool outdoor air is set up in the building. A propeller type fan* is usually recommended as it can handle large volume of air at low pressure efficiently.

(*ii*) **Remote units.** A system in which air handling unit is separated from the condensing unit is called a *remote system*. The conditioning or air handling unit is called *remote unit*. It consists of a fan (either propeller or centrifugal type) with its driving motor, cooling coil, heatin coil, filters, drip pan, louvers etc., with or without the duct connections at the outlet. Remote units are available in capacities ranging from 2 to 100 TR. These units are available for floor mounting or for suspension from ceiling. Some remote units have air-washing and coil wetting features.

(*iii*) **Self contained units.** In a self contained unit the condensing unit and other functional elements (such as coil and fans) are encased in the same cabinet. Fresh air can be introduced if required. The discharge from the casing may be free pressure type (*i.e.,* with or without duct work). Proper means should be adopted to cool the compressor.

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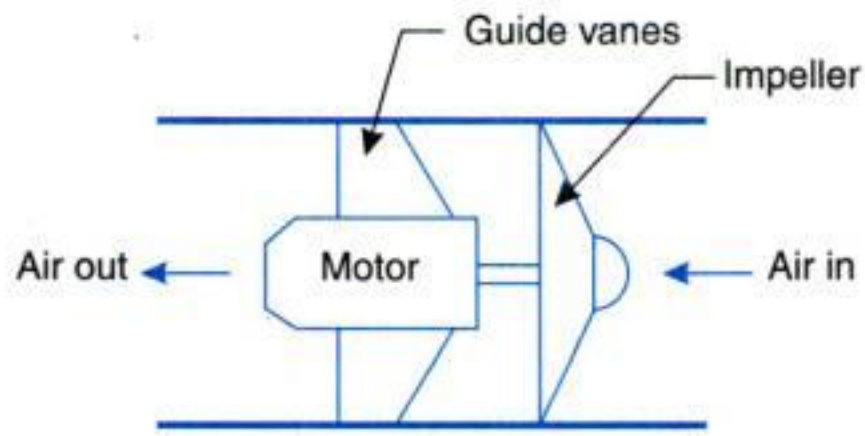

Fig. 27.13. Axial flow fan.

drum-type housing equipped with satisfactory directional vanes, it is called a *vane-axial-flow fan.* If vanes are not included, the term *tube-axial* is used. These fans can be applied in air-conditioning work and operate at good efficiencies and satisfactory noise-level if properly selected for air delivery and system resistance.

Centrifugal or radial-flow fans. In case a *system has duct work, centrifugal fans have to be used as the static pressure drop is considerable*. But when there is no duct work, propellers or axial flow fans can be used. Nevertheless, *in window type and package units drum-type centrifugal fans only are used, whereas most exhaust fans are of axial type*. A centrifugal fan possesses the following *advantages* :

1. Quiet and efficient operation at high pressures.

2. It is easy to connect centrifugal fan inlet to large apparatus sections and its outlet to smaller supply duct sections.

A centrifugal or a radial flow fan consists of an impeller running in a casing, normally of a volute shape. The air enters axially and is discharged at the periphery.

27.3.2.3. Air Washer

Fig. 27.14 shows a schematic diagram of an air washer. Here air flows through a spray of water and during this flow air may be cooled to heated, humidified or dehumidified, or simply adiabatically saturated, depending on the mean surface temperature of water. The water is, accordingly, externally cooled or heated or simply recirculated by a pump. In case of humidification of air loss of water is compensated by make-up water. In order to minimise loss of water droplets eliminator plates are used.

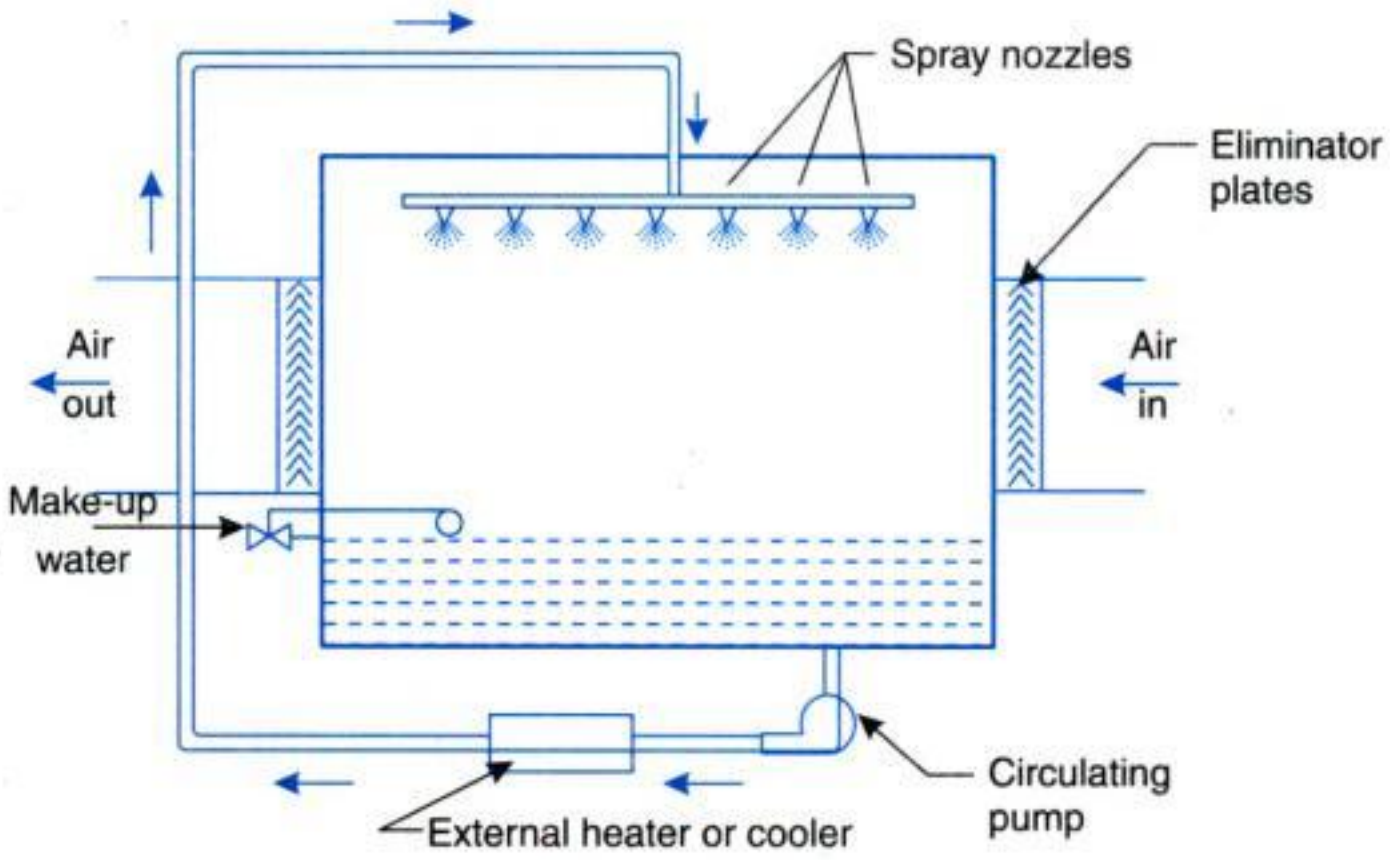

Fig. 27.14. Air washer.

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2. Diffuse or sky radiation. A large part of the sun's radiation is scattered, reflected back into space and absorbed by the earth's atmosphere. A part of this radiation is re-radiated and reaches the earth's surface uniformly from all directions. It is called *diffuse or sky radiation*. It *does not normally change with orientation of surface*.

The total solar radiation reaching a surface is equal to the *sum of the direct and diffuse radiation*.

27.5.5. Solar Heat Gain Through Glass

Glass which is a major cladding material of most buildings, provides the most direct route for entry of solar radiation. For these reasons, the proper estimation of heat gain through glass is necessary.

Heat transmitted through a glass surface depends on the wavelength of radiation and physical and chemical characteristics of the glass. Part of the radiation is *absorbed*, part is *reflected* and the rest is *transmitted*. Glass is opaque to the radiant energy emitted from sources below 200°C. Thus glass has high *transmittivity for short wavelength and low transmittivity for long wavelength radiation.*

Direct solar heat gain can be *reduced* by using different types of glass, glass construction and shades as given below :

(*i*) Double pane glass reduces the solar heat by 10% to 20%.

(*ii*) Special heat absorbing glass reduces the solar heat by 25%.

(*iii*) Stained glass can reduces it upto 65% depending upon its colour.

(*iv*) Shading devices installed on the outside of windows reduce sun load upto 15%.

(*v*) Ventilation blinds and curtain shades reduce it by 30 to 35%.

