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Thermal Engineering

Course Code: UMEL313/23BTMEL401

B. Tech – VI/VII Semester

Prepared by:

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Teaching and Evaluation Scheme

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Course Title: **Thermal Engineering**										
Semester	VI	Teaching Scheme				Evaluation Scheme				
						Theory			Practical	
Term	Even	Th.	Tu.	Pr.	Credits	TAE	CAE	ESE	INT	EXT
Course Category	PE-2	03 Hrs.	-	02	04	20	20	60	25	--
Course Code	UMEL313 UMEP313									
Teaching Mode	Offline	5 Hrs.			Total	100			25	
Duration of ESE	2.5 Hrs.					125				

Teaching and Evaluation Scheme

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Course Title: **Thermal Engineering**											
Semester	**VII**	Teaching Scheme				Evaluation Scheme					
						Theory			Practical		
Term	Odd	L	T	P	Credits	TAE	CAE	ESE	INT	EXT	
Course Category	PCC 12	3 Hrs.	--	2 Hrs.	4	10	15	50	25	-	
Course Code	23BTMEL401 23BTMEP401										
Teaching Mode	Offline	5 Hrs.			Total	75			25		
Duration of ESE	3 Hrs.					100					

Course Objectives and Outcomes

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Course Objectives	1. To use knowledge of thermodynamics for various applications
	2. To understand operation of thermodynamic systems
	3. To study performance parameters of thermodynamic systems for Industrial applications
Course Outcomes	Upon successful completion of this course, student shall be able to:
	CO1. Define various terminologies of compressor.
	CO2. Classify types of internal combustion engine and its components.
	CO3. Evaluate performance of IC engines on various parameters.
	CO4. Analyze performance parameters of gas turbines and jet propulsion
	CO5. Estimate the performance of refrigeration system.

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Mapping of Course Outcomes with Program Outcomes and Program Specific Outcomes:

Course Outcomes	Program Outcomes and Program Specific Outcomes														
	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2	PSO3
CO1	3	3	3	3	-	2	1	1	-	-	-	2	3	3	2
CO2	3	3	3	3	-	-	-	1	-	-	-	2	3	3	2
CO3	3	3	3	3	1	2	3	1	-	-	-	2	3	3	2
CO4	3	3	3	3	2	2	3	1	-	-	-	2	3	3	2
CO5	3	3	3	3	1	2	2	1	-	-	-	2	3	3	2

Syllabus

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Unit	Contents
Unit 1	**Positive Displacement Compressors:** **Reciprocating compressors** : Construction and working, Work done during isothermal, Polytropic and adiabatic compression process, *P-V* diagram, isothermal efficiency, volumetric efficiency, Effect of clearance, Mechanical efficiency, Multistage compression, condition for minimum work input, Actual indicator diagram **Centrifugal compressor:** Construction and working, Velocity diagram, Work input, Efficiency, Effect of blade shape, Slip factor, Surging and choking
Unit 2	**I C. Engines:** Introduction to I.C. Engines: Air standard cycles, Classification of I C Engines, Various parts and working of engines, Difference between SI & CI Engines, Two strokes and four strokes Engines, Basics of Fuel supply systems used in I C Engines
Unit 3	**I C. Engine Testing:** Measurement of Indicated, friction and brake power, fuel and air consumption, Calculation of indicated and brake thermal efficiency, volumetric efficiency and mechanical efficiency, Percentage of excess air, Heat balance sheet, Performance characteristics and factors influencing the performance of I.C. Engines

Syllabus

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Unit	Contents
Unit 4	**Gas Turbines:** Classification of gas turbines, Ideal cycles, Isentropic and small stage efficiency, Applications of gas turbines, Pressure losses, Effect of inter-cooling, reheating and regeneration, Fuel – air ratio, Combustion efficiency, Performance calculation, Open cycle and Closed cycle gas turbine plants
Unit 5	**Refrigeration and Air-conditioning:** Introduction to refrigeration and air conditioning. Vapour compression refrigeration system and its performance, Introduction to Basic psychometric properties and processes.

Study Material

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Text Books	1.	Rathore M.M., Thermal Engineering, TMH Education Private Limited, 5th Edition, 2010
	2.	Ganeshan V., Gas Turbine, Tata McGraw publication, 3rd Edition, 2010
	3.	Gupta H. N., Fundamentals of Internal Combustion Engines, Prentice Hall India Learning Private Limited, 2nd Edition, 2012.
E-Books	1.	Boyace M.B., Gas Turbine Engineering Handbook, 4th Edition, Elsevier. 2011
	2.	Carson J.K., Refrigeration: Theory and Application, 1st Edition, 2012
Reference Books	1.	Sarkar B.K., Thermal Engineering , Tata McGraw publication, 1998
	2.	Heywood J.B., Internal Combustion Engine Fundamentals, McGraw-Hill, 1988
	3.	Cohen H., Rogers G.F.C and Saravanamattoo H.I.H., Gas Turbine Theory, Pearson Education limited, Fifth edition, 2001
	4.	Refrigeration and air conditioning by C.P. Arora, Publisher McGraw Hill Education, 3rd edition 2017

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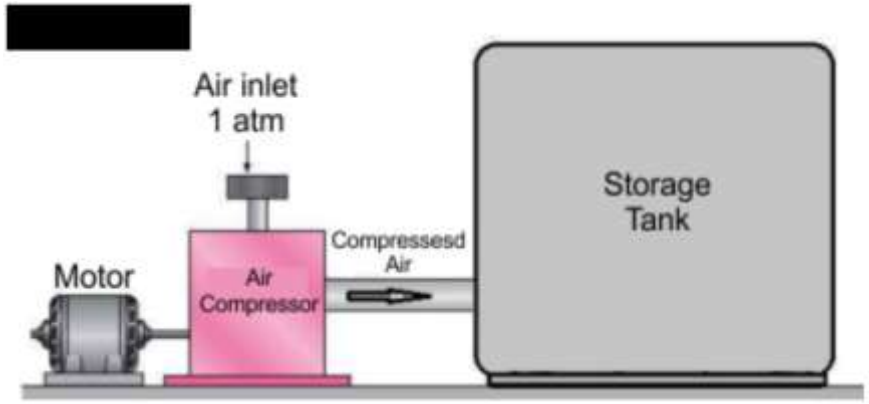
Unit 1

Compressor

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Introduction

An air compressor is a machine to compress the air and to raise its pressure. The air compressor sucks air from the atmosphere, compresses it and then delivers the same under a high pressure to a storage vessel (Receiver).



From the receiver, it may be conveyed by the pipe line to a place where the supply of compressed air is required. Since work is required for compression, a compressor must be driven by some prime mover.

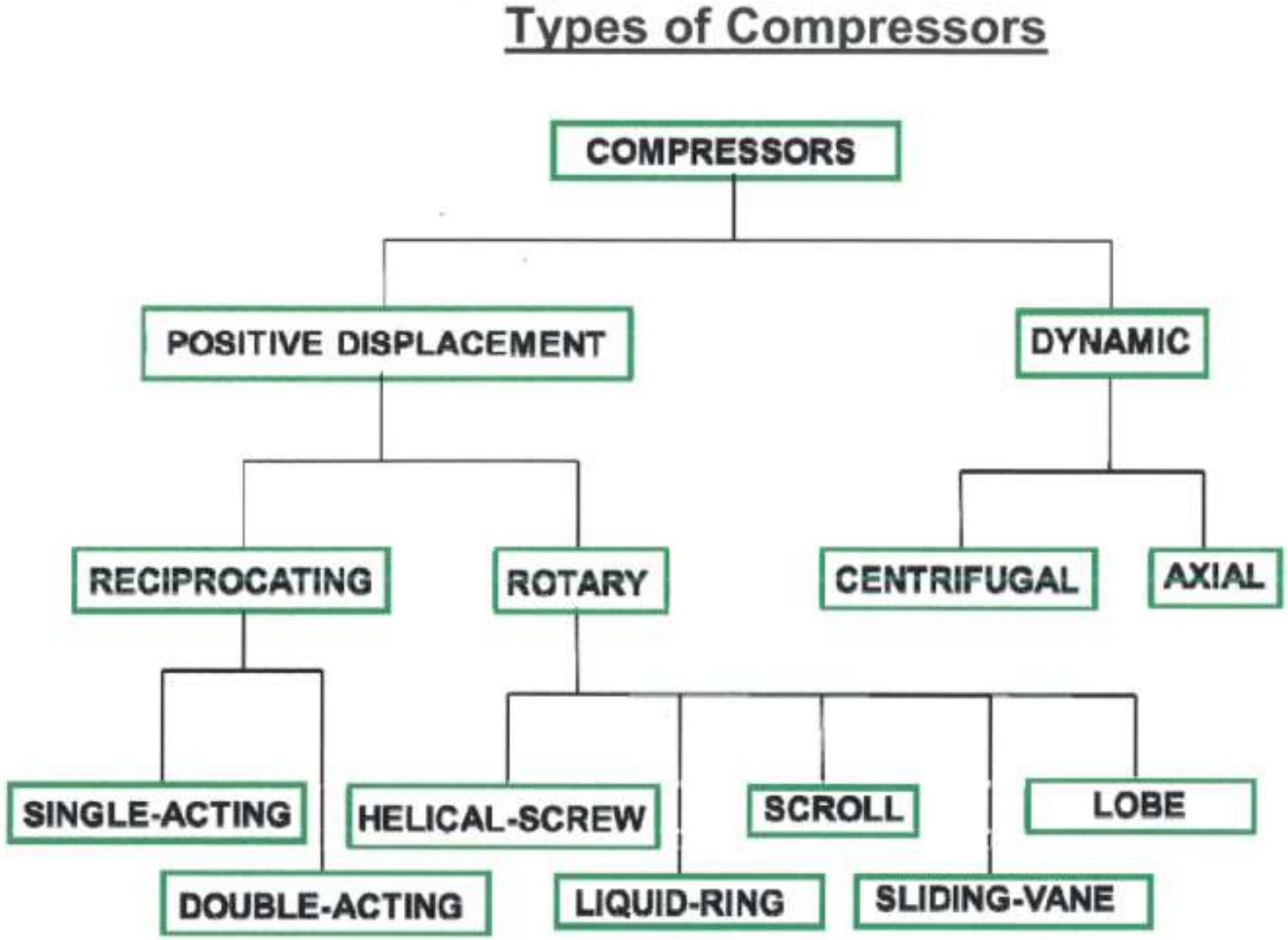
Uses of compressed air

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- The compressed air is used for many purposes such as for operating pneumatic drills, riveters, road drills, paint spraying, in starting & supercharging of I.C. Engines, in gas turbine plants, jet engines and air motors etc.
- It is also utilized in the operation of lifts, rams, pumps and a variety of other devices.
- In industry, compressed air is used for producing blast of air in blast furnaces.

Classification of compressor

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Classification of compressor

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Basic classification

1) According to working principle-

   a) Reciprocating and b) Rotary compressor

2) According to action of air

   a) Single acting and b) Double acting

3) According to no. of stages-

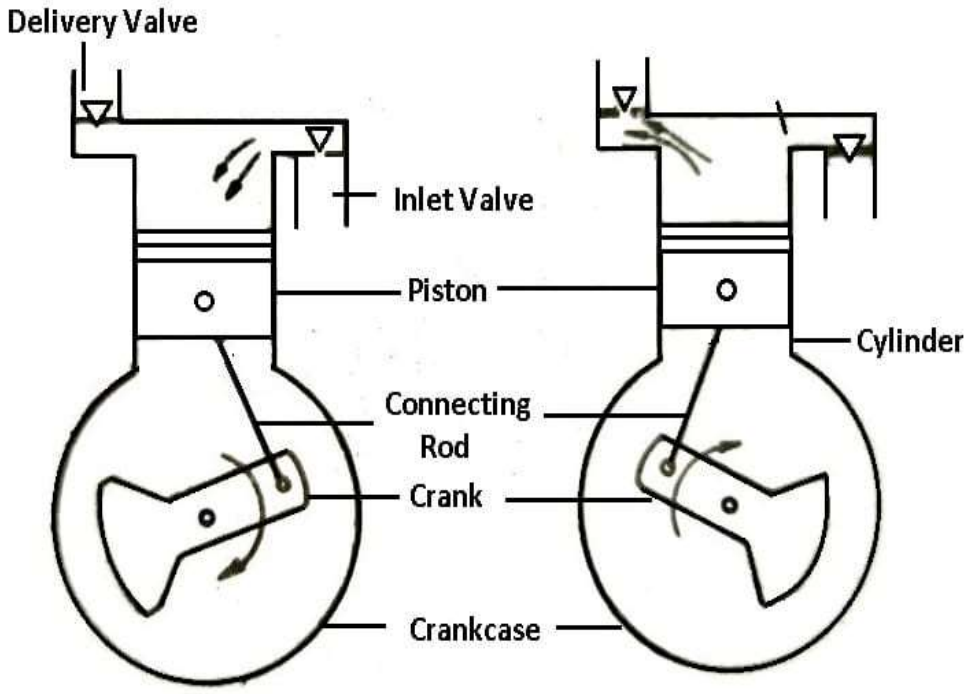
   a) Single stage and b) Multi-stage compressor

Terminology used in compressor

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- **Suction pressure:** Absolute pressure of air at the inlet of the compressor.
- **Discharge pressure:** Absolute pressure at the outlet of compressor
- **Compression ratio:** Ratio of discharge pressure to suction pressure
- **Compressor Capacity:** Volume of air delivered by the compressor.
- **Free Air Delivery (FAD):** It is the actual volume delivered by a compressor when reduced to the normal temp & pressure conditions. The capacity of compressor is generally given in terms of FAD (Free Air Delivery).
- **Swept Volume:** The volume of air sucked by the compressor during its suction stroke.
- **Mean effective pressure:** Air pressure on the piston keeps on changing with the movement of the piston in the cylinder The mean effective pressure of the compressor is found by dividing the work done per cycle by the stroke volume.

Construction and Working of Reciprocating compressor



Reciprocating Air Compressor

- Consists of **cylinder, piston, crankshaft, connecting rod, inlet & delivery valves**
- Driven by a **motor/engine**, converting rotary motion to reciprocating motion
- During **suction stroke**, piston moves down → air enters through inlet valve
- During **compression stroke**, piston moves up → air is compressed
- When pressure exceeds delivery pressure → **delivery valve opens**, air discharged
- Cycle repeats → continuous compressed air supply

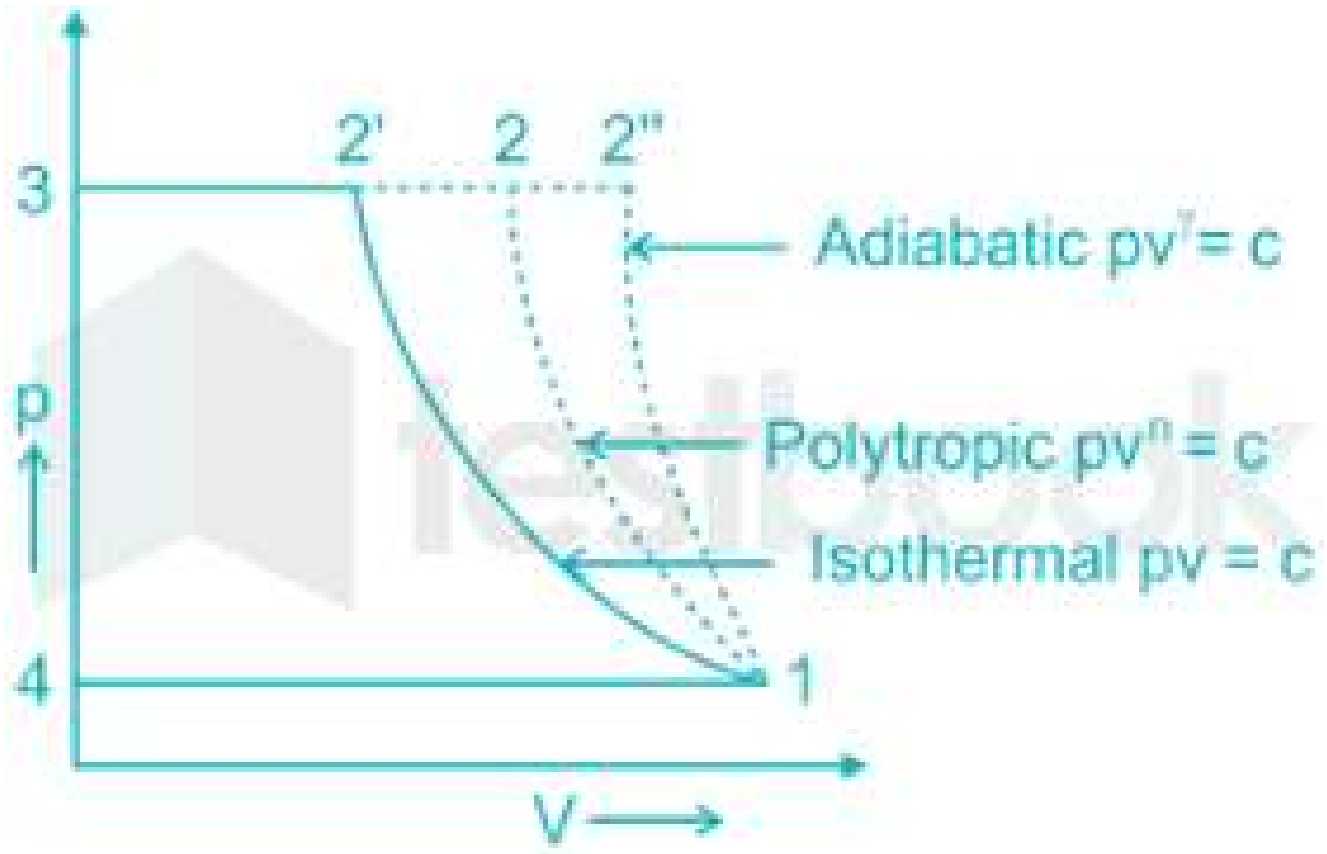
Work done by single stage reciprocating compressor

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Process of compression are-

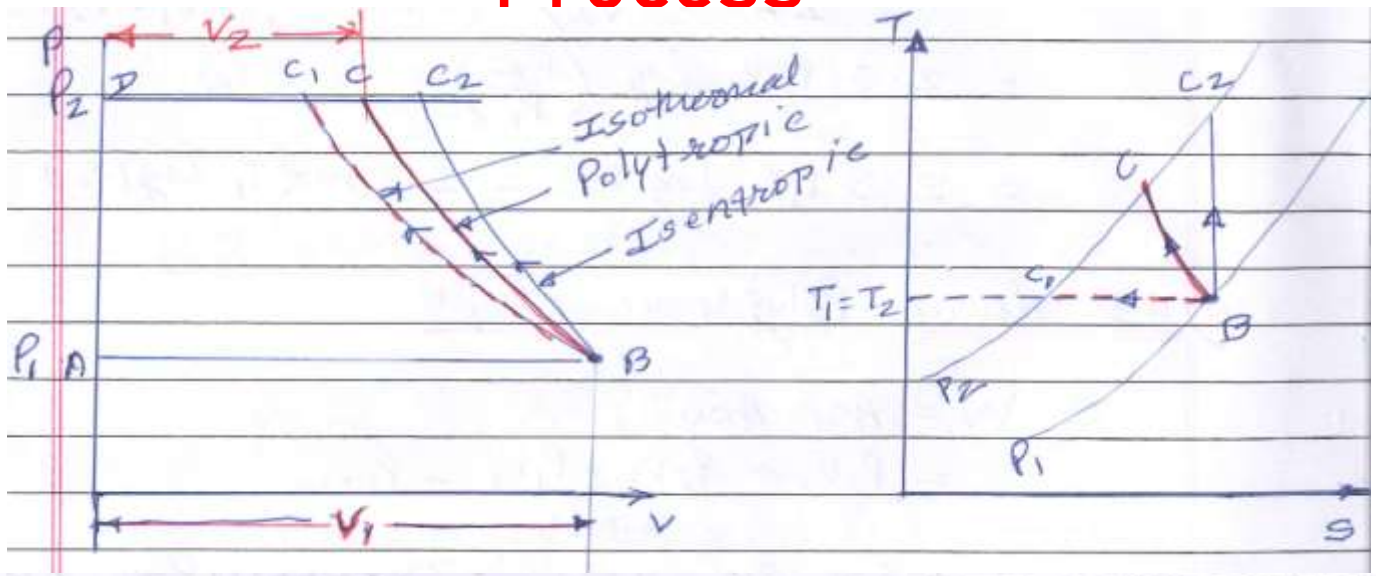
1) **Isothermal Compression**
2) **Polytropic Compression**
3) **Adiabatic compression**

where $1<n<\gamma$



Work done during Isothermal Compression Process

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$$W = \text{Area } ABC_1D$$

$$= P_2V_2 + 2.3 P_2V_2 \log\left(\frac{V_1}{V_2}\right) - P_1V_1$$

$$= 2.3 P_2V_2 \log\left(\frac{V_1}{V_2}\right) \qquad \because P_1V_1 = P_2V_2$$

$$= 2.3 P_1V_1 \log\left(\frac{P_2}{P_1}\right)$$

$$= 2.3 P_1V_1 \log r = 2.3 mRT_1 \log(r)$$

Work done during Polytropic Compression Process

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$$W = \text{Area } ABCD$$

$$= P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_1V_1$$

$$= \frac{(n-1)P_2V_2 + P_2V_2 - P_1V_1 - (n-1)P_1V_1}{n-1}$$

$$= \frac{n}{n-1}(P_2V_2 - P_1V_1)$$

$$= \frac{n}{n-1} P_1V_1 \left[\frac{P_2V_2}{P_1V_1} - 1\right]$$

$$P_1V_1^n = P_2V_2^n \Rightarrow \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} \quad \text{or} \quad \frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$\therefore W = \frac{n}{n-1} P_1V_1 \left[\frac{P_2}{P_1} \times \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} - 1\right]$$

$$W = \frac{n}{n-1} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \quad \text{---(1)}$$

$$W = \frac{n}{n-1} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] = \frac{n}{n-1} mR(T_2 - T_1)$$

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Work done during Adiabatic Compression Process

1) WD during isentropic compression

Replace n by γ

$$W = \frac{\gamma}{\gamma-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$W = \frac{\gamma}{\gamma-1} mR (T_2 - T_1) \quad \text{——} (3)$$

$C_p - C_v = R$, $C_p / C_v = \gamma$ $\therefore R = C_p \left(\frac{\gamma-1}{\gamma} \right)$

(3) $\Rightarrow \therefore W = m C_p (T_2 - T_1)$

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Numerical on Work done

A single stage reciprocating air compressor is required to compress 1 kg of air from 1 bar to 4 bar. The initial temperature is 27 degree Celsius. Compute the work required in the following case.

i) Isothermal compression

ii) Polytropic compression with index n = 1.2

iii) Isentropic compression

$m = 1\ kg, \quad P_1 = 1\ bar \quad P_2 = 4\ bar$

$T_1 = 27^\circ C = 27 + 273 = 300K, \quad n = 1.2$

1) WD for Isothermal comph:-

$W = 2.3\, P_1 V_1 \log\left(\frac{P_2}{P_1}\right)$ $\quad \log = \log_{10}$, $\ln = \log_e$

$= 2.3\, mRT_1 \log\left(\frac{P_2}{P_1}\right)$

$= 2.3 \times 1 \times 287 \times 300 \log\left(\frac{4}{1}\right)$

$= 119230\ J$

$W = 119.23\ KJ$ (Ans.)

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Numerical on Work done

(ii) For polytropic compⁿ with $PV^{1.2} = C$

$$W = \frac{n}{n-1} m R T_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.2}{1.2-1} \times 1 \times 287 \times 300 \left[\left(\frac{4}{1}\right)^{\frac{0.2}{1.2}} - 1\right]$$

$$= 134320 \text{ J}$$

$$W = 134.32 \text{ KJ} \quad \text{(Ans)}$$

(iii) For isentropic compⁿ

$$W = \frac{\gamma}{\gamma-1} m R T_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

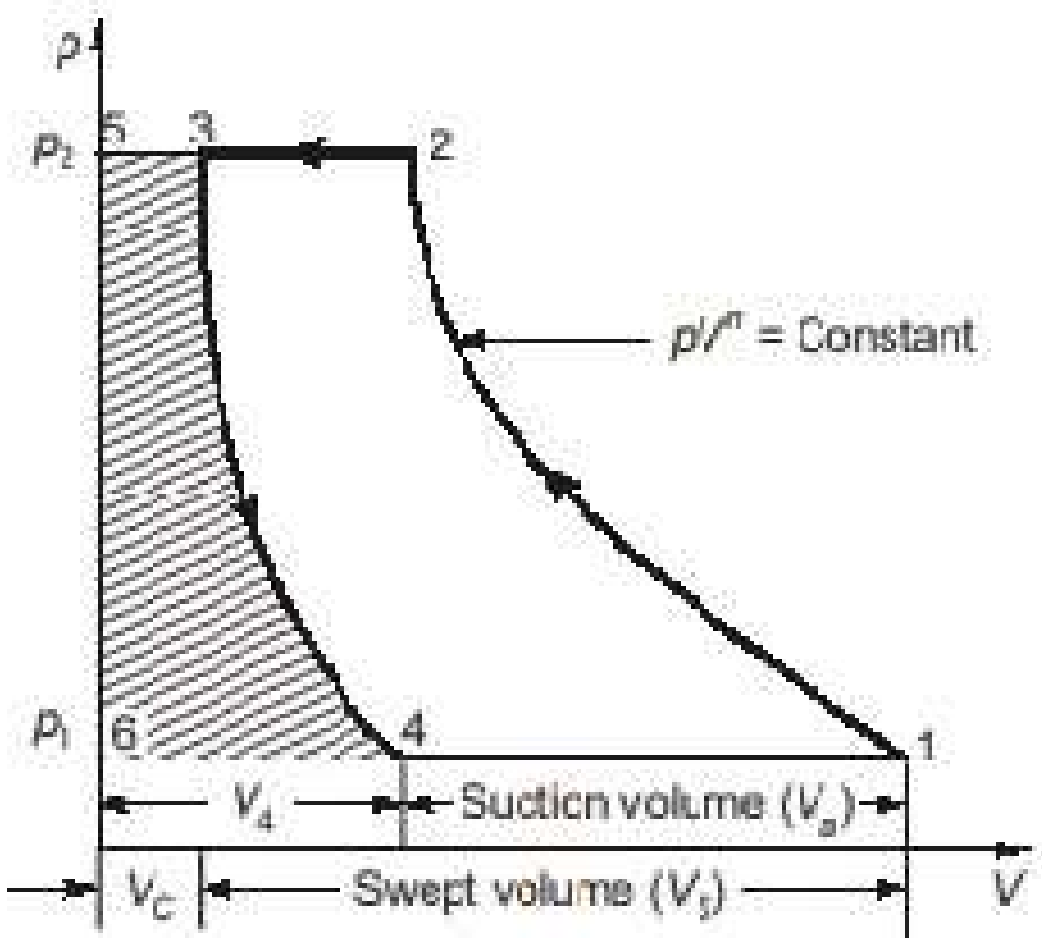
$$= \frac{1.4}{1.4-1} \times 1 \times 287 \times 300 \left[\left(\frac{4}{1}\right)^{\frac{0.4}{1.4}} - 1\right]$$

$$= 146630 \text{ J}$$

$$W = 146.63 \text{ KJ}$$

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Effect of clearance volume on Work done



In the case of compression without clearance volume, the entire volume of air in the compressor cylinder is compressed by the inward stroke of the piston.

However, in actual practice, it is not possible to reduce the clearance Vc to zero. The passages leading to the inlet and outlet valves always contribute to Vc. The Vc is typically expressed as a percentage of the piston displacement

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Effect of clearance volume on Work done

The delivery stroke is completed when the piston reaches point 3. At this stage, some air is left in the clearance volume.

This air will expand (from point 3 to 4) during the return stroke. Compression and expansion processes may be isothermal, polytropic, or isentropic.

Mathematical equation For a Polytropic process, the Work Done (WD) per cycle is calculated as-

$$W = \left(\frac{n}{n-1}\right) P_1(V_1 - V_4)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right], \text{ n = polytropic index}$$

Clearance volume does not affect the work done on the air and the power required for compressing the air. Work required to compress the air in clearance volume is recovered during the expansion process (3-4).

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Multistage Compression

Sometimes, the air is required at high pressure. In such cases, either we employ a large pressure ratio (in a single cylinder) or compress the air in two or more cylinders in series. It is found that if we use single-stage compression for producing high-pressure air (say 8 bar to 10 bar), **it suffers the following disadvantages:**

1) Size of Cylinder: The size of the cylinder will be too large.
2) Due to compression, the temperature of air will rise. It is difficult to reject heat from the air in the small time available during compression.
3) Sometimes, at the end of compression, the temperature of air is too high. It may heat up the cylinder head or burn the lubricating oil.

To overcome the above problems, compression is carried out in stages (2 or 3) connected in series with intercooling between the stages. **Such an arrangement is called Multistage Compression.**

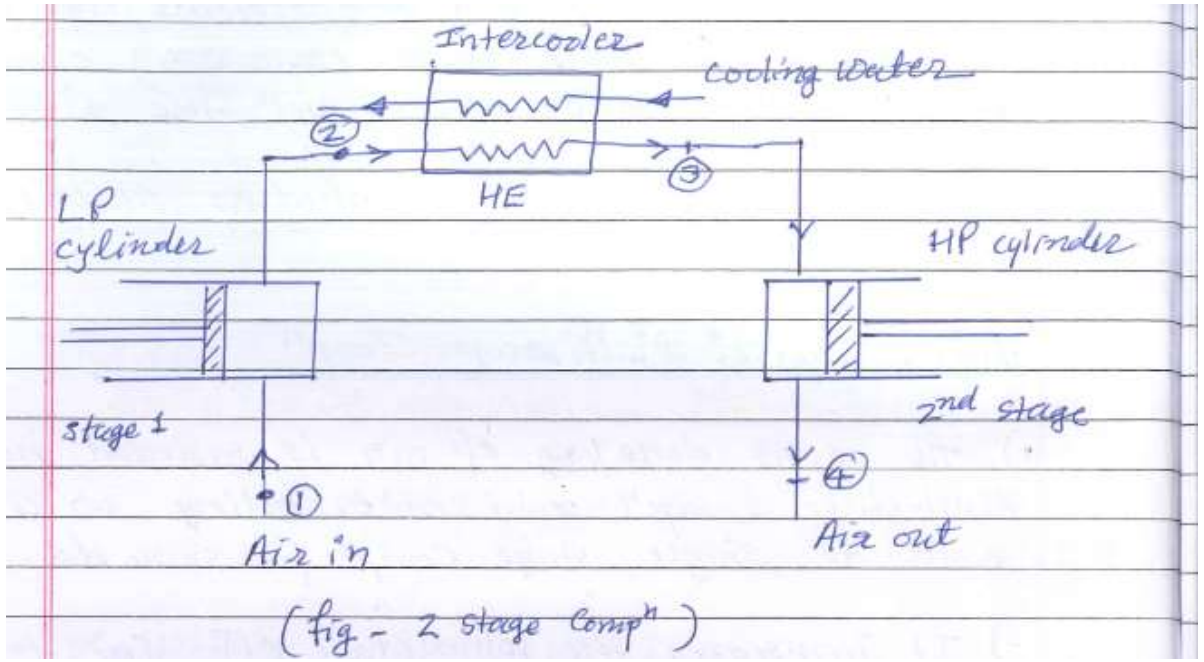
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Advantages of Multistage Compression

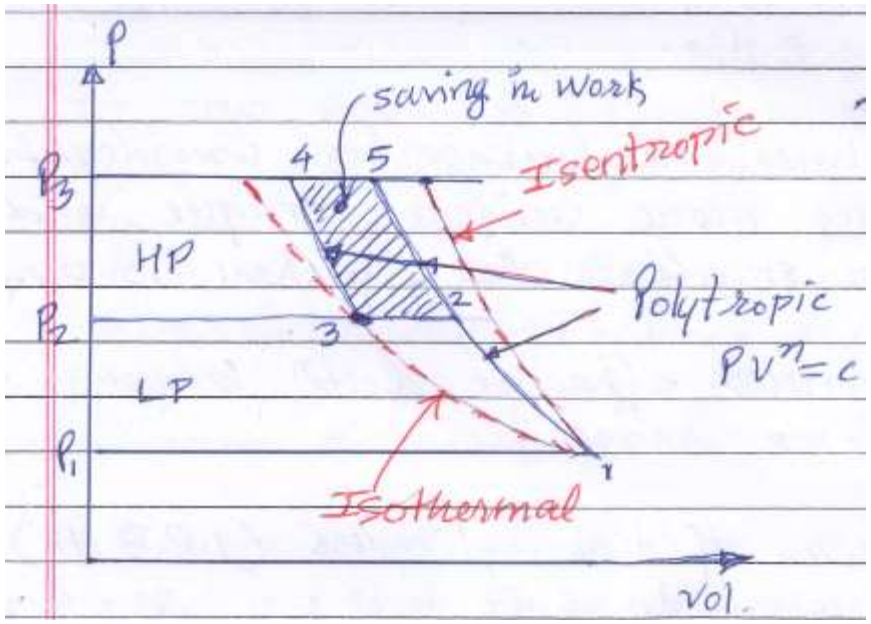
- The work done per kg of air is reduced in multistage compression with intercooling as compared to single-stage compression for the same delivery pressure.
- It improves the volumetric efficiency for a given pressure ratio.
- It reduces the leakage loss considerably.
- It gives more uniform torque, and hence a smaller size flywheel is required.
- It provides effective lubrication because of the low temperature range.
- The sizes of two cylinders (LP & HP — Low Pressure & High Pressure) may be adjusted to suit the volume and pressure of the air.
- It reduces the cost of the unit (compressor).

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Two stage Reciprocating air Compressor with intercooler



Arrangement of components

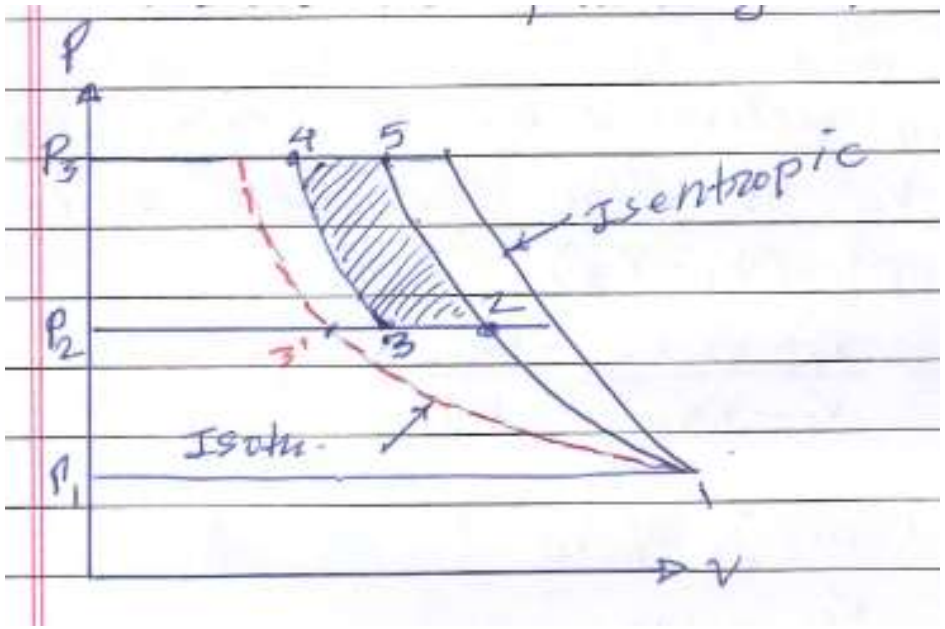


Perfect Intercooling

When the temperature of air leaving the intercooler T3 is equal to the original atmospheric air temperature T1 the intercooling is considered perfect or complete. Point 3 lies on the isothermal curve.

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Two stage Reciprocating air Compressor with intercooler



Imperfect Intercooling

When T3 > T1
Point 3 lies on the RHS (Right Hand Side) of the isothermal curve as shown in the following P-V

The amount of work saved due to intercooling is represented by the shaded area 2-3-4-5.

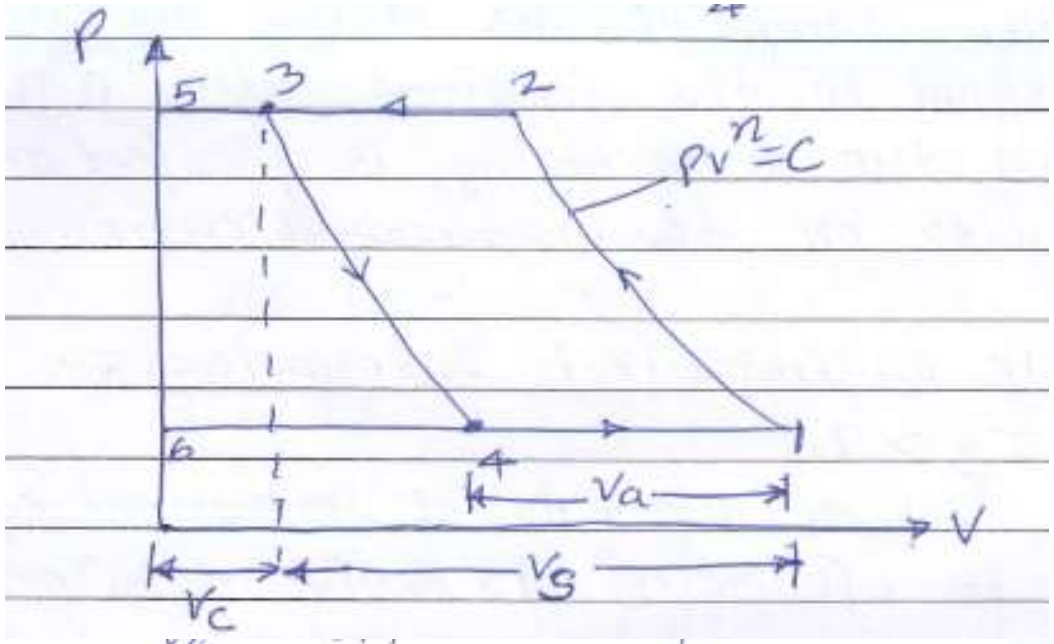
Work saved with perfect intercooling > Work saved with imperfect intercooling

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Volumetric efficiency of Reciprocating air Compressor

The ratio of actual free air delivered (FAD) by the compressor per stroke to the displacement of the compressor is known as the volumetric efficiency of the compressor.

