ATME College of Engineering

Thermo-Fluids Engg
21ME52

Module-1
I C Engines

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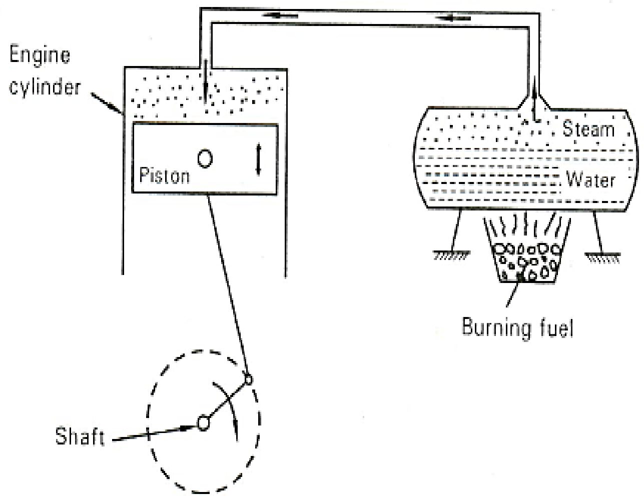
I.C ENGINE

Heat engine can be defined as a device or machine that converts the chemical energy of a fuel into heat energy by combustion of fuel, and utilizes this heat energy to perform useful mechanical work (usually in the form of rotation of shaft)

- Internal combustion engine
- External combustion engine



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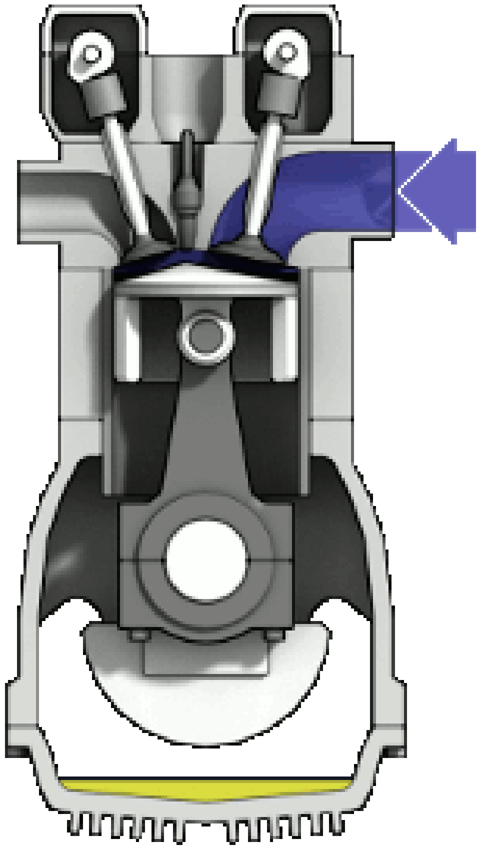


CLASSIFICATION OF IC ENGINE

<u>(1) According to the type of fuel used:</u>

a) Petrol engine

b) Diesel engine

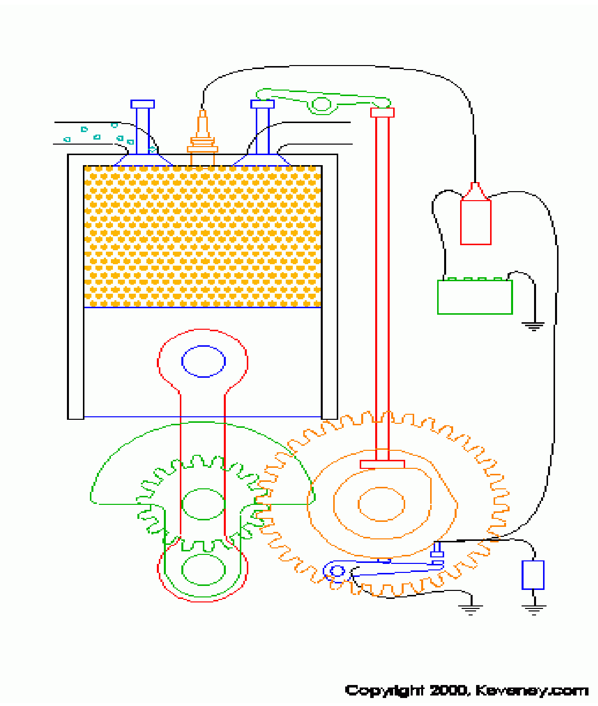
c) Gas engine

d) Bi-fuel (Bio-fuel) engine

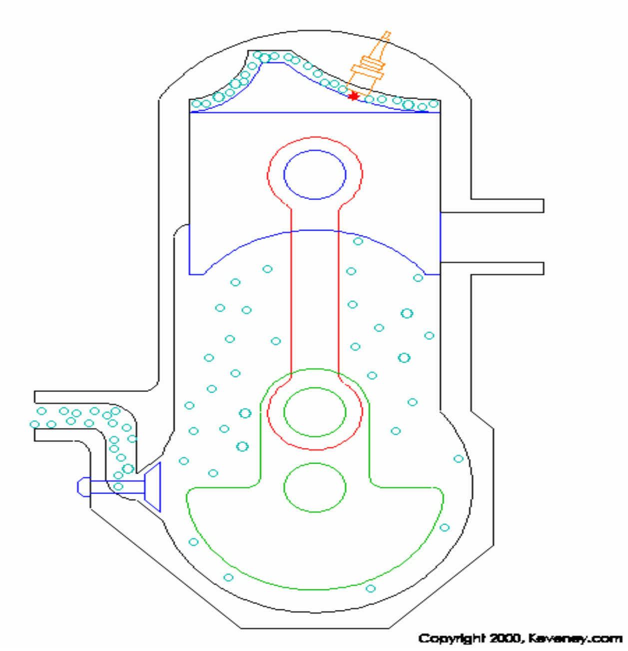
CLASSIFICATION OF IC ENGINE

(2) According to the number of strokes per cycle:

- 4-stroke engine



- 2-stroke engine



CLASSIFICATION OF IC ENGINE

(3) According to the method of ignition:

a) Spark Ignition (SI) engine

b) Compression Ignition (CI) engine

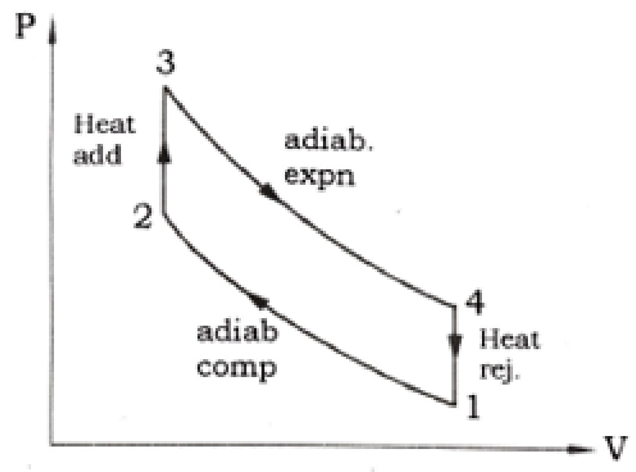
CLASSIFICATION OF IC ENGINE

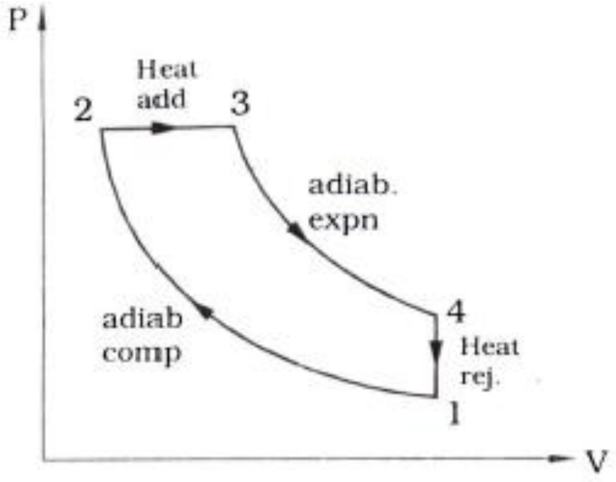
(4) According to the cycle of combustion:

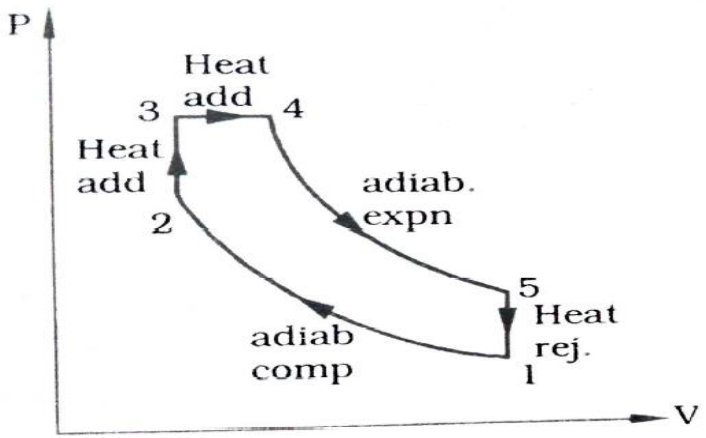
a) Otto cycle engine

b) Diesel cycle engine

c) Dual combustion cycle engine





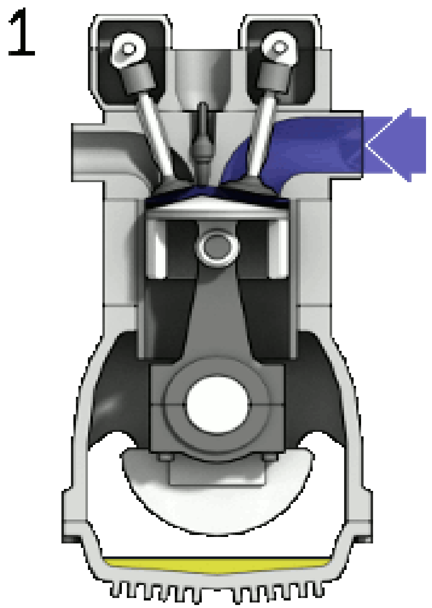


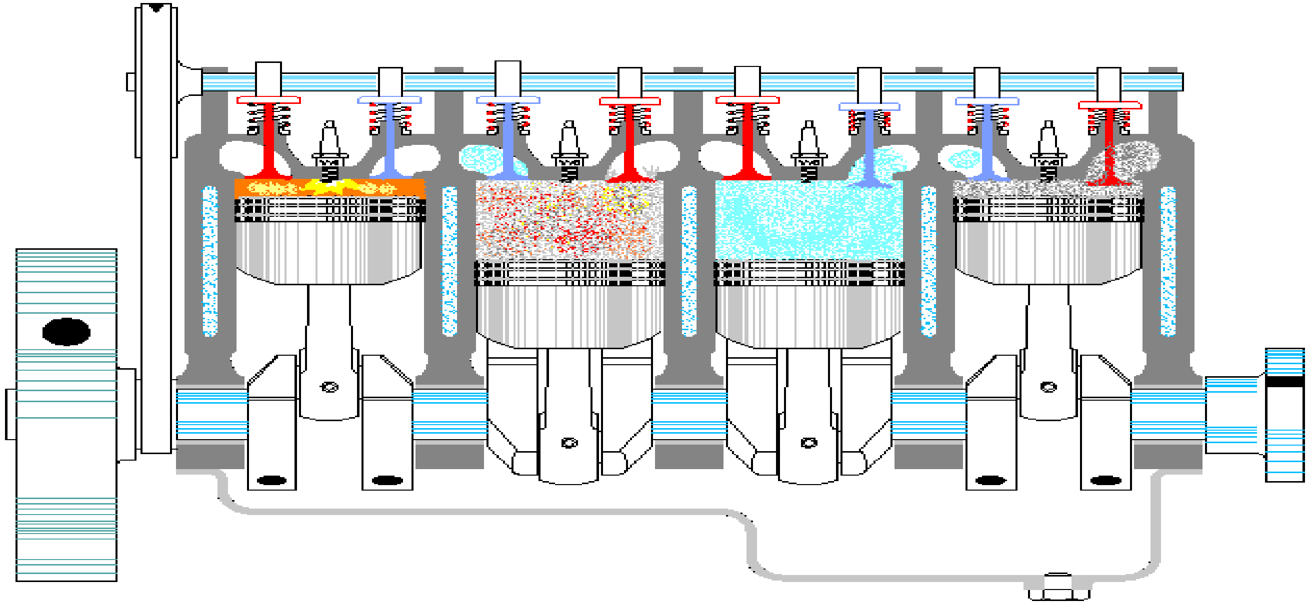
CLASSIFICATION OF IC ENGINE

(5) According to the number of cylinders used:

a) Single cylinder engine

b) Multi-cylinder engine

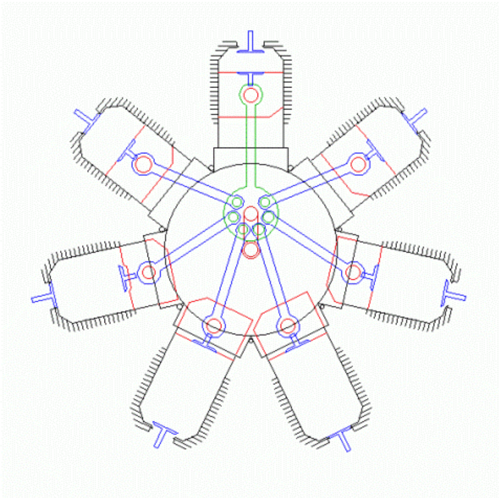




CLASSIFICATION OF IC ENGINE

(6) According to the arrangement of cylinders:

a) Vertical engine
b) Horizontal engine
c) Inline engine
d) Radial engine
e) V-engine
f) Opposed type engine



CLASSIFICATION OF IC ENGINE

(7) According to the method of cooling:

a) Air cooled engine

b) Water cooled engine

(8) According to their uses:

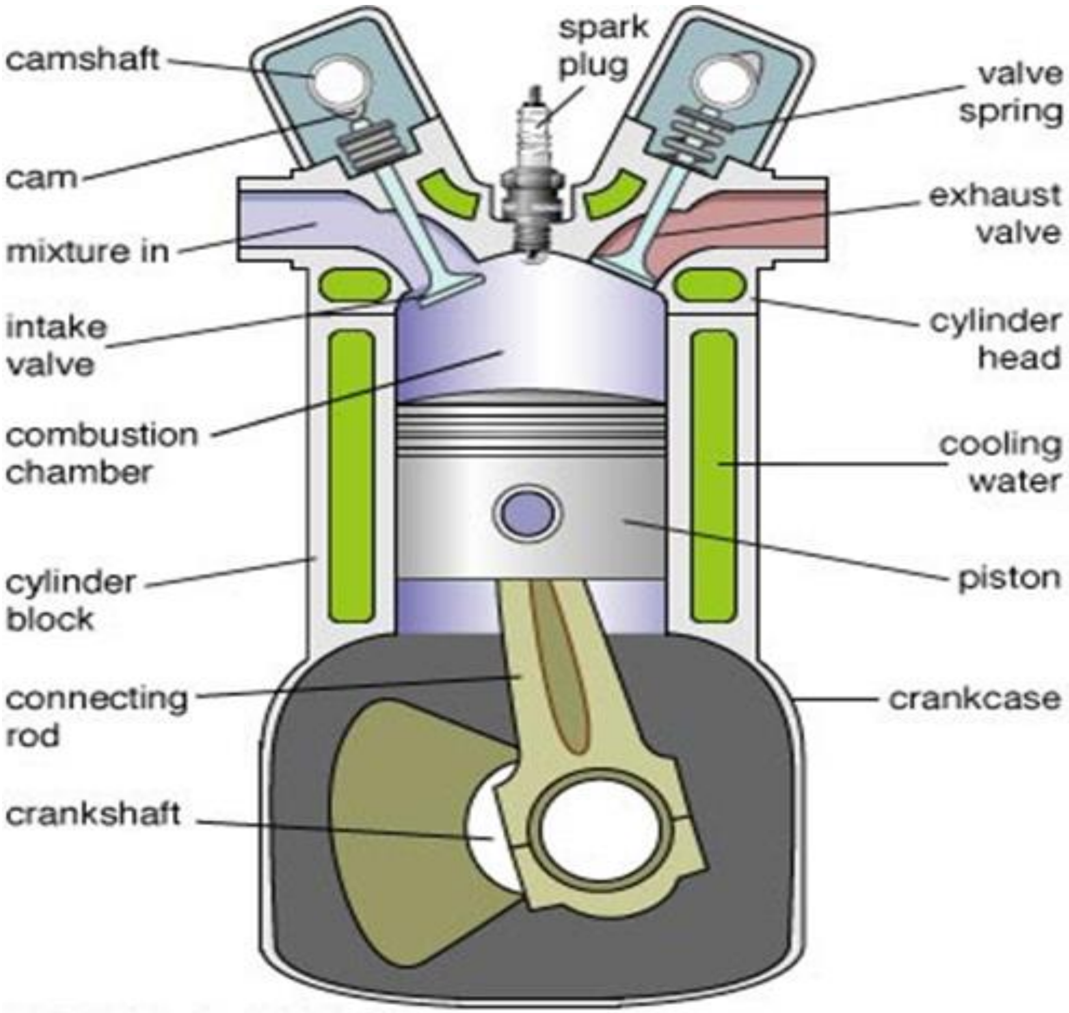
a) Stationary engine

b) Automobile engine

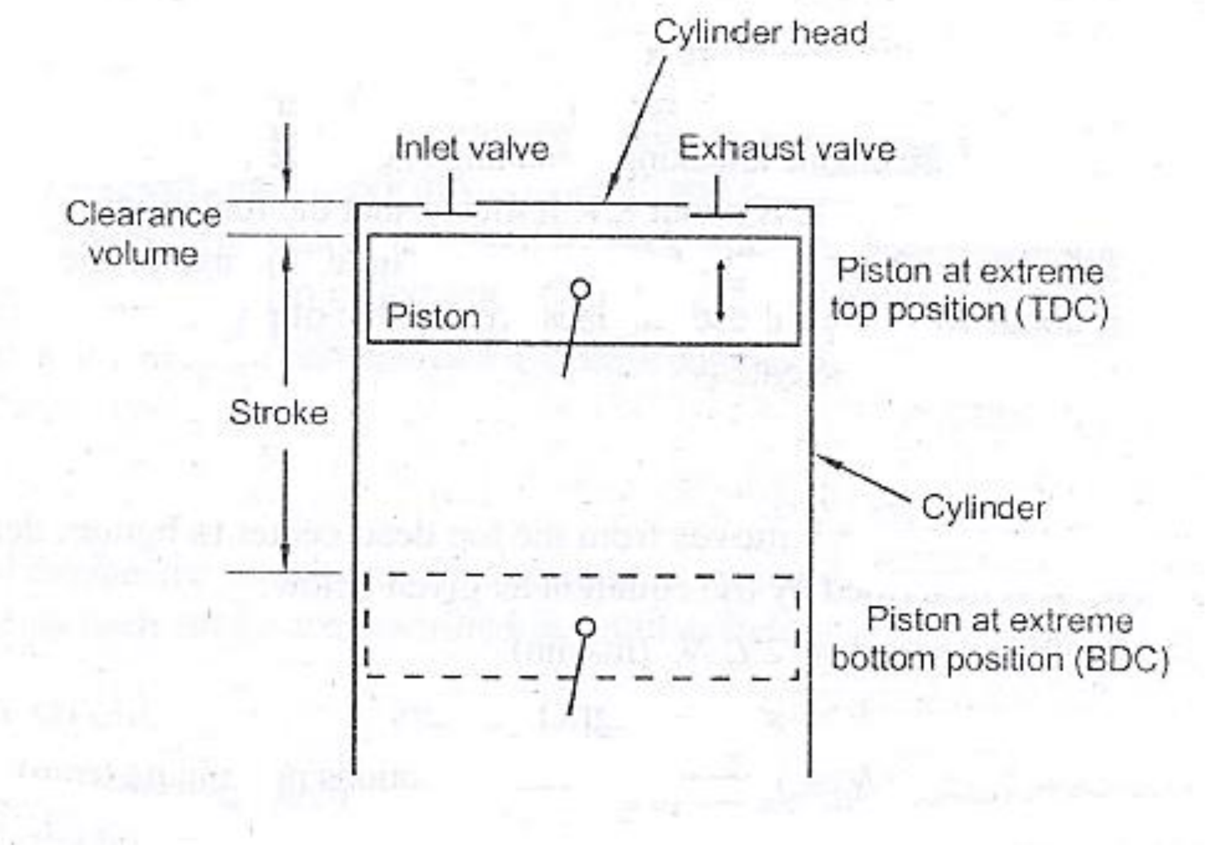
c) Marine engine

d) Aircraft engine, etc

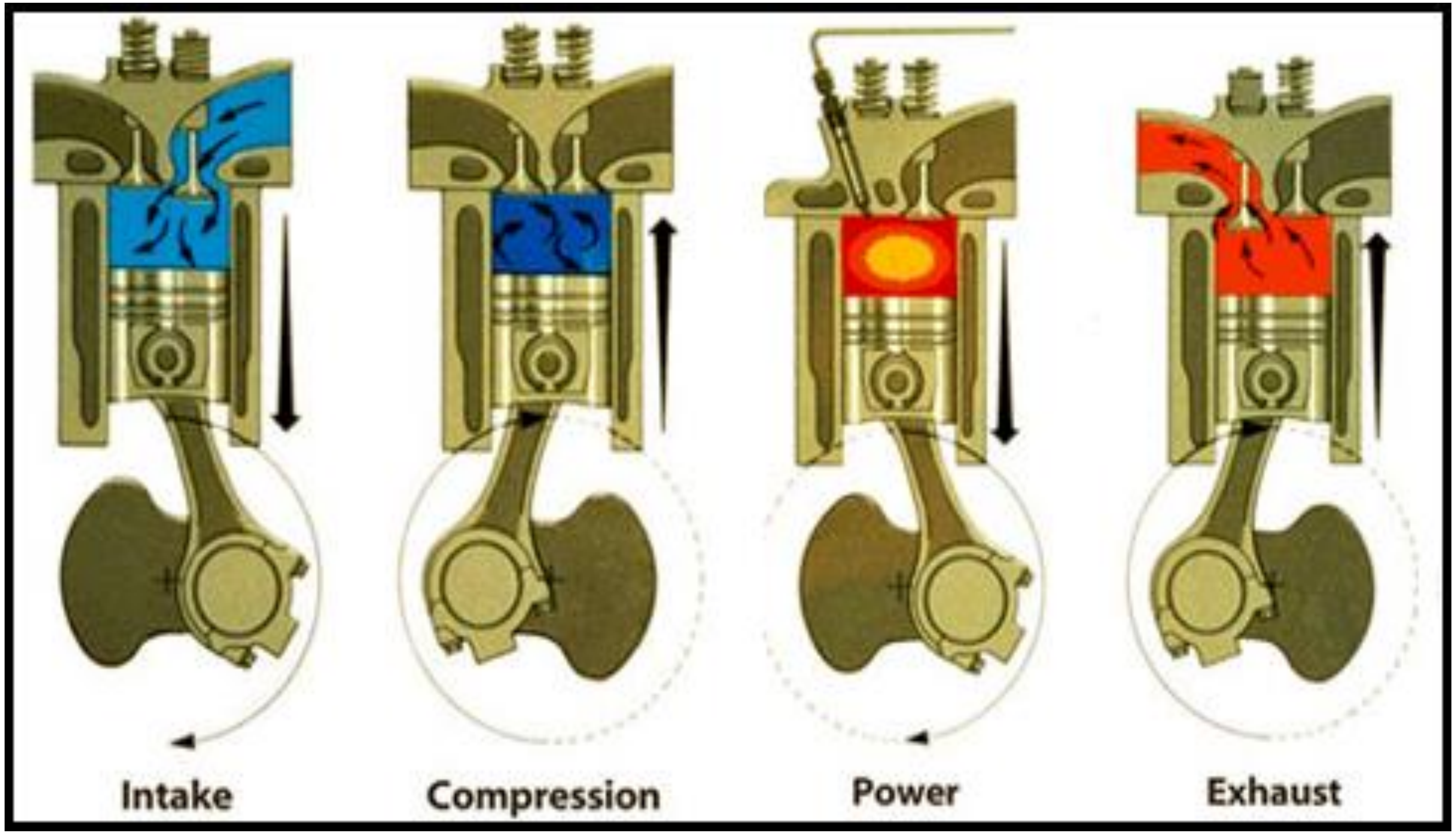
PARTS OF IC ENGINE

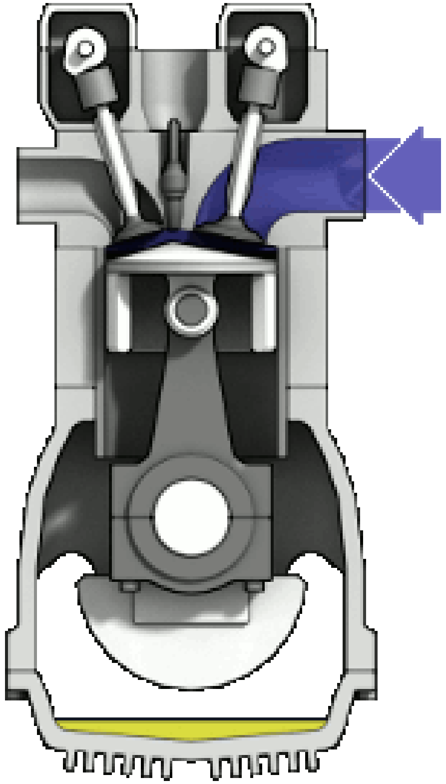