27.5.6. Heat Flow Through Building Structures (Thermal Barriers)

Refer Fig. 27.24. Transfer of heat by conduction (Q) from outside to inside under steady state due to the temperature difference is given by the equation :

$$Q = UA\,(t_0 - t_i) \qquad ...(27.2)$$

where, Q = Heat transfer rate kJ/h,

U = Overall co-efficient of heat transfer,

A = Area, m^2,

t_0 = Outside temperature,

t_i = Inside temperature,

and, *overall thermal resistance*,

$$\frac{1}{U} = \frac{1}{f_0} + \frac{x_1}{K_1} + \frac{x_2}{K_2} + ...+ \frac{1}{f_i} \qquad ...(27.3)$$

where, f_0 = Film co-efficient on outside wall,

x = Thickness of material,

K = Conductivity of material, and

f_i = Film co-efficient on inside wall.

Subscripts 1, 2 etc., refer to successive layers of materials which make up the walls.

The film *co-efficient varies directly with wind velocity*.

While considering the steady state conduction, the outdoor air temperature should be assumed to be the mean average during the day. This assumption of steady state is justified only in cases where

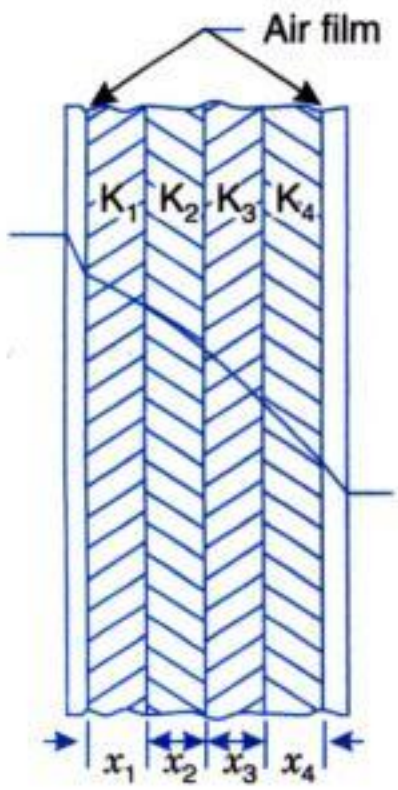

Fig. 27.24

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In liquids, the mechanism of heat is nearer to that of gases. However, the molecules are more closely spaced and intermolecular forces come into play.

(*ii*) **Convection.** *'Convection' is the transfer of heat within a fluid by mixing of one portion of the fluid with another.*

- Convection is possible only in a fluid medium and is *directly linked* with the *transport of medium itself.*
- Convection constitutes the *macroform* of the heat transfer since macroscopic particles of a fluid moving in space cause the heat exchange.
- The effectiveness of heat transfer by convection depends largely upon the mixing motion of the fluid.

This mode of heat transfer is met with in situations where energy is transferred as heat to a flowing fluid at any surface over which flow occurs. This mode is *basically conduction in a very thin fluid layer at the surface and then mixing caused by the flow.* The heat flow depends on the properties of fluid and is independent of the properties of the material of the surface. However, the shape of the surface will influence the flow and hence the heat transfer.

Free or natural convection. *Free or natural convection occurs where the fluid circulates by virtue of the natural differences in densities of hot and cold fluids ; the denser portions of the fluid move downward because of the greater force of gravity, as compared with the force on the less dense.*

Forced convection. *When the work is done to blow or pump the fluid, it is said to be forced convection.*

(*iii*) **Radiation.** *'Radiation' is the transfer of heat through space or matter by means other than conduction or convection.*

Radiation heat is thought of as *electromagnetic waves or quanta* (as convenient) an emanation of the same nature as light and radio waves. *All bodies radiate heat ; so a transfer of heat by radiation occurs because hot body emits more heat than it receives and a cold body receives more heat than it emits.* Radiant *energy* (being electromagnetic radiation) *requires no medium for propagation and will pass through a vacuum.*

Note. The rapidly oscillating molecules of the hot body produce electromagnetic waves in hypothetical medium called *ether.* These waves are identical with light waves, radio waves and X-rays, differ from them only in *wavelength* and travel with an approximate velocity of 3×10^8 m/s. These waves carry energy with them and *transfer it to the relatively slow-moving molecules of the cold body* on which they happen to fall. The molecular energy of the later increases and results in a rise of its temperature. Heat travelling by radiation is known as *radiant heat.*

The properties of radiant heat in general, are similar to those of light. Some of the properties are :

(*i*) It does not require the presence of a material medium for its transmission.

(*ii*) Radiant heat can be reflected from the surfaces and obeys the ordinary laws of reflection.

(*iii*) It travels with velocity of light.

(*iv*) Like light, it shows interference, diffraction and polarisation etc.

(*v*) It follows the law of inverse square.

The wavelength of heat radiations is longer than that of light waves, hence they are invisible to the eye.

28.2. HEAT TRANSFER BY CONDUCTION

28.2.1. Fourier's Law of Heat Conduction

Fourier's law of heat conduction is an empirical law based on observation and states as follows :

"The rate of flow of heat through a simple homogeneous solid is directly proportional to the area of the section at right angles to the direction of heat flow, and to change of temperature with respect to the length of the path of the heat flow".

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But, $$\frac{dt}{dx} = \frac{d}{dx}\left[\left(\frac{t_2 - t_1}{L}\right)x + t_1\right] = \frac{t_2 - t_1}{L}$$

$$\therefore \quad Q = -kA\frac{(t_2 - t_1)}{L} = \frac{kA(t_1 - t_2)}{L} \qquad ...(28.24)$$

Eqn. (28.24) can be written as :

$$Q = \frac{(t_1 - t_2)}{(L/kA)} = \frac{(t_1 - t_2)}{(R_{th})_{\text{cond.}}} \qquad ...(28.25)$$

where, $(R_{th})_{\text{cond.}}$ = Thermal resistance to heat conduction. Fig. 28.3 (*b*) shows the *equivalent thermal circuit* for heat flow through the plane wall.

Let us now find out the condition when instead of space, weight is the main criterion for selection of the insulation of a plane wall.

Thermal resistance (conduction) of the wall, $(R_{th})_{\text{cond.}} = \frac{L}{kA}$ ...(*i*)

Weight of the wall, $W = \rho A L$ ...(*ii*)

Eliminating L from (*i*) and (*ii*), we get

$$W = \rho A.(R_{th})_{\text{cond.}}\, kA = (\rho.k)A^2.(R_{th})_{\text{cond.}} \qquad ...(28.26)$$

The eqn. (28.26) stipulates the condition that, for a specified thermal resistance, the *lightest insulation will be one which has the smallest product of density* (ρ) *and thermal conductivity* (k).

28.2.5.2. Heat Conduction Through a Composite Wall

Refer Fig. 28.4 (*a*). Consider the transmission of heat through a composite wall consisting of a number of slabs.

Let L_A, L_B, L_C = Thicknesses of slabs A, B and C respectively (also called path lengths),

k_A, k_B, k_C = Thermal conductivities of the slabs A, B and C respectively,

t_1, $t_4 (t_1 > t_4)$ = Temperatures at the wall surfaces 1 and 4 respectively, and

t_2, t_3 = Temperatures at the interfaces 2 and 3 respectively.

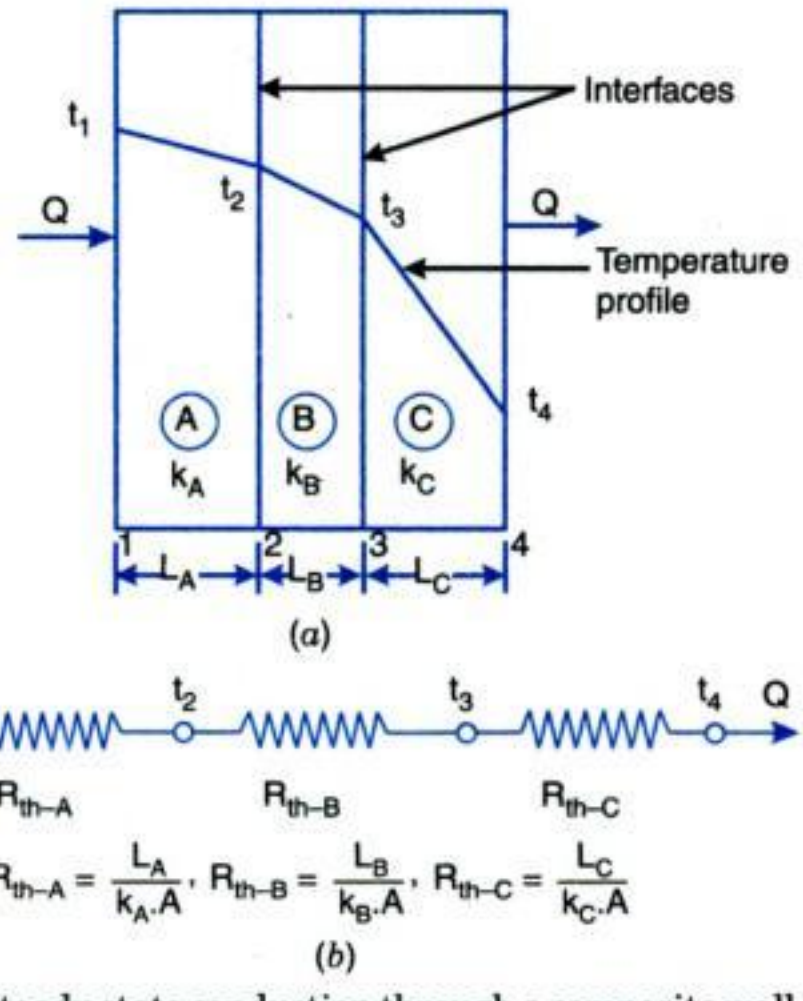

Fig. 28.4. Steady state conduction through a composite wall.