Actual volume of air intake (v1-v4) delivered by the compressor having clearance volume Vc is less than the swept volume of the compressor (V1-V3)



$$\therefore \eta_V = \frac{V_S - V_C\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]}{V_S}$$

$$\eta_V = 1 - \frac{V_C}{V_S}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

$$\eta_V = 1 - \frac{K}{100}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

where $K = \frac{V_C}{V_S} \times 100$

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Volumetric efficiency of Reciprocating air Compressor

$$\eta_V = \frac{V_1 - V_4}{V_1 - V_3} = \frac{V_a}{V_s}$$

$$\therefore \eta_V = \frac{V_s - V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]}{V_s}$$

$$\eta_V = 1 - \frac{V_c}{V_s}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

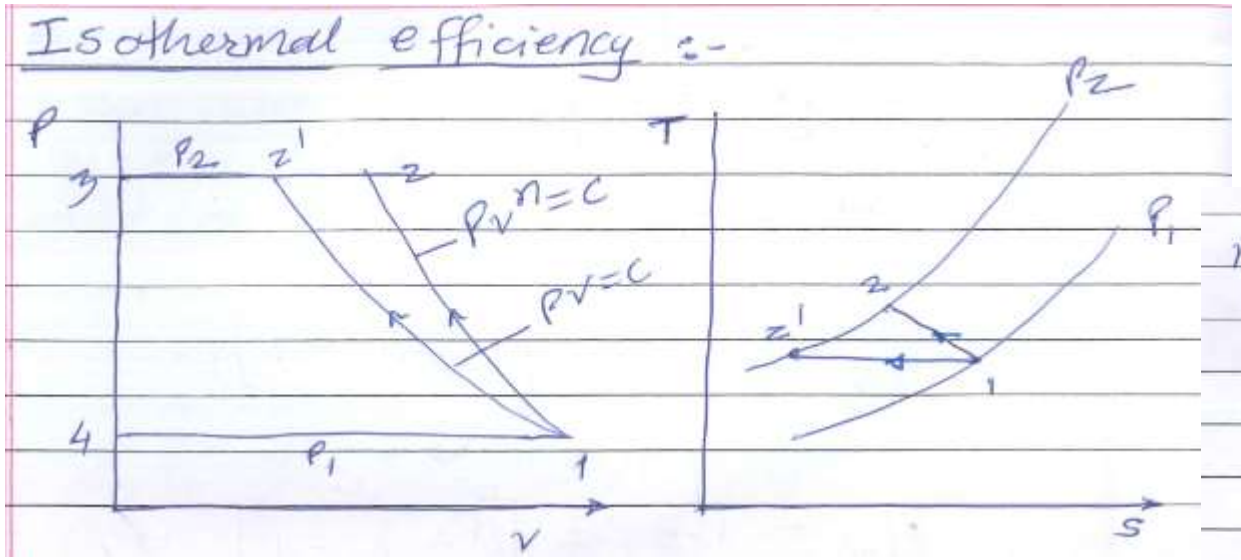
$$\eta_V = 1 - \frac{K}{100}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

where

$$K = \frac{V_c}{V_s} \times 100$$

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Isothermal efficiency of Reciprocating air Compressor



$$\eta_{iso.} = \frac{\text{Isothermal work}}{\text{Actual work}} = \frac{\text{Isentropic Power}}{\text{Indicated Power}}$$

$$= \frac{W_i}{W_a} = \frac{RT_1 \log_e\left(\frac{P_2}{P_1}\right)}{\frac{n}{n-1} RT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]} \quad \text{N.m/kg} \quad \text{N.m/kg}$$

$$\eta_{isoth} = \frac{\log_e (P_2/P_1)}{\frac{n}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

The work required for compression per kg of air can be reduced by decreasing the index of compression. This can be achieved by cooling the compressor by air or water circulated through the cooling jackets.

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Mechanical efficiency of Reciprocating air Compressor

It is the ratio of Indicated Power to the shaft power (or Brake power) of the motor or engine required to drive the compressor.

$$\text{Mechanical efficiency} = \frac{Indicated\ Power}{Brake\ power}$$

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Condition for Minimum work required for 2 stage RA Compressor

Work req^d by 2-stage RAC with perfect intercooling

$$W = \frac{n}{n-1} P_1 V_1 \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 2 \right\}$$

Diff. above eq^n wrt P_2 & equate to zero.

$$\frac{dW}{dP_2} = 0$$

put $\left(\frac{n-1}{n}\right)$ as 'x'.

we can obtain $(P_2)^{2x} = (P_1 \cdot P_3)^x$

$$P_2 = \sqrt{P_1 \cdot P_3}$$

OR $\frac{P_2}{P_1} = \frac{P_3}{P_2} = \left(\frac{P_3}{P_1}\right)^{\frac{1}{2}}$ for 2 stage

OR $\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \cdots\cdots \frac{P_{q+1}}{P_q} = \left(\frac{P_{q+1}}{P_1}\right)^{1/q}$

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Numerical on Multistage Compressor

A three stage compressor delivers air at 70 bar from an atm. press of 1 bar & 30°C. Assuming the intercooling complete, estimate the amount of min. work required to deal with 1 kg of air. Also find the amount of heat rejected in each inter-cooler.
(n = 1.2, Cp = 1.005 KJ/Kgk)

Solution:-

P4 = 70 bar, P1 = 1 bar,

T1 = 300 C =303 K m = 1kg & n =1.2

Amount of minimum work required,

W = 3n/(n-1) *mRT1 [{(P4/P1)^(n-1/3n)} - 1]

= 417070 J

= 417.07 kJ

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Heat rejected in each intercooler,

Let, T2 = Temp. of air at the end of stage

P2/P1 = (P4/P1)^1/3 = (70/1)^1/3 = 4.12

T2/T1 = (P2/P1)^n-1/n = (4.12)^n-1/n = 1.266

T2 = 383.6 K

Since the pr. Ratio & index of compressor is same for each stage and internal cooling is complete, therefore temp. of air will be same on entering each intercooler.

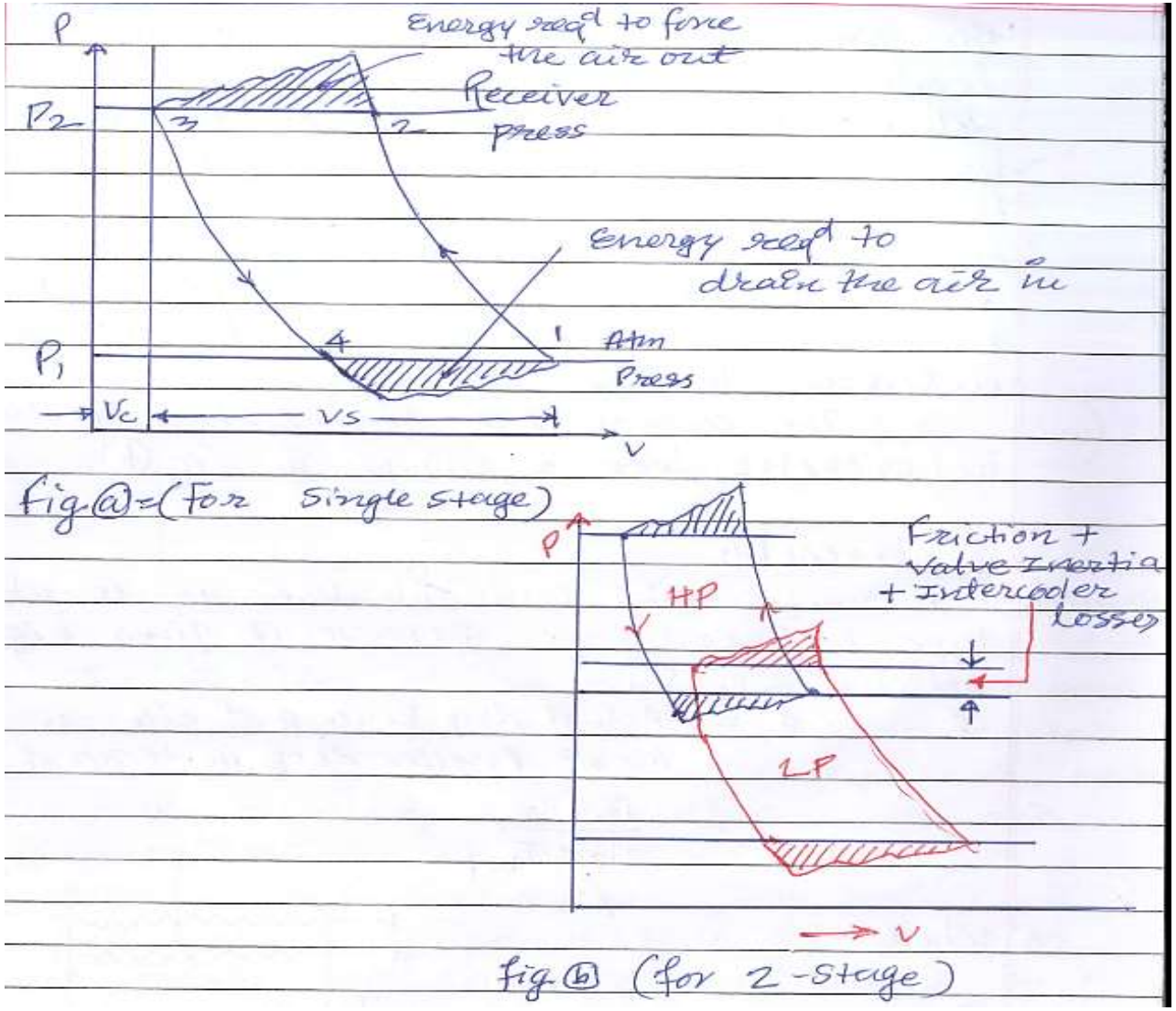
Heat rejected in each intercooler = Cp (T2 – T3)

=1.005 (383.6 -303)

= 81 KJ

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Actual Indicator diagram for RA Compressor



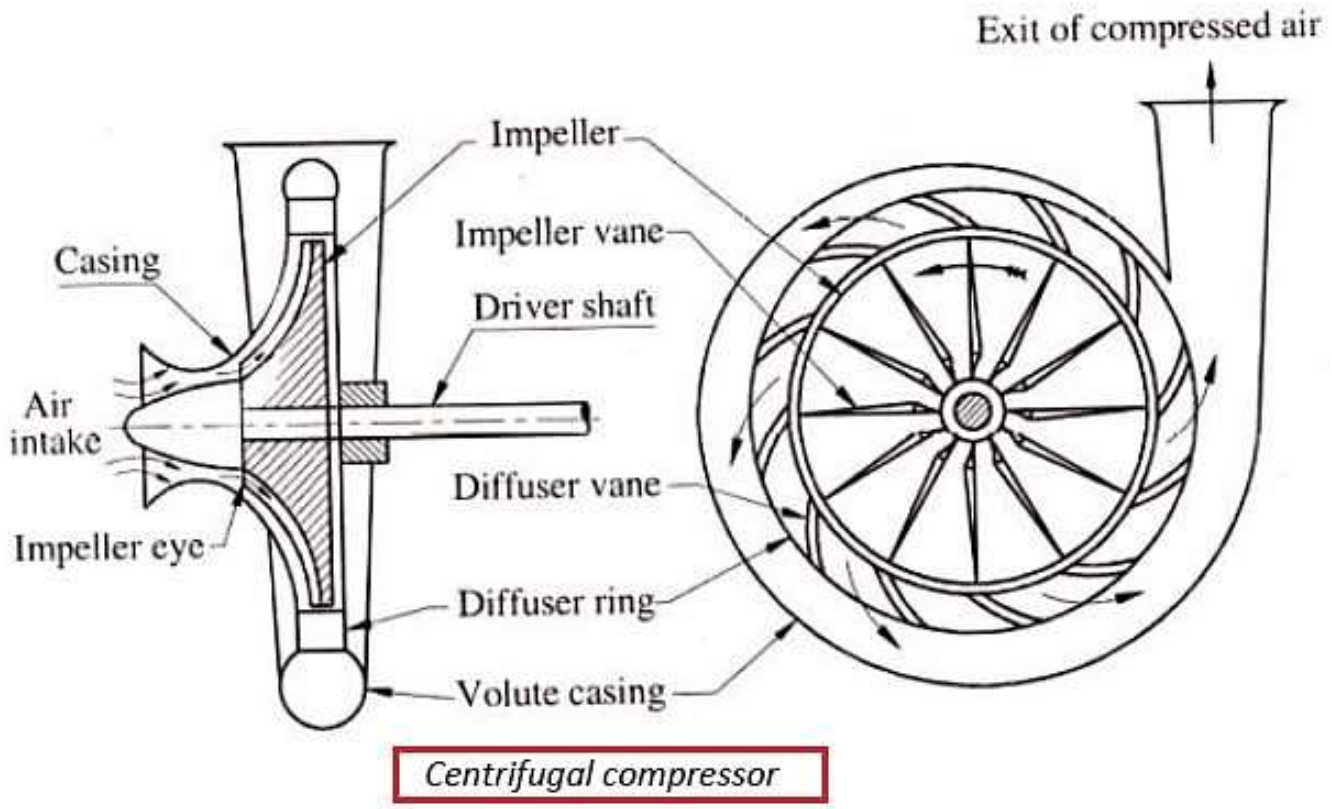
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Actual Indicator diagram

- Actual ID differs from the theoretical as there are intake and discharge losses.
- The intake losses includes the friction loss in pipe, friction loss in inlet valve and valve inertia loss. Theoretically valve should open at 4, but actually it opens little afterwards due to inertia of the valve & the press inside the cylinder falls below atmosphere.
- The oscillating part of the curve indicates the valve flutter due to vibration of spring loaded valve.
- Similar situation occurs in delivery side also. Theoretically the discharge valve thus should open at 2, but actually it opens little afterwards due to inertia of spring loaded discharge valve.
- The effect of these losses is to increase the work required per kg of air delivered to the receiver.

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Centrifugal compressor



Centrifugal compressor

A centrifugal compressor is a dynamic machine that increases gas pressure by accelerating it with a high-speed impeller and converting that kinetic energy into pressure energy through a diffuser.

It consists primarily of an inlet, a rotating impeller, a stationary diffuser, and a collector (volute), using centrifugal forces to achieve high-pressure ratios and mass flow rates.

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Centrifugal compressor

A centrifugal compressor are used to supply large quantities of air but at a lower pressure ratio. It consists of a rotating impeller, diffuser, and casing as shown in Fig.

The impeller consists of a disc on which radial blades are attached. The impeller of a centrifugal compressor can run at speeds of 20,000 to 30,000 r.p.m.

The diffuser is the other important part of the compressor which surrounds the impeller and provides a diverging passage for air flow, thus increasing the air pressure. The air coming out from the diffuser is collected in the casing and taken out from the outlet of the compressor.

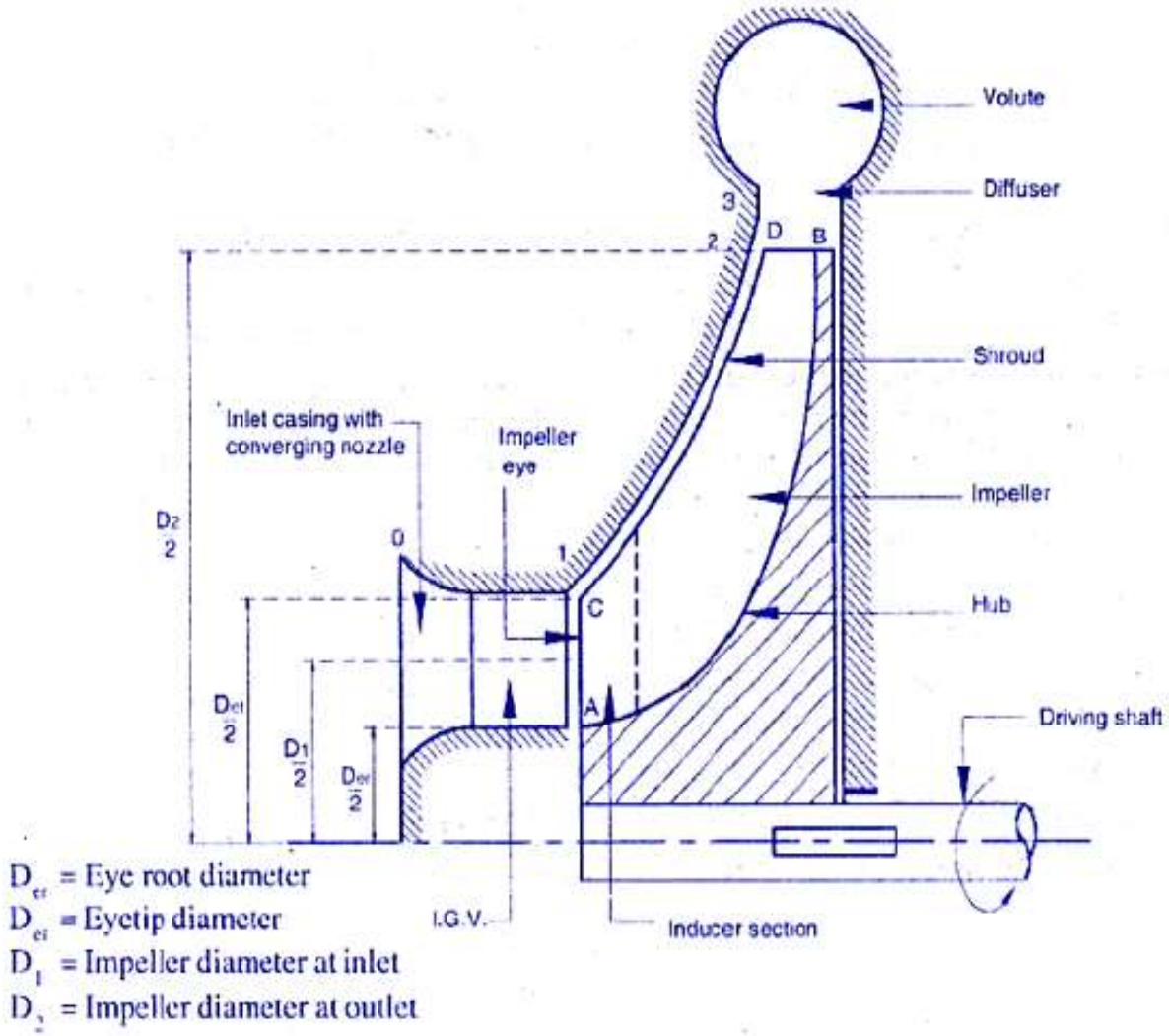
The air enters the eye of the compressor at low velocity and atmospheric pressure. The air moves radially outward passing through the impeller and is guided by the impeller vanes. The impeller increases the momentum of the air flowing through it, causing a rise in pressure and temperature of the air. The air leaving the impeller enters the diffuser where its velocity is reduced by providing more cross-sectional area for the air flow. Part of the kinetic energy of the air is converted into pressure energy passing through diffuser and pressure of the air is further increased.

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Parts of Centrifugal compressor

The principal components of a centrifugal compressor are shown in Fig.

- Inlet casing with accelerating (converging) nozzle
- Inlet guide vanes (IGV)
- Impeller
- Diffuser
- Scroll or volute
- Inducer section



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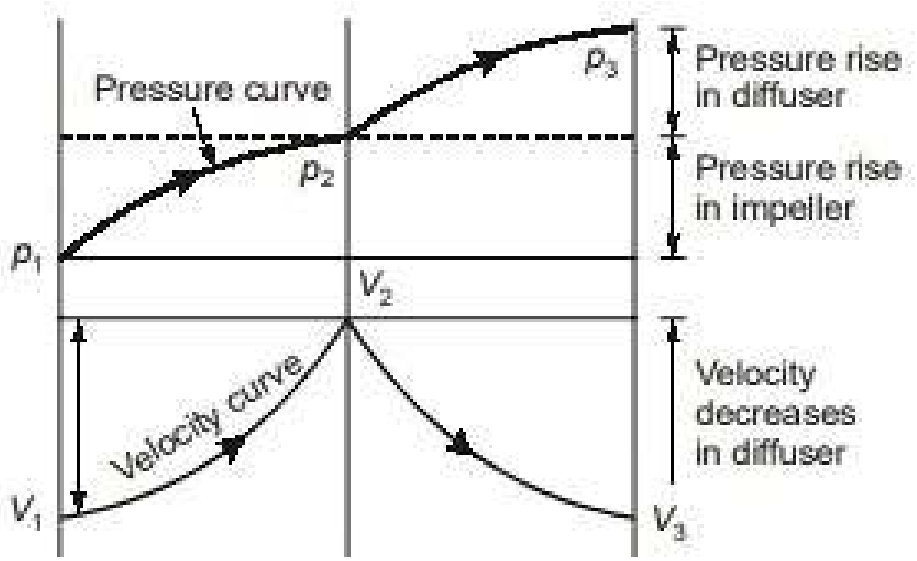
Velocity and pressure variation in Impeller and diffuser

Nearly half of the total pressure rise is achieved in the impeller and remaining half in the diffuser.

A pressure ratio of 4 can be achieved with single stage centrifugal compressor. For higher pressure ratios, multistage compressors are used. In multistage compressors, the outlet of the first stage is passed to the second stage and so on.

A pressure ratio of 12 : 1 is possible with multistage centrifugal compressors.

The change of pressure and velocity of air passing through the impeller and diffuser are shown in Fig.



Impeller and its types

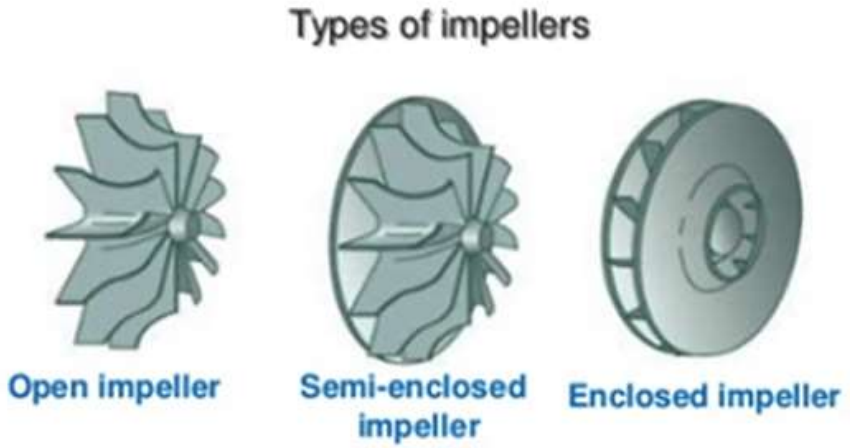
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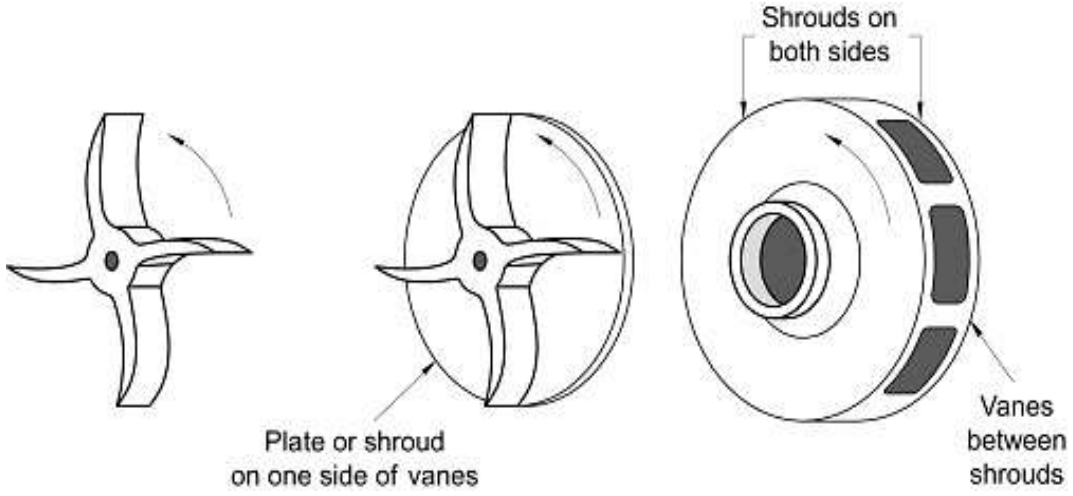
Types of Compressor Impellers by Structure

Enclosed/Shrouded Impeller: Vanes are enclosed by a shroud on both sides. These are highly efficient and are standard for high-pressure, clean gas centrifugal compressors.

Semi-Open or Semi enclosed Impeller: Vanes are attached to a back shroud but free on the front side. These offer a balance between efficiency and handling fluids that might clog a closed impeller.

Open/Un-shrouded Impeller: Vanes are attached directly to the hub with no shrouds. Often used in specialized applications, these are structurally weaker and generally less efficient.



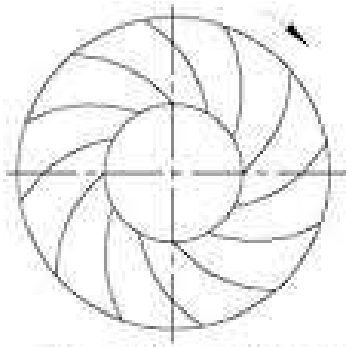


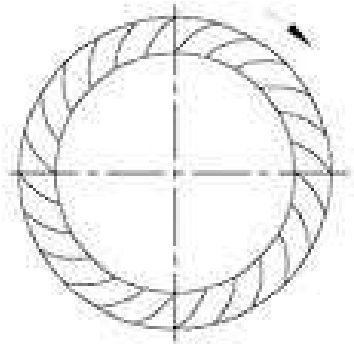
Types of Compressor Impellers by Blade Angle

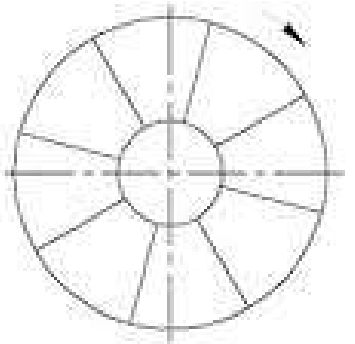
Backward-Curved Vanes (φ< 90 degree): These are the most common in modern high-efficiency compressors because they offer a stable pressure-flow characteristic and high efficiency.

Radial Vanes (φ=90 degree): Blades extend radially from the hub. These are used for lower pressure applications and provide lower efficiency than backward-curved types, but offer high structural strength.

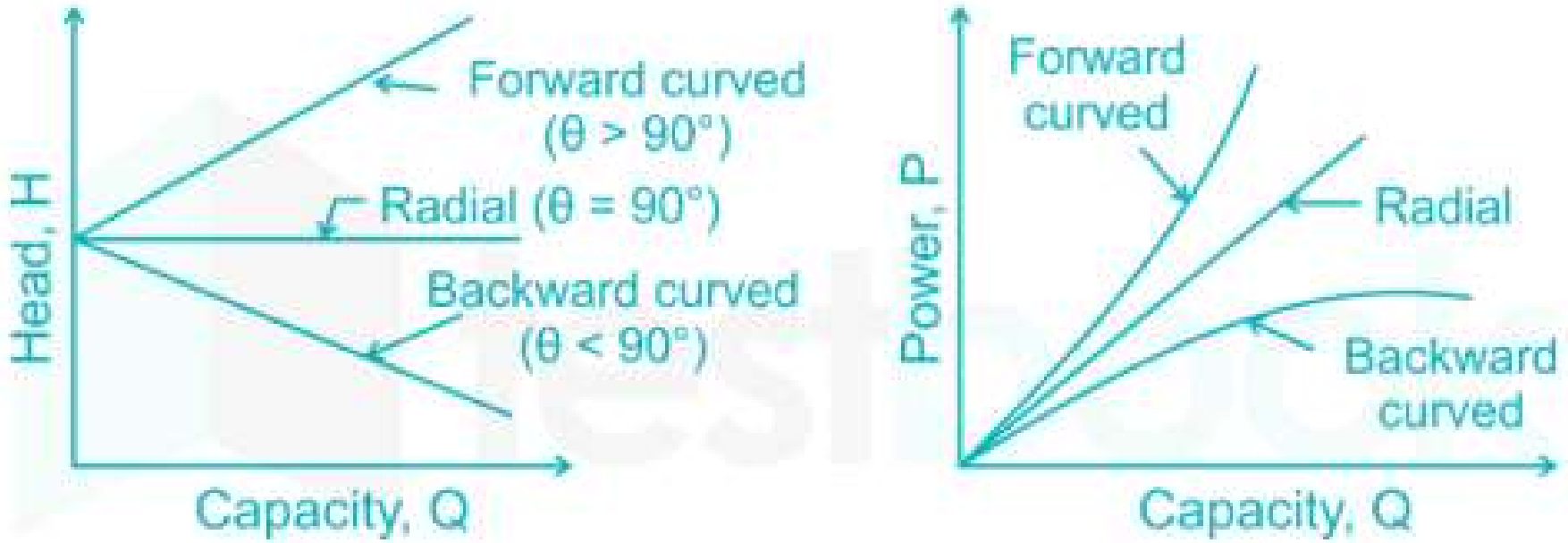
Forward-Curved Vanes (φ>90 degree): These vanes curve in the direction of rotation, providing high pressure at lower speeds, but they are rarely used in high-efficiency compression due to instability.


Backward curved blade


Forward curved blade


Radial blade

Head capacity and power capacity curves of Compressor Impelle...



Head-capacity and power-capacity curves

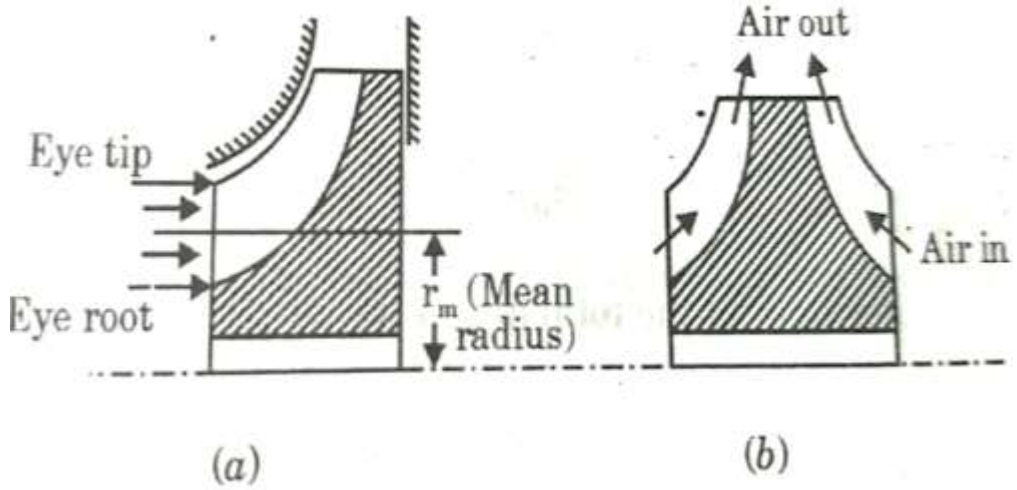
Types of Compressor Impellers by suction side

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Single eye type: Air enters into the compressor from one side.

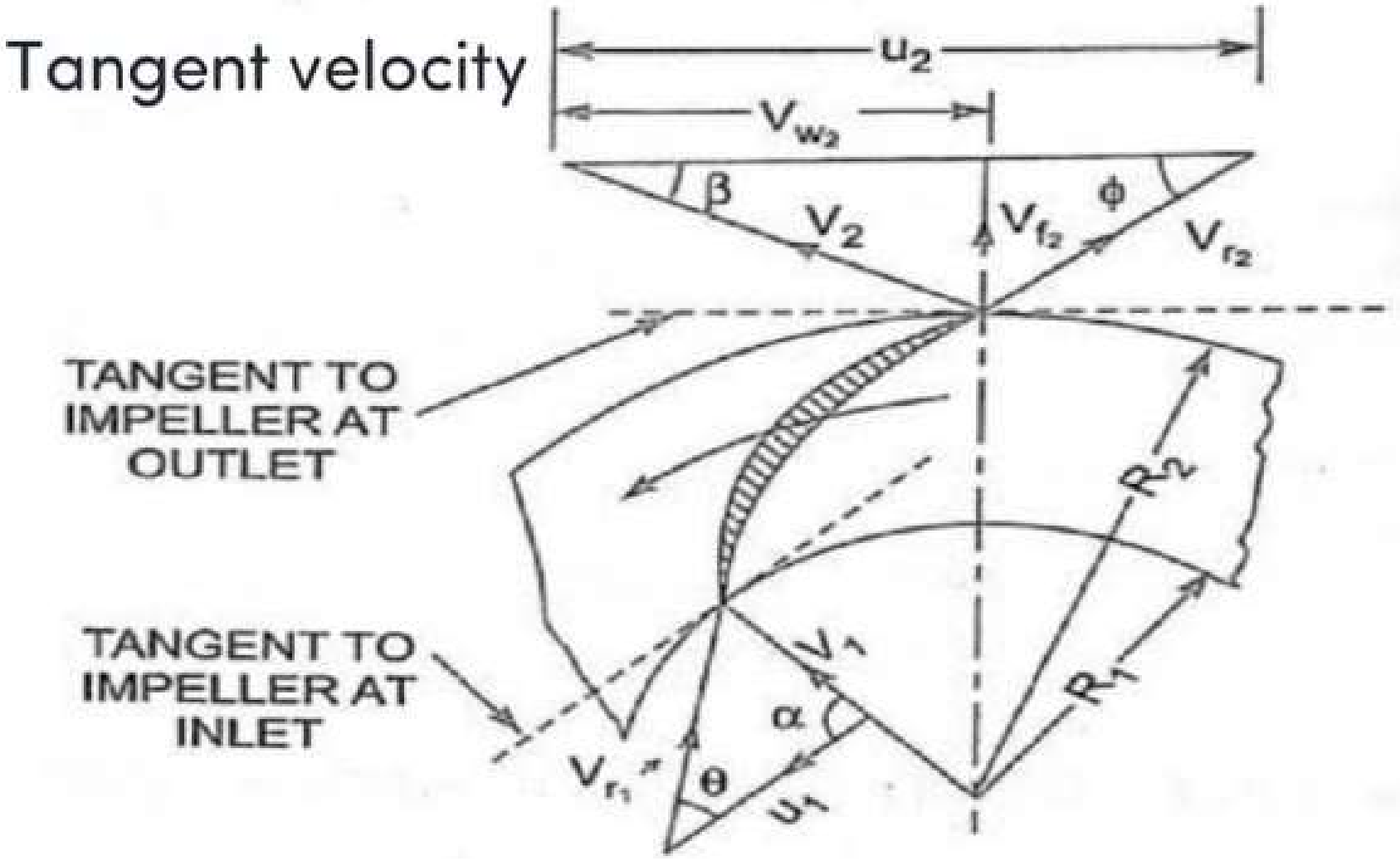
Double eye type: Air enters into the compressor from both sides as shown in figures.

These have equal axial forces in opposite direction which is advantage over single eye impeller



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Velocity diagram for Centrifugal compressor



Work done by Centrifugal Compressor

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The work supplied to fluid in a stage of compressor may be found out by applying the moment of momentum theorem.

Consider 1 Kg of air passing through the impeller.

The torque which must be applied to impeller = Rate of change of angular momentum

$$Torque\ T = V_{W2}.r_2 - V_{W1}.r_1$$

If ω – Angular velocity in rad/s, then Work done per Kg of air

$$W = Theoretical\ Torque * Ang\ Velocity = T * \omega$$

$$W = (V_{W2}.r_2 - V_{W1}.r_1)*\omega$$

$$W = (V_{W2}.U_2 - V_{W1}.U_1)$$

$$W = (h_{o2} - h_{o1})$$

OR $$W = Cp(T_{o2} - T_{o1})$$

This equation is Euler's equation or Euler's Work

Work done by Centrifugal Compressor

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If the working fluid enters radially, V_{W1} = 0, (α = 90 degree)

$$W = V_{W2} * U_2$$

If the exit from the impeller is radial , (φ = 90 degree)

$$V_{W2} = U_2$$

$$W = (U_2)^2 \text{ ----------J/Kg}$$

$$W = m * (U_2)^2 \text{ ----------Joule}$$

$$W = Cp(T_2 - T_1)$$

$$W = Cp * T_1\{Rp^{\frac{\gamma-1}{\gamma}} - 1\}$$

This equation is Euler's equation or Euler's Work

Work done by Centrifugal Compressor

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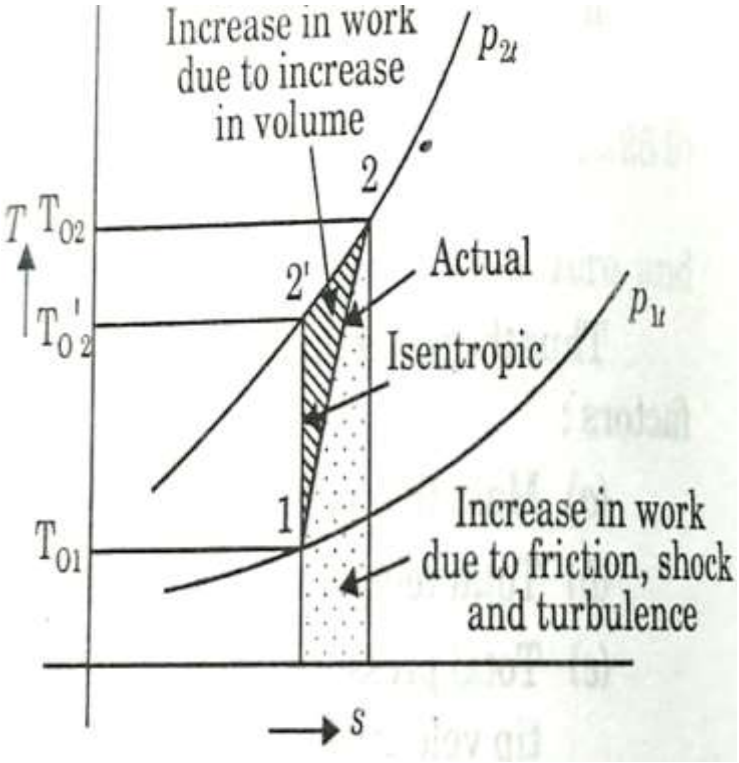
Isentropic efficiency of Centrifugal Compressor

$$\eta_i = \frac{\text{Isentropic work}}{\text{Actual work}} = \frac{h_{02}' - h_{01}}{h_{02} - h_{01}} = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}}$$

provided specific heat at constant pressure remains constant.