IC ENGINE TERMINOLOGY

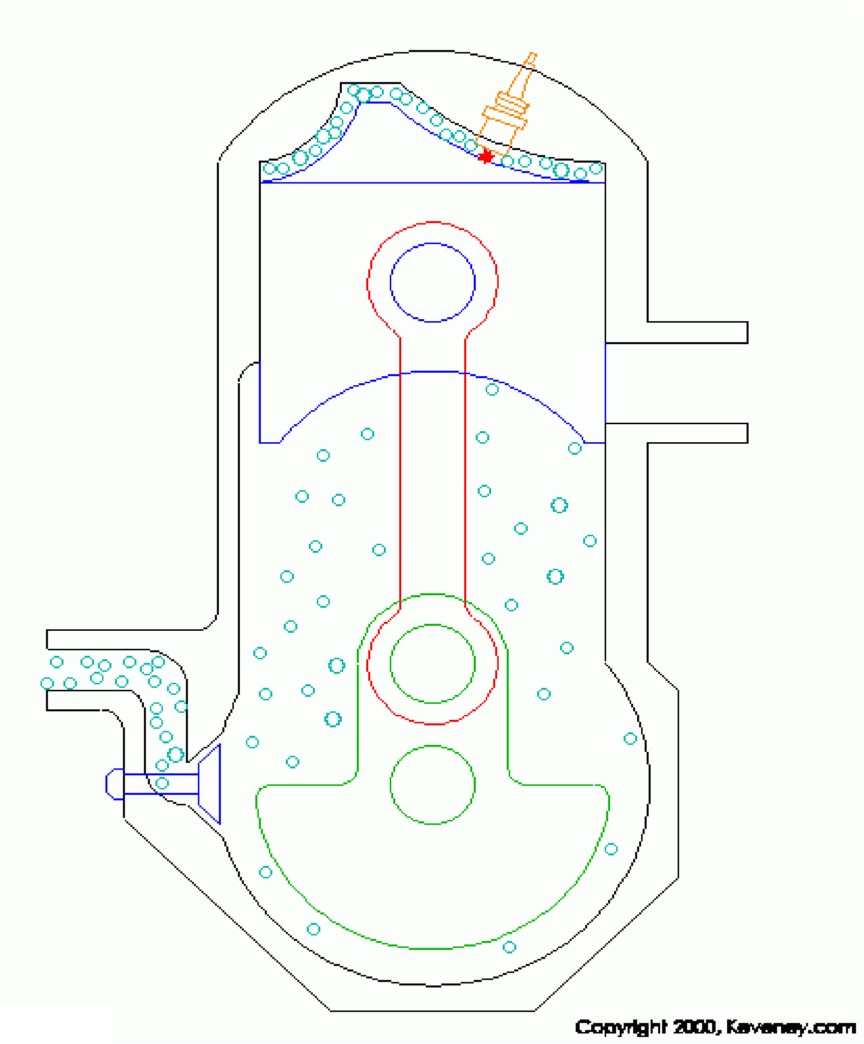


Four - Stroke (4-s) Engine





2-stroke engine



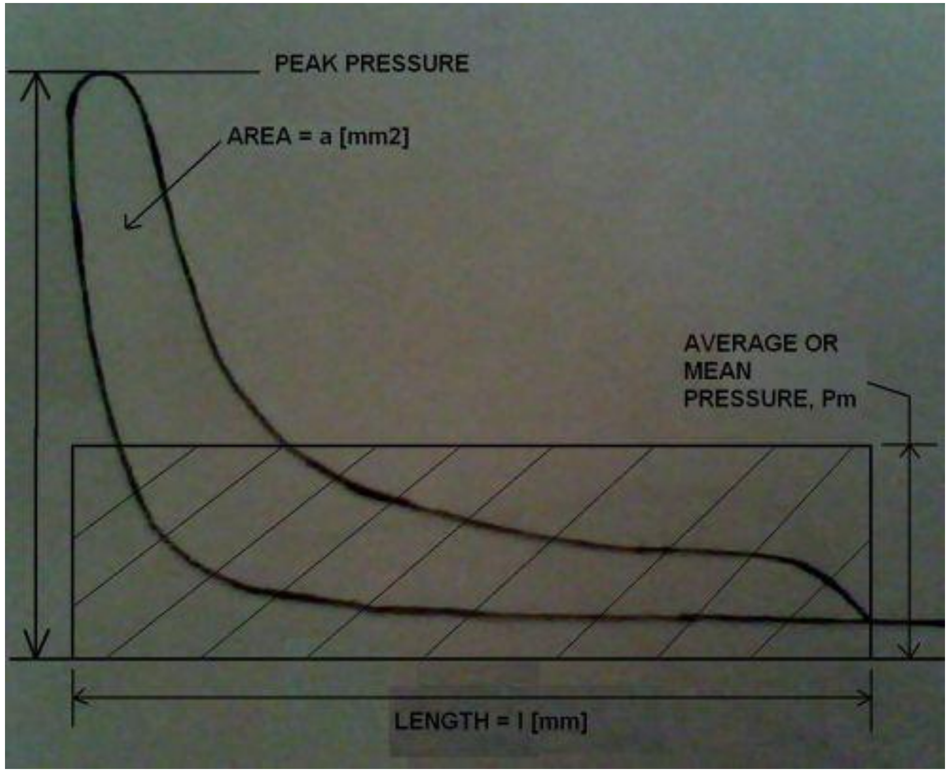
Performance Parameters

- **Mean Effective Pressure (MEP) - P_m**

It is expressed in Bar (1Bar = 10^5 N/m^2)