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Let r_1, r_2 = Inner and outer radii ;

t_1, t_2 = Temperature of inner and outer surfaces, and

k = Constant thermal conductivity within the given temperature range.

Consider an element at radius 'r' and thickness 'dr' for a length of the hollow cylinder through which heat is transmitted. Let dt be the temperature drop over the element.

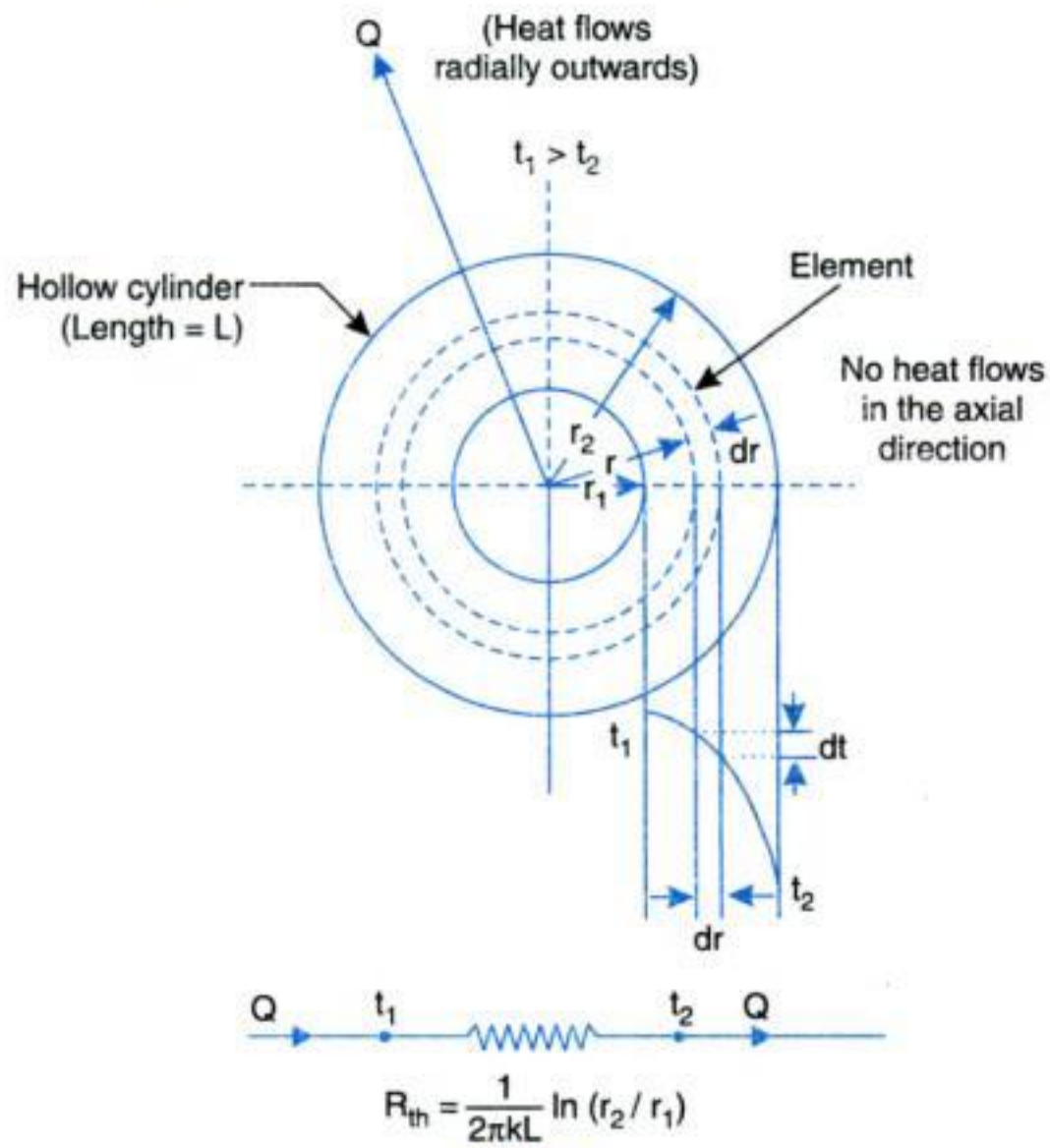

Fig. 28.17

Area through which heat is transmitted. $A = 2\pi\, r.\, L.$

Path length = dr (over which the temperature fall is dt)

$$\therefore \qquad Q = -kA \cdot \left(\frac{dt}{dr}\right) = -k \cdot 2\pi r \cdot L \frac{dt}{dr} \text{ per unit time} \quad \text{or} \quad Q \cdot \frac{dr}{r} = -k \cdot 2\pi L.dt$$

Integrating both sides, we get

$$Q \int_{r_1}^{t_2} \frac{dr}{r} = -k.2\pi L \int_{t_1}^{t_2} dt \quad \text{or} \quad Q \left[\ln (r)\right]_{r_1}^{r_2} = k.2\pi L \left[t\right]_{t_1}^{t_2}$$

or $$Q.\ln(r_2/r_1) = k.2\pi L(t_2 - t_1) = k.2\pi L(t_1 - t_2)$$

$$\therefore \qquad Q = \frac{k.2\pi L(t_1 - t_2)}{\ln (r_2/r_1)} = \frac{(t_1 - t_2)}{\left[\frac{\ln(r_2/r_1)}{2\pi k L}\right]} \qquad \text{...(28.33)}$$

28.2.7.2. Heat Conduction Through a Composite Cylinder

Consider flow of heat through a composite cylinder as shown in Fig. 28.18.

Let t_{hf} = The temperature of the hot fluid flowing inside the cylinder,

t_{cf} = The temperature of the cold fluid (atmospheric air),

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If there are n concentric spheres then the equation (28.38) can be written as follows :

$$Q = \frac{4\pi (t_{hf} - t_{cf})}{\left[\frac{1}{h_{hf} \cdot r_1^2} + \sum_{n=1}^{n=n} \left\{\frac{r_{(n+1)} - r_n}{k_n \cdot r_n \cdot r_{(n+1)}}\right\} + \frac{1}{h_{cf} \cdot r^2_{(n+1)}}\right]} \quad \text{...(28.39)}$$

If inside and outside heat transfer coefficients are considered, then the above equation can be written as follows :

$$Q = \frac{4\pi [t_1 - t_{(n+1)}]}{\sum_{n=1}^{n=n} \left[\frac{r_{(n+1)} - r_n}{k_n \cdot r_n \cdot r_{(n+1)}}\right]} \quad \text{...(28.40)}$$

Example 28.12. *A spherical shaped vessel of 1.4 m diameter is 90 mm thick. Find the rate of heat leakage, if the temperature difference between the inner and outer surfaces is 220°C. Thermal conductivity of the material of the sphere is 0.083 W/m°C.*

Solution. Refer Fig. 28.24.