If $V_1 = V_2$, then

$$\eta_i = \frac{T_2' - T_1}{T_2 - T_1} = \frac{\frac{T_2'}{T_1} - 1}{\frac{T_2}{T_1} - 1} = \frac{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1}, \text{ where } n > \gamma$$



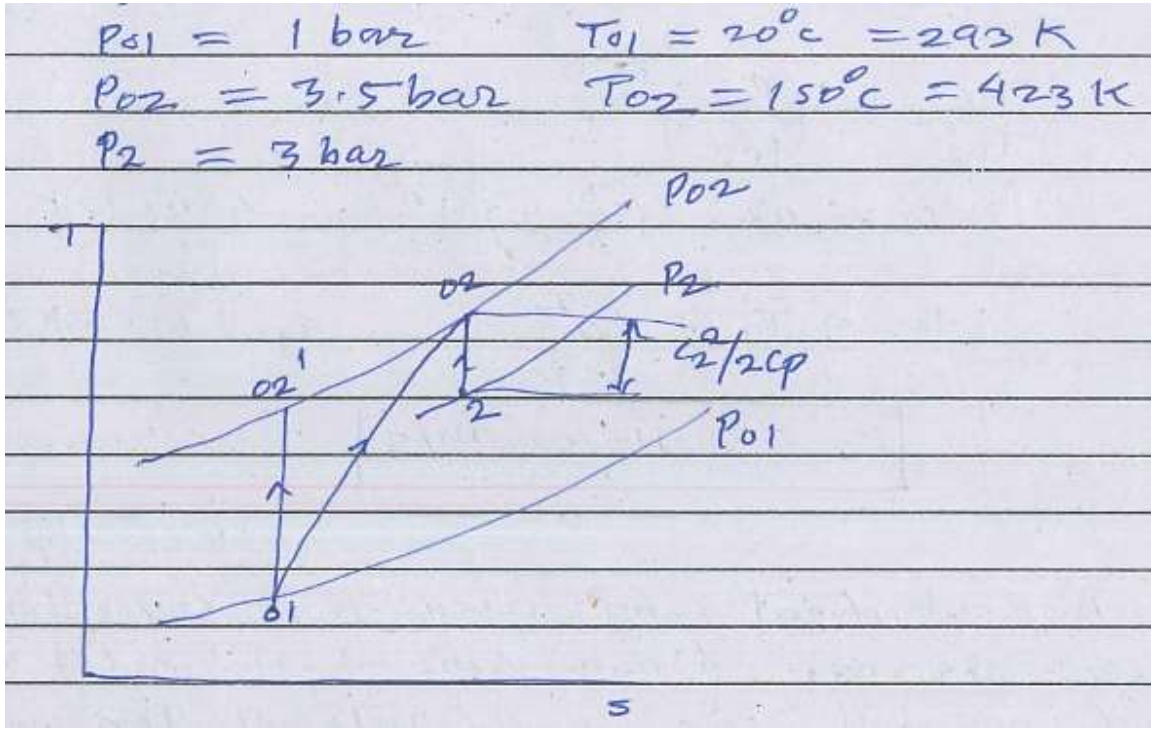
This equation is Euler's equation or Euler's Work

Numerical on Centrifugal Compressor

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The ambient conditions at the inlet are 20 degree Celsius and 1 bar. At the exit the total head temperature and pressure are 150 degree Celsius & 3.5 bar and static pressure at the exit is 3 bar. Calculate-

1) Isentropic efficiency
2) Polytropic efficiency
3) Air velocity at exit



This equation is Euler's equation or Euler's Work

Numerical on Centrifugal Compressor

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(1) $\eta_{is} = \dfrac{T_{02}' - T_{01}}{T_{02} - T_{01}}$

$T_{02}' = T_{01}\left(\dfrac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} = 293\left(\dfrac{3.5}{1}\right)^{\frac{1.4-1}{1.4}} = 419.1$

$\therefore \eta_{is} = \dfrac{419.1 - 293}{423 - 293}$

$= 0.97$

(2) $\dfrac{T_{02}}{T_{01}} = \left(\dfrac{P_{02}}{P_{01}}\right)^{\frac{n-1}{n}} \Rightarrow \dfrac{423}{293} = \left(\dfrac{3.5}{1}\right)^{\frac{n-1}{n}}$

$\therefore n = 1.4147$

$\therefore$ Polytropic eff $\eta_p = \dfrac{n}{n-1} \times \dfrac{\gamma-1}{\gamma}$ (1)

$= 0.9747$

(3) velo. at exit (C_2)

Consider the process (2–02)

(isentropic diffusion)

$\dfrac{T_2}{T_{02}} = \left(\dfrac{P_2}{P_{02}}\right)^{\frac{\gamma-1}{\gamma}}$

$T_2 = 423 \times \left(\dfrac{3}{3.5}\right)^{\frac{0.4}{1.4}} = 404.77\ K$

$T_{02} = T_2 + \dfrac{C_2^2}{2C_p}$ $\quad C_p = 1.005\ KJ/kg\ K$

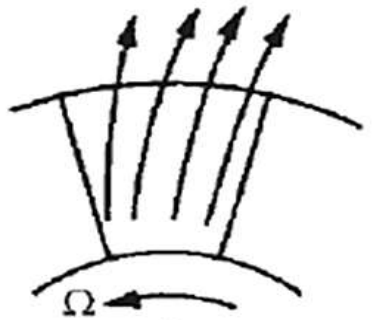
$\boxed{C_2 = 191.42\ m/s}$

This equation is Euler's equation or Euler's Work

Slip and slip factor

Even under ideal (frictionless) conditions the relative flow leaving the impeller does not receive perfect guidance from the vanes and the flow is said to *slip*. Hence, $\beta_2 > \beta_2'$.

average relative flow angle — **vane angle**



The *slip velocity is* defined as

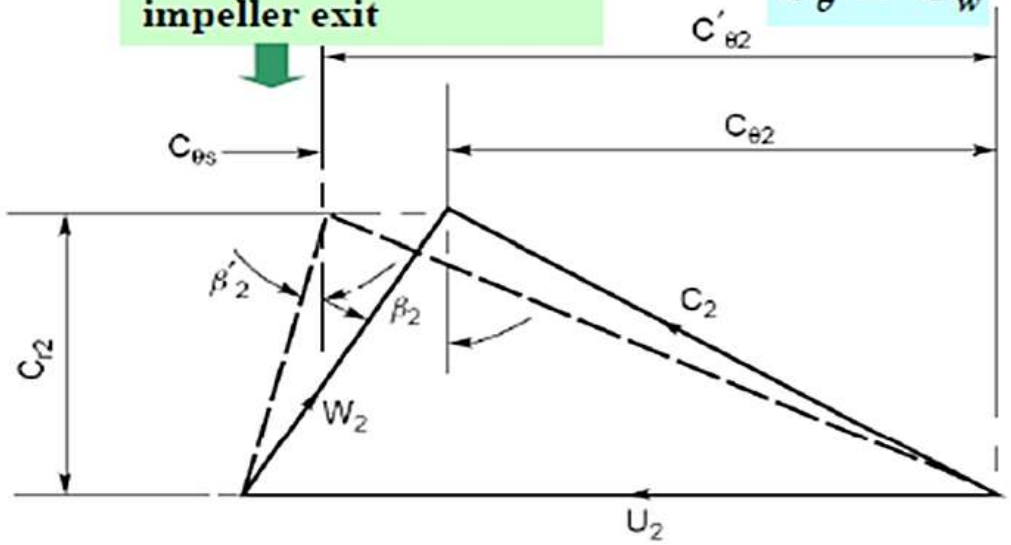
$$C_{ws} = C'_{w2} - C_{w2}$$

and Slip Factor, $\sigma = \dfrac{C_{w2}}{C'_{w2}} < 1$

For radial impeller, $\sigma = \dfrac{C_{w2}}{U_2}$

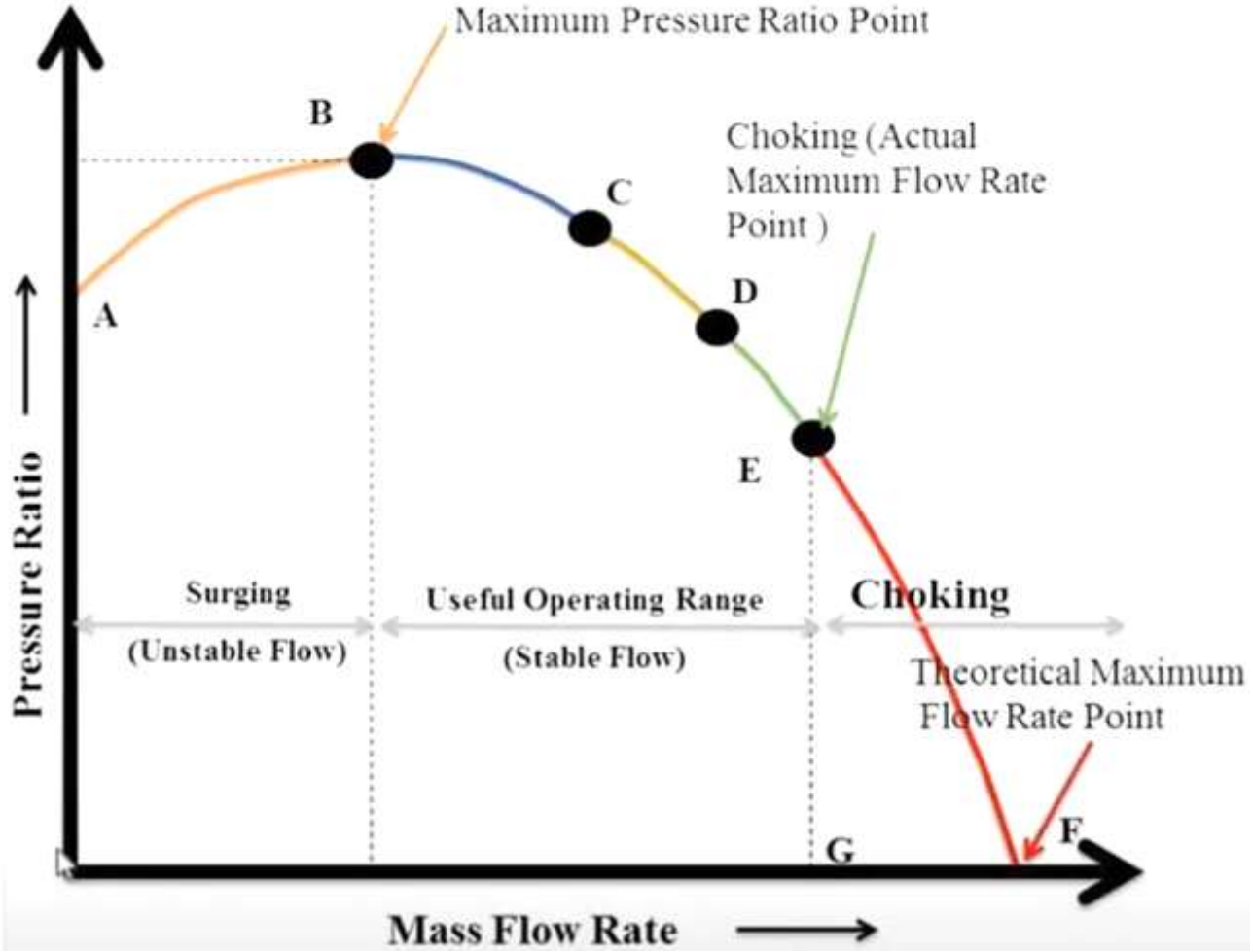
Velocity triangle at impeller exit

$C_\theta \equiv C_w$



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Surging and choking in Centrifugal compressor



The characteristic curves of any turbo compressor define the operating zone for the compressor at different speed lines and is limited by the two phenomenon called choke and surge.

These two opposing constraints can be seen in Figure. Choke conditions occurs when a compressor operates at the maximum mass flow rate.

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Surging and choking in Centrifugal compressor

Surging and choking" primarily refers to two distinct operational instabilities in gas compressors, particularly in centrifugal and axial flow types. These phenomena define the limits of a compressor's stable and efficient operating range.

Surging

Surging is an unstable, complete breakdown of steady flow through the compressor. It is the lower limit of operation, occurring at low flow rates and high discharge pressure.

Choking (Stonewalling)

Choking, also known as stonewalling, is the upper operational limit of a compressor, occurring at a maximum possible mass flow rate.

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Difference between Reciprocating and Centrifugal compressor

S.N.	Reciprocating compressor	Centrifugal Compressor
1	They are suitable for low discharge of air at very high press.	For large discharge of air at low pressure
2	Maxm delivery pr. may be as high as 1000 bar.	10 bar only.
3	Maxm free air discharge is about 300 m^3/min.	as high as 3000 m^3/min.
4	Speed of compressor is low.	The speed is high upto 50000 rpm.
5	The air supply is intermittent, therefore receiver is required.	Air supply is continuous. No receiver is required.

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Difference between Reciprocating and Centrifugal compressor

S.N.	Reciprocating compressor	Centrifugal Compressor
6	size of compressor is large for the given discharge.	Size is small for the same discharge.
7	Balancing is major problem.	There is no balancing problem.
8	The lubrication system is complicated.	Lubrication system is simple.
9	Air delivered is less clean as it comes in contact with lubricating oil.	The air delivered is more clean, as it does not come in contact with lub. oil.
10	Isothermal efficiency is used for all sorts of calculations.	Isentropic eff. is used for all sorts of calculations.

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Unit 2

Internal Combustion Engine

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Introduction to I C Engines:

Engine is a device which converts one form of Energy into another form.

Heat Engine:

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

Heat engines can be broadly classified into two categories:

a) Internal Combustion Engines (IC Engines)
b) External Combustion Engines (EC Engines)

Classification of I C Engine

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I C Engines are classified According to-

1) Nature of thermodynamic cycle as :

i) Otto cycle engine ii) Diesel cycle engine iii) Duel or mixed cycle engine

2) Number of strokes per cycle as

i) Four stroke engine and ii) Two stroke engine

3) Method of ignition as:

i) Spark ignition engine (S.I. engine)- Mixture of air and fuel is ignited by electric spark.

ii) Compression Ignition engine (C.I. engine)- The fuel is ignited as it comes in contact with hot compressed air.

4) Method of cooling as : (i) Air cooled engine and (ii) Water cooled engine

5) Speed of the engine as : (i) Low speed (ii) Medium speed (iii) High speed

Petrol engine are high speed engines & diesel engines are low to medium speed engines

Classification of I C Engine

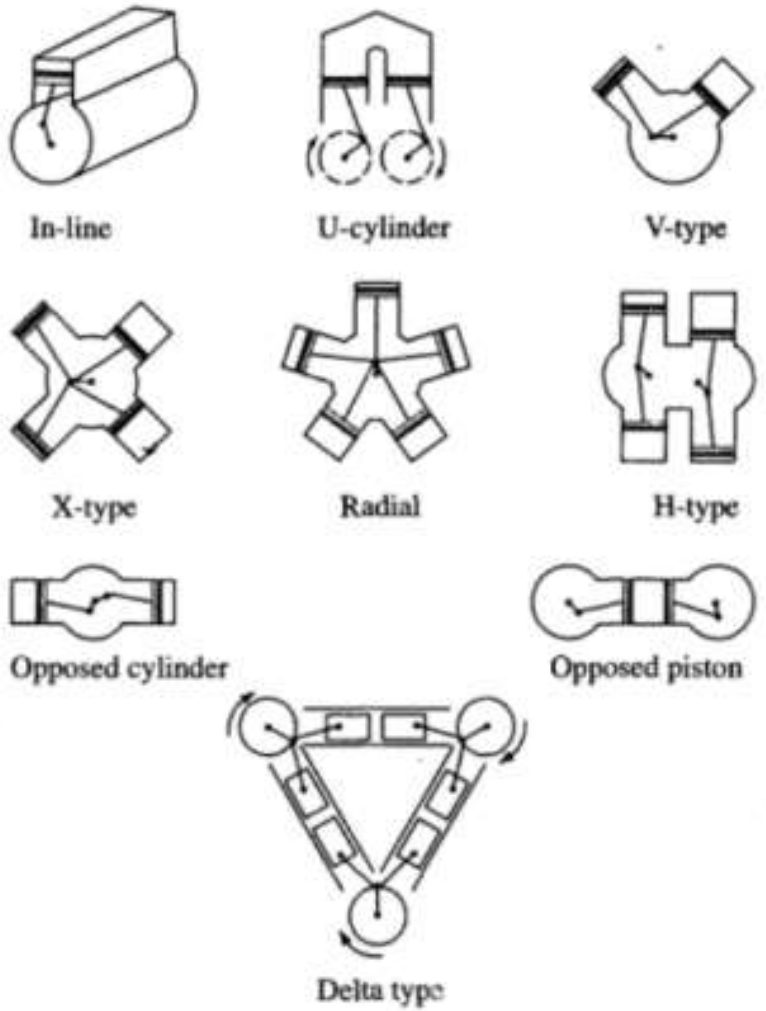
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6) Number of cylinder as :

(i) Single cylinder engine
(ii) Multi cylinder engine

7) Position of the cylinder as :

(i) Inline engines
(ii) V – engines
(iii) Radial engines
(iv) Opposed cylinder engine
(v) Opposed piston engine
(vi) X – Type engine
(vii) H – Type Engine
(viii) U – Type Engine
(ix) Delta type



Air Standard Cycle

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Accurate analysis of I C Engine is complicated. In order to understand them, it is advantageous to analyze the performance of an idealized closed cycle that closely approximates the real cycle. One such approach is Air standard cycle.

An air standard cycle is an idealized thermodynamic cycle in which air is used as the working fluid, and all processes are assumed to be reversible, with heat addition and rejection replacing actual combustion and exhaust, to simplify the analysis of engine performance.

Assumptions in Air Standard Cycle

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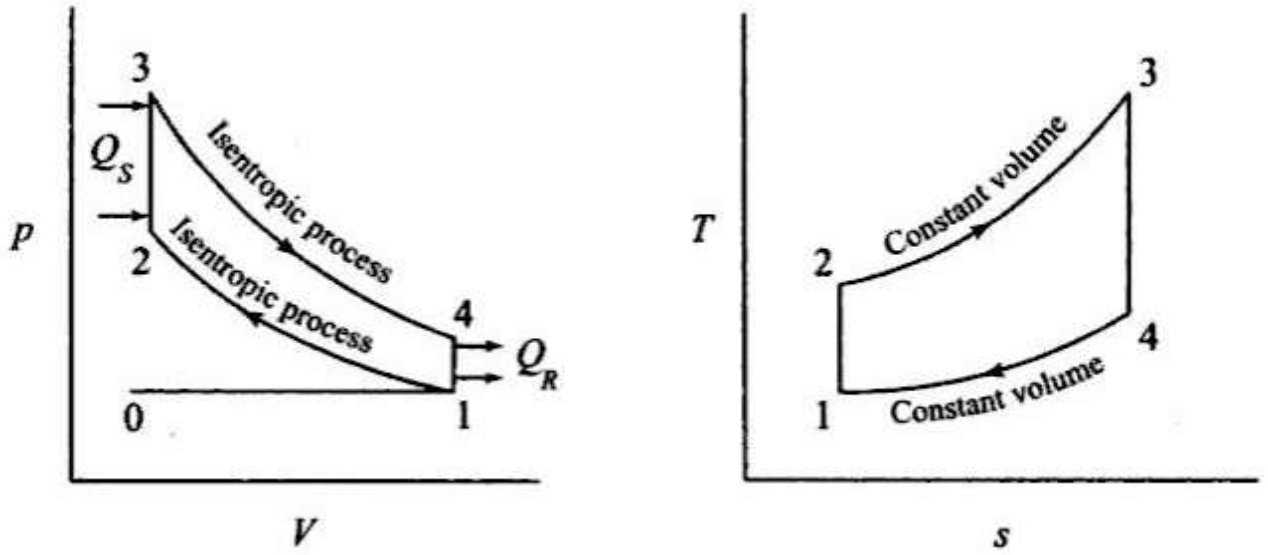
- Working Fluid: The working fluid is considered to be air, behaving as an ideal gas, throughout the entire cycle.
- Closed System: The mass of the air within the cylinder remains constant (no mass leakage).
- Constant Specific Heats: The specific heats of air are assumed to be constant at all temperatures.
- Reversible Processes: All processes are assumed to be internally reversible (no friction, no turbulence).
- Isentropic Compression/Expansion: Compression and expansion processes are assumed to be adiabatic and reversible (isentropic).
- Heat Addition/Rejection: Combustion is replaced by heat addition from an external source, and exhaust is replaced by heat rejection to an external sink, restoring the air to its initial state.
- No Chemical Reaction: No chemical reactions or changes in chemical equilibrium occur in the working fluid.

Air Standard Cycles

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- Otto cycle- Constant Volume cycle
- Diesel cycle- Constant Pressure cycle
- Dual cycle- Limited pressure cycle

Otto Cycle



Process 0 -1 and 1-0 on PV diagram represents the suction and exhaust Process and their effects are nullified

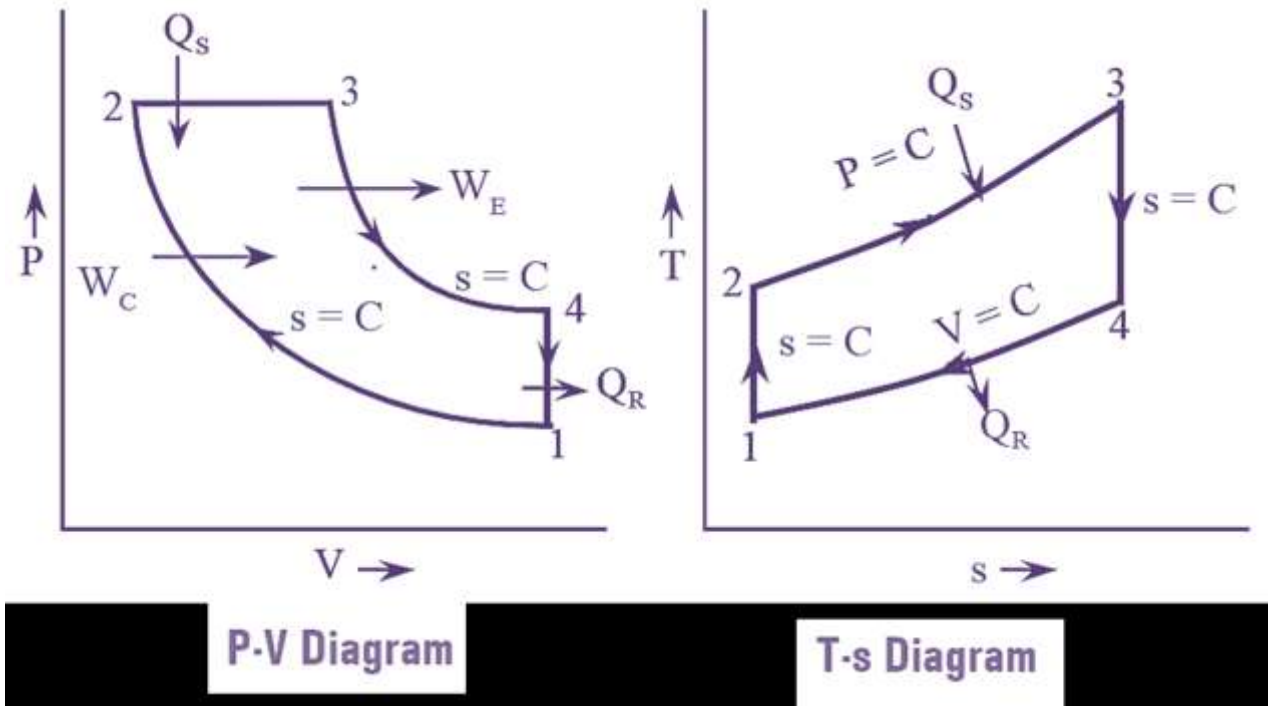
Process 1-2: Isentropic compression
Process 2-3: Heat addition at Constant volume
Process 3-4 Isentropic expansion
Process 4-1: Heat rejection at constant volume

Efficiency of Otto cycle $= 1 - \frac{1}{r^{\gamma-1}}$

Where r = compression ratio= $= \frac{V_1}{V_2}$

γ - Adiabatic Index of Compression & Expansion

Diesel Cycle



- Process 0 -1 and 1-0 on PV diagram represents the suction and exhaust Process and their effects are nullified
- Process 1-2: Isentropic compression
- Process 2-3: Heat addition at Constant Pressure
- Process 3-4 Isentropic expansion
- Process 4-1: Heat rejection at constant volume

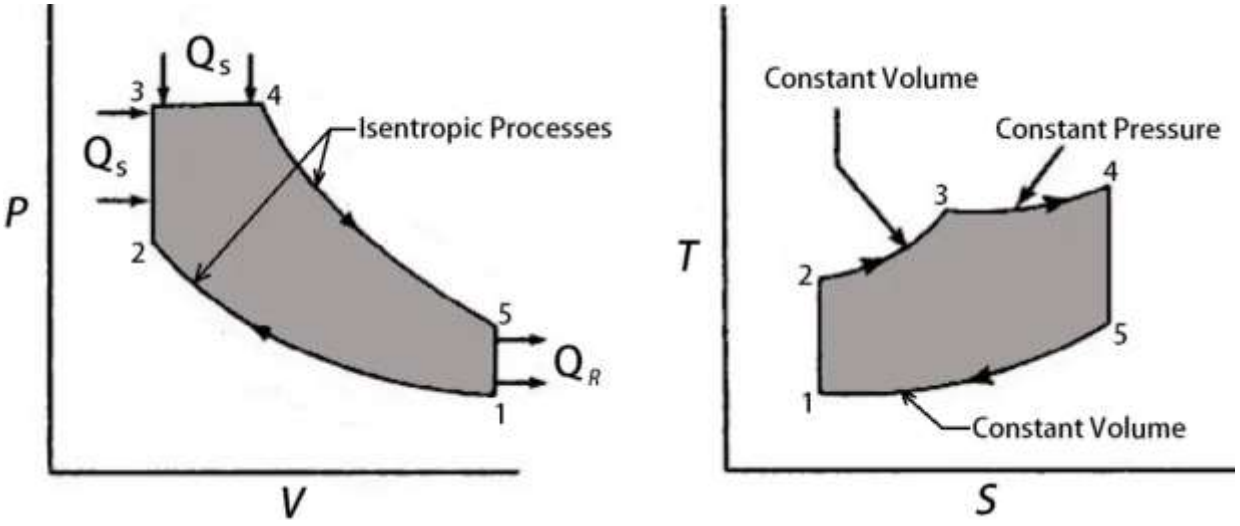
$$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}} \left| \frac{\rho^{\gamma}-1}{\rho-1} \right|$$

Where ***r*** - compression ratio= $= \frac{V_1}{V_2}$

γ - Adiabatic Index of Compression & Expansion

ρ - Cut-off ratio= $\frac{V_3}{V_2}$

Duel Cycle



- Process 1–2: Isentropic compression
- Process 2–3: Constant volume heat addition
- Process 3–4: Constant pressure heat addition
- Process 4–5: Isentropic expansion
- Process 5–1: Constant volume heat rejection

Efficiency of Otto cycle = $\eta = 1 - \frac{1}{r^{k-1}}\left[\frac{r_p r_c^k - 1}{(r_p - 1) + k r_p (r_c - 1)}\right]$

[redacted]

r_c- Cut-off ratio= $\frac{V_4}{V_3}$

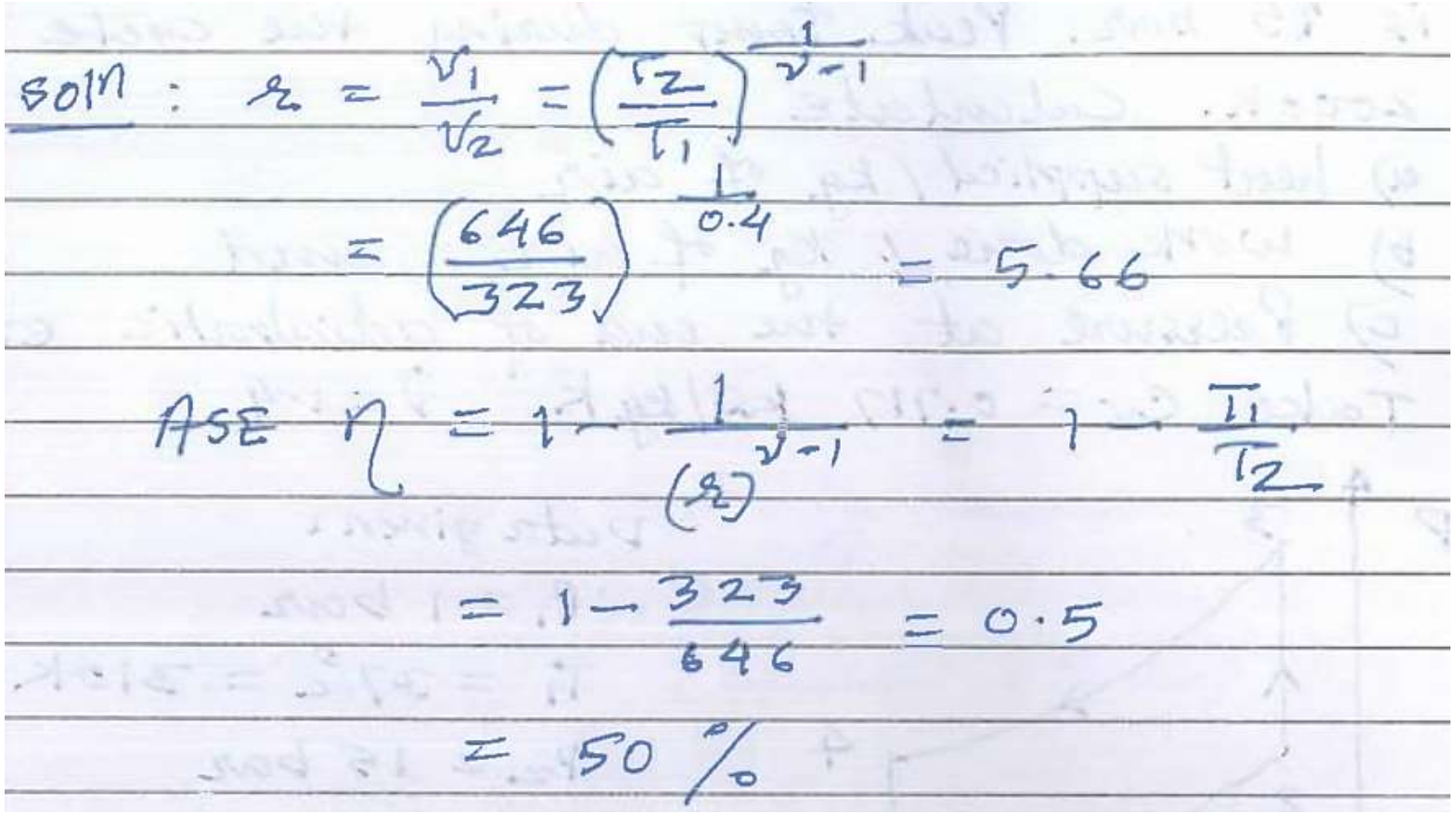
r_p = Pressure ratio= = $\frac{P_3}{P_2}$

k- Adiabatic Index of Compression & Expansion

Numerical on Air standard cycle

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In an engine working on an ideal Otto Cycle the temperatures at the beginning and end of compression are 50 and 323 degree Celsius respectively. Find the compression ratio and the air standard cycle efficiency.



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

$= \left(\frac{646}{323}\right)^{\frac{1}{0.4}} = 5.66$

ASE $\eta = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{T_1}{T_2}$

$= 1 - \frac{323}{646} = 0.5$

$= 50\ \%$

Numerical on Air standard cycle

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In A diesel engine has a compression ratio of 20 and cut-off takes place at 5% of stroke. Find the air standard efficiency. (Take gamma , γ =1.4)

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

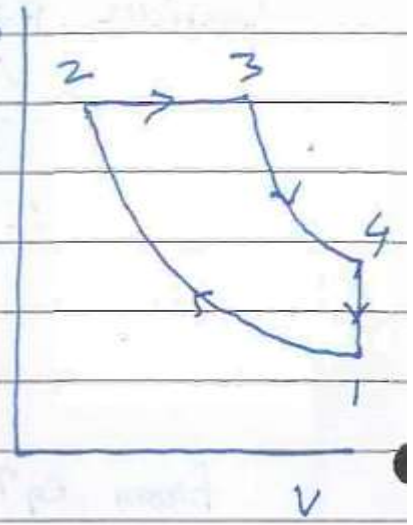
$V_1 = 20 V_2$

$V_s = 20 V_2 - V_2 = 19 V_2$

$V_3 = 0.05 V_s + V_2$

$= 0.05 \times 19 V_2 + V_2$

$V_3 = 1.95 V_2$



cut off ratio $r_c = \frac{V_3}{V_2} = \frac{1.95 V_2}{V_2} = 1.95$

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

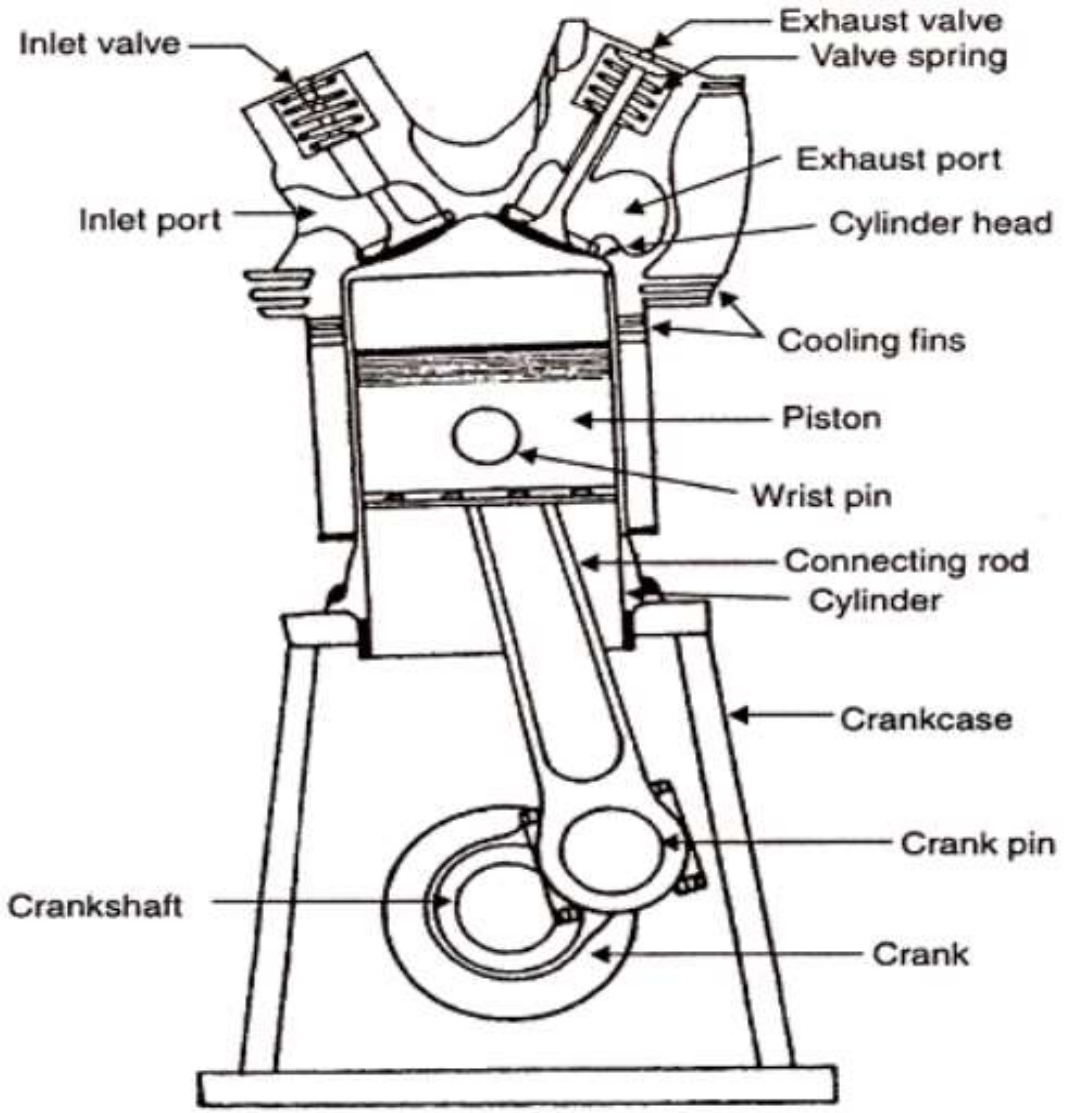
$= 1 - \frac{1}{(20)^{0.4}} \left[\frac{(1.95)^{1.4} - 1}{1.4 \times (1.95 - 1)} \right]$

$= 0.649$

$\eta = 64.9\ \%$

Various Parts of I C Engine

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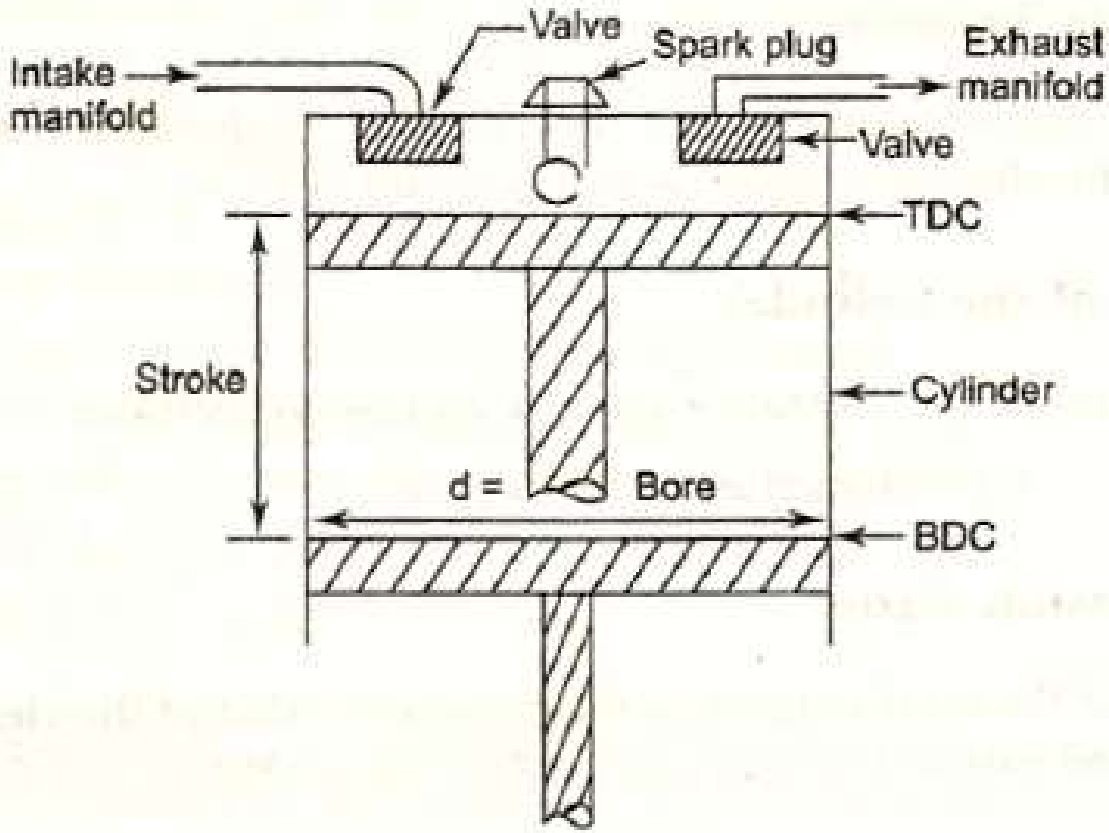


Terminology used in I C Engine

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- Bore- Bore is the diameter of the engine cylinder.
- Stroke - It is the linear distance traveled by the piston from Top dead centre (TDC) to Bottom dead centre (BDC).
- Stroke-bore ratio -The ratio of length of stroke (L) and diameter of bore (D) of the cylinder is called stroke-bore ratio (L/D). In general, this ratio varies between 1 to 1.45
- Swept volume - It is the volume (A x L) displaced by one stroke of the piston where A is the cross sectional area of piston and L is the length of stroke
- Top dead centre - When the piston is at the top of its stroke, it is said to be at the top dead centre (TDC),
- Bottom dead centre - when the piston is at the bottom of its stroke, it is said to be at its bottomdead centre (BDC).