$$MEP = P_m = \frac{\left(\begin{array}{l}\text{spring value of the spring used}\\ \text{in the indicator } (S) \text{ in bar / m}\end{array}\right) \times \left(\begin{array}{l}\text{net area of the indicator}\\ \text{diagram } (a) \text{ in } m^2\end{array}\right)}{\text{length of the indicator diagram } (l) \text{ in m}}$$

$$P_m = \frac{Sa}{l} \text{ Bar}$$



Performance Parameters

Indicated Power (IP)

$$\text{Indicated Power} = \frac{p_m\, L\, A\, N}{60 \times 2 \times 1000} \text{ kW}$$

p_m = Mean Effective Pressure, N/m²

L = Length of Stroke, m

A = Area of Cross section of the Cylinder, sq m

N = RPM of the Crankshaft.

n = Number of cycles per minute.

Brake Power (BP)

BP is given by, $BP = \frac{2\pi NT}{60\times 1000} kW$

where N = speed of engine in *rpm*

T = Torque in *N-m*

Torque (*T*) is measured by using either *belt* or *rope brake dynamometer*.

Friction Power (FP)

$$FP = IP - BP \text{ kW}$$

Performance Parameters

Mechanical Efficiency

$$\eta_{mech} = (BP/IP) \times 100$$

Thermal Efficiency

$$\eta_{th} = \frac{power\ output}{heat\ supplied} \times 100$$

w.k.t. Heat supplied $= (m_f) \times CV$

where m_f = mass of fuel in kg/sec.

CV = calorific value of fuel in kJ/kg.

$$\text{Indicated thermal efficiency} = \eta_{ITH} = \frac{IP}{m_f \times CV} \times 100$$

$$\text{Brake thermal efficiency} = \eta_{BTH} = \frac{BP}{m_f \times CV} \times 100$$

Performance Parameters

Specific Fuel Consumption

$$\text{i.e., SFC} = \frac{m_f\,(kg/hr)}{Power\ developed\ (kW)}\ \ kg/kW\text{-}hr$$

where m_f = mass of fuel (kg/hr)

Power developed can be based on IP or BP.

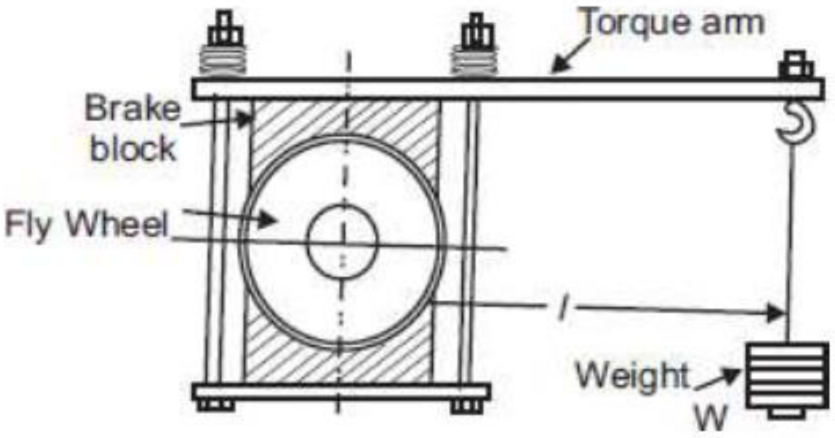
SFC based on IP is termed indicated specific fuel consumption (ISFC), and is given by the equation

$$\text{ISFC} = \frac{Fuel\ consumed\ in\ kg/hr}{IP\ in\ kW}\ \ kg/kWhr$$

While SFC based on BP is termed brake specific fuel consumption (BSFC), and is given by the equation

$$\text{BSFC} = \frac{Fuel\ consumed\ in\ kg/hr}{BP\ in\ kW}\ \ kg/kWhr$$

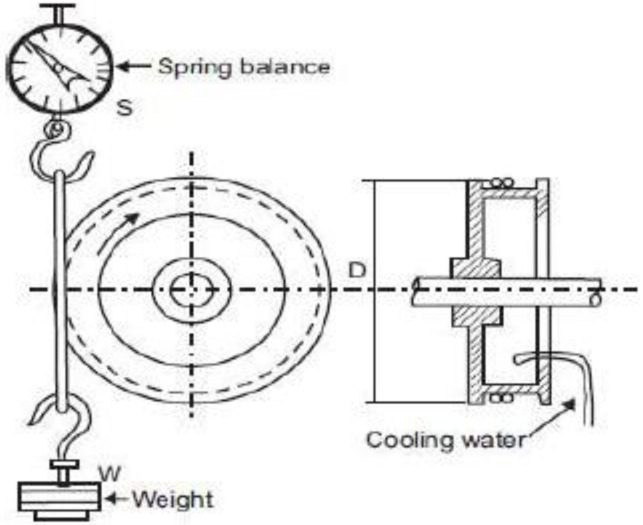
Measurement of brake power



(B) Rope brake dynamometer:

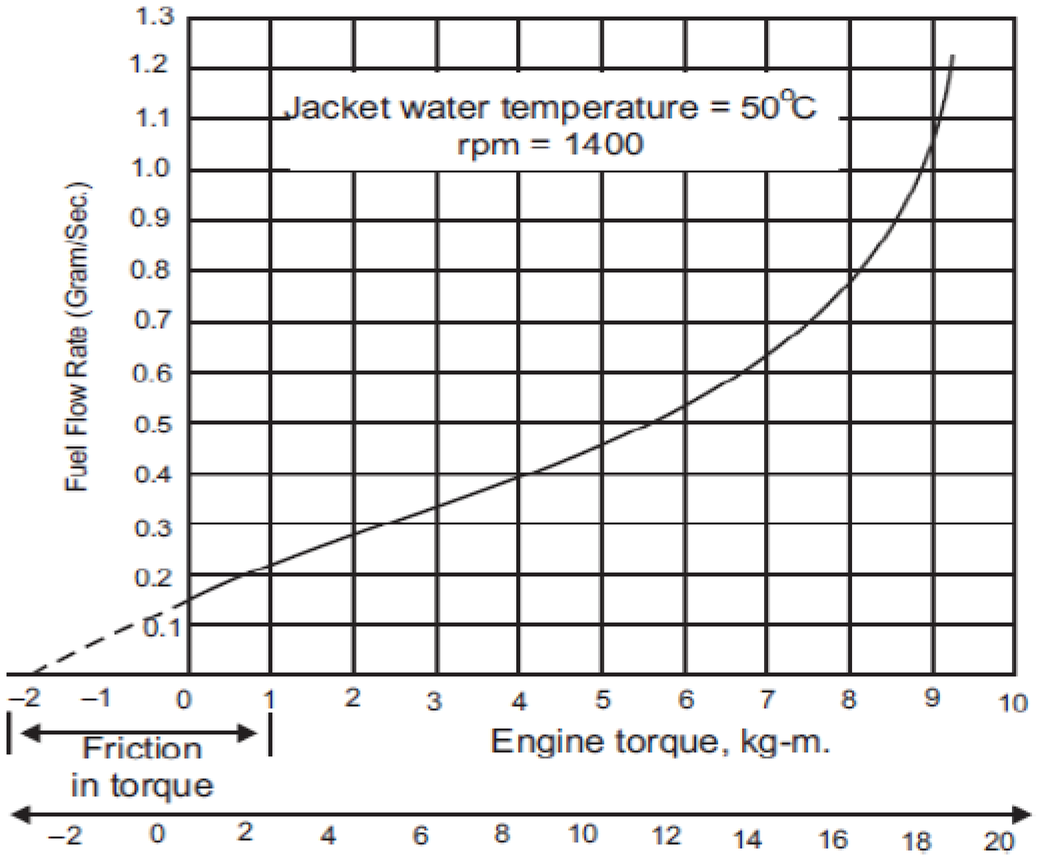
(A) Prony brake dynamometer:

$$BP = \frac{2\pi NT}{60 \times 1000} \text{ kW}$$



Measurement of Friction power

- *Willan's line method.*
- *•Morse test.*
- *•Motoring test.*



Morse test.