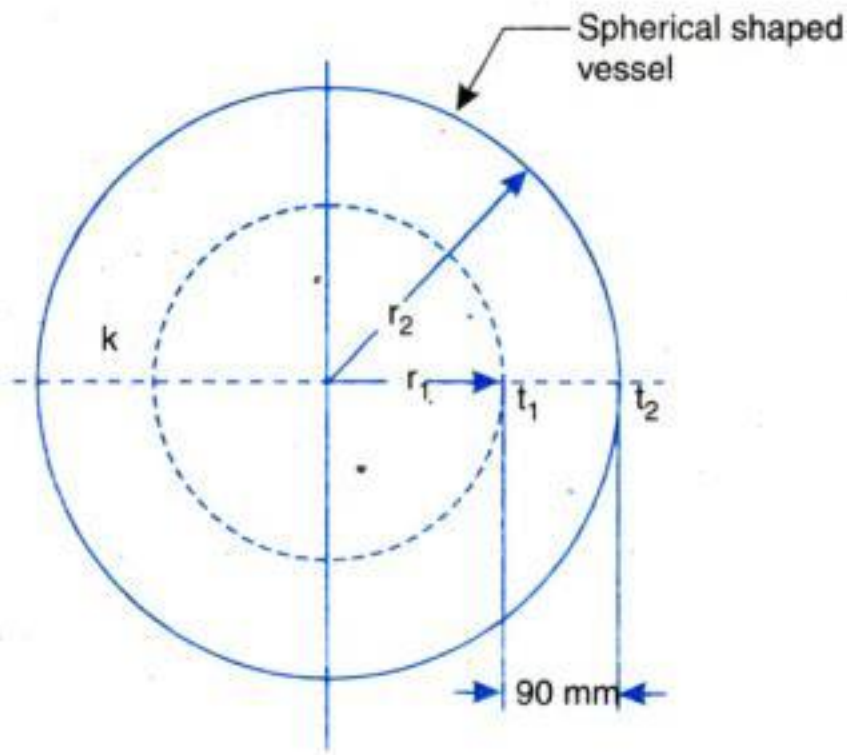

Fig. 28.24

Given : $r_2 = \frac{1.4}{2} = 0.7$ m ;

$$r_1 = 0.7 - \frac{90}{1000} = 0.61 \text{ m} ;$$

$t_1 - t_2 = 220°C$; $k = 0.083$ W/m°C

The rate of heat transfer/leakage is given by

$$Q = \frac{(t_1 - t_2)}{\left[\frac{(r_2 - r_1)}{4\pi k r_1 r_2}\right]} \quad \text{...[Eqn. (28.37)]}$$

$$= \frac{220}{\left[\frac{(0.7 - 0.61)}{4\pi \times 0.083 \times 0.61 \times 0.7}\right]} = 1088.67 \text{ W}$$

i.e., Rate of heat leakage = **1088.67 W.** **(Ans.)**

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Solution. *Given :* $\dot{m}_h = 0.2$ kg/s ; $\dot{m}_c = 0.5$ kg/s ; $t_{h_1} = 75°C$;

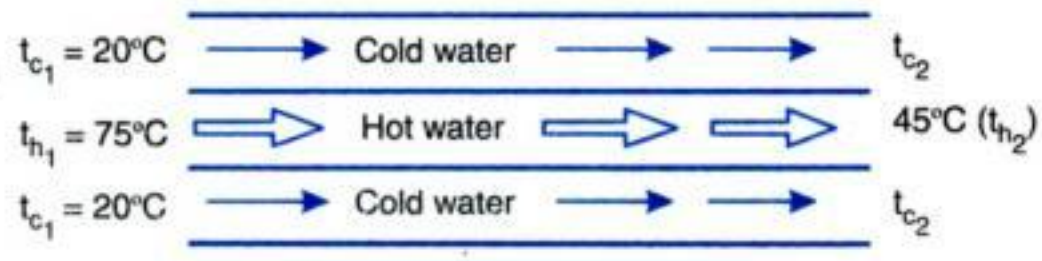
$t_{h_2} = 45°C$; $t_{c_1} = 20°C$; $h_i = h_0 = 650$ W/m^2°C.

The area of heat exchanger, A :

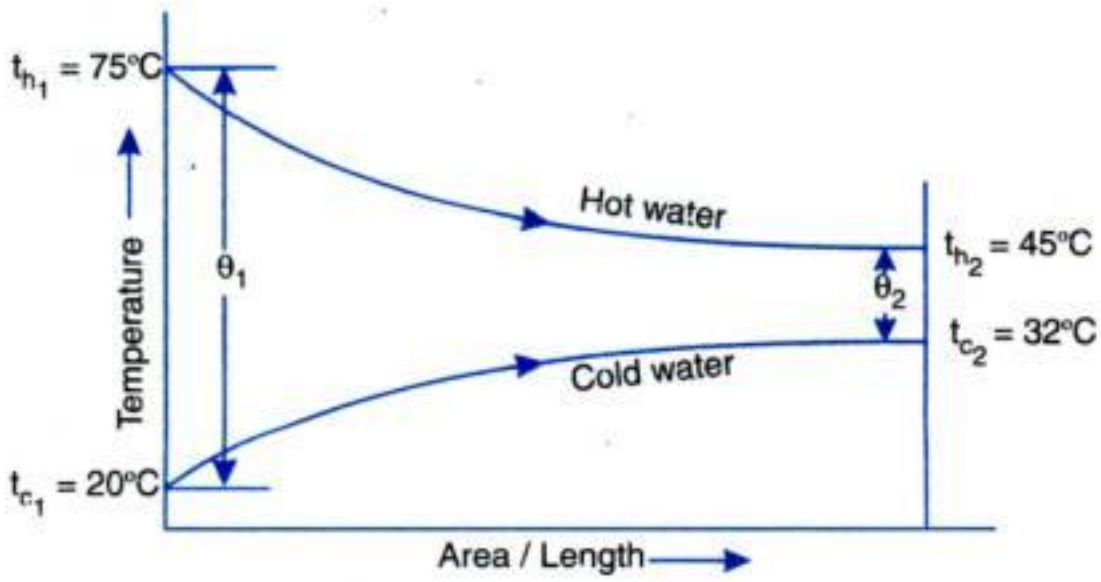
The heat exchanger is shown diagrammatically in Fig. 28.40.

The heat transfer rate, $Q = m_h \times c_{ph} \times (t_{h_1} - t_{h_2})$

$= 0.2 \times 4.187 \times (75 - 45) = 25.122$ kJ/s



(*a*) Flow arrangement.



(*b*) Temperature distribution.

Fig. 28.40. Parallel-flow heat exchanger.

Heat lost by hot water = Heat gained by cold water

$$\dot{m}_h \times c_{ph} \times (t_{h_1} - t_{h_2}) = \dot{m}_c \times c_{pc} \times (t_{c_2} - t_{c_1})$$

$$0.2 \times 4.187 \times (75 - 45) = 0.5 \times 4.187 \times (t_{c_2} - 20)$$

$\therefore$ $$t_{c_2} = 32°C$$

Logarithmic mean temperature difference (*LMTD*) is given by,

$$\theta_m = \frac{\theta_1 - \theta_2}{\ln (\theta_1 / \theta_2)} \quad \text{...[Eqn. (28.54)]}$$

or

$$\theta_m = \frac{(t_{h_1} - t_{c_1}) - (t_{h_2} - t_{c_2})}{\ln [(t_{h_1} - t_{c_1}) / (t_{h_2} - t_{c_2})]}$$

$$= \frac{(75 - 20) - (45 - 32)}{\ln [(75 - 20)/(45 - 32)]} = \frac{55 - 33}{\ln (55/13)} = 29.12°C$$

Overall heat transfer coefficient U is calculated from the relation

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_0} = \frac{1}{650} + \frac{1}{650} = \frac{1}{325}$$

$\therefore$ $$U = 325 \text{ W/m}^2\text{°C}$$

Also, $$Q = U A \theta_m$$

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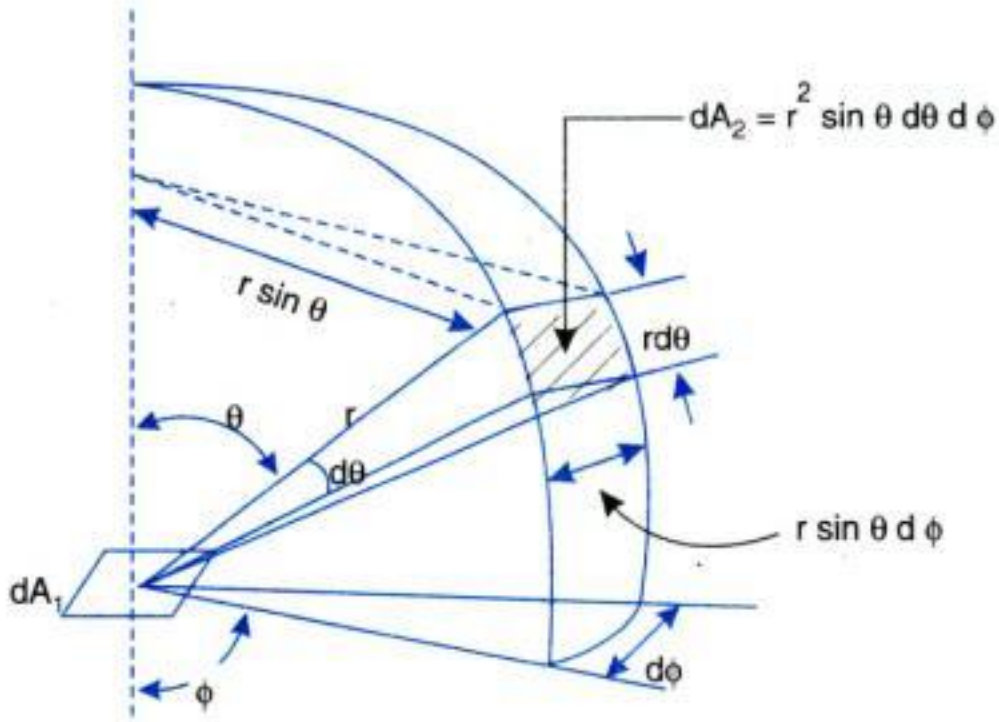

(*b*) Illustration for evaluating area dA_2

Fig. 28.51. Radiation from an elementary surface.