Terminology used in I C Engine



Compression ratio –

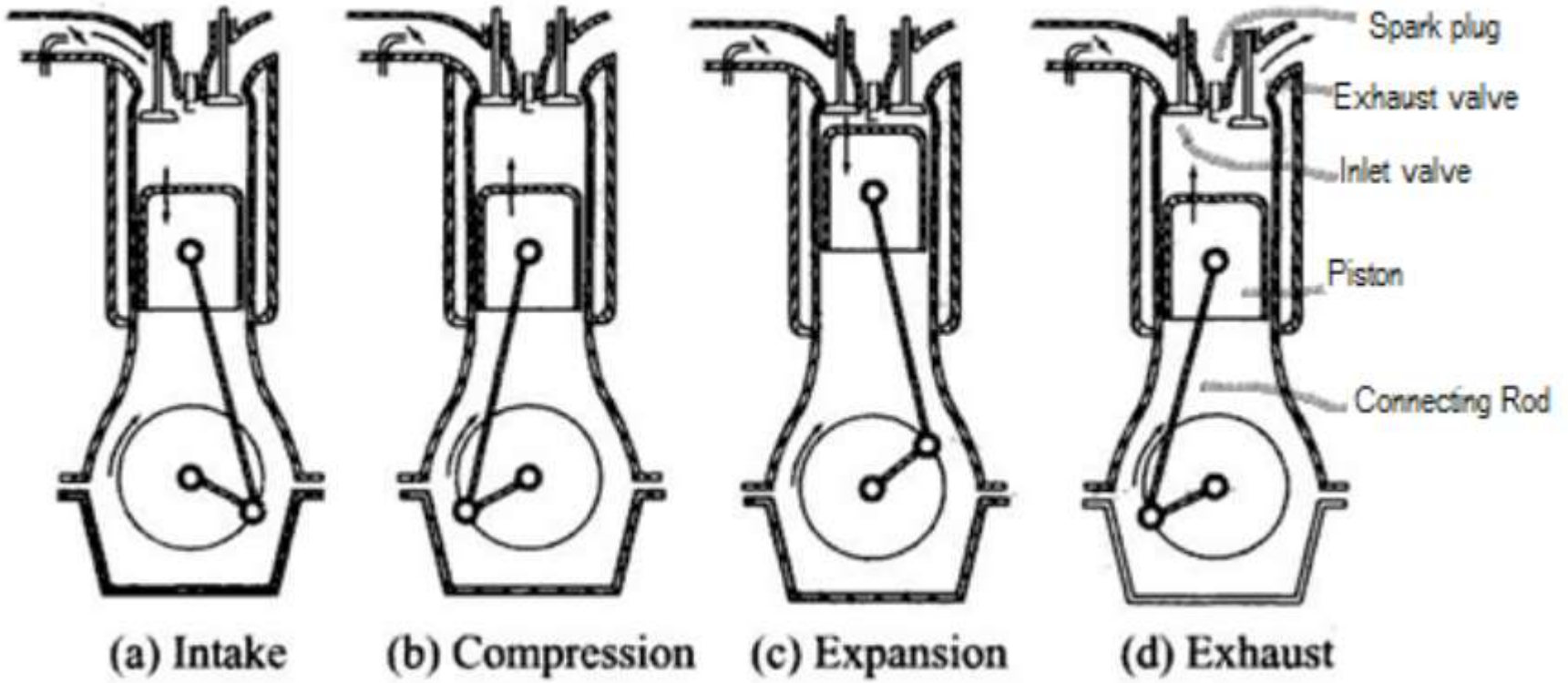
It is the ratio of the volume of the cylinder at the beginning of the compression stroke to that at the end of compression stroke, i.e. ratio of total cylinder volume to clearance volume.

The Compression ratio of diesel engine varies from 14:1 to 22:1 and that of carburetor type engine (spark ignition engine) varies from 6:1 to 10:1.

Working of 4 stroke (SI or CI) Engine

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In four stroke cycle engines the four events namely **suction, compression, power and exhaust** take place inside the engine cylinder. The four events are completed in four strokes of the piston (two revolutions of the crank shaft).



Working of 4 stroke (SI or CI) Engine

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Suction stroke

During suction stroke inlet valve opens and the piston moves downward. Only air or a mixture of air and fuel are drawn inside the cylinder. The exhaust valve remains in closed position during this stroke. The pressure in the engine cylinder is less than atmospheric pressure during this stroke.

Compression stroke

During this stroke the piston moves upward. Both valves are in closed position. The charge taken in the cylinder is compressed by the upward movement of piston. If only air is compressed, as in case of diesel engine, diesel is injected at the end of the compression stroke and ignition of fuel takes place due to high pressure and temperature of the compressed air. If a mixture of air and fuel is compressed in the cylinder, as in case of petrol engine, the mixture is ignited by a spark plug.

Working of 4 stroke (SI or CI) Engine

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Power stroke

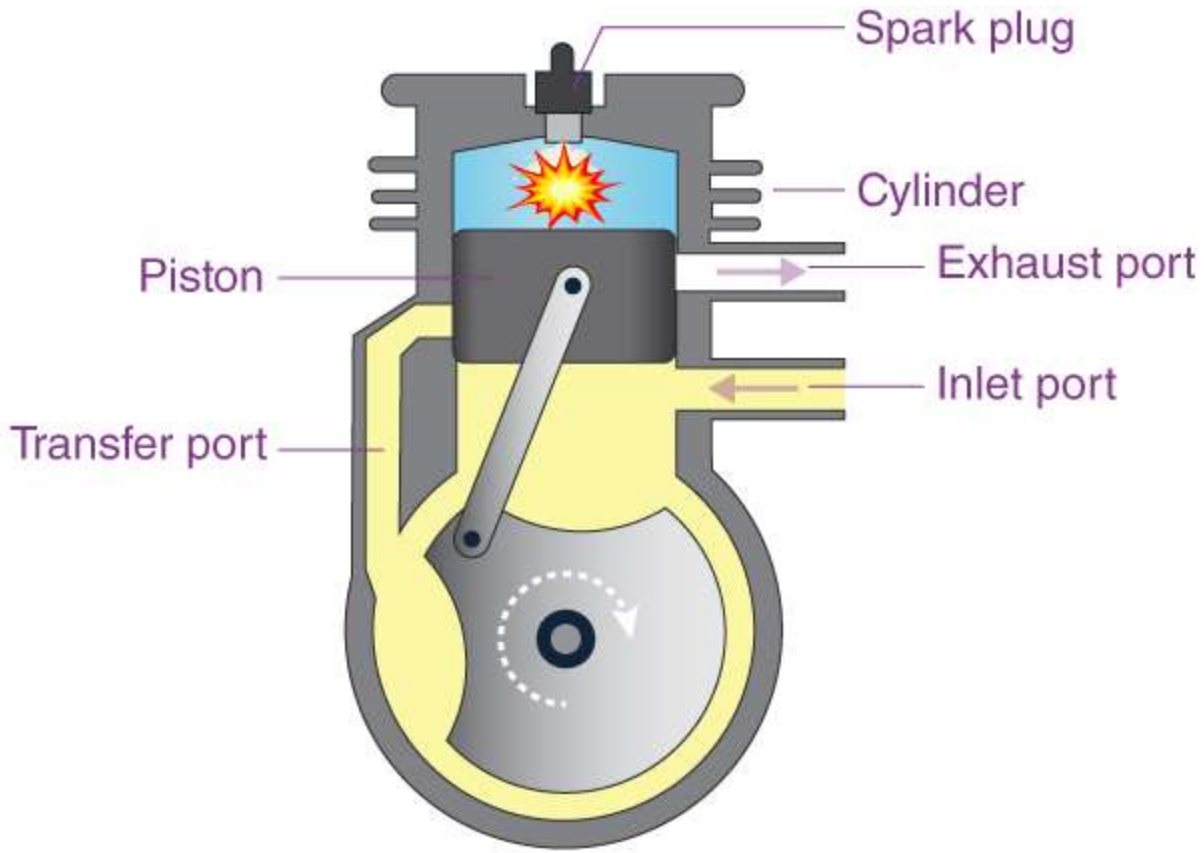
After ignition of fuel, tremendous amount of heat is generated, causing very high pressure in the cylinder which pushes the piston downward. The downward movement of the piston at this instant is called power stroke. The connecting rod transmits the power from piston to the crank shaft and crank shaft rotates. Mechanical work can be taped at the rotating crank shaft. Both valves remain closed during power stroke.

Exhaust stroke

During this stroke piston moves upward. Exhaust valve opens and exhaust gases go out through exhaust valves opening. All the burnt gases go out of the engine and the cylinder becomes ready to receive the fresh charge. During this stroke inlet valve remains closed. Thus it is found that out of four strokes, there is only one power stroke and three idle strokes in four stroke cycle engine. The power stroke supplies necessary momentum for useful work.

Working of 2 stroke (SI Engine)

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Two Stroke engine consist of ports instead of valves.

- **Inlet port**
- **Transfer port**
- **Exhaust port**

- **Cycle is completed in two strokes of piston or one revolution of crankshaft.**

Working of 2 stroke (SI or CI) Engine

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Upstroke (Compression & Intake):

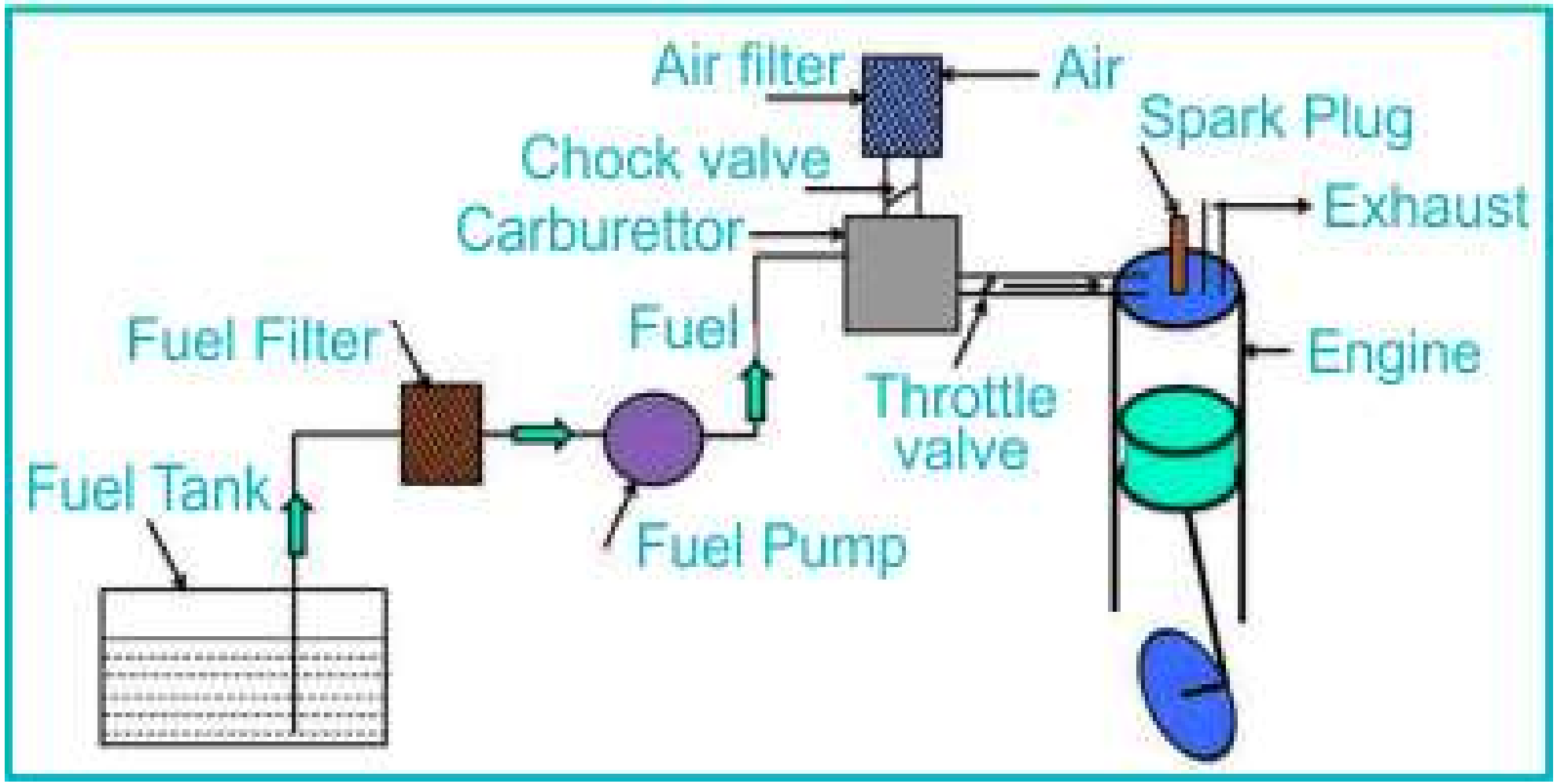
As the piston moves up, it compresses the air-fuel mixture above it in the cylinder, ready for ignition. Simultaneously, a vacuum is created in the crankcase below, which draws a fresh air/fuel/oil mixture into the crankcase through a port.

Down stroke (Power & Exhaust):

The spark plug ignites the compressed fuel, forcing the piston down, which turns the crankshaft. As the piston moves down, it uncovers the exhaust port, allowing exhaust gases to escape, and a transfer port, allowing the fresh mixture in the crankcase to rush into the cylinder (known as scavenging)

Basic Fuel supply system for Engine

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Difference between SI and CI Engines

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S. No.	S. I. Engine	C. I Engine
1	Works on the Otto cycle or constant volume heat addition cycle	Works on Diesel cycle or constant pressure heat addition cycle.
2	Gasoline (petrol), a highly volatile fuel, is used.	Diesel oil, a non-volatile fuel is used.
3	A gaseous mixture of fuel and air is introduced during the suction stroke.	Fuel is injected directly into the combustion chamber at high pressure at the end of the compression stroke.
4	Requires an ignition system with a spark plug in the combustion chamber.	Self ignition occurs due to the high temperature of air (resulting from a high compression ratio).
5	Compression Ratio (CR): Typically ranges from 6 to 10.	16 to 20.

Difference between SI and CI Engines

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S. No.	S. I. Engine	C. I Engine
6	Due to light weight and homogeneous combustion, these are high-speed engines	Due to heavy weight and heterogeneous combustion, they are low speed engines.
7	Thermal efficiency is relatively low (up to 26%) due to the low compression ratio.	Thermal efficiency is up to 40% due to the high compression ratio.
8	Pressure at the end of compression is about 10 bar.	Pressure is about 35 bar.
9	A carburettor is required, though modern engines use petrol injection.	A fuel pump and fuel injector are necessary.
10	Due to light weight and homogeneous combustion, these are high-speed engines.	Due to heavy weight and heterogeneous combustion, they are low speed engines.

Difference between 2 Stroke and 4 stroke Engines

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S. No.	2 stroke Engine	4 stroke Engine
1	Working cycle is completed in two strokes of piston or one revolution of the crankshaft	4 strokes of piston or 2 revolutions of crankshaft.
2	For same speed, it develops 1.7 to 1.8 times the power developed by 4 strokes.	Less power developed compared to 2-stroke.
3	For same power developed, they are lighter & occupy less space.	Comparatively heavy & occupies more space
4	Turning moment diagram is more uniform. Small flywheel is required.	TMD (Turning Moment Diagram) is comparatively less uniform & more size of flywheel is required.
5	Mechanism is much simpler.	Mechanism is not as simple as 2-stroke engine.

Difference between 2 Stroke and 4 stroke Engines

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S. No.	2 stroke Engine	4 stroke Engine
6	Easy to start.	Less easy to start compared to 2-stroke.
7	Thermal efficiency is less.	Thermal efficiency is more.
8	Capacity of cooling system must be higher.	Capacity is less.
9	Consumption of lubricating oil is more.	Consumption of lubricating oil is Less.
10	Exhaust gases create noise.	Comparatively less noise.

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Unit 3

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I C Engine testing

Engine Testing includes the measurement of the following parameters

- Indicated Power (IP)
- Brake Power (BP)
- Frictional Power (FP)
- Mechanical Efficiency
- Thermal Efficiency
- Brake Specific Fuel Consumption (bsfc)
- Heat Balance Sheet

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I C Engine testing

Brake Power (BP)-Power available at output shaft/ crank shaft

Mechanical Losses/Frictional Power(FP)-Sum of frictional losses and pumping losses including power required to operate engine accessories like water pump, dynamo etc.

Indicated Power (IP)/Theoretical Power-Power produced within engine cylinder

$$IP = BP + FP$$

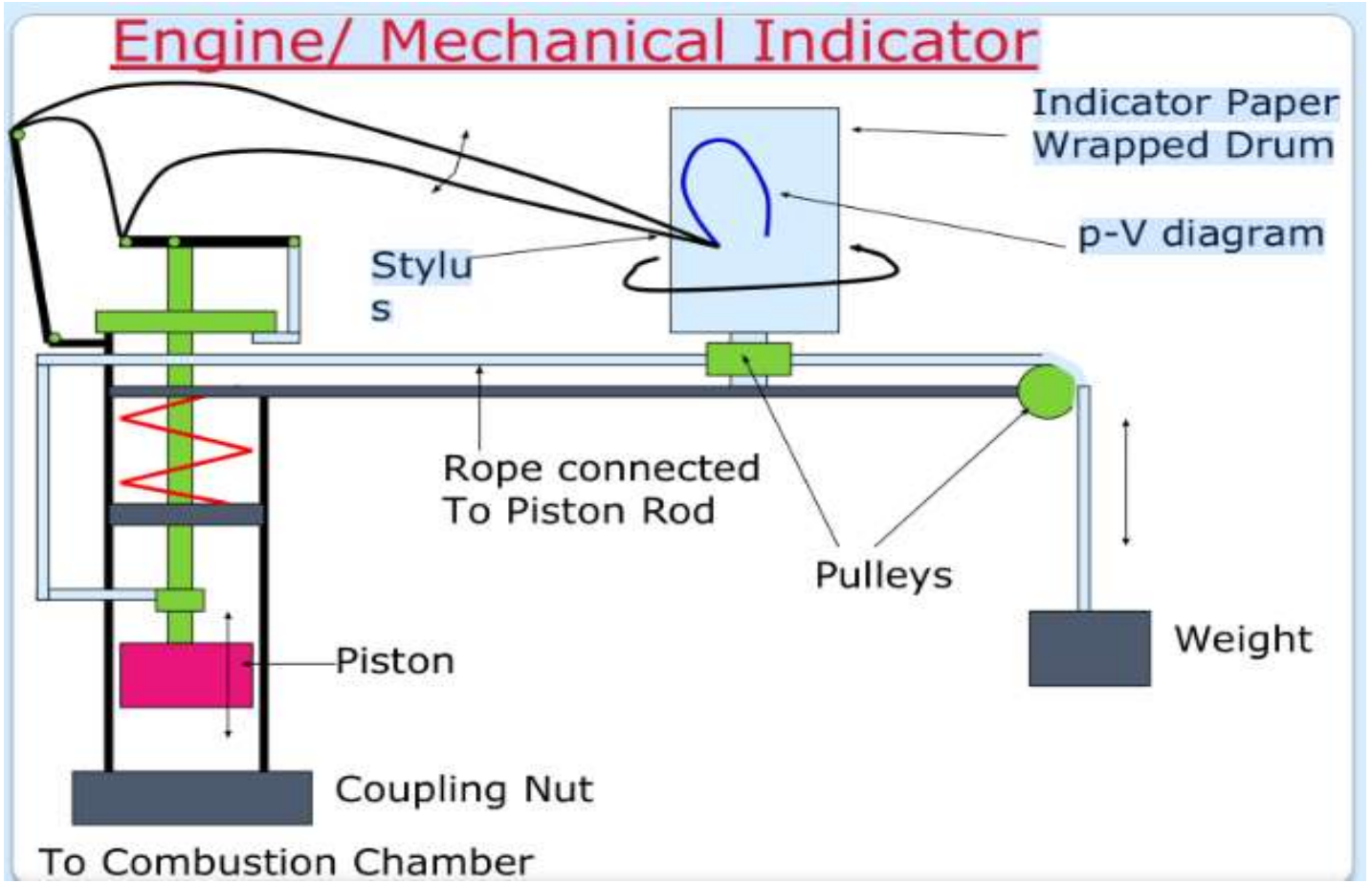
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I C Engine testing

Tests Performed on IC Engines

- Indicated Power (IP)
- Brake Power (BP)
- Frictional Power (FP)
- Mechanical Efficiency
- Air-Fuel Ratio
- Brake Specific Fuel Consumption (bsfc)
- Thermal Efficiency
- Working out Heat Balance Sheet

Measurement of Indicated power (I.P.)



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Measurement of IP on Mech/Eng Indicator

- To determine IP, p-V diagram is required, the area of which represents work developed by engine per cycle
- Apparatus used for drawing actual p-V diagram is called Mechanical/ Engine Indicator
- Eng indicator consists of a cylinder, piston, piston rod coupling nut, straight line linkage with stylus, spring of required stiffness, indicator card wrapped drum, pulley, rope and weights .
- Vertical movement of stylus and horizontal movement of the cord combines to produce a closed figure called Indicator diagram
- Area of indicator diagram can be measured by Planimeter to a definite scale giving work developed

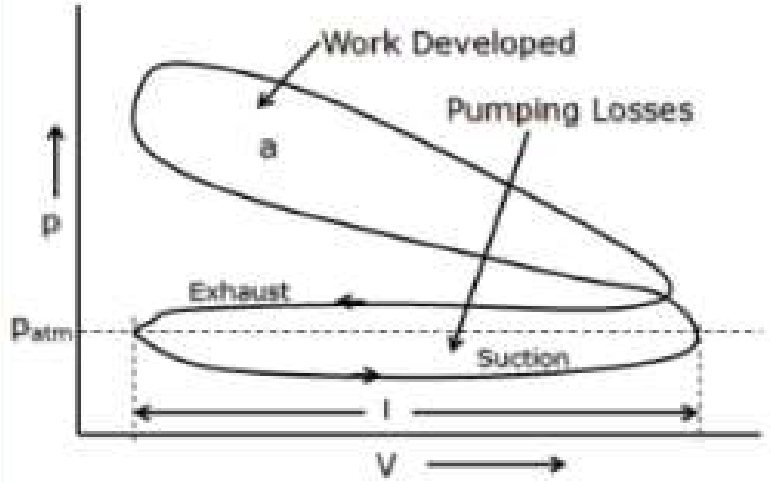
Mean Effective Pressure

Indicated Mean Effective Pressure (imep)

- imep is the average pressure, which if acted over the entire stroke length, would produce the same work done by the piston as is actually produced by the engine during a cycle
- Let 'a' be the net area of indicator diagram (cm^2)
  'l' be length of diagram (cm) and
  'k' be spring stiffness $N/cm^2/cm$

Hence, mean height of diagram = a/l

$$\therefore\ p_m\,(imep) = \frac{a}{l}\,xk \quad N/cm^2$$



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Indicated Power (IP)

Let p_m = imep (N/cm^2); L = Length of stroke (m)

A = Piston top area (cm^2) = $\Pi d^2/4$; N = RPM

n = Power stroke /min (=N/2 for 4 S eng as one power stroke per 2 rev & N for 2S eng)

Hence, Force on Piston = P_m x A (Newton)

WD per Cycle = p_m x A x L (Nm)

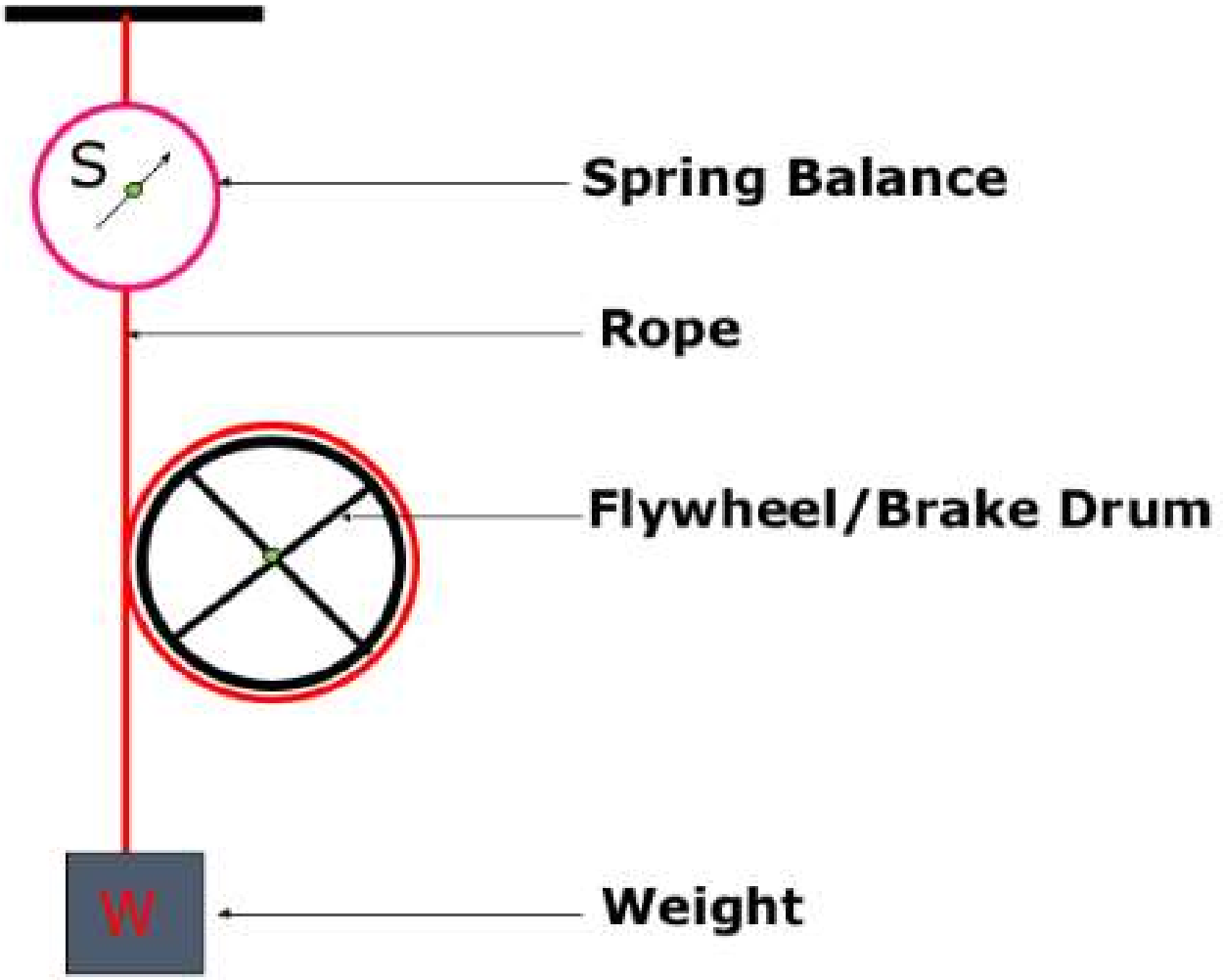
Hence, IP = p_m x A x L x n (Nm/min)/Cycle/Cylinder

$$= \frac{p_m L A n}{60} \quad (Nm/s/cylinder)$$

$$\therefore IP = \frac{p_m L A n}{60,000} \times No\ of\ Cylinders \quad (kW)$$

Measurement of BP

1. Rope Brake Friction Dynamometer



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Rope Brake Friction Dynamometer (Contd.)

Let W=Dead Weight in Newton (N)

S=Spring Balance Reading (N)

R_b=Radius of Brake Drum (D+d)/2 (m)

D=Brake Drum dia and 'd' rope dia

N=Engine RPM

Hence, net Brake Load= (W – S)

Braking Torque = (W-S) x R_b

Hence, Brake Power (BP) $= \frac{(W-S)xR_b x2\pi N}{60{,}000}\ (kW)$

Brake Mean Effective Pressure (bmep)

$$BP = \frac{P_{bmep} LAn}{60{,}000} x No\ of\ Cylinders\ (kW)$$

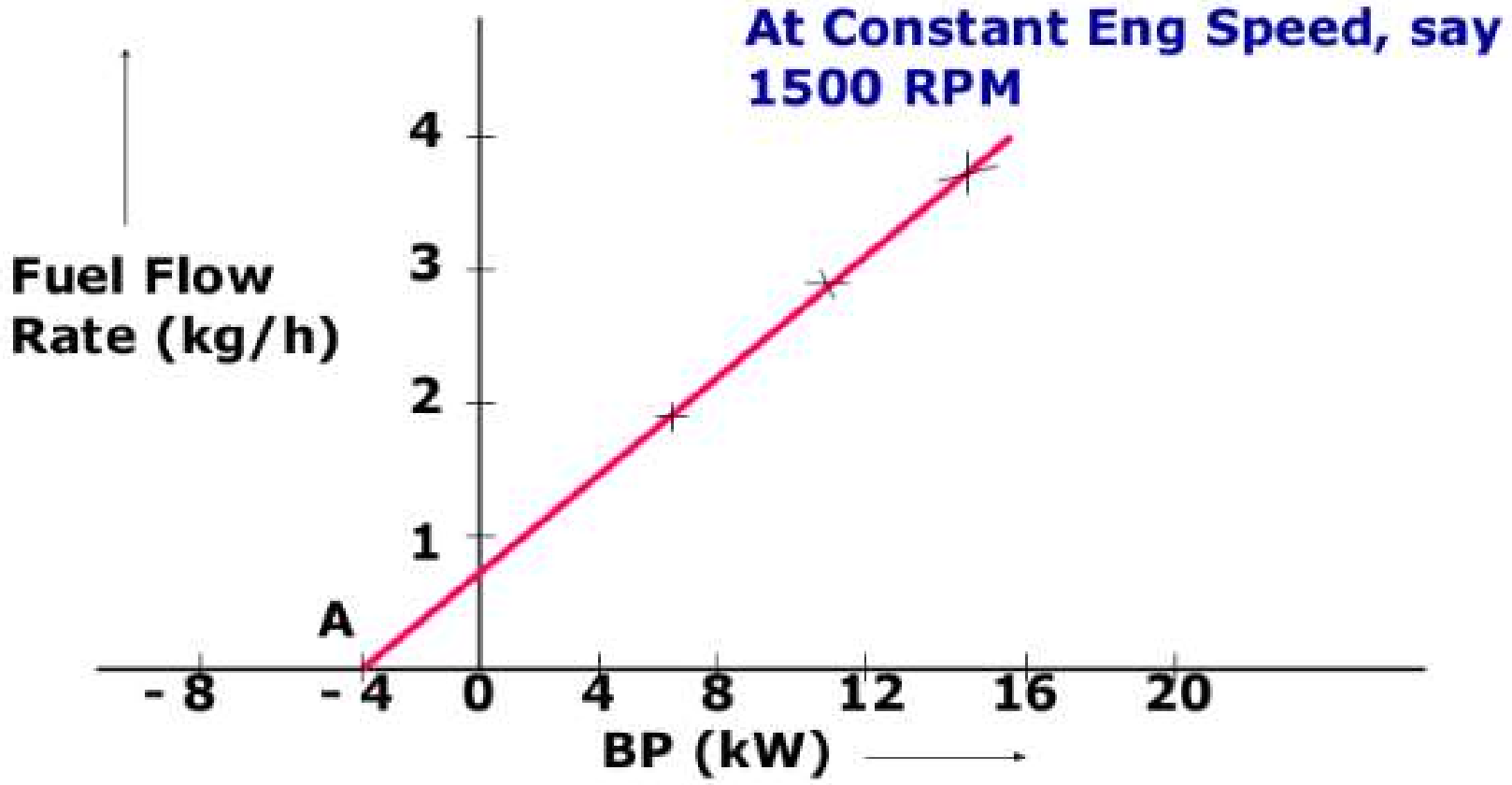
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Methods of Measurement of FP

1. By measurement of IP and BP
2. Motoring Test
3. Morse Test
4. Willan's Line Method

FP by Willan's Line Method

(Fuel Rate Extrapolation Method)



FP by Willan's Line Method

- A graph between fuel consumption rate (kg/h) taken on y-axis and BP (kW) on x-axis is drawn, while engine is made to run at some constant speed, say 1500 RPM
- The graph is extrapolated back to zero fuel consumption, which cuts on –ve x-axis at point 'A'
- The –ve intercept on x-axis represents FP at that speed of the engine
- Although when BP=0, some fuel consumption is there. This fuel is consumed to overcome engine friction
- Only for CI engine to be run at constant speed as Fuel consumption rate v/s BP plot is almost straight line in case of diesel engine, hence can be extrapolated

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FP by Morse Test

- Morse Test can be used for determining FP/IP of multi-cylinder IC engines, generally 3 cyl and more by cutting off each cylinder in turn
- In SI engines, each cylinder is rendered in-operative by short-circuiting the SP or cutting off fuel supply in MPFI systems. In CI engines, fuel supply is cut off
- Consider 4 stroke, 4 cylinder SI engine coupled with dynamometer
- Engine is run at constant speed N throughout one set of test parameters, as FP $\propto N^2$
- It is assumed that pumping & mech losses are same whether a cylinder is working or not
- Throttle position is kept fixed, however, to attain same speed N, load is decreased by dynamometer

FP by Morse Test

- Let B=BP of eng when all cylinders are working

  B_1=BP of eng when Cylinder No 1 is cut off

Similarly, B_2=BP of eng when Cylinder No 2 is cut off

B_3=BP of eng when Cylinder No 3 is cut off

B_4=BP of eng when Cylinder No 4 is cut off

- Let I_1, I_2, I_3 & I_4 be the IP_s developed by Cylinder Nos 1, 2, 3 & 4 respectively and their corresponding FP_s be F_1, F_2, F_3 & F_4

- Total BP(B) = $(I_1+I_2+I_3+I_4) - (F_1+F_2+F_3+F_4)$

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FP by Morse Test

$BP(B) = (I_1+I_2+I_3+I_4) - (F_1+F_2+F_3+F_4)$

Hence, $B_1=(I_2+I_3+I_4) - (F_1+F_2+F_3+F_4)$

On subtracting; $B - B_1 = I_1$

Similarly, $B - B_2 = I_2$

$B - B_3 = I_3$

$B - B_4 = I_4$

On adding; $IP = I_1 + I_2 + I_3 + I_4$

$= 4B - (B_1+B_2+B_3+B_4)$

B, B_1, B_2, B_3 & B_4 can be measured by Dynamometer,
Hence IP can be calculated

Therefore, FP = IP - BP

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Thermal Efficiencies:

i) Indicated Thermal Efficiency $\eta_i = \frac{IP}{m_f \, x CV}$

ii) Brake/Overall Thermal Efficiency $\eta_b = \frac{BP}{m_f \, x CV}$

- Where m_f is fuel consumed in kg/sec
- CV is Calorific Value of fuel in kJ/kg
- IP/BP is in kW

Some Definitions

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Mechanical Efficiency:

$$\eta_{mech} = \frac{BP}{IP} x100$$

Relative Efficiency : Defined as the ratio of Brake Thermal Efficiency to Air Standard Efficiency at same Compression Ratio(CR)

$$\eta_r = \frac{\eta_b}{\eta_a} x100$$

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Some Definitions

Volumetric Efficiency:

- Ratio of actual mass of charge inducted during suction stroke to mass of charge corresponding to swept volume of the engine at atm pr & temp

$$\eta_v = \frac{Mass\ of\ actual\ charge\ inducted}{Mass\ of\ charge\ corresponding\ to\ swept\ vol\ at\ Atm\ \mathrm{Pr}\ \&\ Temp}$$

- Reduced Volumetric Efficiency causes reduction in Power Output

$$Also,\ \eta_v = \frac{Actual\ Vol\ of\ Charge\ inhaled\ at\ suction\ conditions}{Swept\ Volume}$$

- Volumetric Efficiency puts a limit on the amt of fuel that can be burnt and hence on its power, since output of eng depends on amt of air inducted

Some Definitions

(Brake) Specific Fuel Consumption (bsfc/sfc):

- bsfc is defined as the amount of fuel required to be supplied to eng to develop 1kW of power per hour at crankshaft

$$\therefore bsfc = \frac{m_{fx}\,3600}{BP}\ (kg/kWh)$$

Specific Output:

- BP per unit of piston displacement

$$Specific\ Output = \frac{BP}{AxL}$$

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Heat Balance Sheet

- Heat Balance Sheet is an account of heat released on combustion of fuel in the combustion chamber and its utilization in the engine
- To draw heat balance sheet, tests are carried out on engine, while it is run at some constant speed

Heat Supplied:

Heat Supplied = m_f x CV (kJ/min)

Where m_f = mass flow rate of fuel (kg/min)

CV = Calorific Value of fuel (kJ/kg)

Heat Balance Sheet

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Heat Expenditure/Utilization:

1) Heat Equivalent to BP:

Heat Equivalent to BP = BP x 60 (kJ/min)

2) Heat Rejected to Cooling Water:

Heat carried away by water $= m_w C_{pw}(T_{wo} - T_{wi})$ kJ/min

Where m_w = cooling water circulation kg/min
& C_{pw} = 4.187 kJ/kgK

3) Heat carried away by Exhaust gases

Heat carried by exh gases $= m_g C_{pg}(T_{ge} - T_{sa})$ kJ/min

Where $m_g = (m_a + m_f)$ = flue gases flow rate (kg/min)

4) Unaccounted Heat:

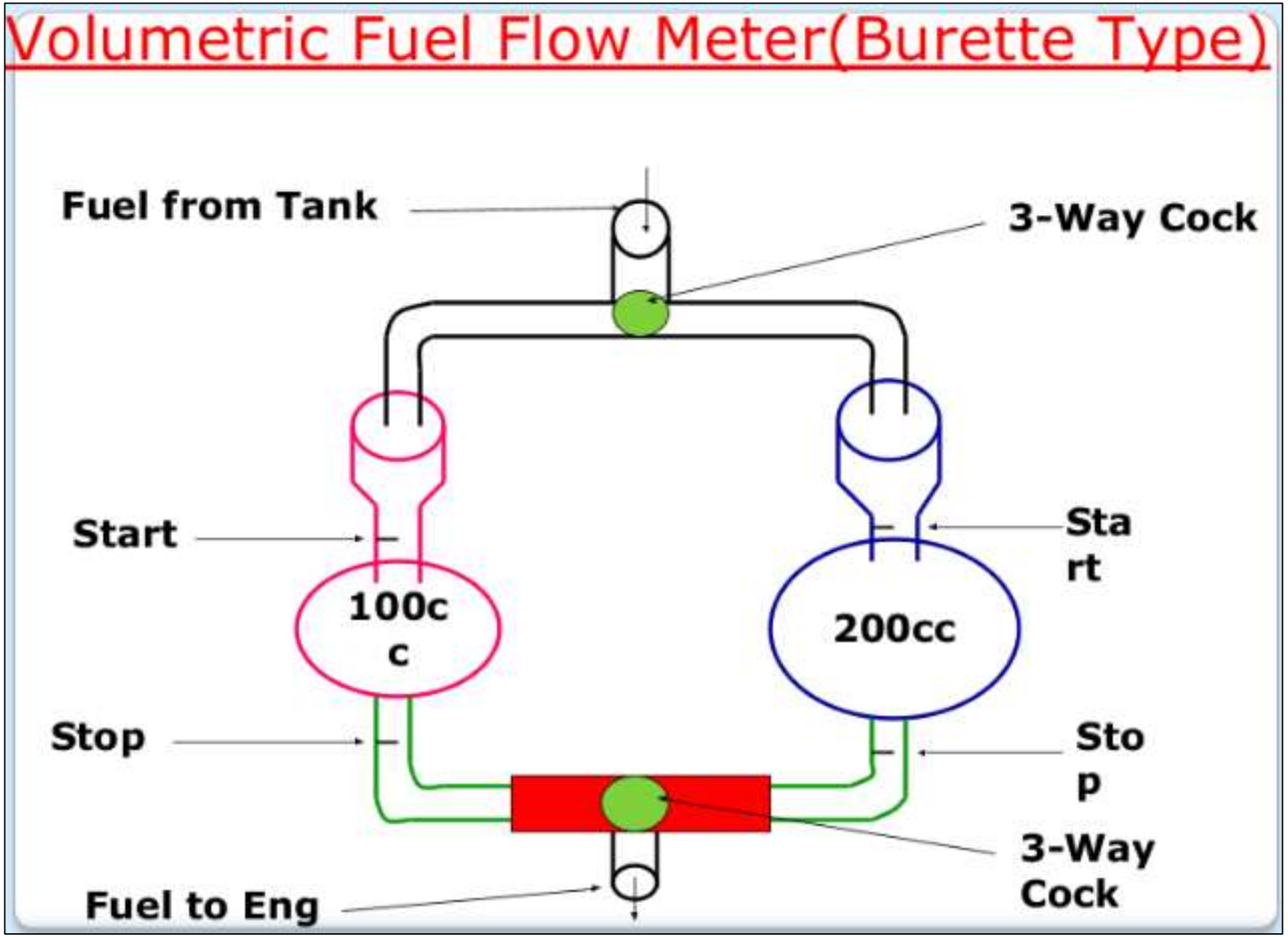
By difference

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Heat Balance Sheet

Heat Supplied	kJ/ min	%	Heat Utilization	kJ/ min	%
Heat Supplied by comb of Fuel $=m_f \times CV$		100	a) Heat to BP=BPx60		
			b) Heat to water $=m_w x C_{pw}(T_{wo} - T_{wi})$		
			c) Heat carried away by exhaust gases $=m_g x C_{pg}(T_{ge} - T_{sa})$		
			d) Heat Unaccounted (By difference)		
Tota		100			100

Volumetric Fuel Flow Meter(Burette Type)



Fuel Measurement

- Time required to supply given volume of fuel is noted
- Mass Flow Rate of Fuel Supply:

$$m_f = \frac{Volume}{Time} x Density\ of\ Fuel$$

- Density of Fuel = Sp Gravity of fuel x Density of water

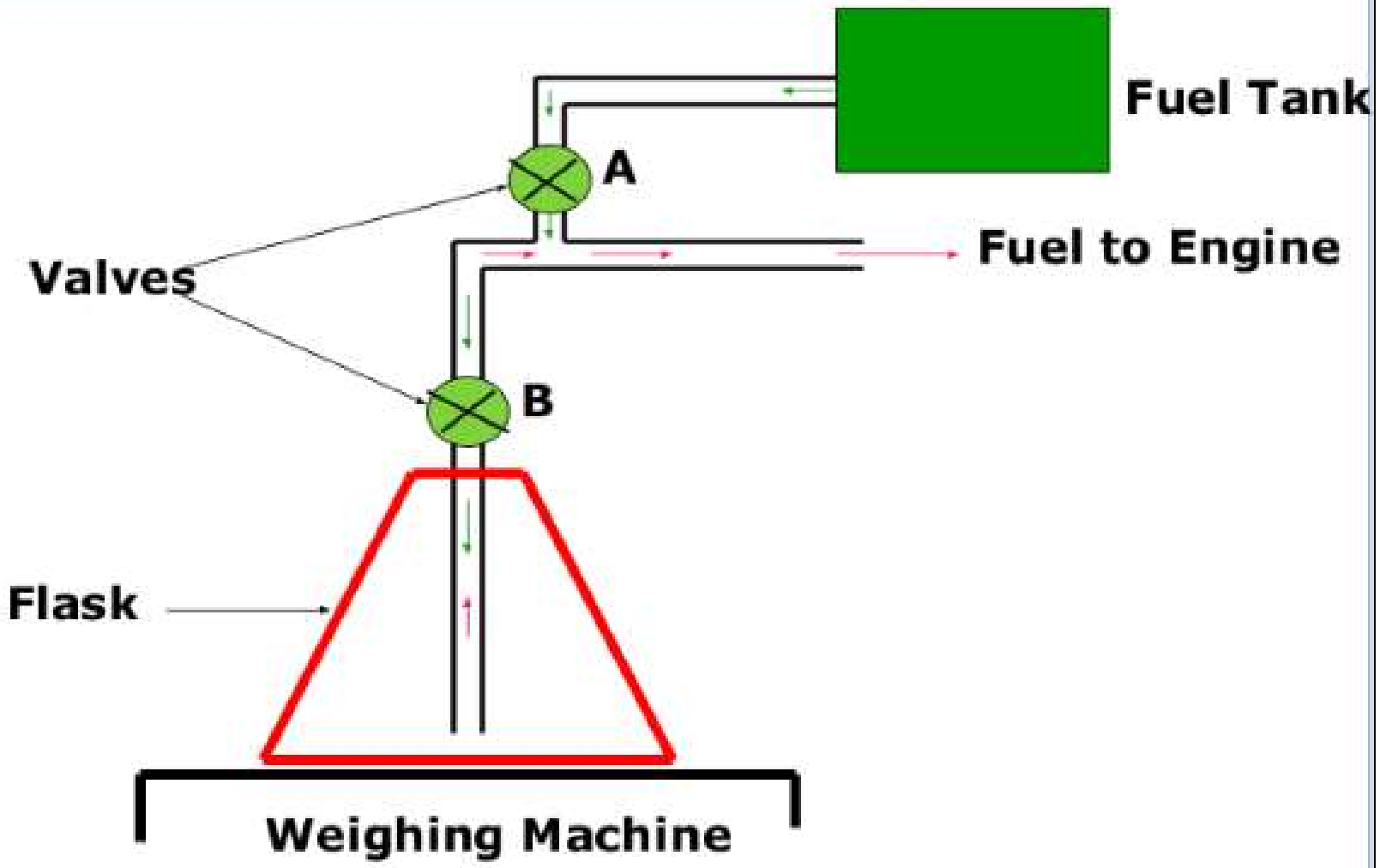
$$Specific\ Gravity = \frac{Density\ of\ material}{Density\ of\ Water}$$

- This method does not give very accurate mass flow rate due to variation in density with temp

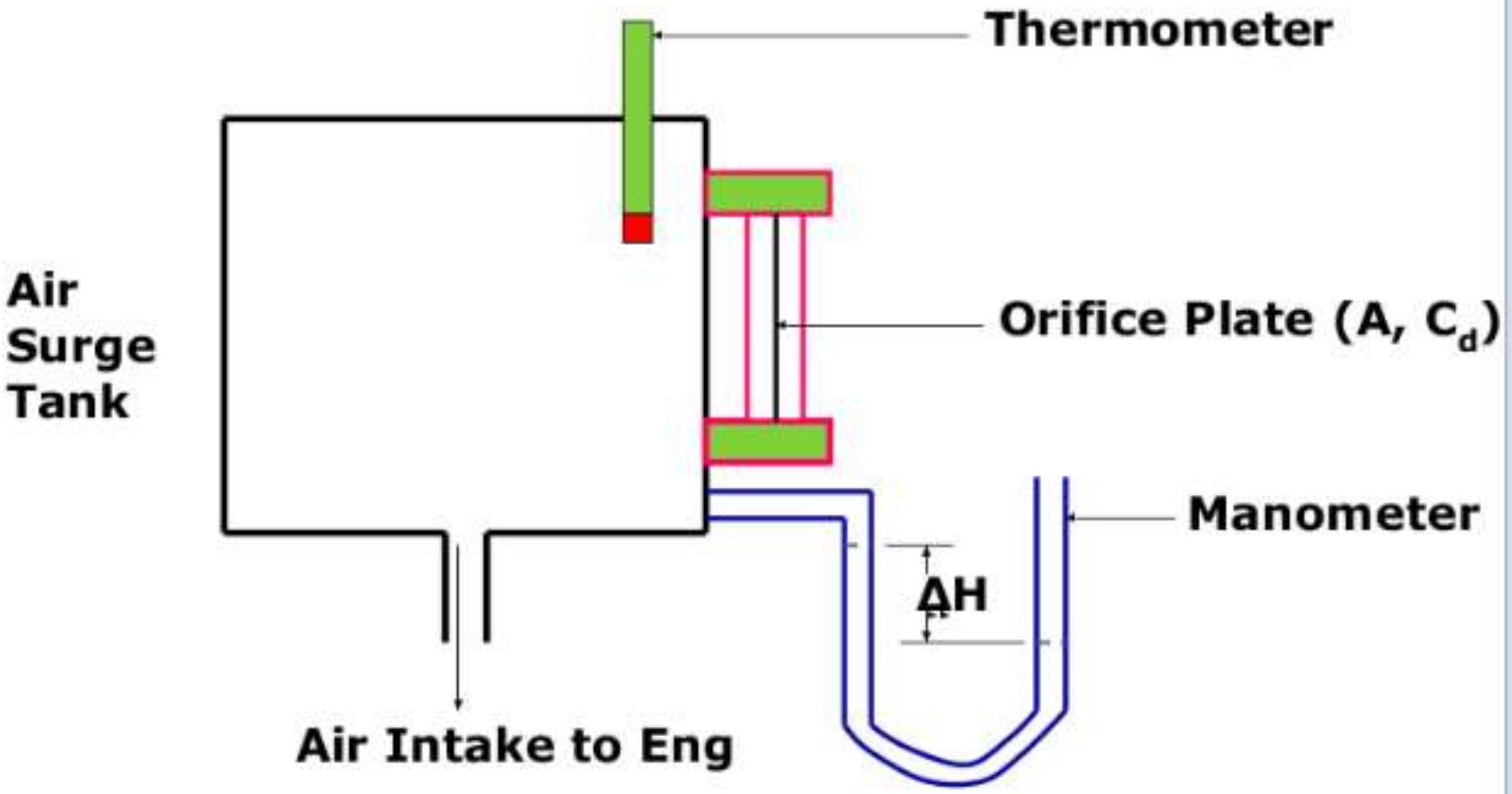
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Gravimetric Fuel Flow Meter



Air Flow Meter



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Measurement of Air Consumption by Air Flowmeter

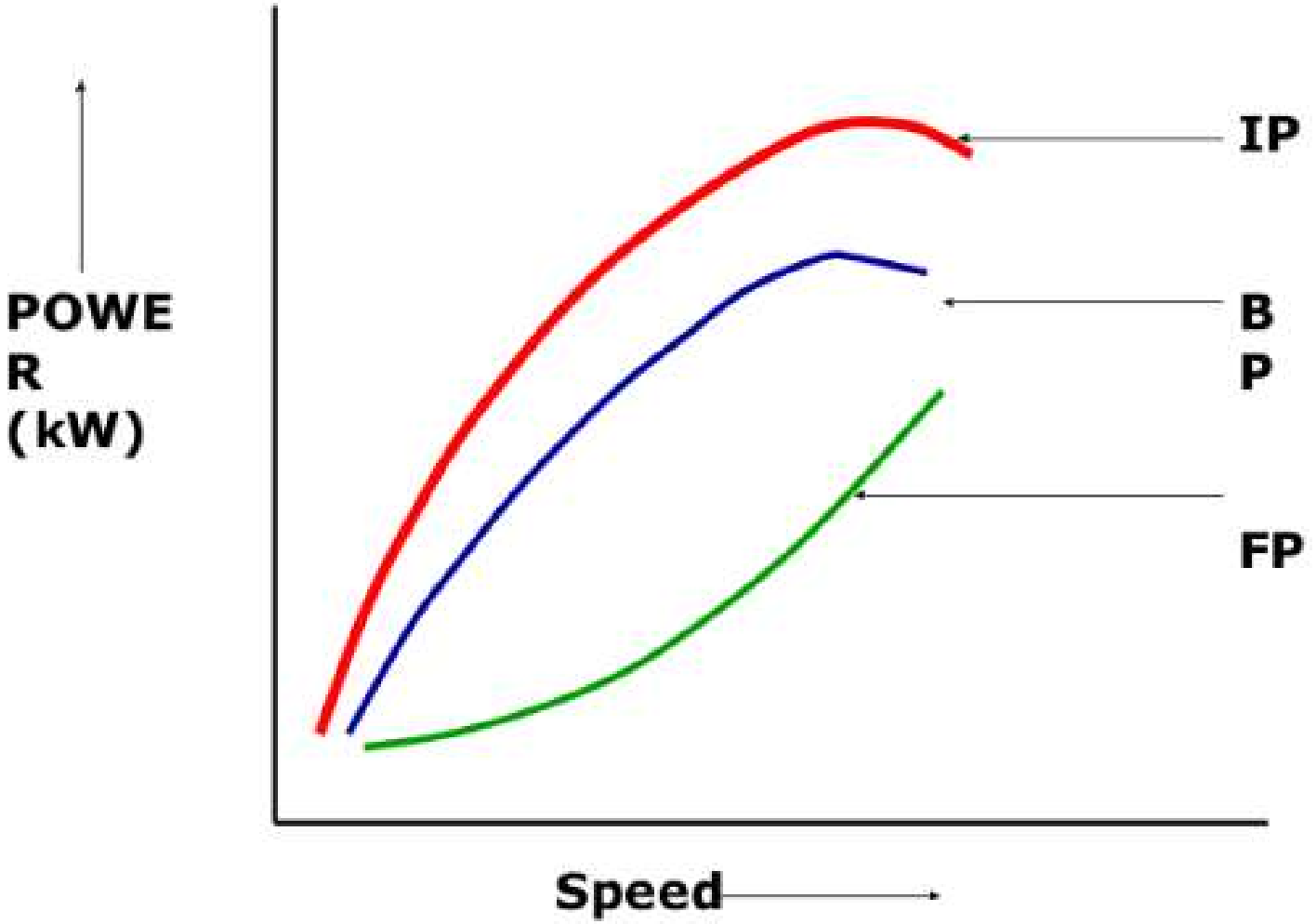
- Surge tank is connected to intake side of the engine
- Manometer measures the pressure difference

- Vol Flow Rate $= C_d \, x \, A \, x \sqrt{2g.\Delta H}_w$

-C_d – Coeff of discharge for given orifice
-A – Orifice cross sectional area
-ΔH_w-Head of water to be converted to air head

SI Engine Characteristic Curves

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SI Engine Characteristic Curves

- Lab tests carried out to determine eng performance
- During tests, throttle is kept full (full /rated load, max fuel consumption) and speed is varied by adjusting the brake load
- IP, BP, FP, bsfc, mechanical & volumetric efficiencies etc are worked out
- Same tests can be repeated at half load

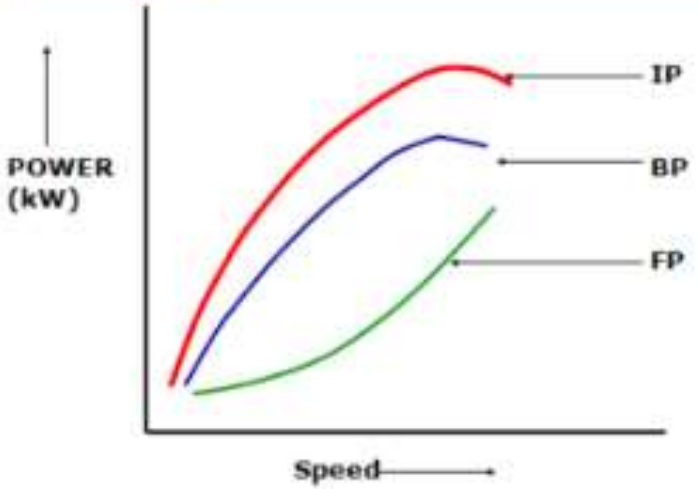
$$hp = \frac{Tx2\pi N}{60{,}000} kW$$

- At rated output, max p-V diagram area, hence max imep; For given torque; power ∞ N

SI Engine Characteristic Curves

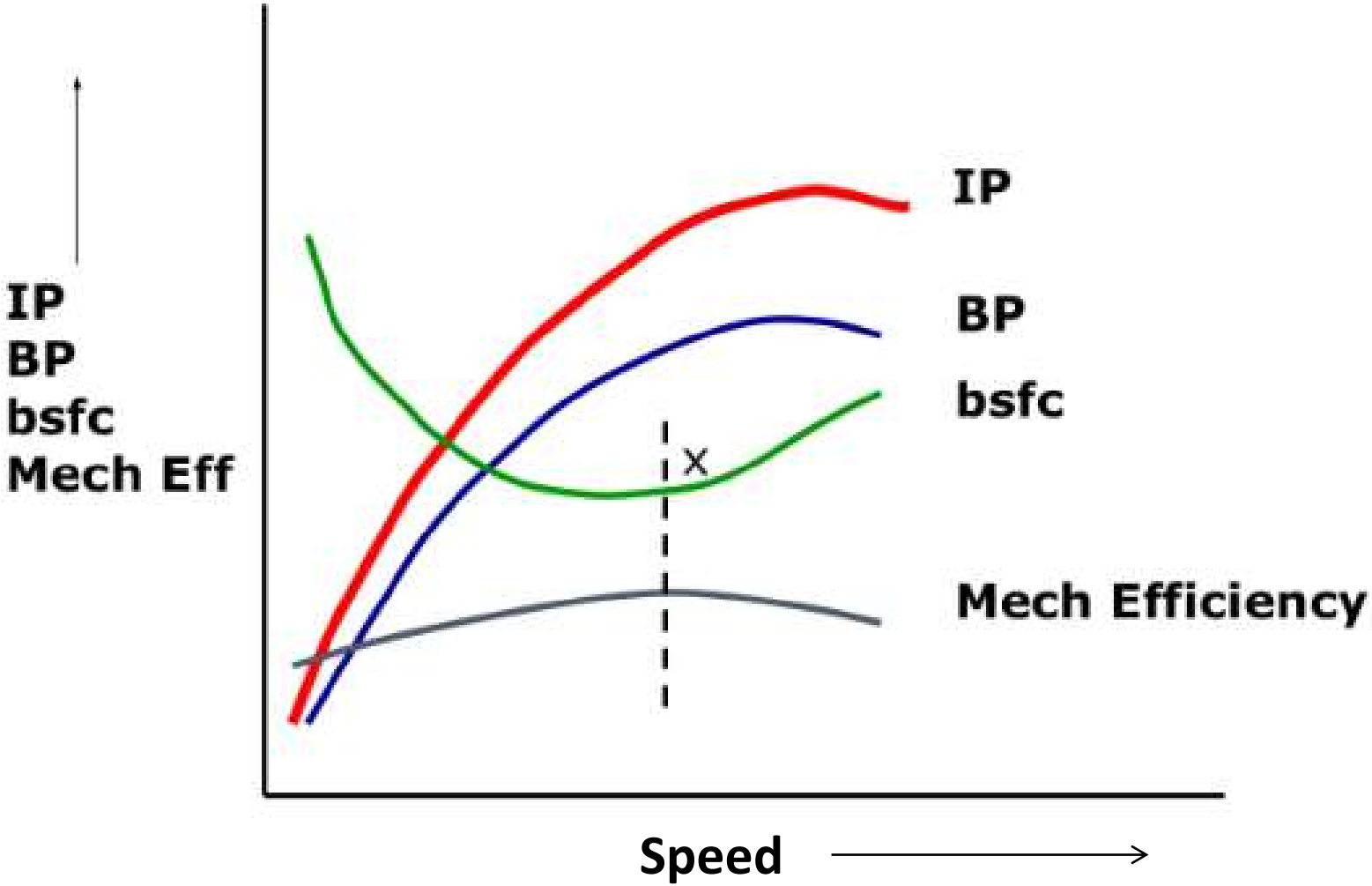
- IP increases when imep or speed or both increase
- IP initially increases faster with speed, if inlet conditions are kept constant
- However, after certain limit, rate of increase of IP reduces with speed due to reduction in vol efficiency as air/charge velocity increase results in inlet pr drop
- Mech losses increase with increase in speed(FP∞ N^2) due to which increase in IP is off-set by steep increase in FP

SI Engine Performance Curves



SI Engine Characteristic Curves

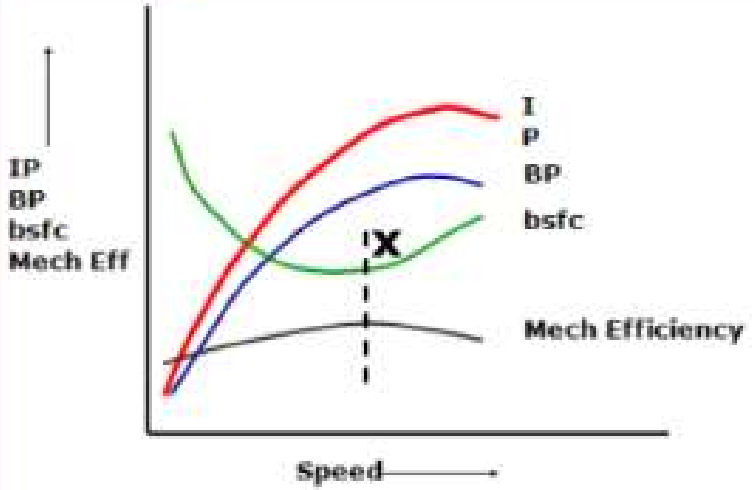
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SI Engine Characteristic Curves

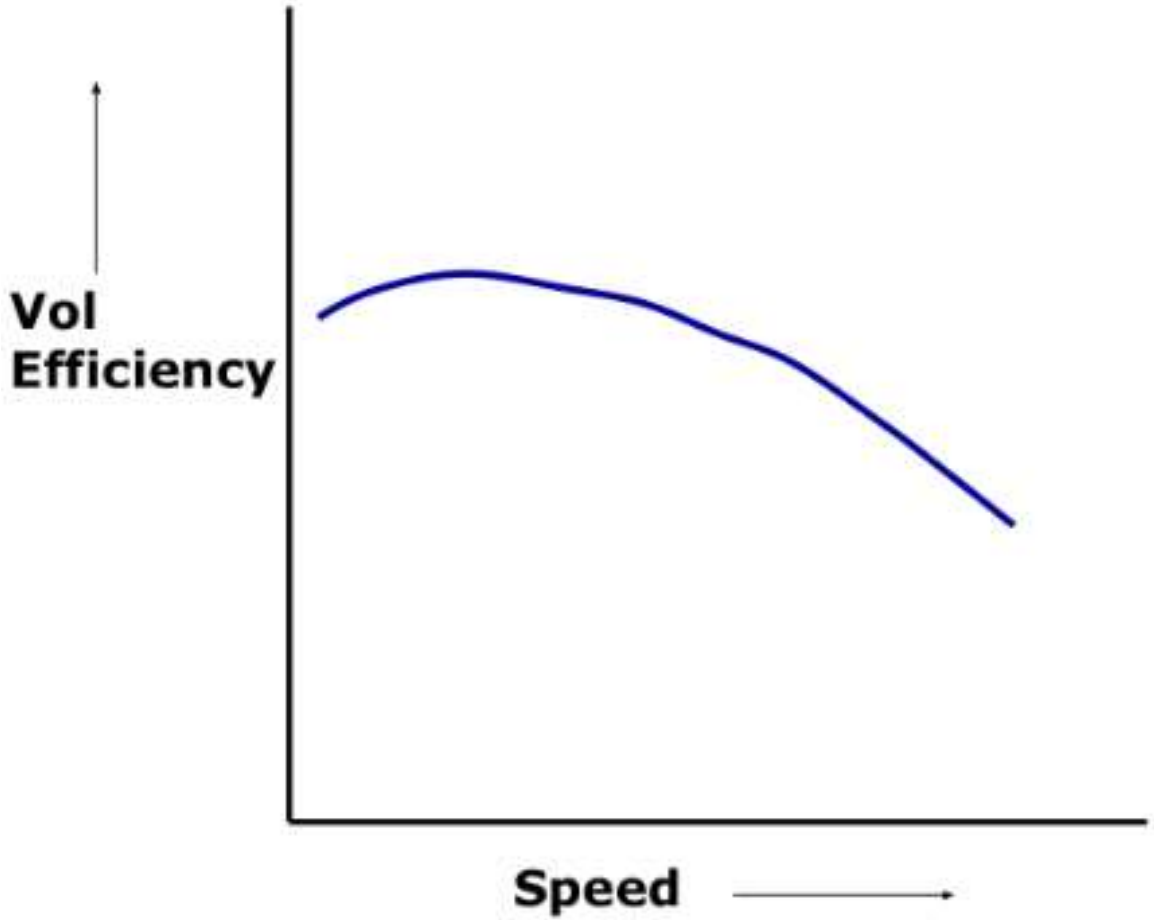
- As FP ∞ N^2, mech efficiency reduces due to steep increase in FP
- At lower speeds, due to lower charge velocity because of low piston speed, bsfc reduces since volumetric efficiency increases and mech efficiency also increases
- After certain speed, bsfc increases due to reduction in volumetric efficiency and increase in mech losses
- Point x represents economical speed of eng for min fuel consumption

SI Engine Performance Curves



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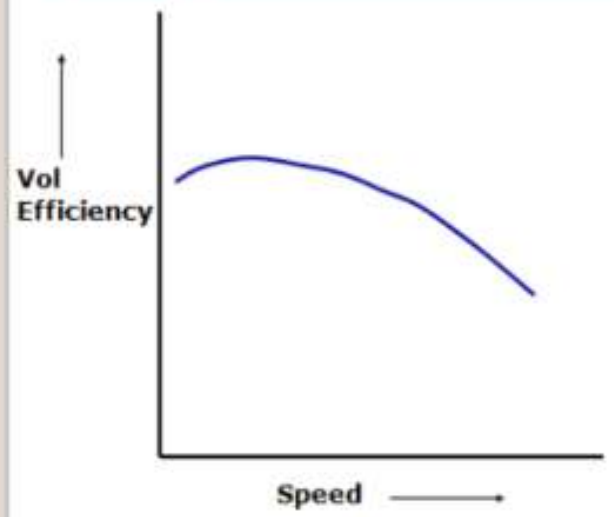
SI Engine Characteristic Curve



SI Engine Characteristic Curve

- Volumetric Efficiency reduces with increase in speed due to increase in intake velocity resulting in drop of suction pressure
- Higher the speed, lesser the time available for induction of charge
- Suction valve fully opens only when pressure inside cylinder slightly below the surrounding pressure, thus reducing effective suction stroke

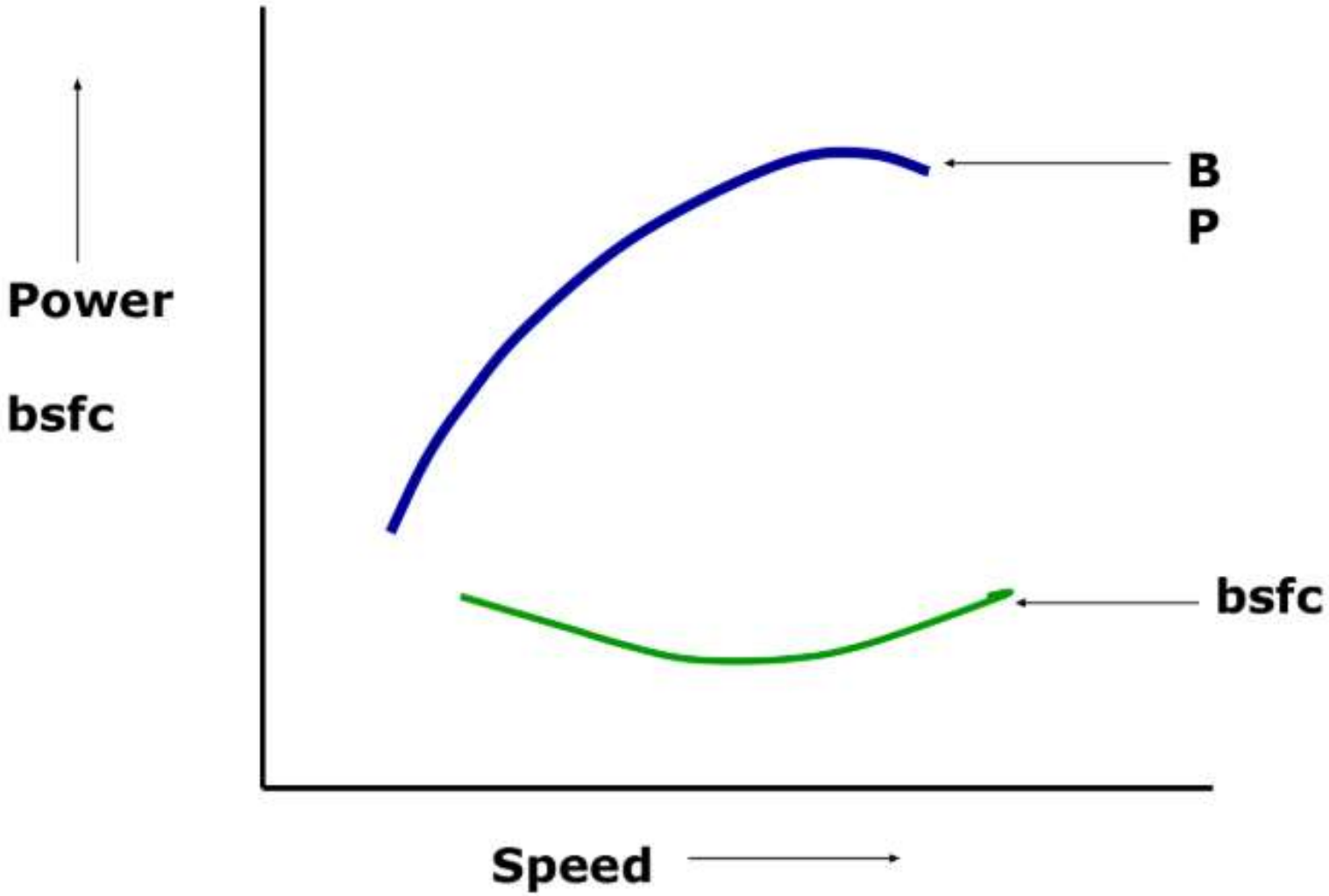
SI Engine Performance Curve



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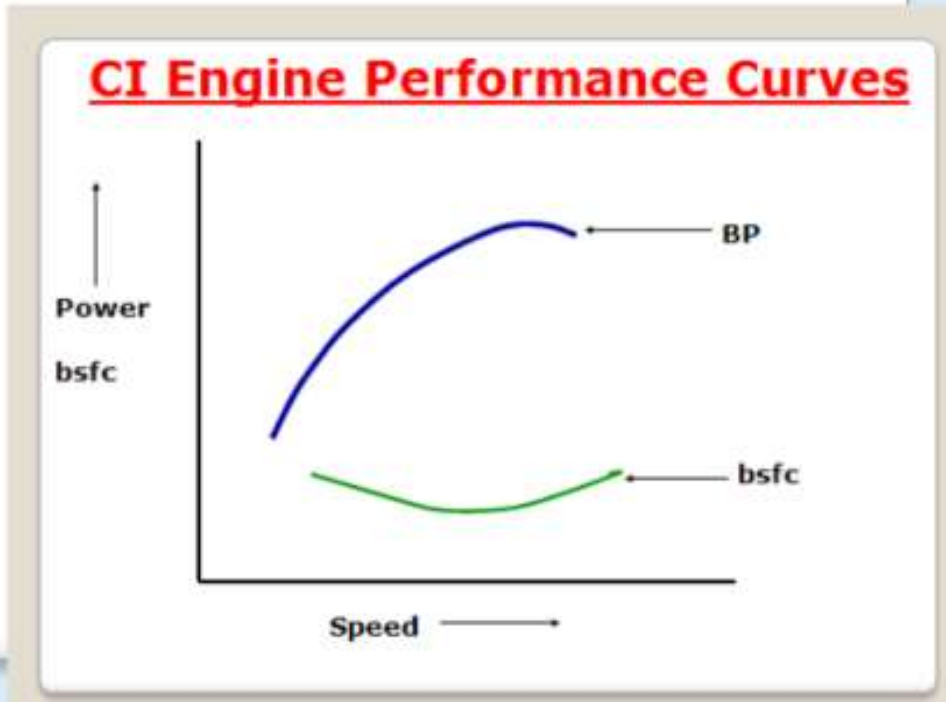
CI Engine Characteristic Curves

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CI Engine Characteristic Curves

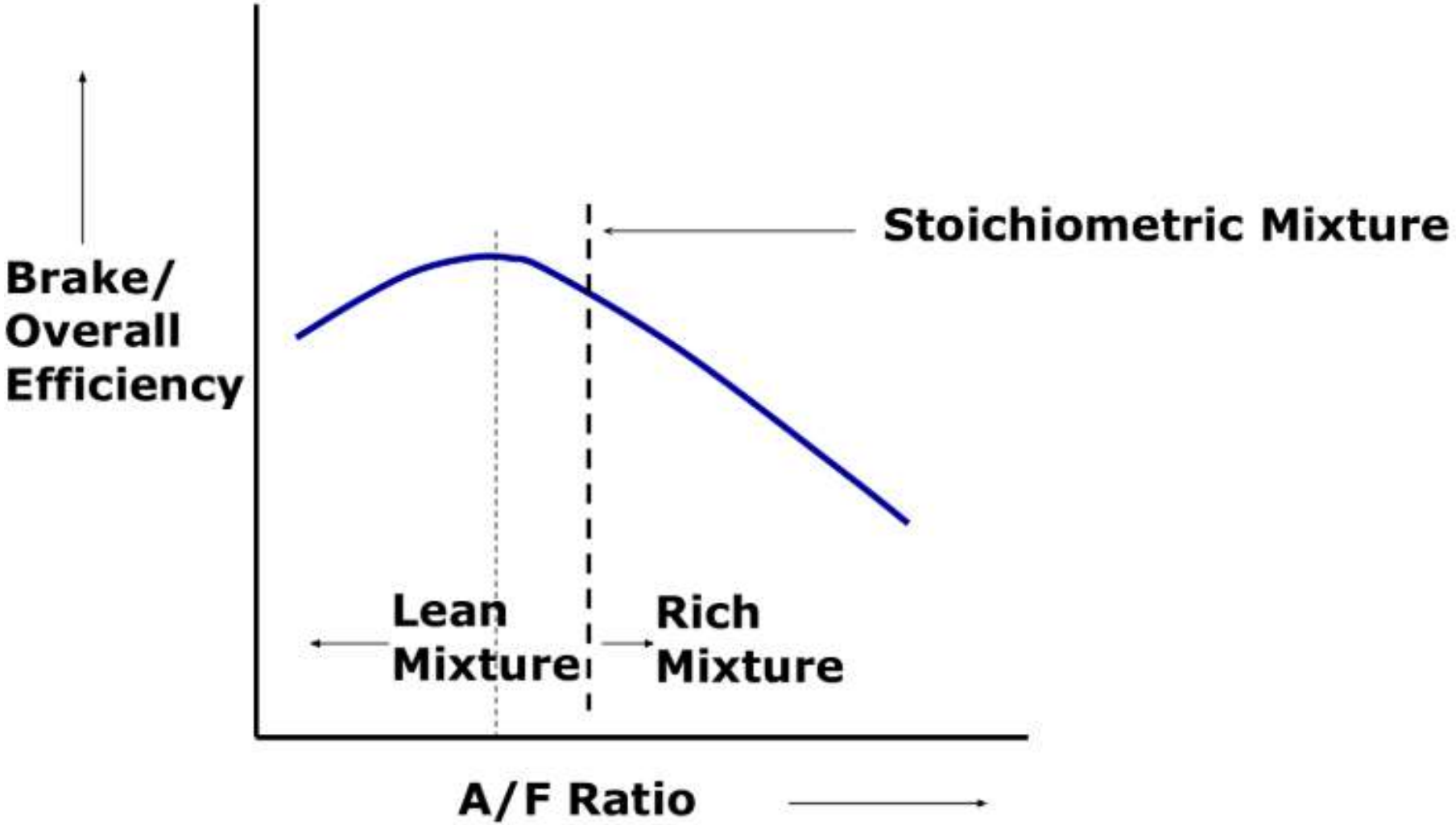
- IP and BP increase with speed but due to steep increase in FP, IP and BP start coming down
- For bsfc curve, same reasons as in SI engine



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CI Engine Characteristic Curve

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Numerical on I C Engine testing

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A 6 cylinder gasoline engine operates on 4 stroke cycle. Bore of cylinder is 80mm and stroke 100mm.Clearance volume per cylinder is 70CC. At 4000 RPM, Fuel consumption is 20kg/hr and the torque developed is 150Nm.