- The Morse test is applicable only to multicylinder engines.

- With all cylinder firing = BP kW
- Cut off at 1st cylinder = BP1 kW
- Cut off at 2nd cylinder = BP2 kW
- Cut off at 3rd cylinder = BP3 Kw

IP1 = BP-BP1, IP3 = BP-BP3

$$IP = IP_1 + IP_2 + IP_3$$

Heat Balance sheet

- The heat balance sheet from the above data can be drawn as follows:

Particulars	*kJ/s or kJ/min or kJ/hr*	*Percent*
a) Heat supplied by fuel	----	------
b) Heat absorbed in B.P. **c) Heat taken away by cooling water** **d) Heat carried away by the exhaust gases** **e) Heat unaccounted for (a-(b+c+d))**	---- ---- ---- ----	------ ------ ------ ------
Total	----	------

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Module-1:
RECIPROCATING COMPRESSORS

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CONTENTS

- Introduction
- Classification
- Working of single stage Reciprocating compressor
- Workdone in single stage compressor
- Efficiencies of a compressor
- Double acting Reciprocating compressor
- Limitations of single stage compressor
- Multistage compressor
- Advantages of Multistage compressor

Introduction

- Compressors are work absorbing devices which are used for increasing pressure of fluid at the expense of work done on fluid.
- The compressors used for compressing air are called air compressors.
- Compressors are invariably used for all applications requiring high pressure air.

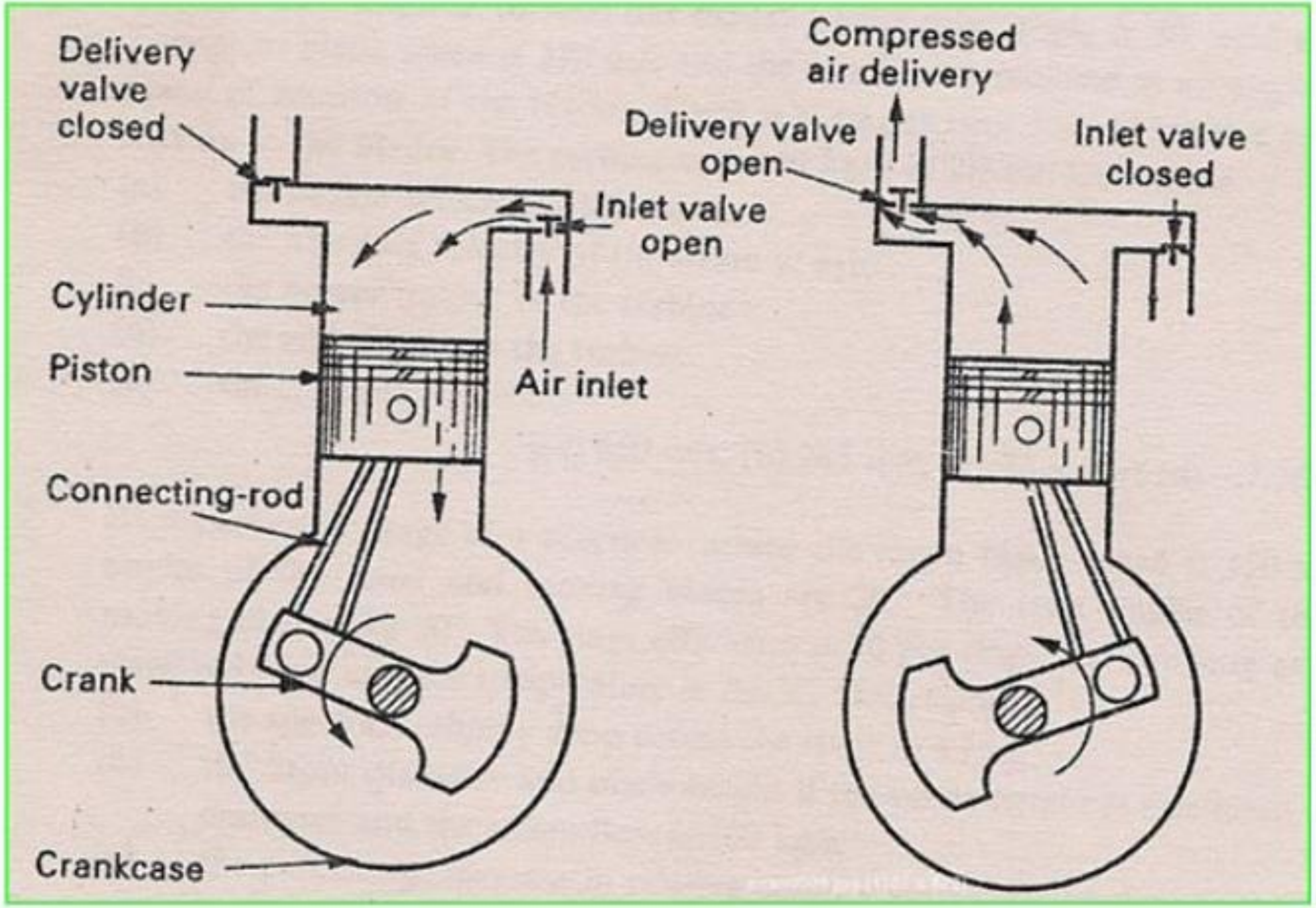
Applications:

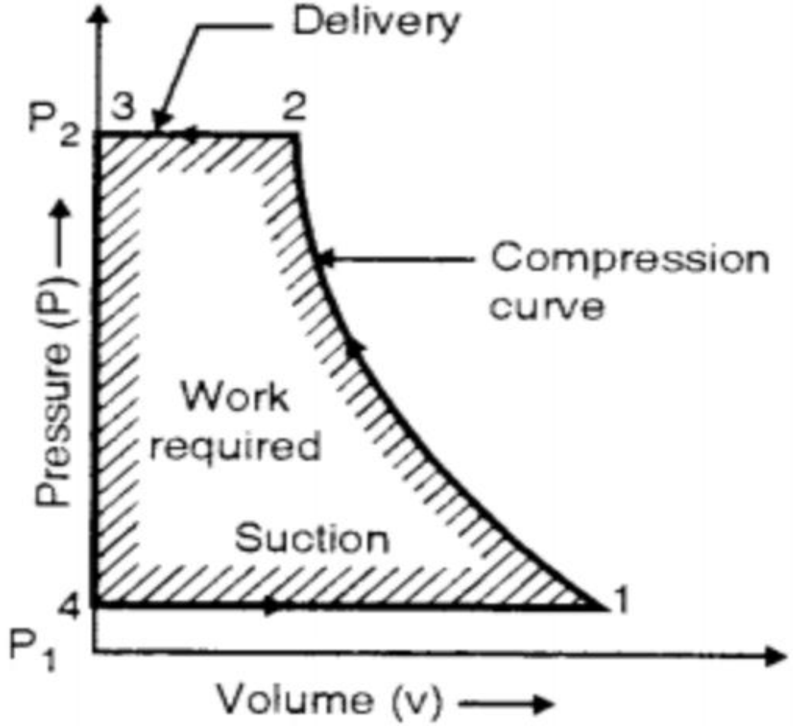
- Pneumatic tools
- Spray painting
- Compressed air engine
- Supercharging in internal combustion engines
- Material handling (for transfer of material)
- Surface cleaning
- Refrigeration and air conditioning
- chemical industry
- Aircraft Industry etc.