The solid angle subtended by $dA_2 = \dfrac{dA_2}{r^2}$

$\therefore$ The intensity of radiation, $I = \dfrac{dQ_{1-2}}{dA_1 \cos\theta \times \dfrac{dA_2}{r^2}}$ ...(28.82)

where dQ_{1-2} is the rate of radiation heat transfer from dA_1 to dA_2.

It is evident from Fig. 28.51 (*b*) that,

$$dA_2 = r\, d\theta\, (r \sin\theta\, d\phi)$$

or $$dA_2 = r^2 \sin\theta . d\theta . d\phi \quad ...(28.83)$$

From eqns. (28.82) and (28.83), we obtain

$$dQ_{1-2} = I\, dA_1 . \sin\theta . \cos\theta . d\theta . d\phi$$

The total radiation through the hemisphere is given by

$$Q = I\, dA_1 \int_{\theta=0}^{\theta=\frac{\pi}{2}} \int_{\phi=0}^{\phi=2\pi} \sin\theta \cos\theta\, d\theta\, d\phi$$

$$= 2\pi I\, dA_1 \int_{\theta=0}^{\theta=\frac{\pi}{2}} \sin\theta \cos\theta\, d\theta$$

$$= \pi I\, dA_1 \int_{\theta=0}^{\frac{\pi}{2}} 2 \sin\theta \cos\theta\, d\theta$$

$$= \pi I\, dA_1 \int_{\theta=0}^{\frac{\pi}{2}} \sin 2\theta\, d\theta$$

or $$Q = \pi I\, dA_1 \quad ...(28.84)$$

Also $$Q = E.dA_1$$

$\therefore$ $$E\, dA_1 = \pi I\, dA_1$$

or $$E = \pi I$$

i.e., The *total emissive power of a diffuse surface is equal to π times its intensity of radiation.*

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(*ii*) **When the body is not gray :**

Absorptivity when source is at 500°C = Emissivity when body is at 500°C

i.e., absorptivity, $\alpha = 0.72$

Then, energy emitted = $\varepsilon\sigma T_1^4 = 0.42 \times 5.67 \times 10^{-8} \times (1273)^4$

and, Energy absorbed = $\alpha\sigma T_2^4 = 0.72 \times 5.67 \times 10^{-8} \times (773)^4$

i.e., q = Energy emitted – Energy absorbed

$= 0.42 \times 5.67 \times 10^{-8} \times (1273)^4 - 0.72 \times 5.67 \times 10^{-8} \times (773)^4$

$= 62538 - 14576 = 47962$ W

i.e., Heat loss per m^2 by radiation = **47.962 kW. (Ans.)**

Example 28.27. *A long steel rod, 22 mm in diameter, is to be heated from 420°C to 540°C. It is placed concentrically in a long cylindrical furnace which has an inside diameter of 180 mm. The inner surface of the furnace is at a temperature of 1100°C, and has an emissivity of 0.82. If the surface of the rod has an emissivity of 0.62, find the time required for the heating operation.*

Take for steel : c = 0.67 kJ/kg K, ρ= 7845 kg/m³.

Solution. Refer Fig. 28.53.

Diameter of the steel rod

$= 22$ mm $= 0.022$ m

Inside diameter of the furnace

$= 180$ mm $= 0.18$ m

Emissivity $\varepsilon_1 = 0.62$

Emissivity $\varepsilon_2 = 0.82$

Specific heat of steel, $c = 0.67$ kJ/kg K

Density of steel, $\rho = 7845$ kg/m³

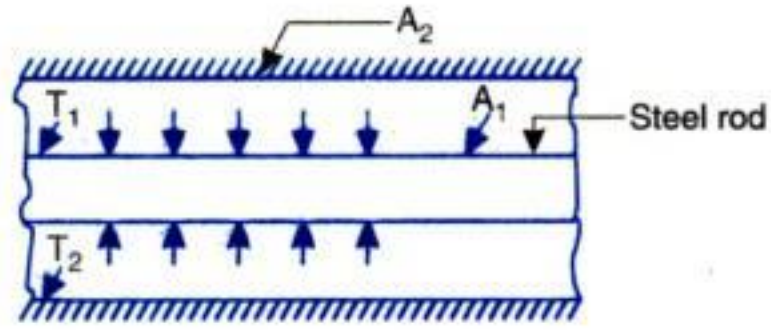

Fig. 28.53

$T_1 = 420 + 273 = 693$ K ..... 1st case

and $= 540 + 273 = 813$ K ..... 2nd case

$T_2 = 1100 + 273 = 1373$ K

The surface area of the rod, $A_1 = \pi \times 0.022 \times l$ m²

The surface area of the furnace, $A_2 = \pi \times 0.18 \times l$ m²

Time required for the heating operation, t_h :

Initial rate of heat absorption by radiation, when the rod is at 420°C or 693 K

$$Q_i = \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)}$$

$$= \frac{\pi \times 0.022 \times 1 \times 5.67 \times 10^{-8}\,(693^4 - 1373^4)}{\frac{1}{0.62} + \left(\frac{\pi \times 0.022 \times l}{\pi \times 0.18 \times l}\right)\left[\frac{1}{0.82} - 1\right]}$$

$$= \frac{-13022.5}{1.64} = -7940.5 \text{ W/m} \qquad [\because\ l = 1\,m \ldots \text{assumed}]$$

Rate of heat absorption at the end of the heating process, when the rod is at 540°C or 813 K

$$Q_e = \frac{A_2\,\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)}$$

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16. A spherical shaped vessel of 1.2 m diameter is 100 mm thick. Find the rate of heat leakage, if the temperature difference between the inner and outer surfaces is 200°C. Thermal conductivity of the material is 0.3 kJ/m-h-°C. [**Ans.** 2262 kJ/h]

17. Exhaust gases flowing through a tubular heat exchanger at the rate of 0.3 kg/s are cooled from 400°C to 120°C by water initially at 10°C. The specific heat of exhaust gases and water may be taken as 1.13 and 4.19 kJ/kg K respectively, and overall heat transfer co-efficient from gases to water is 140 W/m² K. Calculate the surface area required when the cooling water flow is 0.4 kg/s.

(*i*) For parallel-flow ; (*ii*) For counter-flow. [**Ans.** (*i*) 4.0 m², (*ii*) 3.37 m²]

18. Water flows inside a tube 50 mm in diameter and 3 m long at a velocity of 0.8 m/s. Determine the heat transfer co-efficient and the rate of heat transfer if the mean water temperature is 50°C and the wall is isothermal at 70°C. For water at 60°C, take k = 0.66 W/m K, ν (kinematic viscosity) = 0.478×10^{-6} m²/s, and Prandtl number = 2.98. [**Ans.** 4075 W/m²K ; 38.39 kW]

19. Liquid air at – 153°C is stored in the space of two concentric spheres of 21 cm and 30 cm diameters. The surface emissivities are 0.03. Assume the outer surface temperature is 27°C. Considering only radiation heat transfer and taking the latent heat of liquid air of 209 kJ/kg, find the rate of evaporation. Take $\sigma = 2.04 \times 10^{-4}$ kJ/h-m² K⁴. [**Ans.** 21.7 kg/h]

20. A body at 1100°C in black surroundings at 550°C has an emissivity of 0.4 at 1100°C and an emissivity of 0.7 at 550°C. Calculate the ratio of heat loss by radiation per m²,

(*i*) when the body is assumed to be grey with $\varepsilon = 0.4$

(*ii*) when the body is not grey. [**Ans.** (*i*) 70.22 kW, (*ii*) 62.42 kW]

21. A long steel rod, 20 mm in diameter, is to be heated from 427°C to 538°C. It is placed concentrically in a long cylindrical furnace which has an inside diameter of 160 mm. The inner surface of the furnace is at a temperature of 1093°C, and has an emissivity of 0.85. If the surface of the rod has an emissivity of 0.6, find the time required for the heating operation.