Calculate- (a) BP (b) Brake mean effective pressure (c) Brake thermal efficiency

If CV of the fuel is 43000kJ/kg, find relative efficiency on brake power basis, assuming engine works on Constant volume cycle and gamma for air =1.4.

$$BP = \frac{Tx2\pi N}{60{,}000}kW \quad = \frac{150x2\pi x4000}{60000} = 62.83kW$$

$$bmep = ?$$

$$BP = \frac{bmep\ x\ AxLxn}{60000}\ x\ no\ of\ cylinders$$

$$62.83 = \frac{bmep\ x\ \frac{\pi}{4}(0.08)^2\ x\ 0.1\ x\ \frac{4000}{2}}{60000}\ x6$$

$$= 6.25x10^5\ N/m^2 = 6.25\ bar$$

Numerical on I C Engine testing

$$\eta_b = \frac{BP}{m_f x CV} = \frac{62.83}{\frac{20}{3600} x 43000} = 0.263 = 26.3\%$$

$$\eta_r = \frac{\eta_b}{\eta_a}; \qquad \eta_a = 1 - \frac{1}{r^{\gamma - 1}}$$

$$r = \frac{V_s + V_c}{V_c}; \quad V_s = \frac{\pi}{4} 0.8^2 x 10 = 502.6CC$$

$$r = \frac{502.6 + 70}{70} = 8.18; \; \eta_a = 1 - \frac{1}{8.18^{1.4-1}} = 0.5686$$

$$\eta_r = \frac{0.263}{0.5686} = 0.4625 = 46.25\%$$

Numerical on I C Engine testing

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During trial of a single cylinder, 4 stroke oil engine the following results were obtained: Cyl bore=200mm, Stroke=400mm, mep=6 bar, Torque=407Nm, speed=250 RPM, Oil consumption=4kg/hr, CV of fuel=43MJ/kg, Cooling water rate=4.5kg/min, Air used per kg of fuel=30kg, Rise in cooling water temp=45°C, Temp of Exhaust gases=420°C, Room temp=20°C, mean sp. heat of exhaust gases=1kJ/kgK, Sp. Heat of water= 4.18kJ/kgK, Barometric pressure=1.01325 bar Find IP, BP and draw up heat balance sheet in kJ/hr.

$$IP = \frac{imep\ x\ AxLxn}{60{,}000}\ x\ no\ of\ cyl\ (kW)$$

$$IP = \frac{6x10^5\ x\ \frac{\pi}{4}(0.2)^2\ x0.4x\ \frac{250}{2}}{60{,}000}\ x1 = 15.7\ kW$$

$$BP = \frac{T\ x2\pi N}{60000} = \frac{407\ x2\pi\ x250}{60000} = 10.65kW$$

Heat Balance Sheet

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1. Heat supplied by fuel to eng $= m_f \times CV$

$= 4 \times 43000 kJ/hr = 172,000 kJ/h$

2. Heat utilized

(i) Heat to Power Output = BP x 3600kJ/h

$= 10.65 \times 3600 = 38,358 kJ/h$

(ii) Heat to cooling water $= m_w \times C_{pw} \times \Delta T_w$

$= 4.5 \times 60 \times 4.18 \times 45 = 50,787$ kJ/h

(iii) Heat to exhaust gases $= m_g \times C_{pg} \times (T_e - T_a)$

To find m_g : m_a = 30kg/kg of fuel; hence m_g = 31kg
Since fuel consumption is 4kg/h; mg = 31x4kg/h

Hence, Heat to exhaust gases = 31x4x1(420-20)

$= 49,600 kJ/h$

(iv) Unaccounted Heat = 33,255kJ/h (by difference)

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Heat Balance Sheet

Heat Supplied	kJ/hr	%	Heat Utilized	kJ/hr	%
Heat supplied by fuel= m_fxCV	172,000	100	To BP= BPx3600	38,358	22.3
			To Cooling Water $=m_w.C_{pw}.\Delta T_w$	50,787	29.5
			To exhaust gases $=m_g.C_{pg}.\Delta T_g$	49,600	28.8
			Unaccounted Heat	33,255	19.3
Total	**172,000**	**100**	**Total**	**172,000**	**100**

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Numerical on I C Engine testing

During A 4 cylinder 4 stroke SI engine has a bore of 5.7cm and stroke 9cm. Its rated speed is 2800 RPM and it is tested at this speed against a brake which has a torque arm of 356mm. The net brake load is 155N and fuel consumption is 6.74 lit/hr. Sp gravity of petrol is 0.735 and CV is 44200kJ/kg. A Morse test is carried out and cylinders are cut off in order of 1, 2, 3 & 4 with corresponding brake loads of 111, 106.5, 104.2 and 111N. Determine engine torque, bmep,brake thermal efficiency, sfc, mech efficiency and imep.

Engine torque= $(W-S).R_b = WxL$

$= 155x0.356 = 55.18Nm$

Numerical on I C Engine testing

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$$BP = \frac{bmep\,x\,AxLxn}{60{,}000} x\,no\,of\,cyl = \frac{Tx2\pi N}{60000}(kW)$$

$$BP = \frac{55.18x2\pi.2800}{60000} = 16.17kW$$

$$\frac{bmep\,x\frac{\pi}{4}(0.057)^2 x0.09x\frac{2800}{2}}{60{,}000} x\,4 = 16.17kW$$

$$\therefore bmep = 7.55\,bar$$

$$\eta_b = \frac{BP}{m_f xCV}$$

$$m_f = \frac{6.74(lit/h)x10^{-3}(m^3)x0.735x1000}{3600}$$

$$= 1.376x10^{-3}\,kg/s$$

$$\eta_b = \frac{BP}{m_f xCV} = \frac{16.17}{1.376x10^{-3}\,x44200}$$

$$= 0.2658\,or\,26.58\%$$

$$bsfc = \frac{m_f}{BP} = \frac{1.376x10^{-3}\,x3600}{16.17}$$

$$= 0.306kg/kWh$$

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We know that IP = 4BP - (BP_1+BP_2+BP_3+BP_4)

$$BP = \frac{WxR_b x2\pi N}{60000}$$

$$IP = [4W - (W_1 + W_2 + W_3 + W_4)]\frac{R_b x2\pi N}{60000}$$

$$IP = [4x155 - (111 + 106.5 + 104.2 + 111)]\frac{0.356x2\pi\ x2800}{60000}$$

$$= 19.54kW$$

$$\eta_m = \frac{BP}{IP} = \frac{16.17}{19.54}x100 = 82.75\%$$

$$\eta_m = \frac{bmep.LAn}{imep.LAn} = \frac{7.55}{imep} = 0.8275 \quad \therefore imep = 9.12\,bar$$

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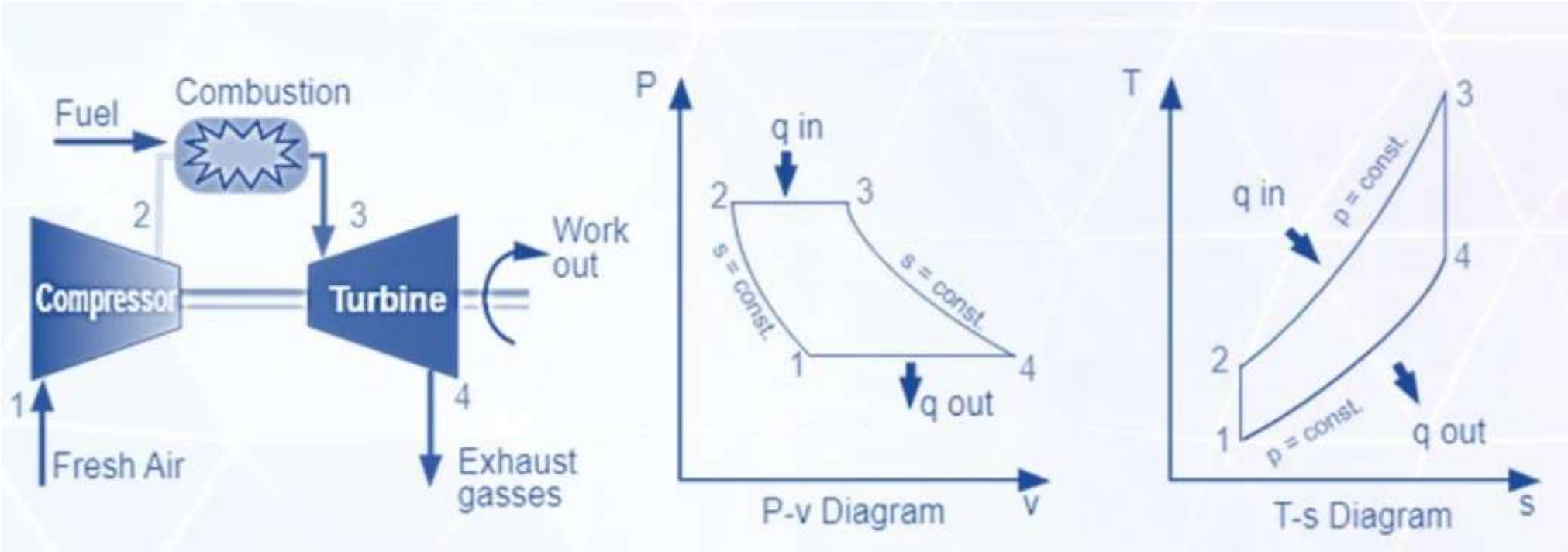
Unit 4

Gas Turbines

A gas turbine converts chemical energy of fuel (natural gas, diesel, or aviation fuel) into mechanical energy or electricity by burning fuel with compressed air.

It operates on the Brayton or Joule Cycle consisting of compressor, combustor and turbine stages commonly used for power generation.

Brayton or Joule cycle for Gas Turbines



Process 1-2: Isentropic compression process
Process 2-3: Constant Pressure heating
Process 3-4: Isentropic Expansion process
Process 4-1: Constant Pressure cooling

Brayton or Joule cycle for Gas Turbines

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Work done by Turbine per Kg of air

$$W_T = C_P(T_3 - T_4)$$

Work required by compressor per Kg of air

$$W_C = C_P(T_2 - T_1)$$

Net Work done per Kg of air

$$W_{Net} = W_T - W_T$$
$$= [C_P(T_3 - T_4) - C_P(T_2 - T_1)]$$

Classification of Gas Turbines

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According to the path of working substance-

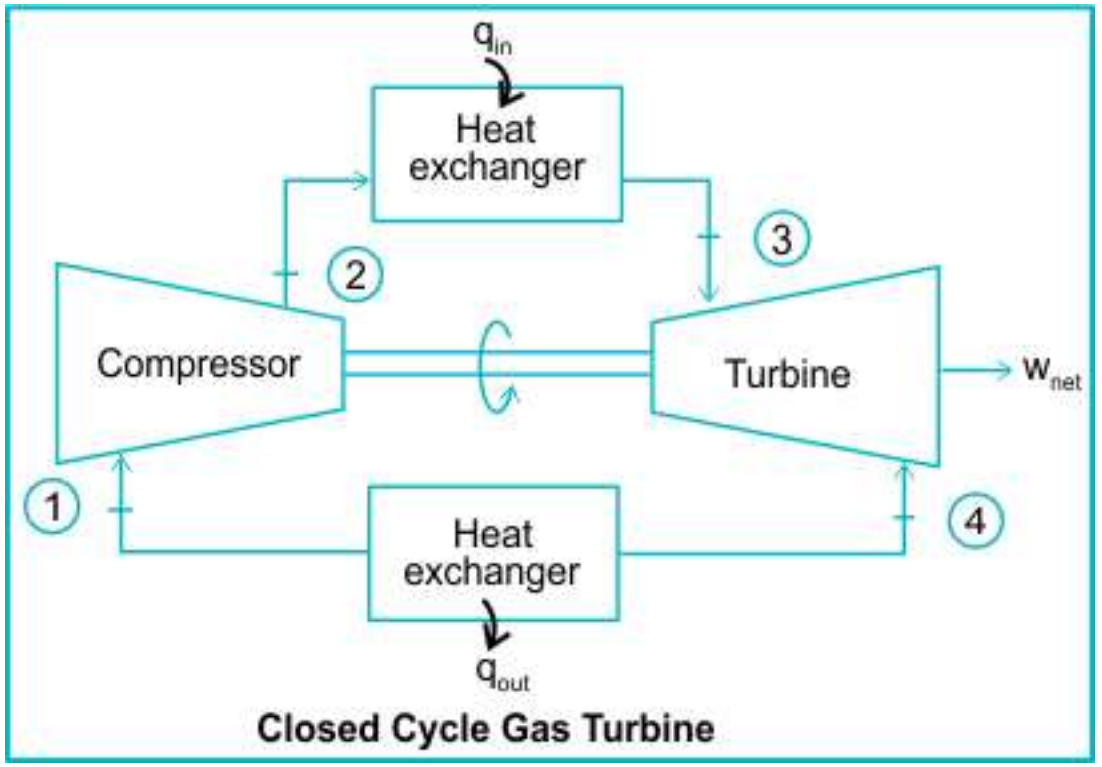
1) Closed cycle Gas turbines
2) Open cycle Gas turbines
3) Semi-Closed cycle Gas turbines

According to the process of heat absorption-

1) Constant Pressure Gas turbines
2) Constant Volume Gas turbines

Closed cycle Gas Turbine

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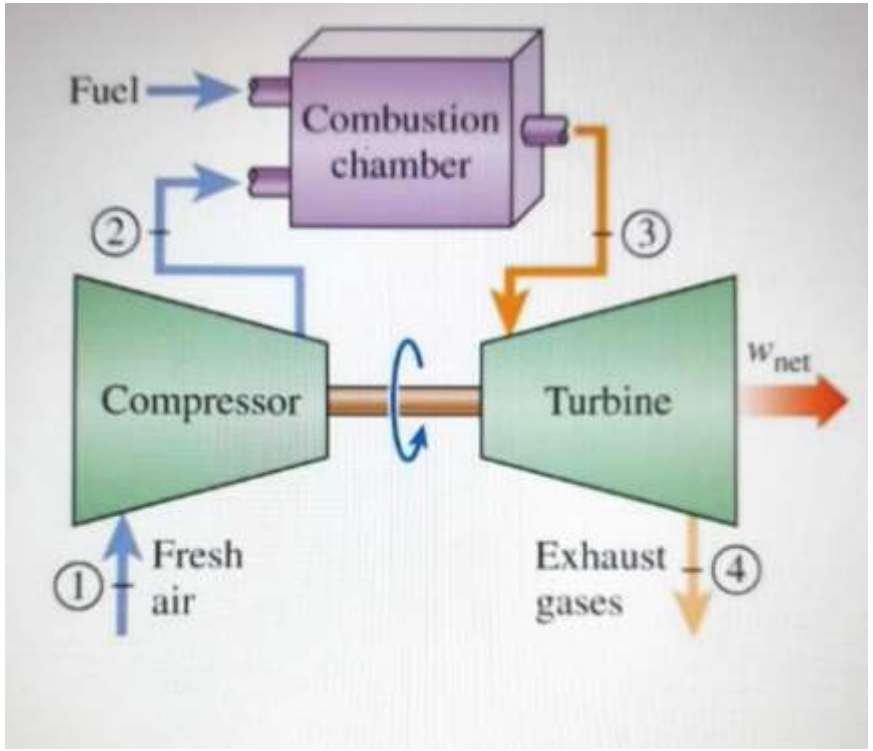
Closed Cycle Gas Turbine

A closed-cycle gas turbine operates by continuously recirculating a pressurized working fluid (such as air, helium, etc.) through a compressor, heat exchanger (heater), turbine and cooling chamber.

It uses an external heat source to heat the gas indirectly, keeping the working fluid clean, reducing corrosion, and allowing for high-efficiency.

Open cycle Gas Turbine

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It consists of Compressor, Combustion chamber and Gas turbine.

The compressed air is heated by the combustion of fuel and the products of combustion gets mixed up with compressed air, thus increasing the mass of compressed air.

The hot gas is then made to flow over the turbine blades. The expansion of gas will takes place & finally gas is exhausted to atmosphere.

Unlike closed-cycle turbines, which recirculate the working fluid, open-cycle systems exhaust the gas after a single pass.

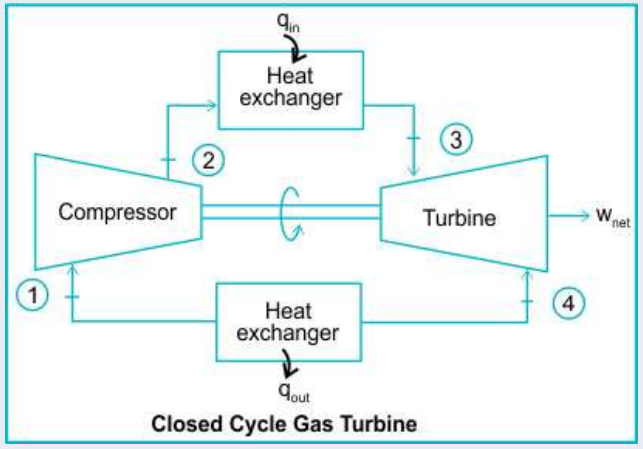
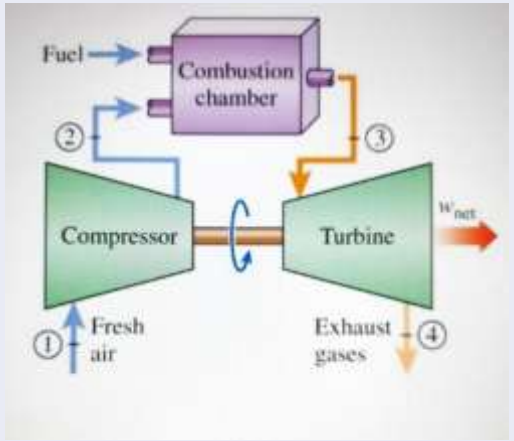
Difference between Closed and Open cycle Gas Turbine

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S. No.	Closed cycle Gas turbine	Open cycle Gas Turbine
1	The compressed air is heated in a heating chamber, since the gas is heated by an external source, the amount of gas remains the same.	The compressed air is heated in a combustion chamber. The products of combustion get mixed up in the heated air.
2	The gas from the turbine is passed into the cooling chamber.	The gas from the turbine is exhausted into the atmosphere.
3	The working fluid is circulated continuously.	The working fluid is replaced in each cycle.
4	Any fluid with better thermodynamic properties can be used.	Only air can be used as working fluid.
5	The turbine blades do not wear away earlier as the enclosed gas does not get contaminated while flowing through the heating chamber.	Turbine blades wear away earlier, as the air from atmosphere gets contaminated while flowing through the combustion chamber.

Difference between Closed and Open cycle Gas Turbine

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S. No.	Closed cycle Gas turbine	Open cycle Gas Turbine
6	Since the air from the turbine is cooled by water, it is best suited for stationary installations or marine uses.	Since the air is discharged into the atmosphere, it is best suited for moving vehicles
7	Mass of installation per kW is more.	Mass of installation per kW is less.
8	Its maintenance cost is high.	Its maintenance cost is low.
9	 Closed Cycle Gas Turbine	

Uses of Gas Turbines

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Important uses of Gas Turbines are:

- **Generation of electric power:** They are extensively used in the generation of electric power.
- **Turbojet & Turbo propeller engines:** Gas turbines are used to drive compressors in turbojets. They are also used to drive air compressors and turbines in turbo-propeller engines.
- **Superchargers:** They are used to drive superchargers fitted in aviation gasoline engines as well as for heavy-duty diesel engines.
- **Marine Engines:** G.T. are used in marine engines. They do not require water storage tanks or distillation plants.
- **Railway Engines:** They have also entered the field of railway engines.

Thermal Efficiency of Gas turbine

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The following assumptions are made for the analysis of GT:

1. Compression & expansion processes are considered isentropic.
2. The pressure losses in the system are neglected.
3. The specific heat of working fluid remains constant throughout the cycle
4. The heat losses are neglected.
5. The KE (Kinetic Energy) of the fluid entering the compressor& leaving the turbine is same.

Thermal Efficiency

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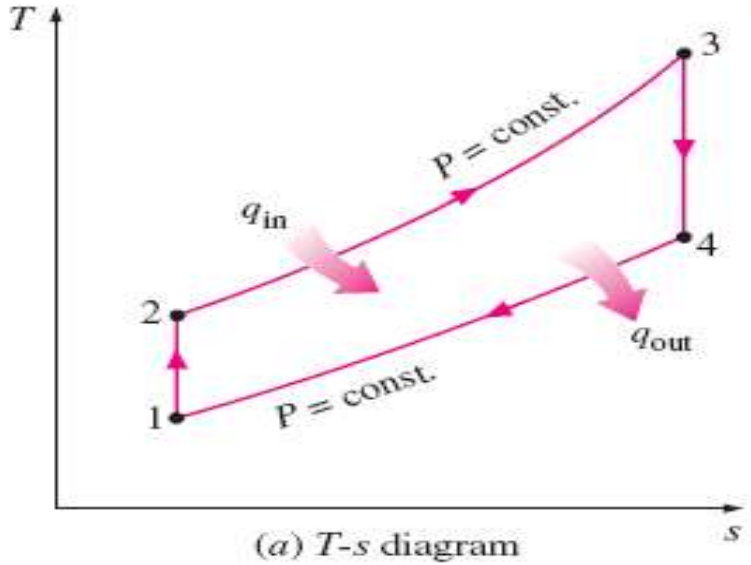
The **energy balance** for a steady-flow process can be expressed, on a unit–mass basis, as

$$(q_{\text{in}} - q_{\text{out}}) + (w_{\text{in}} - w_{\text{out}}) = h_{\text{exit}} - h_{\text{inlet}}$$

The **heat transfers** to and from the working fluid are:

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2)$$

$$q_{\text{out}} = h_4 - h_1 = c_p(T_4 - T_1)$$



(a) T-s diagram

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k} = \frac{T_3}{T_4}$$

The **thermal efficiency** of the ideal Brayton cycle,

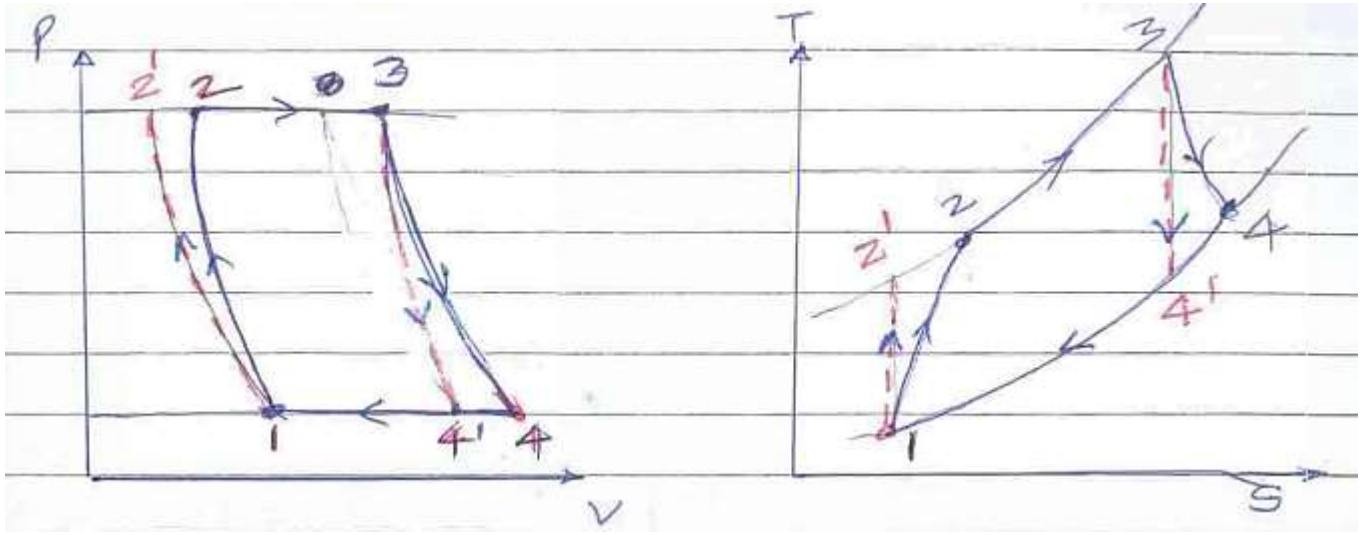
$$\eta_{\text{th,Brayton}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

$$\eta_{\text{th,Brayton}} = 1 - \frac{1}{r_p^{(k-1)/k}}$$ Constant specific heats

where $r_p = \frac{P_2}{P_1}$ is the **pressure ratio**.

Turbine and compressor internal Efficiencies of Gas turbine

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$$\eta_{Compr} = \frac{T_2' - T_1}{T_2 - T_1}$$

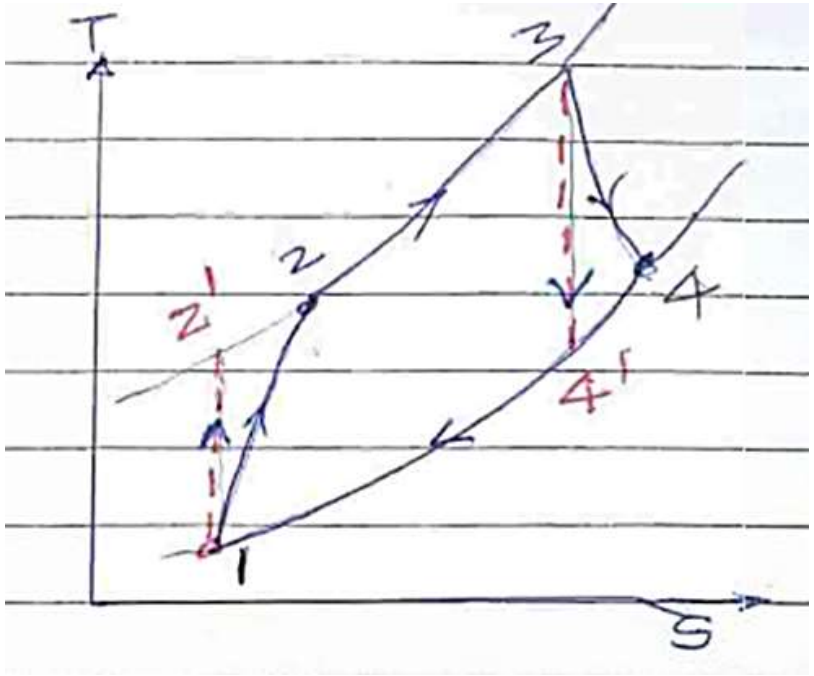
$$\eta_{turbine} = \frac{T_3 - T_4}{T_3 - T_4'}$$

Numerical on Gas turbine

A gas turbine plant with a pressure ratio of 1: 5 takes in air at 15 degree Celsius. The maximum temperature is 600 degree Celsius and develops 2200 Kw. The turbine and compressor efficiencies are equal to 0.85. Take Cp = 1 KJ/Kg-K and Cv = 0.714 KJ/Kg-K. Determine-

a) Actual overall efficiency of the turbine,

b) Mass of air circulated by the turbine



Solution:

Numerical on Gas turbine

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Given :- $\frac{P_2}{P_1} = \frac{P_3}{P_4} = \frac{5}{1}$, $T_1 = 15^\circ C = 288 K$

$T_3 = 600^\circ C = 873 K$

$P = 2200 KW$

$\eta_c = \eta_t = 0.85$ $C_p = 1$, $C_v = 0.714 KJ/kgK$

$\gamma = \frac{C_p}{C_v} = \frac{1}{0.714} = 1.4$

$\left(\frac{T_2'}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$ $T_2' = 456 K$

Also $\frac{T_3}{T_4'} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4' = 551 K$

$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} \Rightarrow \frac{456 - 288}{T_2 - 288} = 0.85$

$\therefore \boxed{T_2 = 486 K}$

$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'} \Rightarrow \frac{873 - T_4}{873 - 551} = 0.85$

$\boxed{T_4 = 599 K}$

$\therefore W_T = C_p(T_3 - T_4) = 1(873 - 599)$

$= 274 KJ/kg$ of air

$W_c = C_p(T_2 - T_1)$

$= 1(486 - 288) = 198 KJ/kg$ of air

$W_{net} = W_T - W_c$

$= 274 - 198 = \underline{76 KJ/kg}$

Heat supplied/kg of air $= C_p(T_3 - T_2)$

$= 1(873 - 486)$

$= \underline{387 KJ/kg}$

$\therefore$ Overall eff of turbine $= \frac{WD \text{ or } W_{net}}{\text{Heat supplied}} = \frac{76}{387}$

$= 0.1964$

$\boxed{\eta_o = 19.64\%}$ (Ans)

Let m_a = mass of air circulated by the turbine in Kg/s

We know that the net work done by turbine

$W_n = 76 KJ/kg$ of air

$\therefore W_n = 76 \times m_a$ KW ——— ①

We have $P = 2200 KW$ ——— ②

$\therefore P = 76 m_a$

$\therefore m_a = \frac{P}{76} = \frac{2200}{76} = 28.95 Kg/s$

$\boxed{m_a = 28.95 Kg/s}$ Ans

Methods to improve the thermal efficiency of Gas Turbines plant

The thermal efficiency and specific output of the Gas Turbines plant can be increased with the help of following methods:

1) **Regeneration**

2) **Intercooling and**

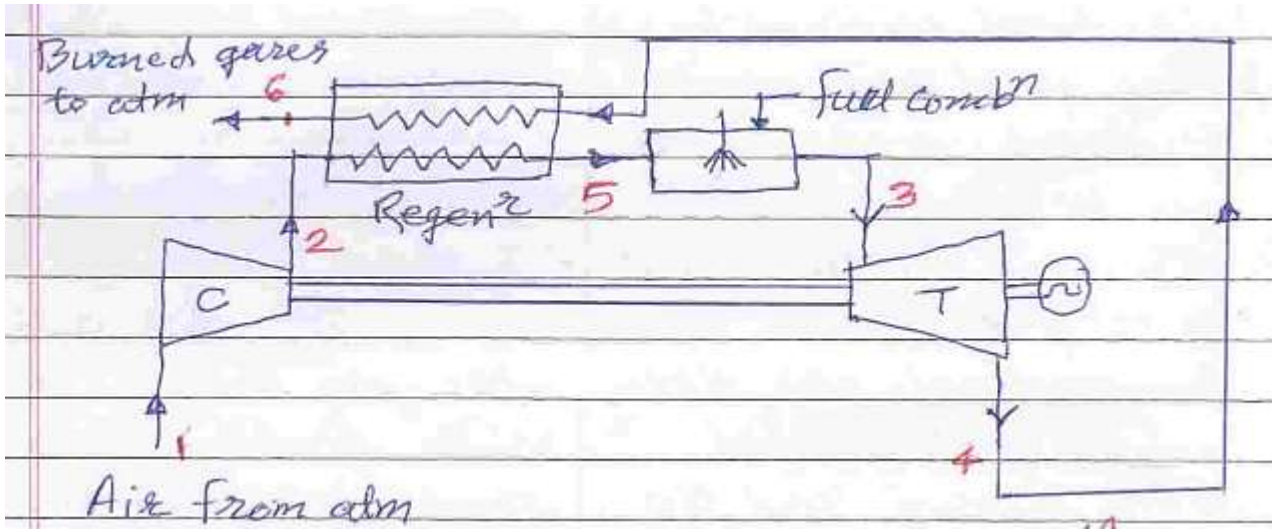
3) **Reheating**

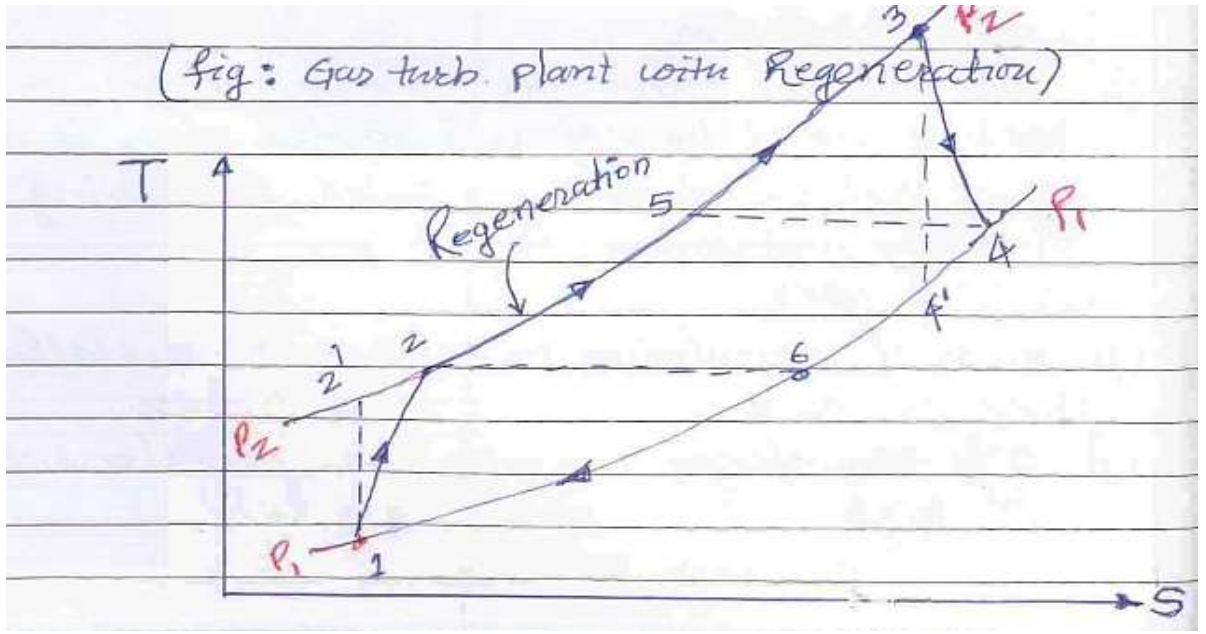
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Regeneration

Regeneration in gas turbines is a heat recovery process that uses hot turbine exhaust gases to preheat compressed air before it enters the combustion chamber.

By utilizing this waste heat via a regenerator, fuel consumption is reduced and the overall thermal efficiency of the cycle is improved.



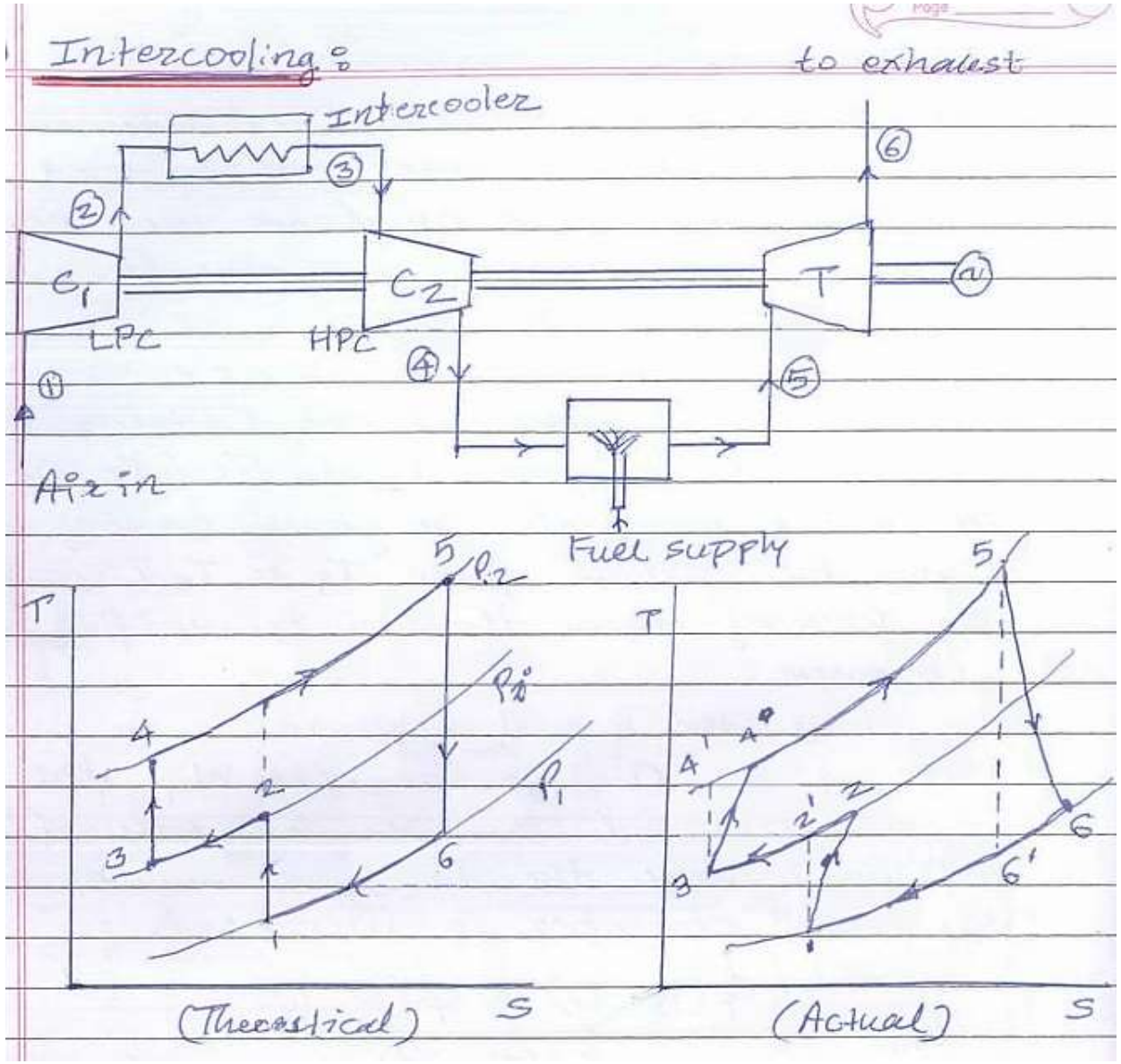


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Intercooling-

Intercooling in gas turbines is a technique that breaks compression into multiple stages, using a heat exchanger (intercooler) to cool the air between them.

This process reduces the compressor's specific volume and temperature, decreasing the work required to drive it, which increases the net power output of the plant.