Take for steel : ρ = 7845 kg/m³ and c = 0.67 kJ/kg K. [**Ans.** 29.88 s]

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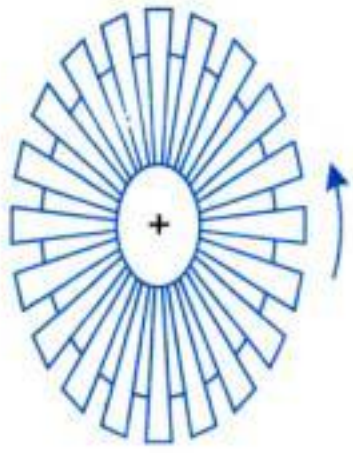
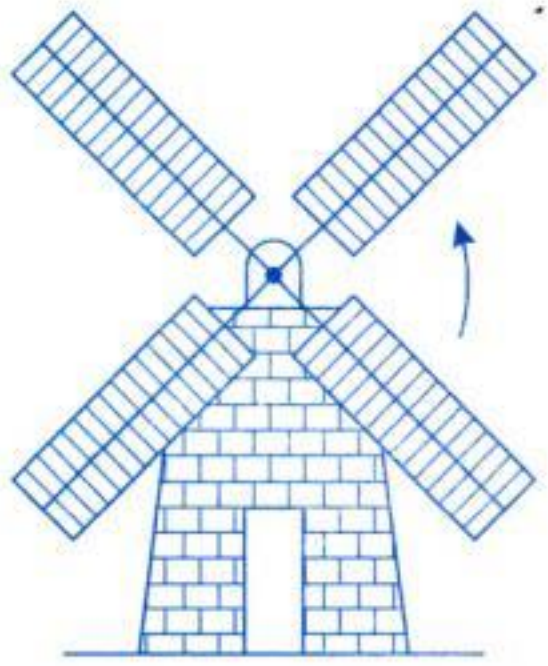
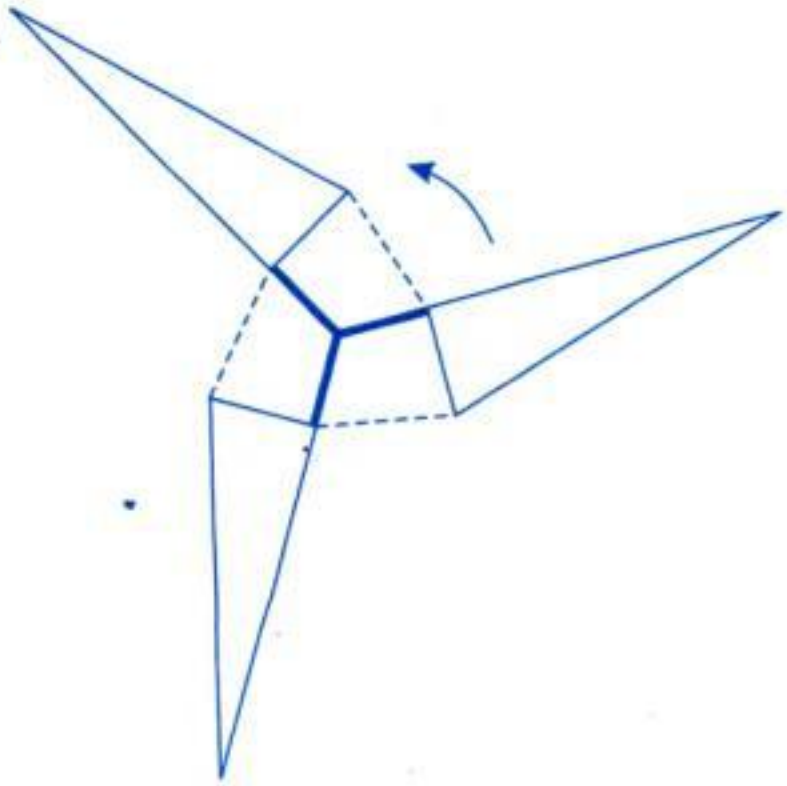


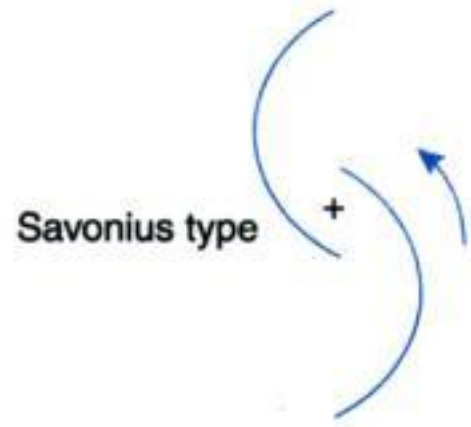
Multiblade type



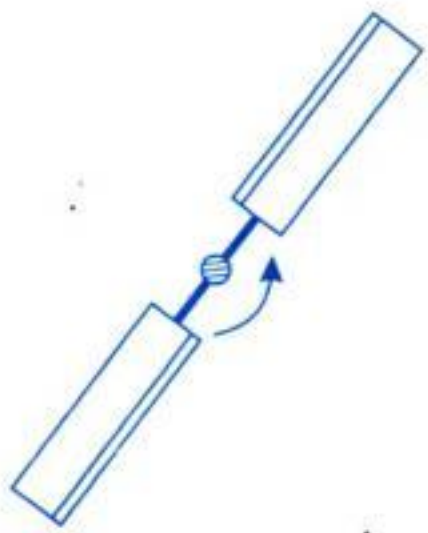
Four-blade dutch wind mill



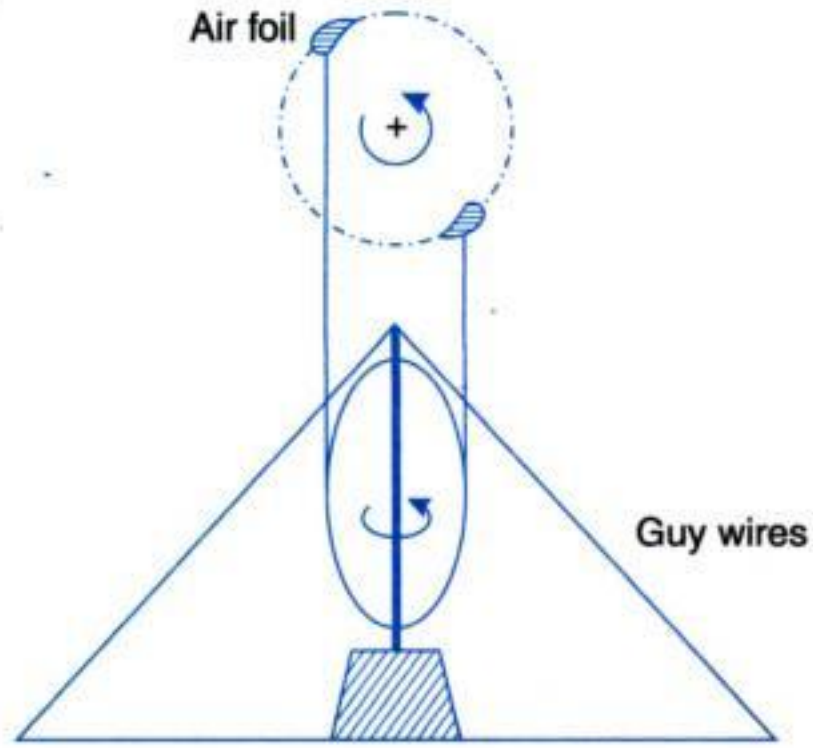
Sail type



Savonius type



Propeller type



Darrieus type

Fig. 29.1. Types of wind mills.

Fig. 29.2 shows a plot between K_p and tip speed ratio U_{bt}/U_w where U_{bt} = speed of blade tip.

It can be seen that K_p is the *lowest of Savonius* and *Dutch types* whereas the *propeller types have the highest value*.

In the designing of wind mills, it is uppermost to keep the power to weight ratio at the lowest possible level.

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Declination. *It is the angular position of the sun at solar moon with respect to the plane of the equator (north positive).*

Solar Energy—Introduction

The surface of the earth receives from the sun about 10^{14} kW of solar energy which is approximately five order of magnitude greater than currently being consumed from all resources. It is evident that sun will last for 10^{11} years. Even through the sunlight is filtered by the atmosphere one square metre of the land exposed to direct sunlight receives the energy equivalent of about 1 H.P. or 1 kW. However, this last amount of solar energy reaching earth is not easily convertible and certainly is not "free".

There are two obvious obstacles to harnessing solar energy. Firstly it is not constantly available on earth. Thus, some form of storage is needed to sustain a solar power system through the night and during periods when local weather conditions obscure the sun. Secondly the solar energy is diffused. Although the total amount of energy is enormous, the collection and conservation of solar energy into useful forms must be carried out over a large area which entails a large capital investment for the conversion apparatus.

Solar energy, therefore, most likely will be developed not because it is cheaper than alternative energy sources but because these alternative sources sooner or later (*i*) will be exhausted, (*ii*) will become increasingly more expensive, (*iii*) will continue to political and economical control by the nations possessing them, and (*iv*) will produce undesirable yet incompletely understood environmental consequences, especially on large scale that will be required to meet projected demands even with controlled growth.

Solar energy has some good advantages in comparison to the other sources of power. *Solar radiation does not contaminate environment or endanger ecological balance. It avoids major problems like exploration, extraction and transportation.*

29.4.2. Collectors in Various Ranges and Applications

Following list gives the thermal applications of solar energy and possible temperature ranges :

1. Low temperature (t = 100°C) :

(*i*) Water heating
(*ii*) Space heating
(*iii*) Space cooling
(*iv*) Drying
} ... Flat plate

2. Medium temperature (100 to 200°C) :

(*i*) Vapour engines and turbines
(*ii*) Process heating
(*iii*) Refrigeration
(*iv*) Cooking
} ... Cylindrical parabola

3. High temperature (> 200°C) :

(*i*) Steam engines and turbines
(*ii*) Stirling engine
(*iii*) Thermoelectric generator
} ... Parabolloid mirror arrays

The above classification of low, medium and high temperature ranges is somewhat arbitrary. Heating water for domestic applications, space heating and cooling and drying of agricultural products (and industrial products) is generally at temperature below 100°C, achieved using flat

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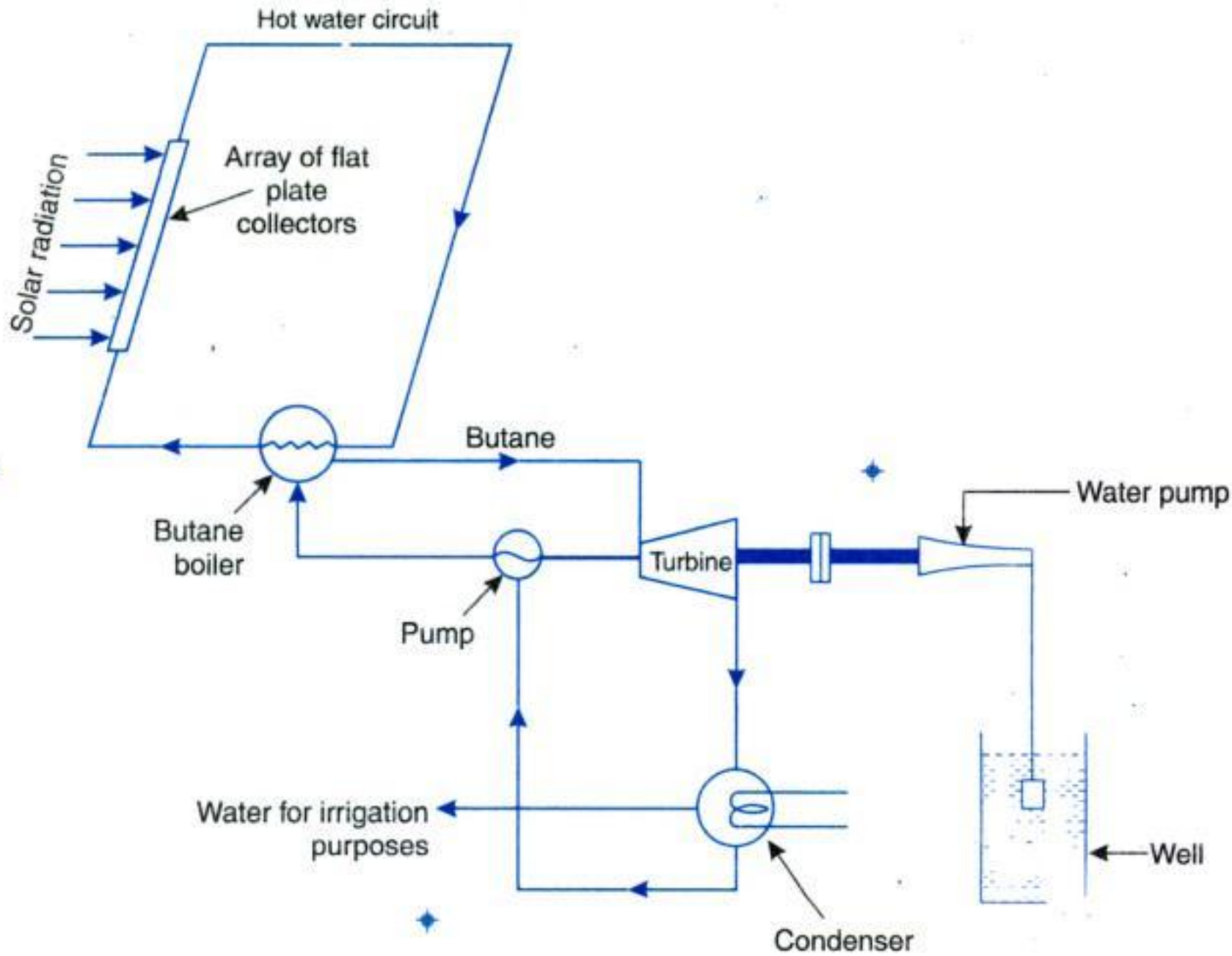

Fig. 29.14. Low temperature solar power plant.

29.4.7. Medium Temperature Systems Using Focusing Collectors

A circular or rectangular parabolic mirror can collect the radiation and focus it on to a small area, a mechanism for moving the collector to follow the sun being necessary. Such devices are used for *metallurgical research* where *high purity and high temperatures are essential,* an example being a 55 m diameter collector giving about 1 MW at Mont Louis in Pyrenees. Smaller units having 20 m diameter reflector can give temperatures of 300°C over an area of about 50 m^2. The collector efficiency is about 50%. On a small scale, units about 1 m diameter giving temperatures of about 300°C have been used for cooking purposes.

Fig. 29.15 shows a concave solar energy collector focusing sun's rays on boiler at a focal point. Generation of steam at 250°C could give turbine efficiencies up to 20-25 per cent.

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January, 1976, the first geothermal power plant at Tiwi produced 55 MW. A geothermal plant with a capacity of 11 MW has been in operation for nearly 20 years in Russia. The construction of another power plant at Mutnovsky with a capacity of 200 MW has been completed.

29.6. BIO-GAS PLANTS

In our country, there is a great potential for application of biomass as an **alternate source of energy.** We have plenty of agricultural and forest resources for reproduction of biomass.

The following are the biomass resources :

1. Concentrated wastes

(*i*) Municipal solid
(*ii*) Sewage wood products
(*iii*) Industrial waste
(*iv*) Manure at large lots.

2. Dispersed waste residue

(*i*) Crop residue
(*ii*) Logging residue
(*iii*) Disposed manure.

3. Harvested bio-mass

(*i*) Standing biomass
(*ii*) Biomass energy plantations.

Bio-gas

The main source for production of bio-gas is wet **cowdung.** Some of the other sources are :

(*i*) Sewage
(*ii*) Crop residue
(*iii*) Vegetable wates
(*iv*) Water hyacinth
(*v*) Poultry droppings
(*vi*) Pig-manures
(*vii*) Algae
(*viii*) Ocean kelp.

Bio-gas, a mixture *containing 55-65% methane, 30-40% carbon dioxide* and the rest being the impurities hydrogen (hydrogen sulphide and some nitrogen), can *be produced from the decomposition of animal, plant and human waste.* It is a clean but slow burning gas and usually has a calorific value 5000 to 5500 kcal/kg. It can be used directly in cooking, reducing the demand for firewood. Moreover, the material from which the bio-gas is produced retains its value as a fertilizer and can be returned to soil.

Bio-gas is produced by digestion, pyrolysis or hydrogasification. *Digestion* is a biological process that occurs in the absence of oxygen and in the presence of anaerobic organisms at ambient pressures and temperatures of 35-70°C. The container in which this digestion takes place is known as the *digester*.

Bio-gas plants have been built in various designs. Janata model gobar gas plant is described below :

Janata Model Gobar Gas Plant

Constructional Features. A Janata model gobar gas plant comprises of following parts :

Refer Fig. 29.23 :

1. Foundation
2. Digester
3. Dome
4. Inlet chamber
5. Outlet chamber
6. Mixing tank
7. Gas outlet pipe.

1. Foundation. The foundation is the amply compacted base of the digester made of cement concrete and brick ballast. Its construction is so carried out that it may provide a stable foundation to the digester walls and bear full load of slurry filled in the digester. It should be water proof so that no percolation or water leakage takes place.

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40. Consider the following statements :

The injector nozzle of a CI engine is required to inject fuel at a sufficiently high pressure in order to

1. be able to inject fuel in a chamber of high pressure at the end of the compression stroke.
2. inject fuel at high velocity to facilitate atomisation.
3. ensure that penetration is not high.

Of the above statements

(*a*) 1 and 2 are correct (*b*) 1 and 3 are correct

(*c*) 2 and 3 are correct (*d*) 1, 2 and 3 are correct.

41. Match List I with List II and select the correct answer using the codes given below the lists :

List I (*SI engine operating mode*)	*List II* (*Desired air-fuel ratio*)
A. Idling	1. 13.0
B. Cold starting	2. 4.0
C. Cruising	3. 16.0
D. Full throttle	4. 9.0

Codes :

	A	*B*	*C*	*D*
(*a*)	4	3	2	1
(*b*)	2	4	1	3
(*c*)	4	2	1	3
(*d*)	2	4	3	1

42. Compensating jet in a carburetter supplies almost constant amount of petrol at all speeds because

(*a*) the jet area is automatically varied depending on the suction.