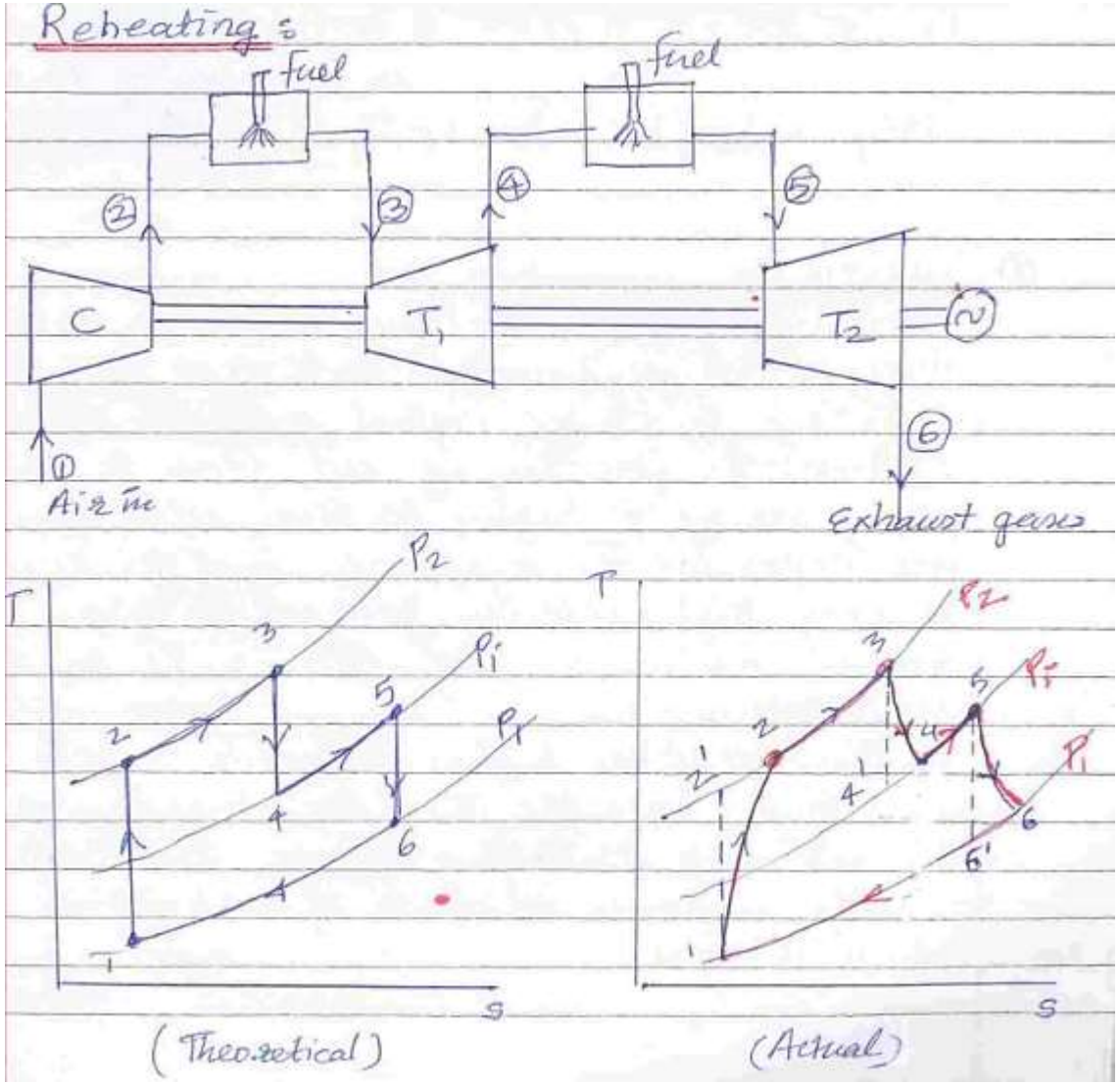


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Reheating-

Reheating in a gas turbine is a method that increases total net work output by reheating exhaust gases from the first turbine stage in a second combustor before further expansion, without increasing compressor work.

This process raises the average temperature of heat addition, increasing thermal efficiency

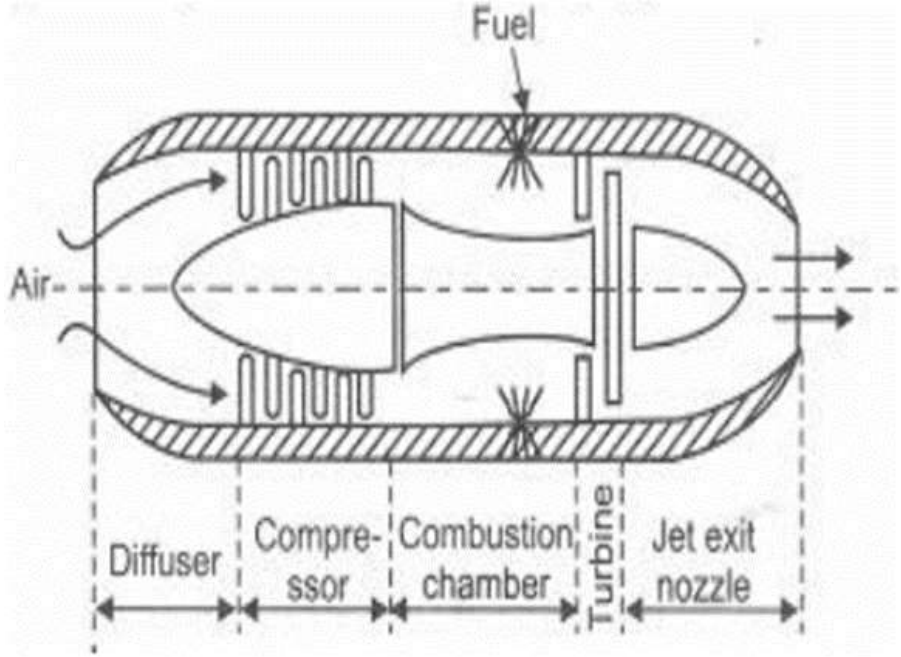


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Turbojet engine

In a turbojet unit, the turbine of the gas turbine power plant produces just sufficient power to drive the compressor by partial expansion in the turbine.

The gases coming out from the turbine, which are at a higher pressure than atmospheric pressure, are expanded in a nozzle and produce a very high-velocity jet. This provides forward motion to the aircraft through jet reaction.



Key Characteristics

Efficiency: Turbojets provide higher propulsion efficiency at higher speeds.

Suitability: They are best suited for aircraft traveling above 800 km/hr.

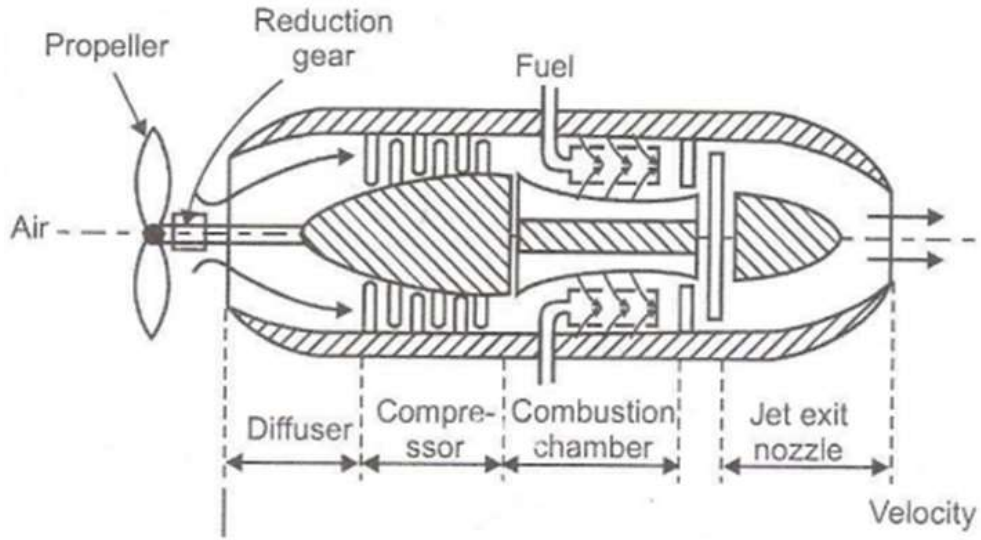
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Turbo propeller engine

In this system, the gases are partially expanded in the turbine and partially in the nozzle. Nearly 80% expansion takes place in the turbine and remaining 20% in the nozzle.

The total power developed by the turbine is used to run the compressor as well as propeller. The forward motion to the aircraft is given partly by the propeller and partly by the jet produced by the nozzle.

The turbo-prop combines the merits of a turbojet (i.e. low specific weight and simplicity in design) and the merits of a propeller (i.e. high power for take-off and high propulsive efficiency at speeds less than 600 km/hr.



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Unit 5

What is a Refrigeration

- Refrigeration is defined as an art of producing and maintaining the temperature in a space below the temperature of surrounding.
- Refrigerants are used as working substances in a Refrigeration systems.
- Refrigerant absorbs heat from the place to be cooled, carries and reject it to the atmosphere.
- The heat transfer generally takes place through a phase change of the refrigerant.
- *"Refrigerant is the fluid used for heat transfer in a refrigerating system that absorbs heat during evaporation from the region of low temperature, low pressure and rejects heat during condensation to a region of higher temperature.*

What is a Refrigeration

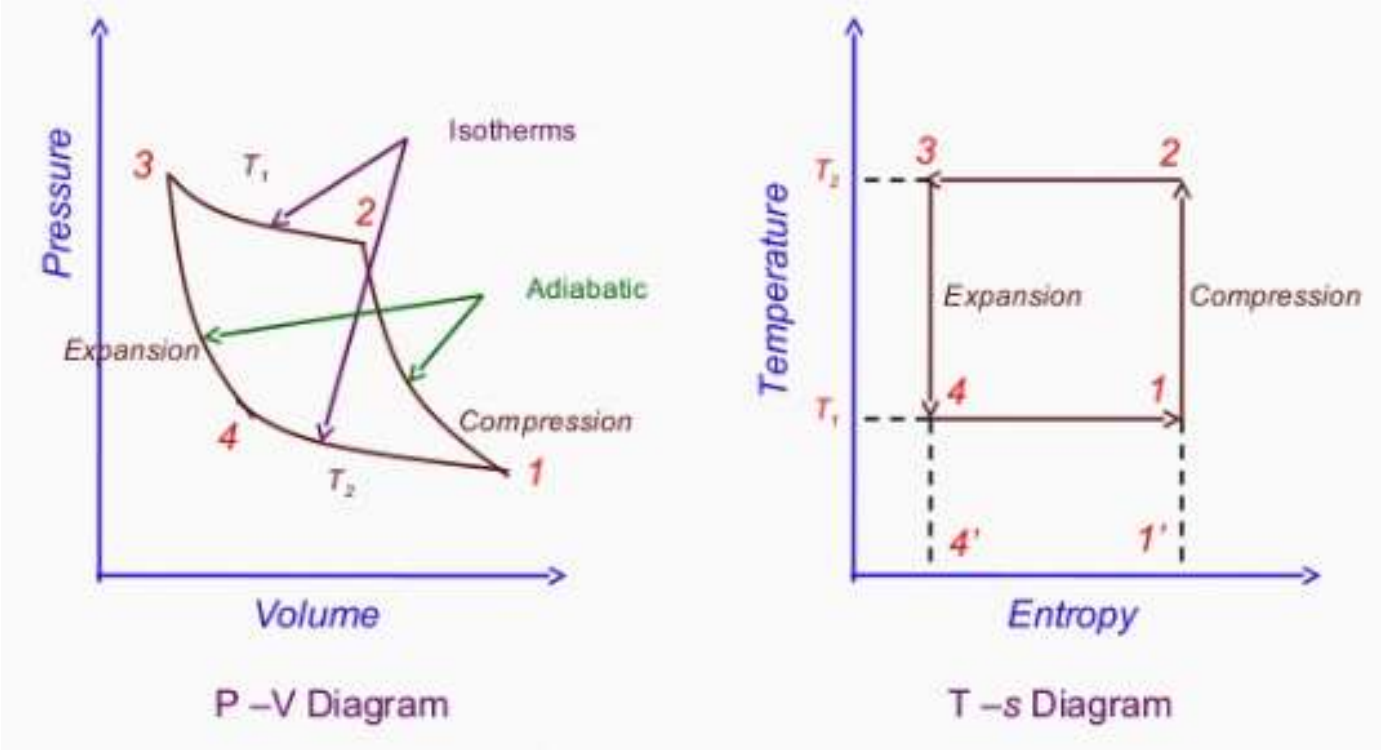
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- Refrigerants absorb heat at low temperature and low pressure and release heat at a higher temperature
- Most refrigerants undergo phase change during heat absorption (evaporation) and heat release (condensation)
- The thermodynamic efficiency of a refrigeration system depends mainly on its operating temperatures.

Reversed Carnot cycle for Refrigeration system

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Refrigerator works on **Reverse Carnot Cycle** as shown in PV & TS diagram



The Coefficient of performance (COP) for a reversed Carnot refrigeration cycle is defined as the ratio of refrigeration effect to work input

It is expressed using absolute temperatures as

$$COP = \frac{T_2}{T_1 - T_2}$$

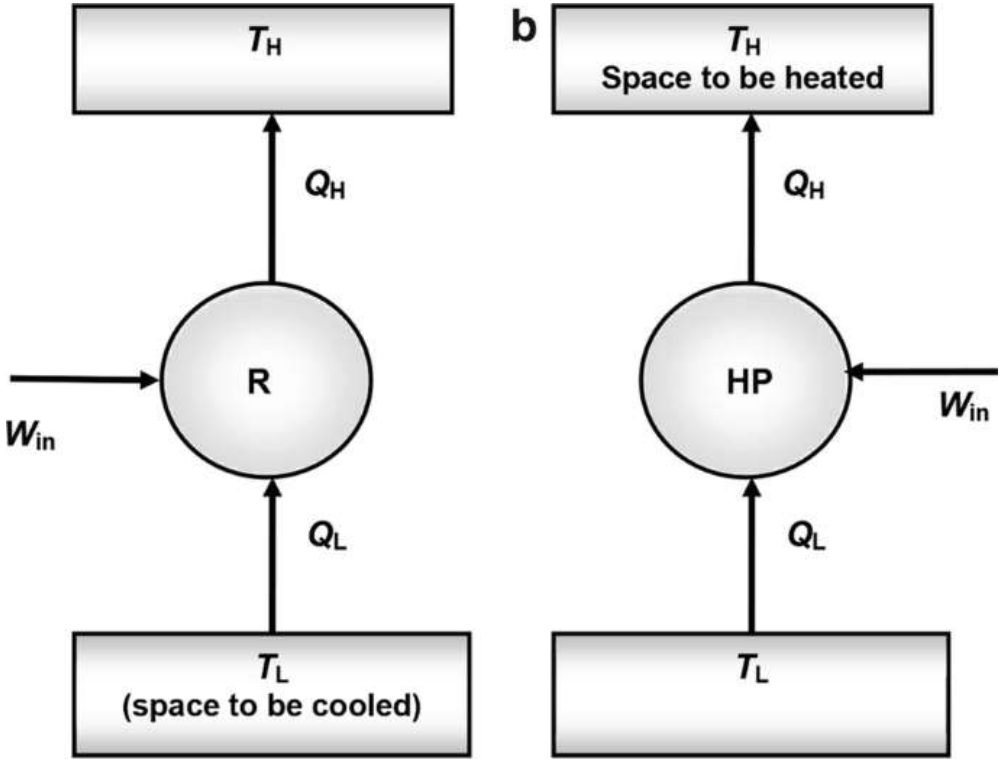
Where-
T1 - Condenser temperature (Higher)
T2 - Evaporators temperature (Lower)

Refrigerator and Heat Pump

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Refrigeration is an equipment used to remove the heat continuously from the space and maintain the temp below atmospheric temp & reject the heat to the atmosphere with the help of external work.

Heat Pump is an equipment used to supply the heat continuously to the space & maintain its temp above atmospheric temp with the help of external work.



COP of Refrigerator and Heat Pump

$$COP_{REF} = \frac{T_2}{T_1 - T_2}$$

$$COP_{HP} = \frac{T_1}{T_1 - T_2}$$

Carnot cycle for Refrigeration system

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Refrigerator works on Reverse Carnot Cycle as shown in above PV & TS diagrams

① Process 1-2: Isentropic compression of working fluid (air) by using external work. Temperature rises from T_2 to T_1 (T_H)

② Process 2-3: Isothermal compression, heat is rejected at constant temperature T_1 (T_H)

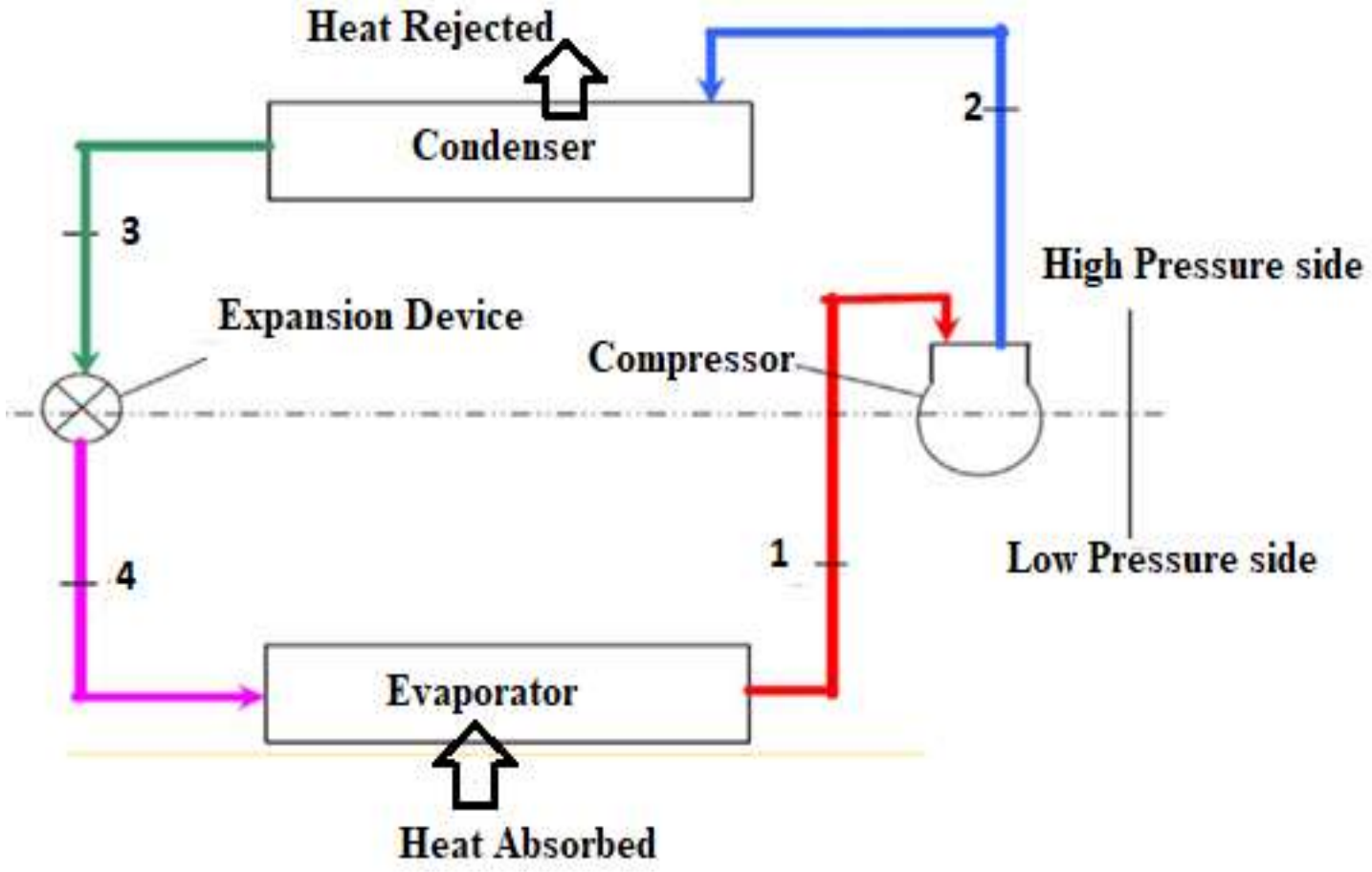
③ Process 3-4: Isentropic expansion. Temperature falls from T_1 to T_2

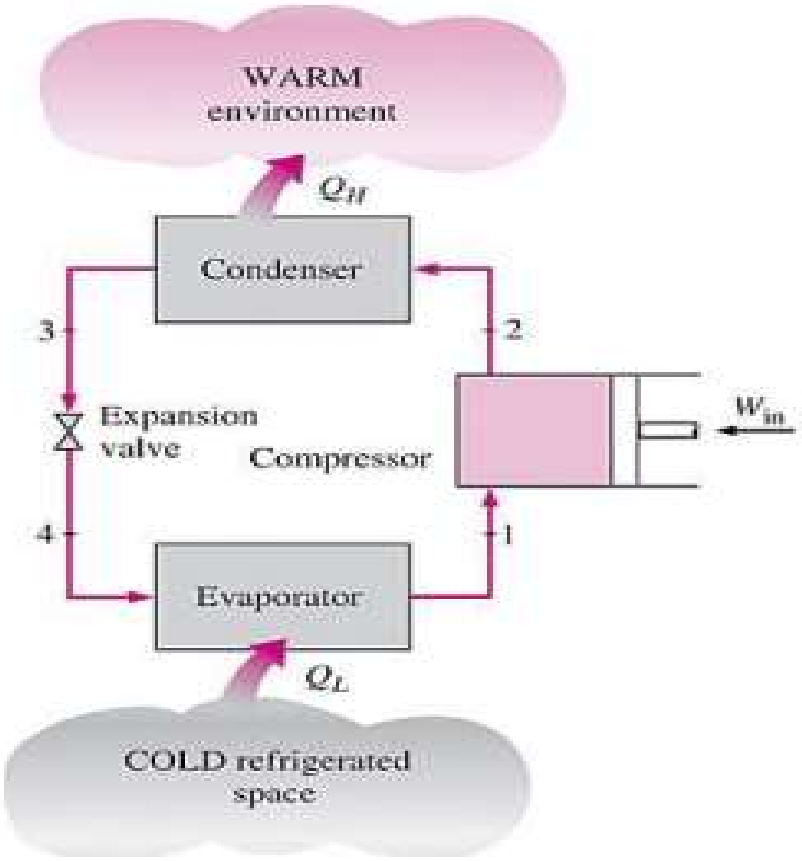
④ Process 4-1: Absorption of heat from refrigerator at constant temperature (isothermal expansion) $T_{2.}$ (T_L)

COP of a refrigeration system depends mainly on its operating temperatures.

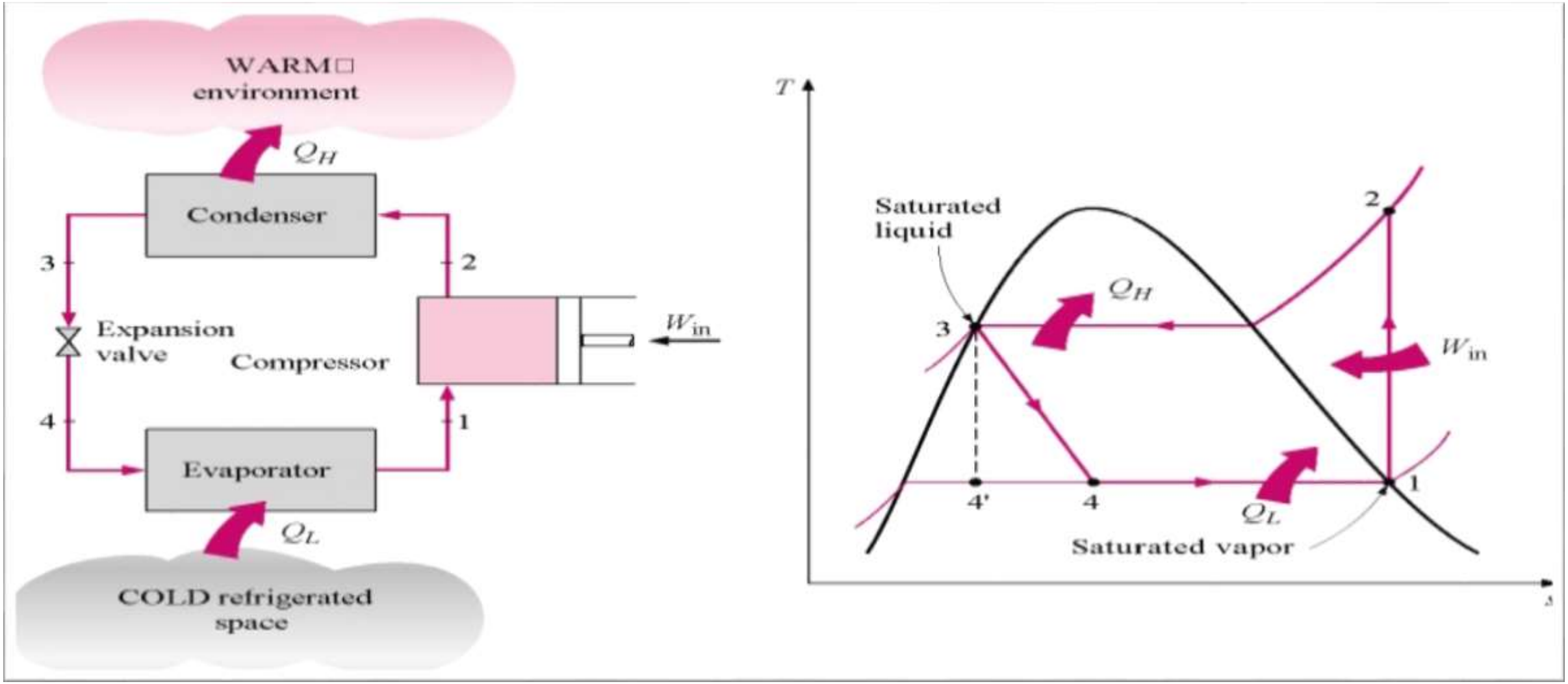
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Simple Vapour Compression Refrigeration Cycle (VCRC)





Vapour Compression Refrigeration Cycle



Unit of Refrigeration

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- The rate at which heat is absorbed from the space to be cooled is known as the refrigerating effect

- The standard unit used for expressing the capacity of refrigerating machine is TON of refrigeration (TR)
- Capacity of a refrigerating machines is expressed by its cooling capacity.

- **A ton of refrigeration is equivalent to the rate of heat transfer needed to produce 1 tone of ice at 0 degree Celsius from water at 0 degree Celsius in 24 hours.**

1 TR = 210 kJ/min = 3.516 kW

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Ton of refrigeration

- A ton of refrigeration is equivalent to the rate of heat transfer needed to produce 1 ton of ice at 0 °C from water at 0 °C in 24 hours
- A machine capable of producing a net refrigeration effect of 210 kJ/min or 3.516 kW is called a 1 ton machine.
- Heat extracted from water at 0^{o} C = Latent heat of ice

  Latent heat of ice = 335 kJ/kg
- i.e., 335 kJ of heat should be extracted from 1kg of water at 0^{o} C to convert it into ice at 0^{o} C .
- One ton of refrigeration (1TR) = 335 x 1000 kJ/24 hrs.

  = 335 x 1000 /(24 x 60) kJ/min

  = 210 kJ/min

  = 3.516 kJ/s = kW

Psychrometry

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Psychrometry is the branch of science which deals with the study of properties of moist air and its behavior under different conditions

Dry Air:

Air consisting of N2, O2, CO2, H2 Argon etc.is called dry air;

(Air without water vapour)

Moist Air:

It is a homogeneous mixture of dry air and water vapour. The amount of wv present in the air depends on absolute pressure and temp. of the mixture

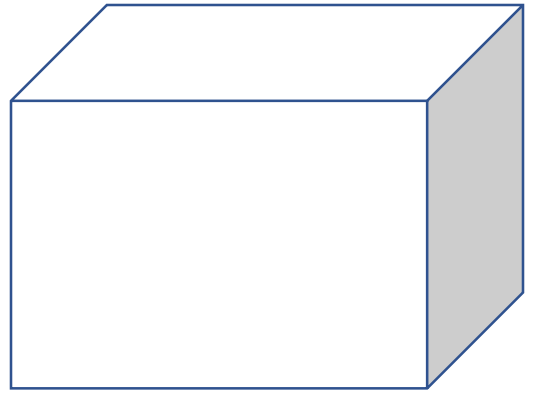
Dalton's Law of Partial Pressure:

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The total pressure exerted by the moist air is equal to the sum of pressures, which each constituent exerts, if it occupies the same space by itself.

$P_{total}=P_{barometric}$ (P_t or P_b)

$P_b= P_{da}+P_{wv}$ or $P_b= P_a+P_v$



P_{da}=Partial pressure of dry air

P_v=Partial pressure of water vapour

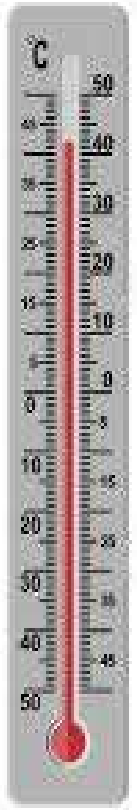
✓Since P_v is very small ,the saturation temperature of w v is also low thus the w v in the air exists in superheated state (superheated vapour)

Psychrometric Properties of Air

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Dry Bulb Temperature (DBT):

It is the temperature of air/moist air measured by a thermometer.



Wet Bulb Temperature (WBT):

It is the temperature of air/moist air measured by a thermometer when its bulb is covered with wet cloth or wick over which air is moving at a velocity of 2.5m/s to 10m/s

Wet bulb temperature is the lowest temperature that can be reached by the evaporation of water only.

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Wet bulb depression

The difference between DBT and WBT is known as wet bulb depression

- Wet bulb depression=DBT-WBT
- If wet bulb depression is zero

  DBT=WBT, air is saturated (RH=100%)
- For saturated air DBT=WBT= DPT

Saturated Air/Unsaturated Air

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Saturated Air:

- The moisture holding capacity of the air depends upon its temperature and pressure. Air at higher temperature can hold more moisture than the air at low temperature
- If air is saturated with water vapour, is known as saturated air.
- It is the condition of air that holds maximum quantity of water vapour into it at a given temperature.







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Degree of saturation:

It is the ratio of actual mass of w.v. (water vapour) in a unit mass of dry air to the mass of w.v. in the same mass & pressure of dry air when it is saturated at the same temp.

Humidity / Specific humidity / Humidity Ratio:

It is the mass of w.v. in 1 Kg of dry air. Expressed in gm/Kg of d.a

Absolute humidity:

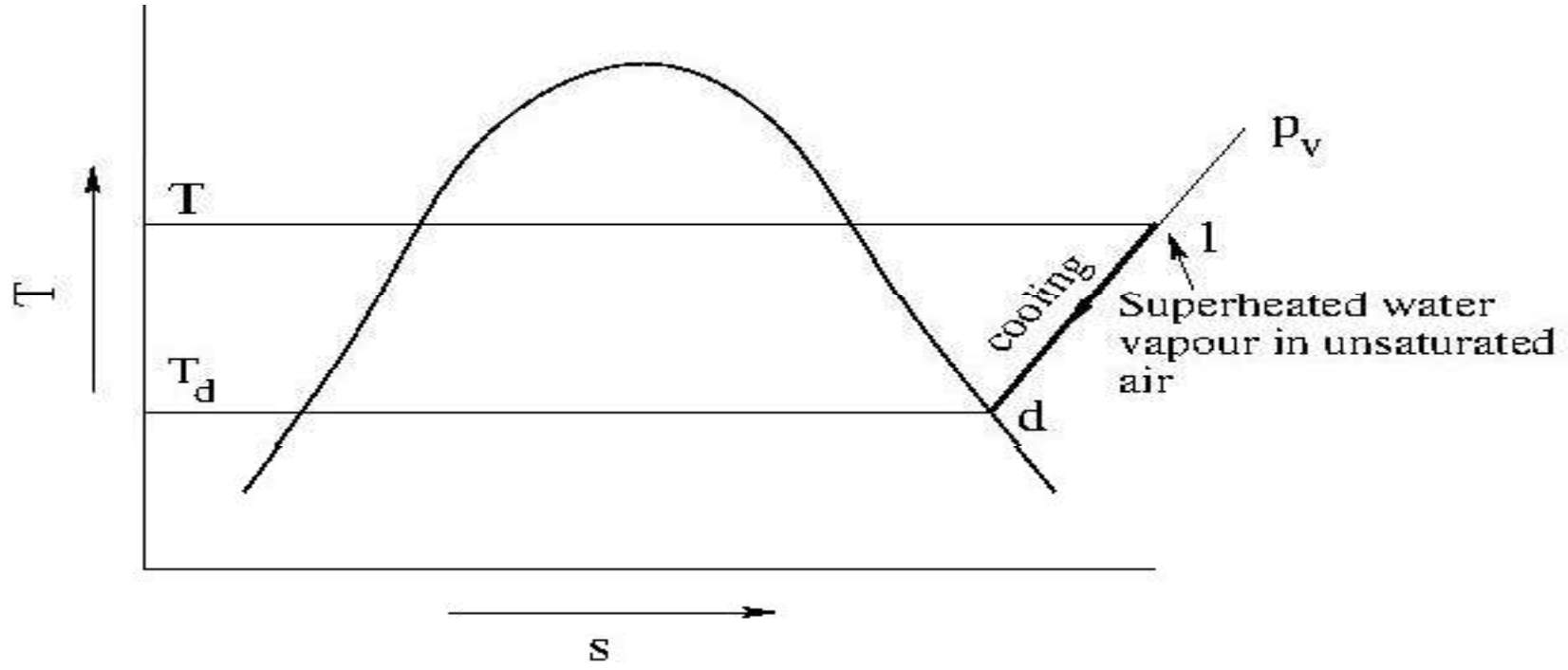
It is the mass of w.v. present in $1m^3$ of dry air. Expressed in gm/ m^3 of d.a.

Relative Humidity

It is the ratio of the actual mass of water vapor (w.v.) in a given volume of moist air to the mass of water vapor in the same volume of saturated air at the same temperature and pressure.

Dew Point Temperature (DPT):

- If unsaturated air is cooled at constant pressure, its temperature goes on decreasing. The temperature at which the first drop of dew is formed is called dew point temperature(DPT).
- Condensation occurs at the Dew Point Temperature.



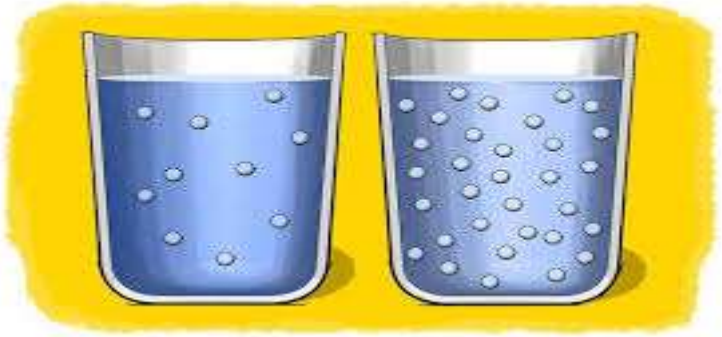
Dew Point Temperature (DPT):

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Dew Point Temperature (DPT):

- The temperature at which moisture starts to condense out of the air is known as **DEW POINT**.
- Dew point is also known as saturation temperature corresponding to the partial pressure of water vapour (P_v)







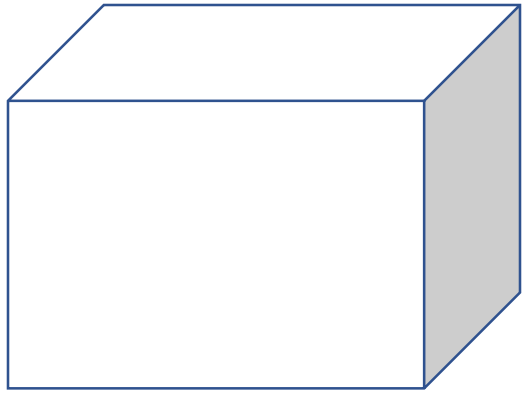
Humidity ratio or specific humidity

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Humidity ratio or specific humidity(w):

- The moisture content or the amount of water vapour in a moist air is known as specific humidity.
- It is the ratio of mass of w v to the mass of dry air contained in the sample of air.
- ✓

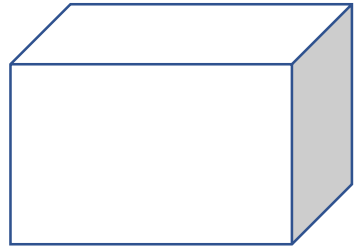
$$w = \frac{mass\ of\ wv}{mass\ of\ da} = \frac{m_{wv}}{m_{da}}$$



- it is expressed in g/kg of d. a.

Humidity ratio or Specific humidity

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- Dry Air $P_aV_a=m_aR_aT_a$ ----------(1)
- Water vapour $P_vV_v=m_vR_vT_v$ ----------(2)
- Here $V_a=V_v$, and $T_a=T_v$
- Equation (1)/Equation (2)

$$\frac{P_aV_a}{P_vV_v}=\frac{m_aR_aT_a}{m_vR_vT_v} \quad \text{and} \quad \frac{P_a}{P_v}=\frac{m_aR_a}{m_vR_v} \text{ ----------(3)}$$

- R_a, R_v= Characteristic gas constant
- R=Universal gas constant=8.314kJ/kgK
- M=Molecular weight, $M_a=29$, $M_v=18$

Humidity ratio or specific humidity

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$$R_a = \frac{R}{M_a} = \frac{8.314}{29} = 0.287 \text{ kJ/kgK} \quad \text{and} \quad R_v = \frac{R}{M_v} = \frac{8.314}{18} = 0.461 \text{ kJ/kgK}$$

- Put in equation (3) $\dfrac{P_a}{P_v} = \dfrac{m_a R_a}{m_v R_v}$

$\dfrac{P_a}{P_v} = \dfrac{m_a R_a}{m_v R_v}$ Rearranging equation (3) we get $\dfrac{m_v}{m_a} = \dfrac{P_v R_a}{P_a R_v}$

$$\frac{m_v}{m_a} = w = \frac{P_v}{P_a}\frac{0.287}{0.461}$$

$$w = 0.623\frac{P_v}{P_a} \quad \text{or} \quad w = 0.623\frac{P_v}{P_b - P_v}$$

- $P_b = P_a + P_v$
- At saturation ($P_v = P_s$)

$$w_{max} = 0.623\frac{P_s}{P_b - P_s}$$

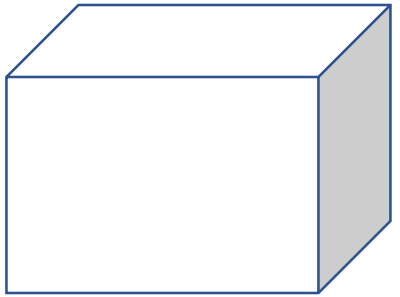
Relative Himidity

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Relative Humidity (RH= Ø):

- It is the ratio of actual mass of w.v. (m_v) in a given volume of moist air to the mass of w.v. if the air is saturated at the same P&T

$$RH = \phi = \frac{m_v}{(m_v)_{sat}}$$



w v in unsaturated air $P_v V_v = m_v R_v T_v$ ----------(1)

w v in saturated air $(P_{vs})(V_{vs}) = (m_{vs})(R_{vs})(T_{vs})$ --------(2)

Here $V_v = V_{vs}$ and $R_v = R_{vs}$ and $T_v = T_{vs}$

Relative Humidity

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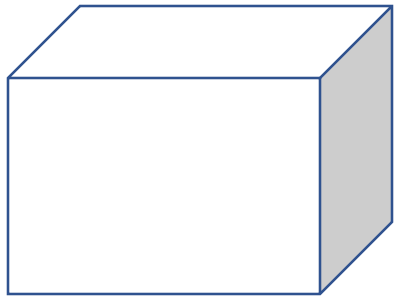
- Equation (1)/Equation (2)

$$\frac{P_v V_v}{P_{vs} V_{vs}} = \frac{m_v R_v T_v}{m_{vs} R_{vs} T_{vs}}$$

- Here $V_v = V_{vs}$ and $R_v = R_{vs}$ and $T_v = T_{vs}$

$$\frac{P_v}{P_{vs}} = \frac{m_v}{m_{vs}}$$

$$RH = \phi = \frac{P_v}{P_{vs}}$$



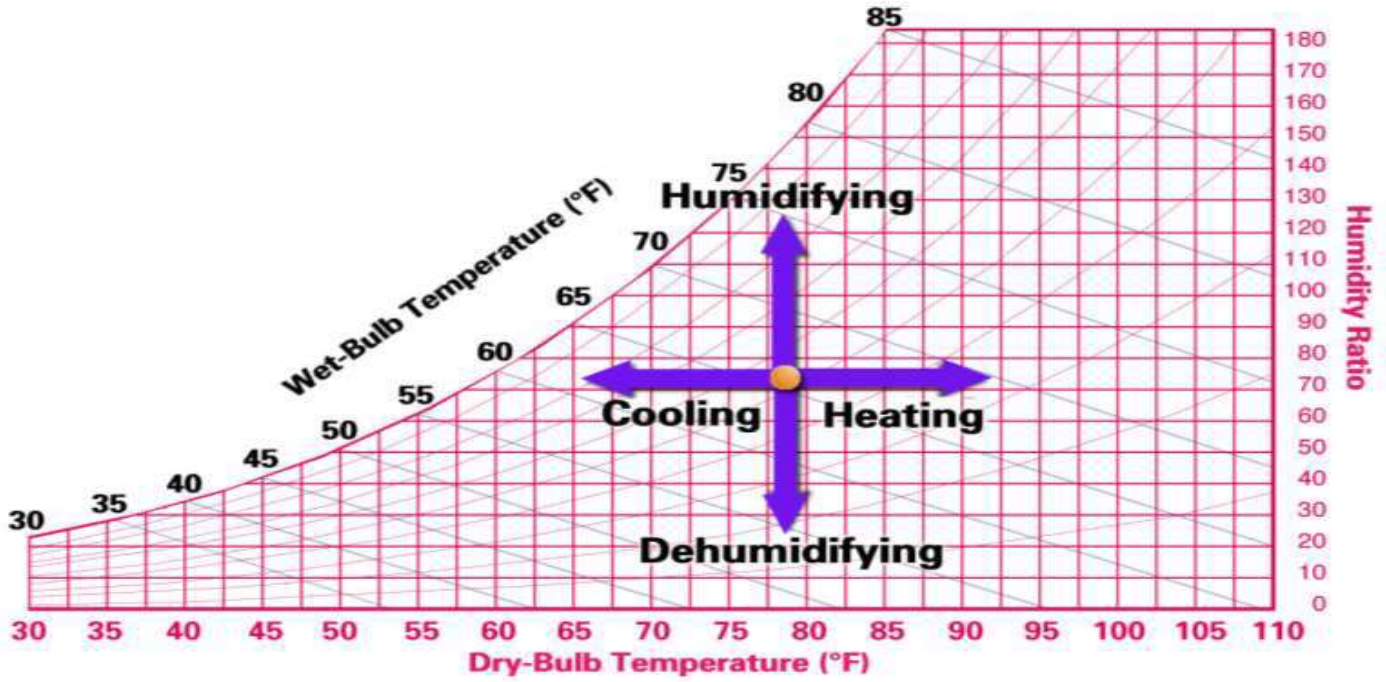
- P_v=Partial Pressure of wv (actual air/unsaturated air)
- P_{vs}=Partial Pressure of wv if air is saturated

Psychrometric Chart

Psychrometric chart is a graphical representation of the thermodynamic properties of moist air.

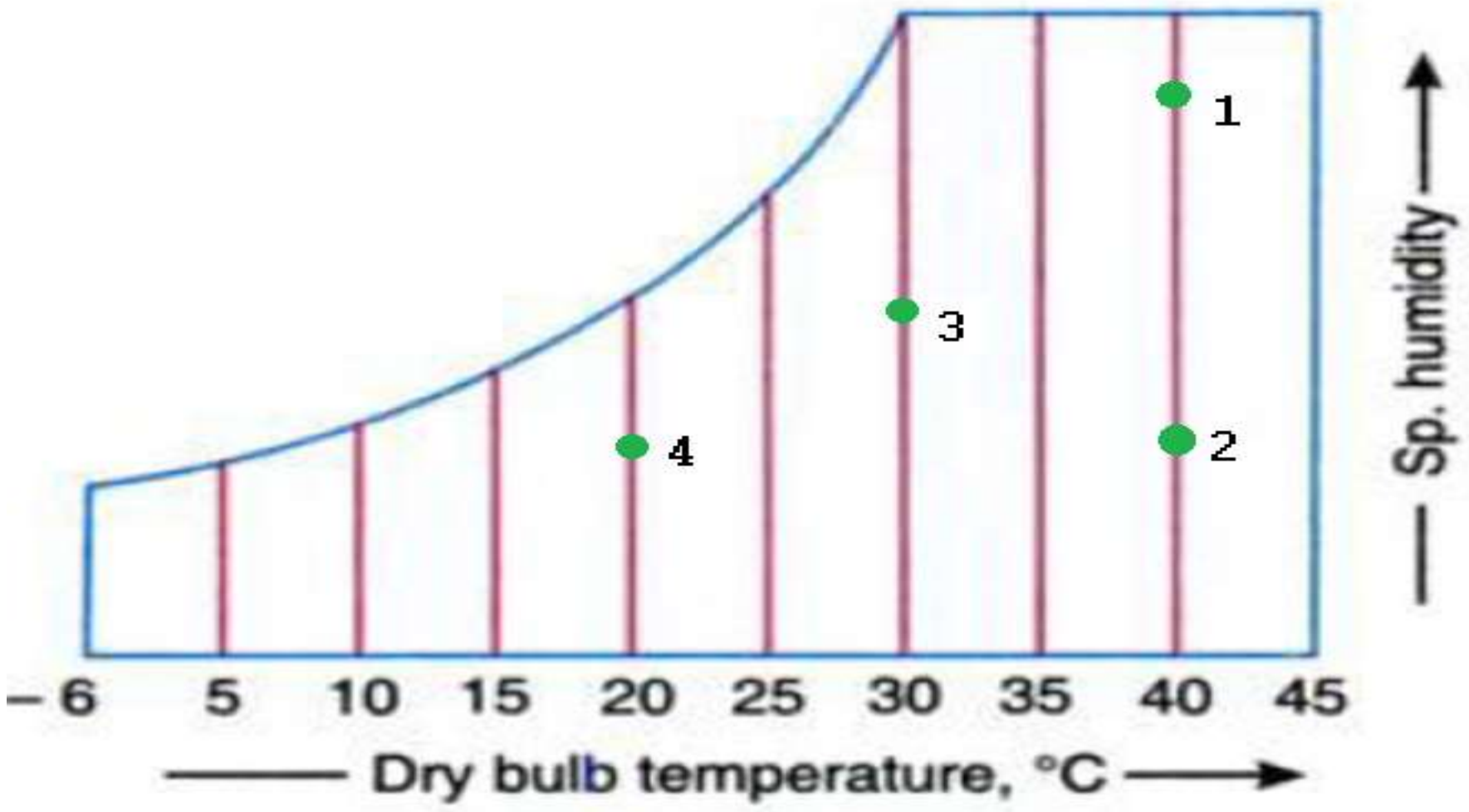
The chart is normally drawn for standard atmospheric pressure of 760 mm Hg or 101.325 kPa.

1. DBT
2. DPT
3. WBT
4. Enthalpy
5. Sp humidity
6. RH
7. Specific volume



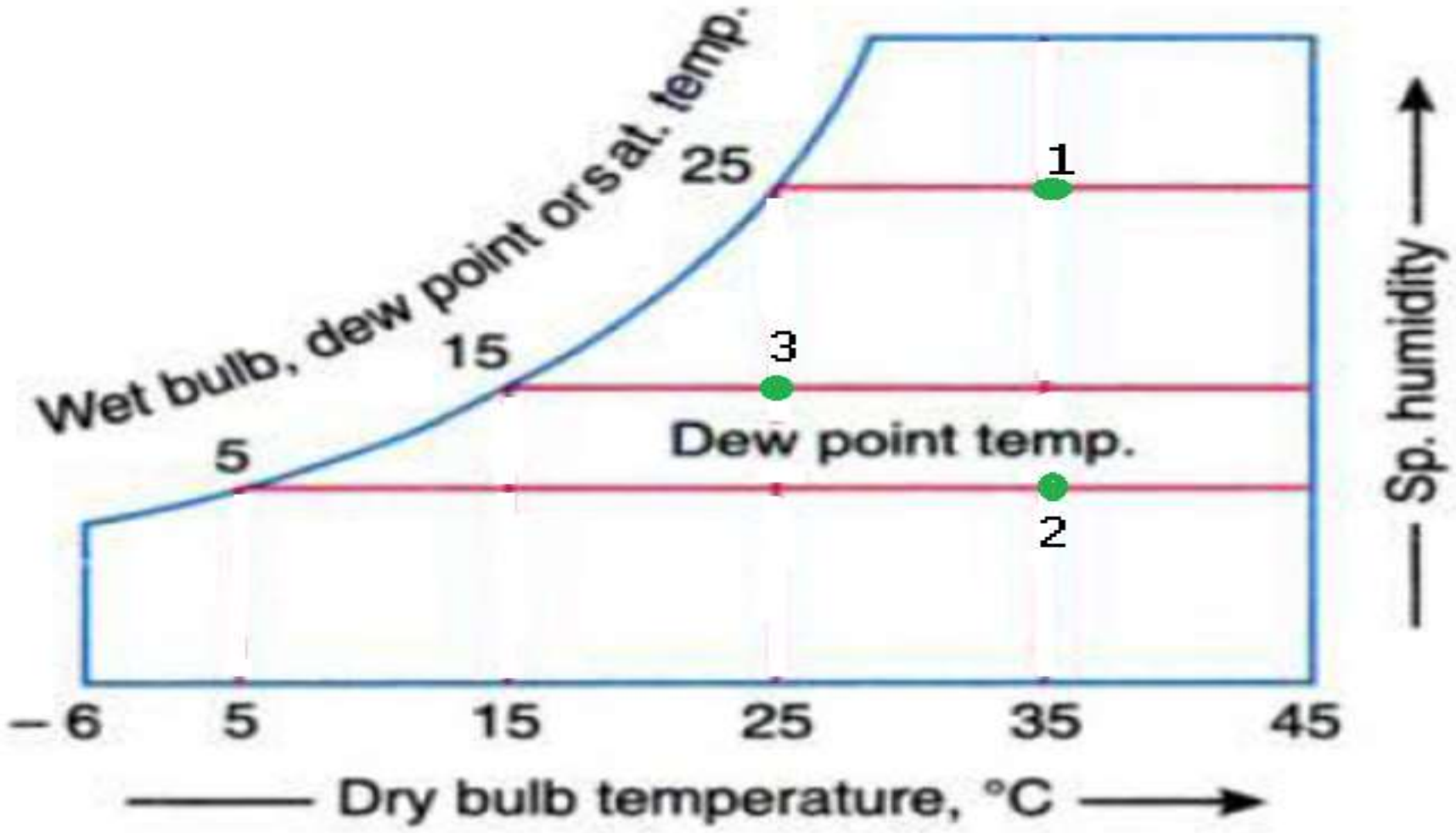
1. Dry Bulb Temperature (DBT) lines

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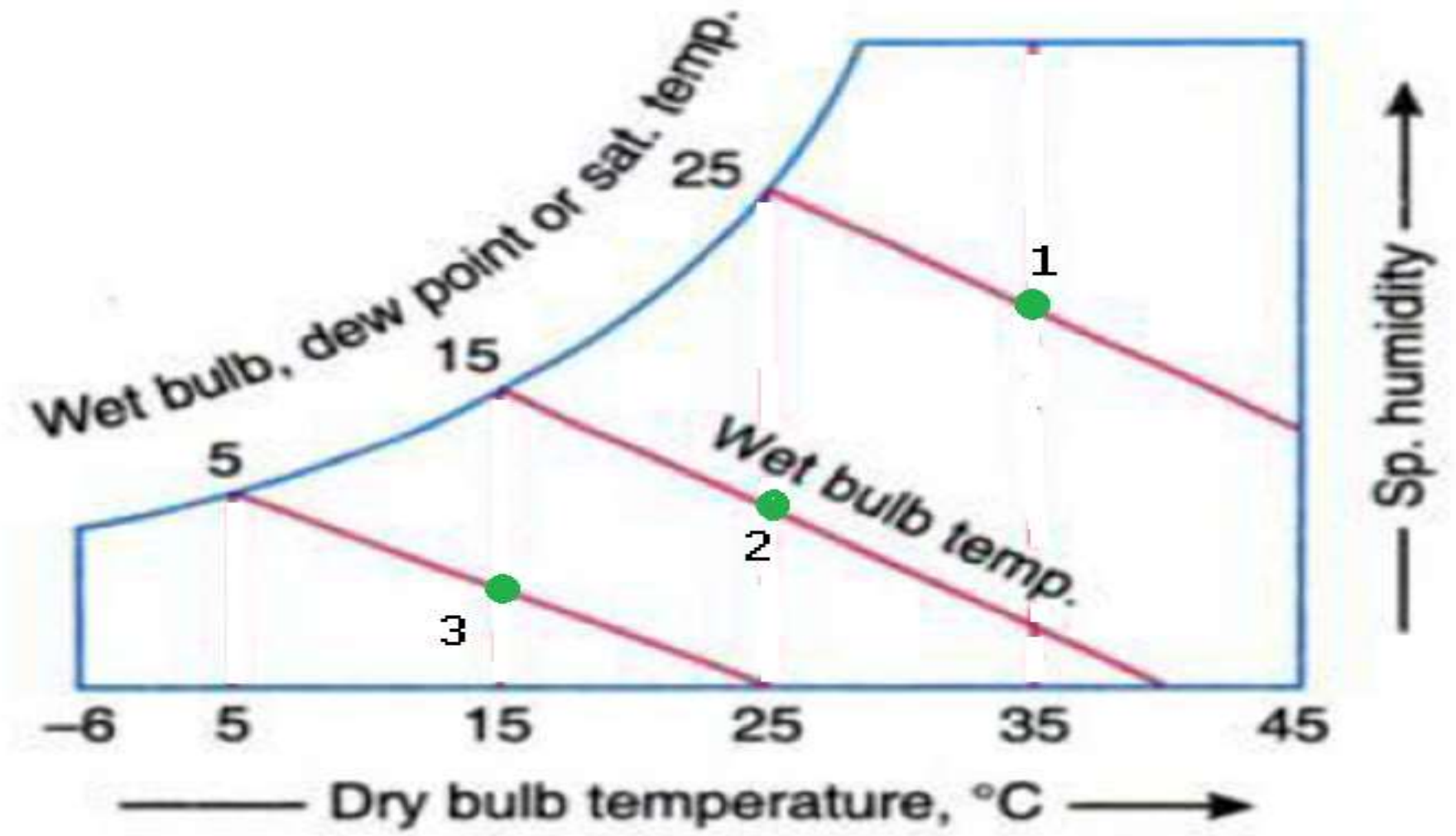
2. Dew Point Temperature(DPT) lines

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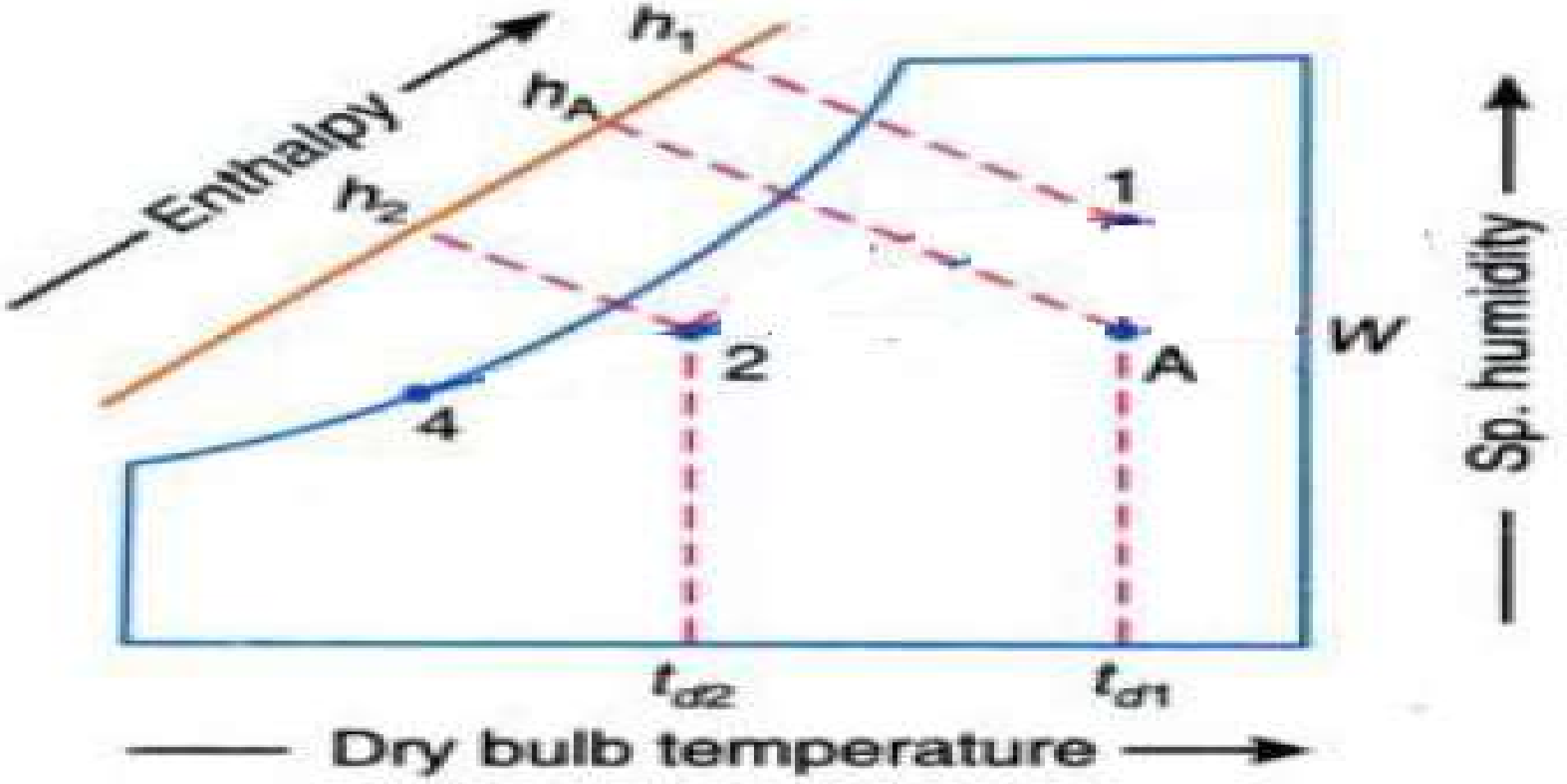
3. Wet Bulb Temperature(WBT) lines

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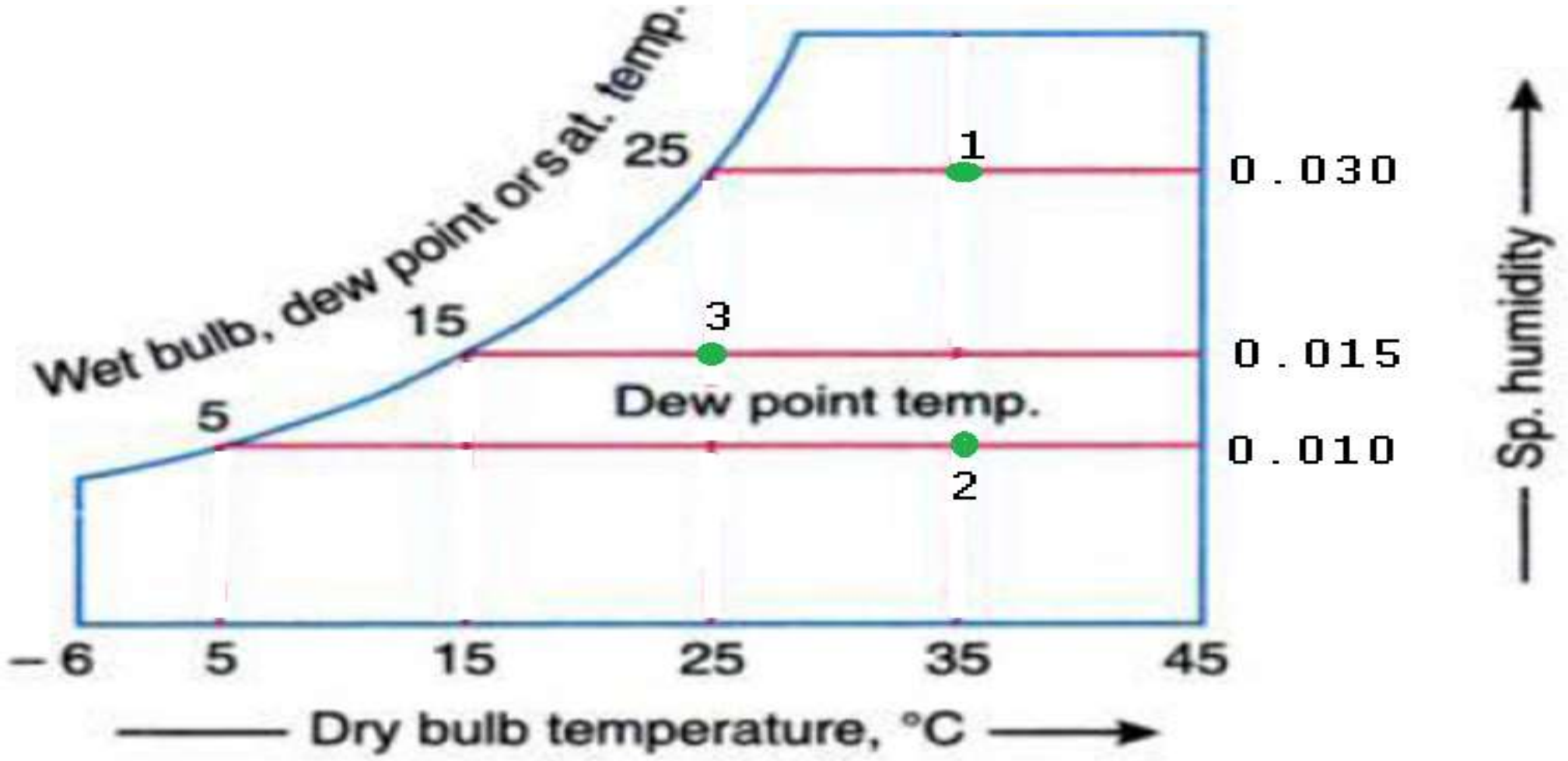
4. Enthalpy(total heat) lines

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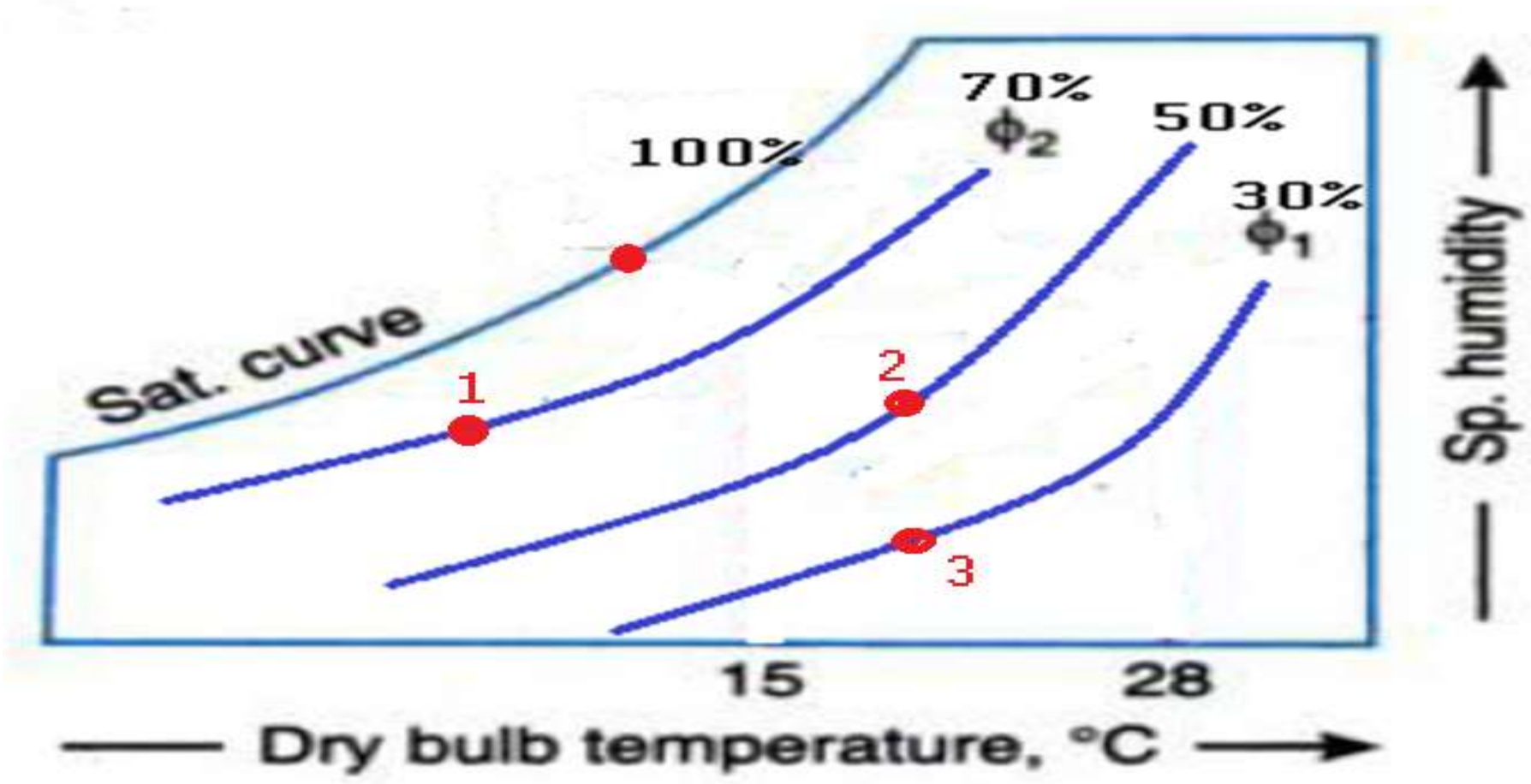
5. Specific humidity (w)

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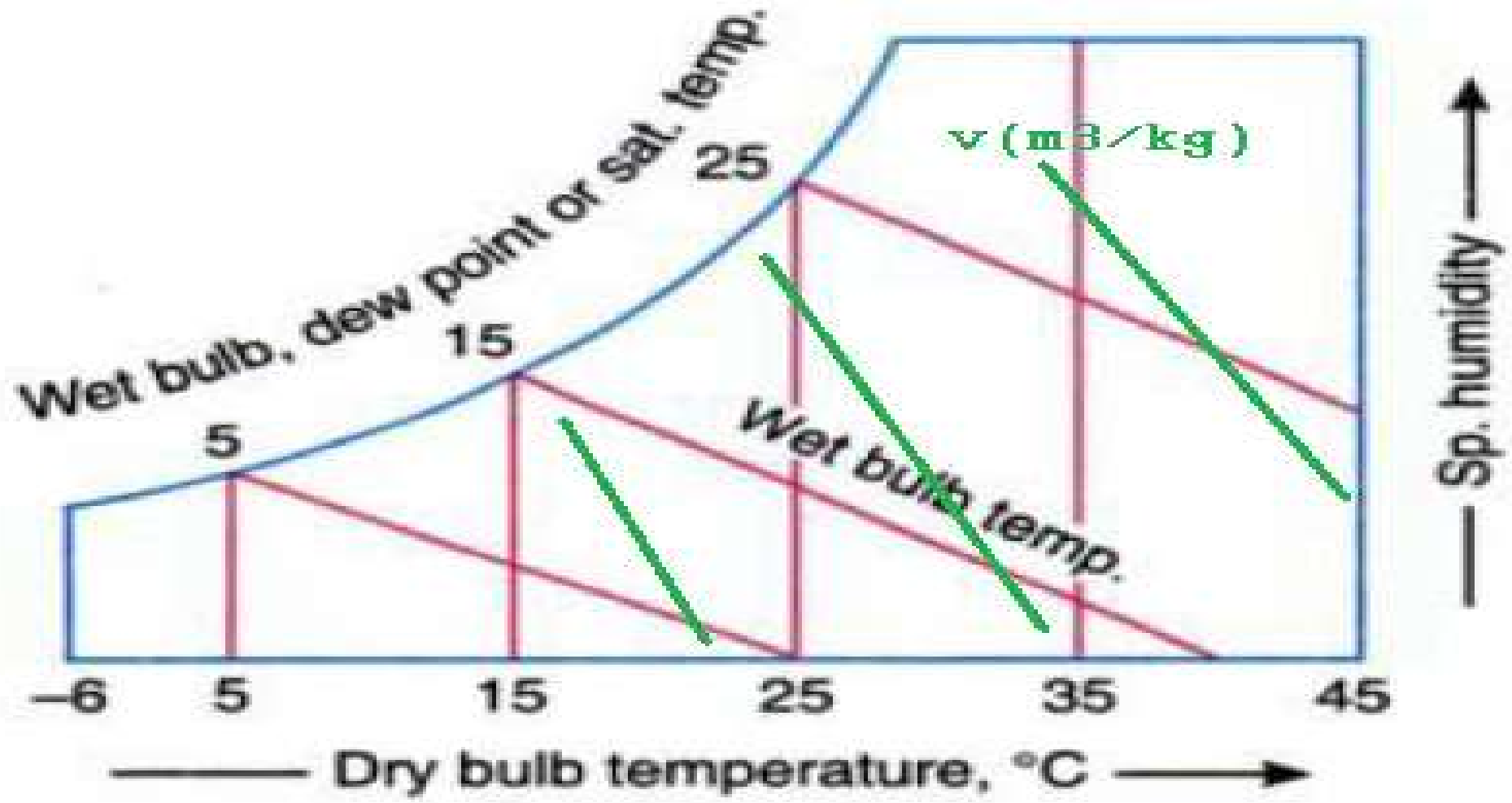
6. Relative Humidity (RH) lines

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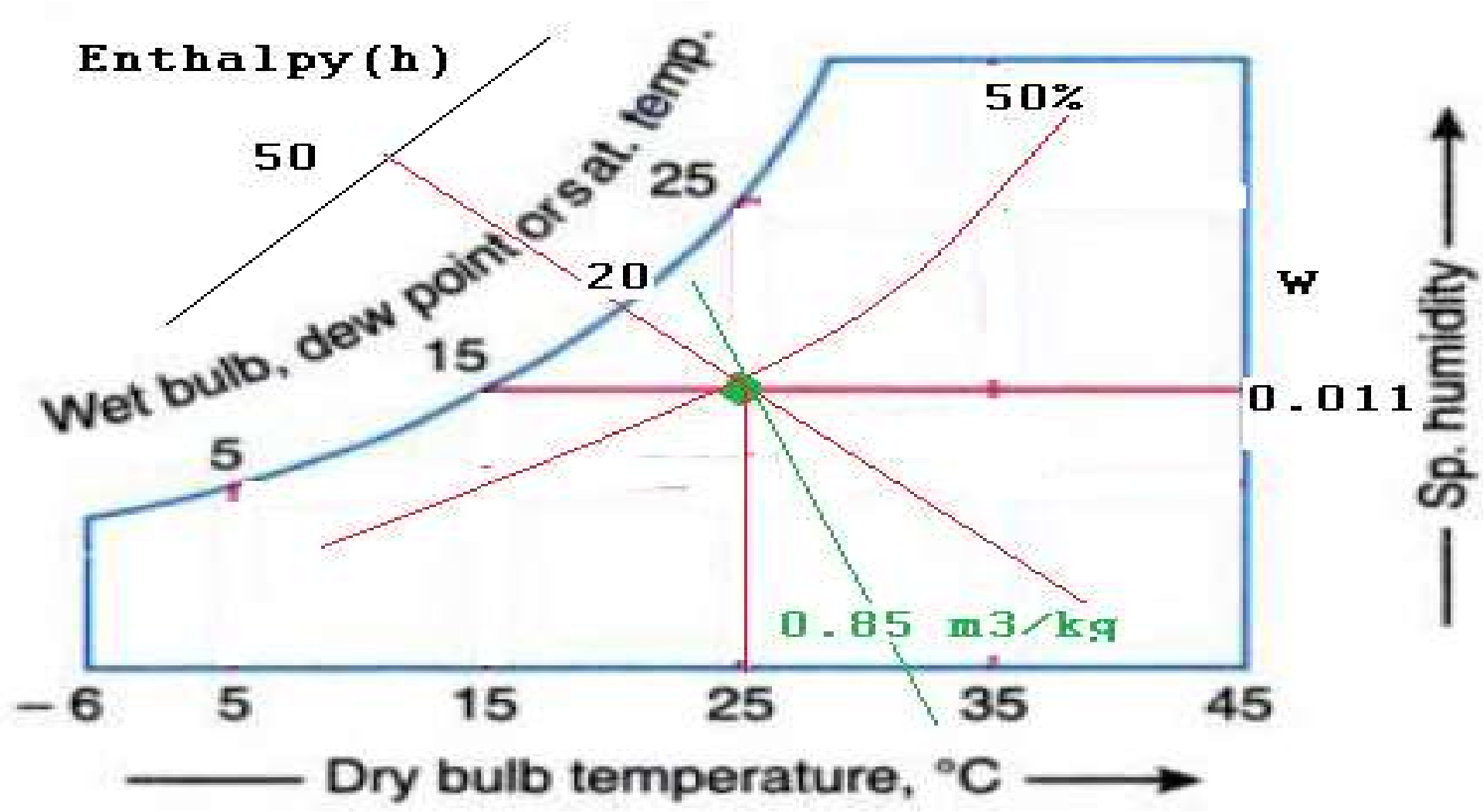
7. Specific volume lines (v)

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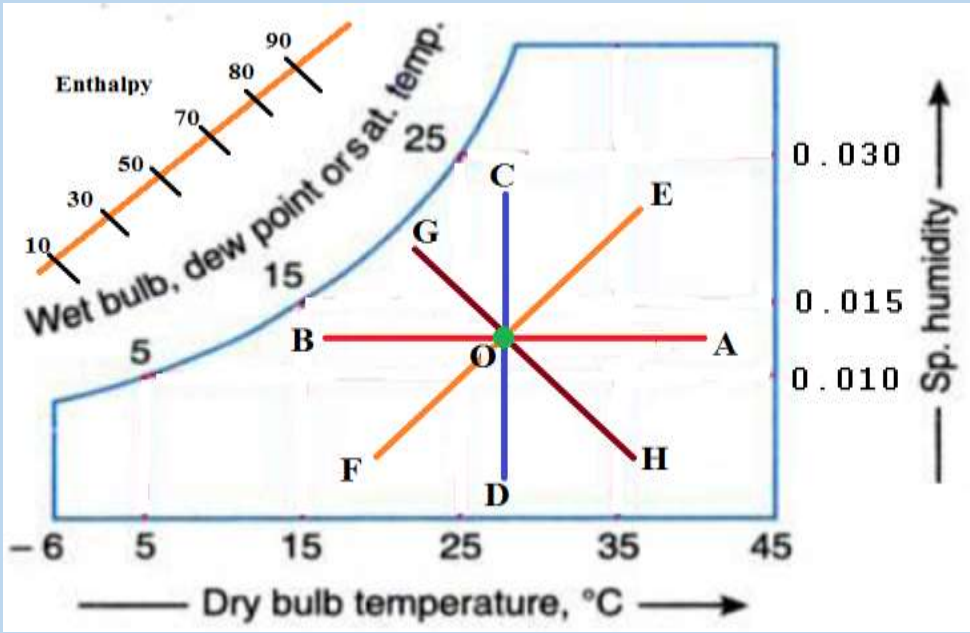
Psychrometric Chart

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Psychrometric Processes

- O-A: Sensible heating
- O-B: Sensible Cooling
- O-C: Humidification
- O-D: Dehumidification



- O-E: Heating and Humidification (Winter AC)
- O-F: Cooling and Dehumidification (Summer AC)
- O-G: Cooling and Humidification
- O-H: Heating and dehumidification

Numerical

1) A machine working on a Carnot cycle operated between 305 K and 260 K. Determine the COP when it is operated as refrigerator, heat pump and heat engine.

Solution: $T_1 = 305$ K $T_2 = 260$ K

$$COP_{REF} = \frac{T_2}{T_1 - T_2} = \frac{260}{305 - 260} = 5.78$$

$$COP_{HP} = \frac{T_1}{T_1 - T_2} = \frac{305}{305 - 260} = 6.78$$

$$COP_{Engine} = \frac{T_1 - T_2}{T_1} = \frac{305 - 260}{305} = 0.147 \text{ or Efficiency is } 14.7\%$$

Numerical

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2) A Vapour compression refrigerator uses R40 (Methyl chloride) as a refrigerant and operates between temperature limits of – 10 and 45 degree Celsius. At entry to the compressor, the refrigerant is dry saturated and after compression it acquires a temperature of 60 degree Celsius. Find the COP of the refrigerator. The required properties OF r40 are as follows-

Sat. temp (^{0}C)	**Enthalpy in KJ/Kg**		**Entropy in KJ/Kg-K**	
	Liquid	Vapour	Liquid	Vapour
-10	45.4	460.7	0.183	1.637
45	133.0	483.6	0.485	1.587

Solution

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Given data

$T_1 = T_A = -10^\circ C = 263 K$

$T_2' = T_3 = 45^\circ C = 318 K$

$T_2 = 60^\circ C = 333 K$

$h_1 = 460.7$ $\quad h_2' = 483.6$

$h_{f_1} = 45.4$ $\quad h_{f_3} = 133$ kJ/kg

$S_2 = S_1 = 1.637$ $\quad Sf_1 = 0.183$

$S_2' = 1.587$ kJ/kgK $\quad Sf_3 = 0.485$ kJ/kgK

$$S_2 = S_2' + c_p \log_e\left(\frac{T_2}{T_2'}\right)$$

$\therefore c_p = 1.09$ sp heat for superheated vap.

$$h_2 = h_2' + c_p(T_2 - T_2')$$

$= 483.6 + 1.09(333 - 318)$

$= 500$ kJ/kg

$$\therefore (COP)_R = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{460.7 - 133}{500 - 460.7}$$

$= 8.34$ Ans.