(*b*) the flow from the main jet is diverted to the compensating jet with increase in speed.

(*c*) the diameter of the jet is constant and the discharge coefficient is invariant.

(*d*) the flow is produced due to the static head in the float chamber.

43. In the context of performance evaluation of I.C. Engine, match List I with List II and select the correct answer using the codes given below lists :

List I (*Parameter*)	*List II* (*Equipment for measurement*)
A. Brake power (B.H.P.)	1. Bomb calorimeter
B. Engine speed	2. Electrical tachometer
C. Calorific value of fuel	3. Hydraulic dynamometer
D. Exhaust emissions	4. Flame ionisation detector.

Codes :

	A	*B*	*C*	*D*
(*a*)	3	1	2	4
(*b*)	4	2	1	3
(*c*)	3	2	1	4
(*d*)	2	3	4	1

44. Consider the following statements :

In open cycle turbo-jet engines used in military aircraft, reheating the exhaust gas from the turbine by burning, more fuel is used to increase

1. thrust 2. the efficiency of engine 3. the range of aircraft.

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*74. In thermal power plants, the deaerator is used mainly to
(*a*) remove air from condenser (*b*) increase feed water temperature.
(*c*) reduce steam pressure (*d*) reduce dissolved gases from feedwater.

75. In high pressure natural circulation boilers, the flue gases flow through the following boiler accessories
1. Superheater 2. Air heater
3. Economiser 4. I.D. fan.
The correct sequence of the flow of the gases through these boiler accessories is
(*a*) 1, 3, 4, 2 (*b*) 3, 1, 4, 2
(*c*) 3, 1, 2, 4 (*d*) 1, 3, 2, 4.

76. Consider the following components
1. Radiation evaporator 2. Economiser
3. Radiation superheater 4. Convection superheater.
In the case of a Benson boiler, the correct sequence of the entry of water through these components is
(*a*) 1, 2, 3, 4 (*b*) 1, 2, 4, 3
(*c*) 2, 1, 3, 4 (*d*) 2, 1, 4, 3.

77. Coal fired power plant boilers manufactured in India generally use
(*a*) pulverised fuel combustion (*b*) fluidised bed combustion
(*c*) circulating fluidised bed combustion (*d*) moving stoker firing system.

78. The net result of pressure-velocity compounding of steam turbine is
(*a*) less number of stages (*b*) large turbine for a given pressure drop
(*c*) shorter turbine for a given pressure drop (*d*) lower friction loss.

79. Consider the following statements
When dry saturated or slightly superheated steam expands through a nozzle
1. the coefficient of discharge is greater than unity.
2. it is dry upto Wilson's line.
3. expansion is isentropic throughout.
Of these statements
(*a*) 1, 2 and 3 are correct (*b*) 1 and 2 are correct
(*c*) 1 and 3 are correct (*d*) 2 and 3 are correct.

80. The total and static pressures at the inlet of a steam nozzle are 186 kPa and 178 kPa respectively. If the total pressure at the exit is 180 kPa and static pressure is 100 kPa, then the loss of energy per unit mass in the nozzle will be
(*a*) 78 kPa (*b*) 8 kPa
(*c*) 6 kPa (*d*) 2 kPa.

81. Given, V_b = Blade speed
V = Absolute velocity of steam entering the blade
α = Nozzle angle,
the efficiency of an impulse turbine is maximum when
(*a*) $V_b = 0.5\ V \cos \alpha$ (*b*) $V_b = V \cos \alpha$
(*c*) $V_b = 0.5\ V^2 \cos \alpha$ (*d*) $V_b = V^2 \cos \alpha$.

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105. Consider the following statements :
Across the normal shock, the fluid properties change in such a manner that the
1. velocity of flow is subsonic.
2. pressure increases.
3. specific volume decreases.
4. temperature decreases.
Of these statements
(*a*) 2, 3 and 4 are correct (*b*) 1, 2 and 4 are correct
(*c*) 1, 3 and 4 are correct (*d*) 1, 2 and 3 are correct.

106. When a system undergoes a process such that $\int \frac{dQ}{T} = 0$ and $\Delta S > 0$, the process is
(*a*) irreversible adiabatic (*b*) reversible adiabatic
(*c*) isothermal (*d*) isobaric

107. Consider the following statements :
When a perfect gas enclosed in cylinder-piston device executes a reversible adiabatic expansion process.
1. its entropy will increase. 2. its entropy change will be zero
3. the entropy change of the surroundings will be zero.
Of these statements
(*a*) 1 and 3 are correct (*b*) 2 alone is correct
(*c*) 2 and 3 are correct (*d*) 1 alone is correct.

108. The heat rejection by a reciprocating air compressor during the reversible compression process AB, shown in the following temperature entropy diagram, is represented by the area

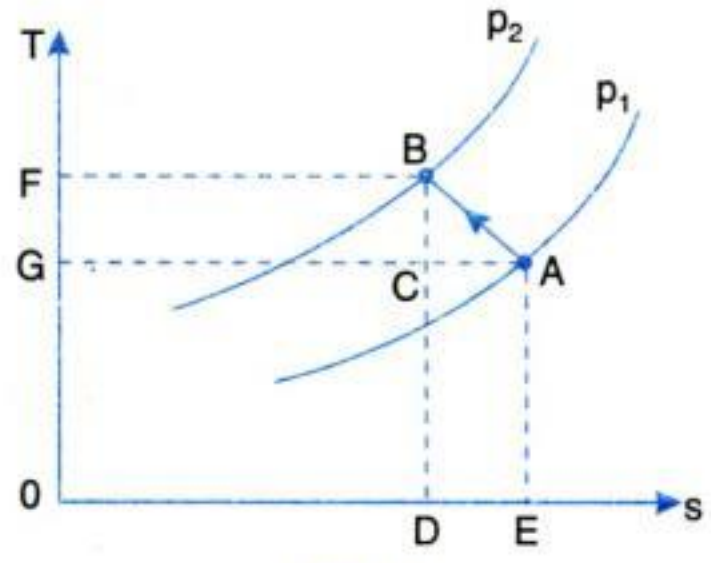

Fig. 18

(*a*) *ABC* (*b*) *ABDE*
(*c*) *ABFG* (*d*) *ABFOE*.

109. Centrifugal compressors are suitable for large discharge and wider mass flow range, but at a relatively low discharge pressure of the order of 10 bar, because of
(*a*) low pressure ratio (*b*) limitation of size of receiver
(*c*) large speeds (*d*) high compression index.

110. Given : V_{w_2} = Velocity of whirl at outlet, and
u_2 = Peripheral velocity of the blade tips.

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ABOUT THE BOOK

This book on **"Thermal Engineering"** (*printed in two colours*) has been written for the students preparing the subject for B.E. examinations of various Indian Universities, A.M.I.E. and competitive examinations (*e.g.*, U.P.S.C., GATE etc.). The Book contains 29 chapters in all, and deals the subject matter exhaustively.

Salient Features:

- The presentation of the subject matter is very systematic and the language of the text is lucid, direct and easy to understand.
- Each chapter of book is saturated with much needed text supported by neat and self-explanatory diagrams to make the subject self-speaking to a great extent.
- A large number of solved examples, questions selected from various universities, U.P.S.C., GATE etc., examination question papers, properly graded, have been added in various chapters to enable the students to attempt different types of questions in the examination without any difficulty.
- At the end of each chapter *Highlights, Objective Type Questions, Theoretical Questions* and *Unsolved Examples* have been added to make the book a complete unit in all respects.

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Er. R. K. Rajput, born on 15th September, 1944 (coincident with Engineer's Day) is a multi-disciplinary engineer. He obtained his *Master's degree* in ***Mechanical Engineering*** (with Hons.-Gold Medal) from Thapar Institute of Engineering and Technology, Patiala. He is also a *Graduate Engineer* in ***Electrical Engineering***. Apart from this he holds memberships of various professional bodies like Member Institution of Engineers (MIE); Member Indian Society of Technical Education (MISTE) and Member Solar Energy Society of India (MSESI). He is also a Chartered Engineer (India). He has served for several years as Principal of "Punjab College of Information Technology, Patiala" and "Thapar Polytechnic College, Patiala".

He has more than 40 years of experience in *teaching* different subjects of Mechanical and Electrical Engineering disciplines. He has published/presented a large number of technical papers. He is the author of several books on the important subjects of Mechanical as well as Electrical Engineering disciplines.

He has earned, by dint of hard work and devotion to duty, the following **awards/honours:** **** Best Teacher (Academic) Award * Jawahar Lal Nehru Memorial Gold Medal for an outstanding research paper (Institution of Engineers) * Distinguished Author Award * Man of Achievement Award.***

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