Classification of compressor

- Based on principle of operation
  - ➢ Reciprocating type compressors
  - ➢ Rotary type compressors
- Based on number of stages:
  - ➢ Single stage compressor
  - ➢ Multistage compressor
- Based on capacity of compressors:
  - ➢ Low capacity compressors
  - ➢ Medium capacity compressors
  - ➢ High capacity compressors
- Based on highest pressure developed:
  - ➢ Low pressure compressor,
  - ➢ Medium pressure compressor,
  - ➢ High pressure compressor

Working of single stage Reciprocating compressor





Workdone in single stage compressor (without clearance volume)

Workdone/cycle,

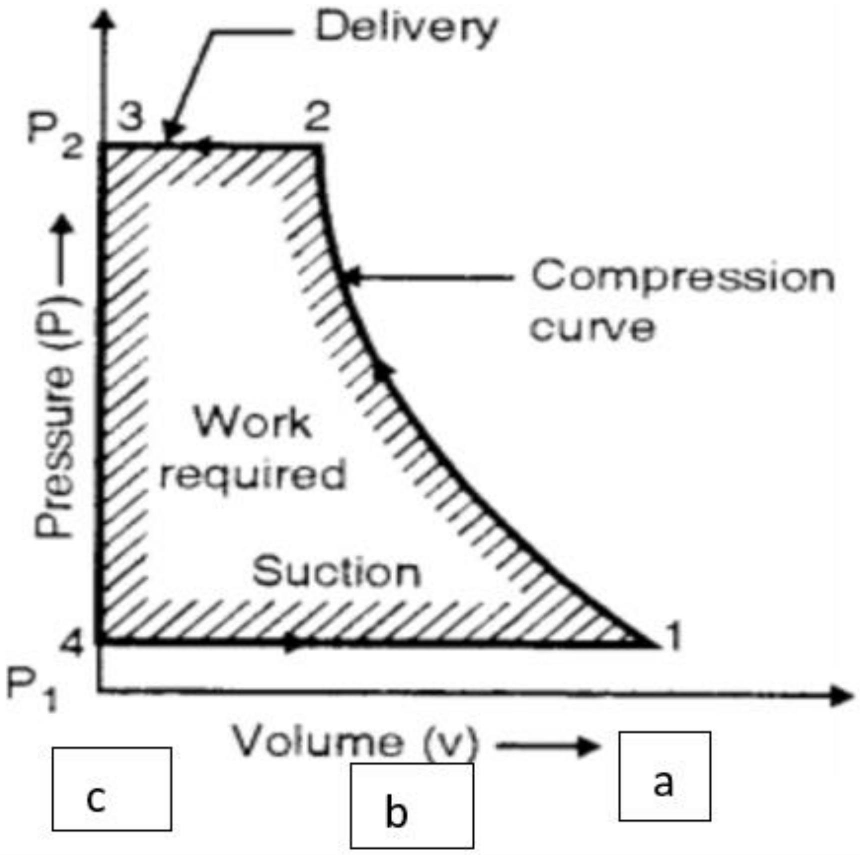
W c = Area on p-V diagram (1-2-3-4-1)

= area (c-3-2-b-c)+area(b-2-1-a-b)-area(c-4-1-a-c)

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

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

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



Workdone in single stage compressor (contd.)

For Polytropic process

$$p_1V_1^n = p_2V_2^n$$

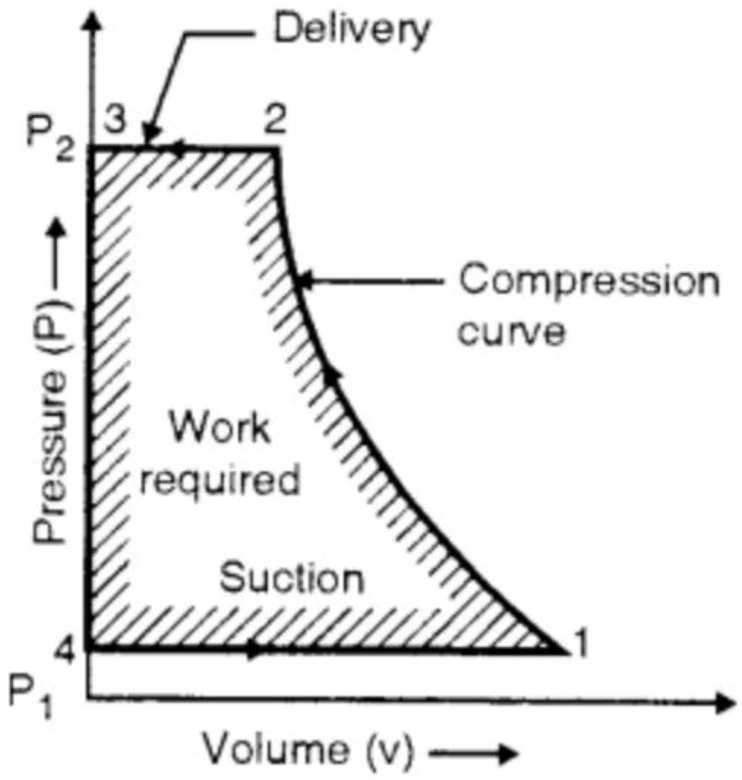
$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

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

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

$$w_c = \left(\frac{n}{n-1}\right) mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

$$w_c = \left(\frac{n}{n-1}\right) mR[T_2 - T_1]$$



Workdone for isothermal and adiabatic compression

In case of compressor having isothermal compression process, n = 1, i.e. $P_1 V_1 = P_1 V_2$

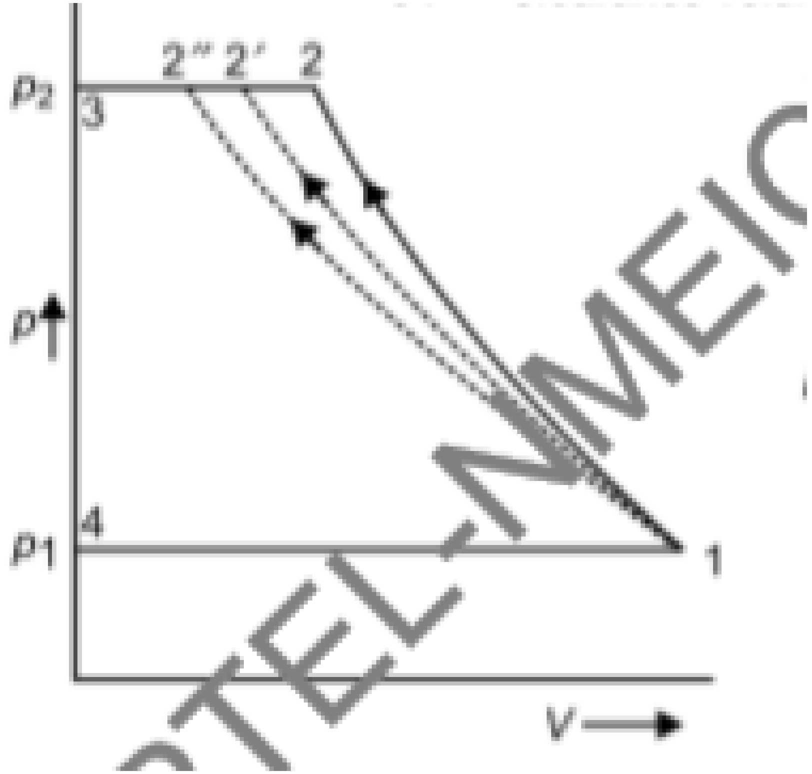
$$W_{c,\,iso} = P_2 V_2 + P_1 V_1 \ln r - P_1 V_1$$

$$W_{c,\,iso} = P_1 V_1 \ln r, \quad \text{where } r = \frac{V_1}{V_2}$$

1 – 2 = Adiabatic process

1 – 2' = Polytropic process,

1 – 2'' = Isothermal process



In case of compressor having adiabatic compression process, n = γ

$$W_{c\,adiabatic} = \left(\frac{\gamma}{\gamma-1}\right) mR[T_2 - T_1]$$

Formulae:

For Polytropic process

$$p_1V_1^n = p_2V_2^n$$

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

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

$$W = \left(\frac{n}{n-1}\right)mRT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W = \left(\frac{n}{n-1}\right)p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Indicated work / cycle $W = \left(\frac{n}{n-1}\right)mR(T_2 - T_1)$

$$IP = \left(\frac{n}{n-1}\right)\dot{m}R(T_2 - T_1)$$

Assuming the air as a perfect gas,

$$p_1V_1 = mRT_1 \qquad p_2V_2 = mRT_2$$

Volumetric efficiency of compressor

It is the ratio between FAD at standard atmospheric condition in one delivery stroke (Actual air intake) to the swept volume (theoretical air intake)

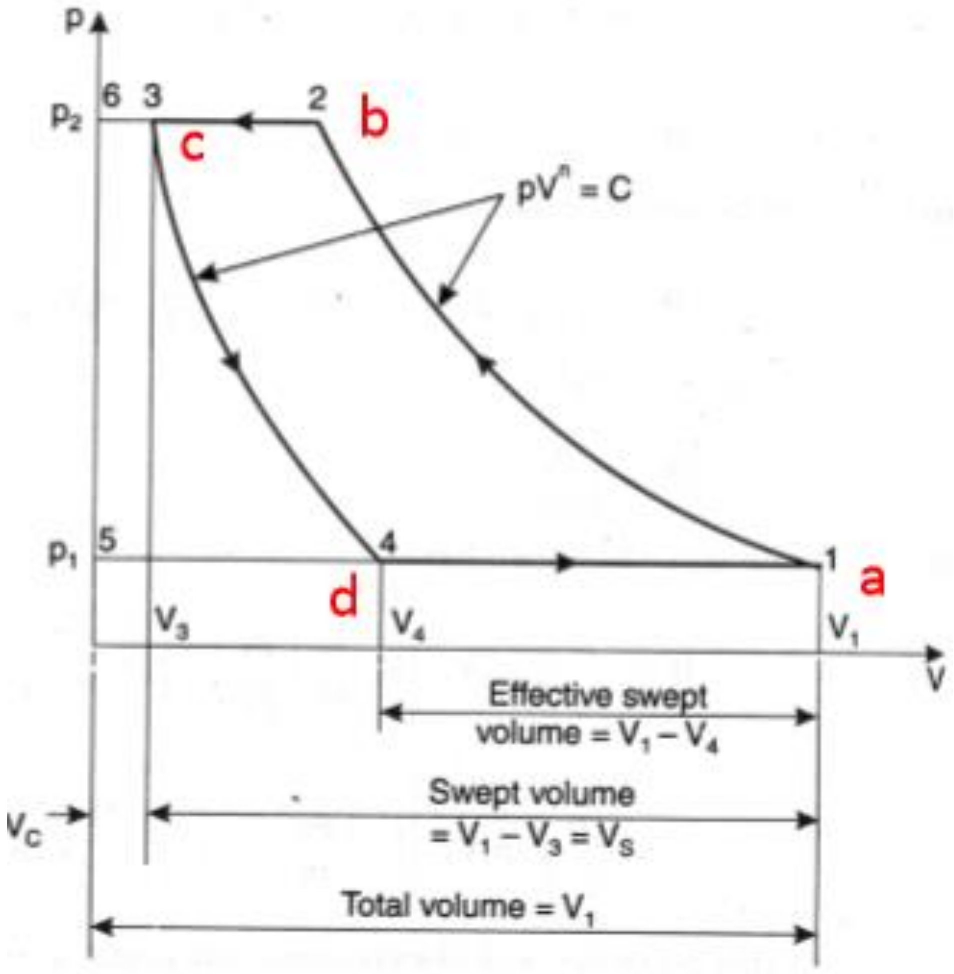
$$\eta_v = \frac{FAD}{V_s}$$

$$V_s = \frac{\pi}{4} D^2 L.$$

Volumetric efficiency= $\frac{V_1 - V_4}{V_1 - V_3}$

$$\text{Volumetric efficiency } (\eta_v) = 1+k-k\left(\frac{P^2}{P_1}\right)^{1/n}$$

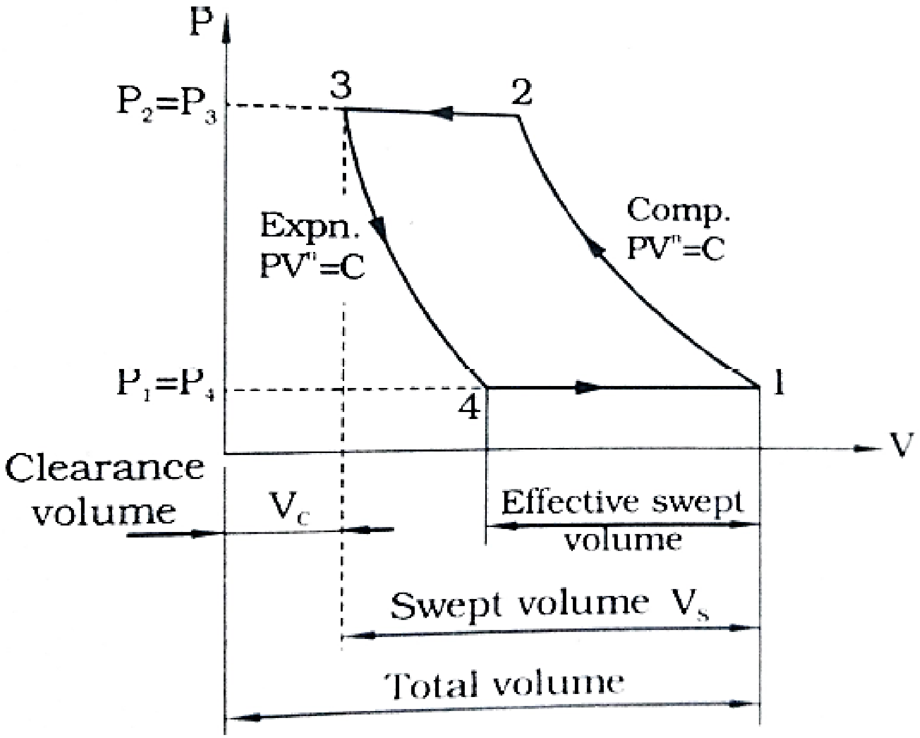
$$k = \text{Clearance Raito} = \frac{V_c}{V_s}$$



Volumetric efficiency of compressor

Volumetric efficiency $\eta_{vol} = \dfrac{\text{Actual free air delivered or effective swept volume}}{\text{Displacement or swept volume}}$

$$\eta_{vol} = \frac{V_1 - V_4}{V_1 - V_3} \quad \text{-----(1)}$$



Adding and subtracting V_3 in the numerator of equation (1), we have

$$\eta_{vol} = \frac{V_1 - V_4 + V_3 - V_3}{V_1 - V_3} = \frac{(V_1 - V_3) + (V_3 - V_4)}{V_1 - V_3} = 1 + \frac{(V_3 - V_4)}{V_1 - V_3}$$

Volumetric efficiency of compressor

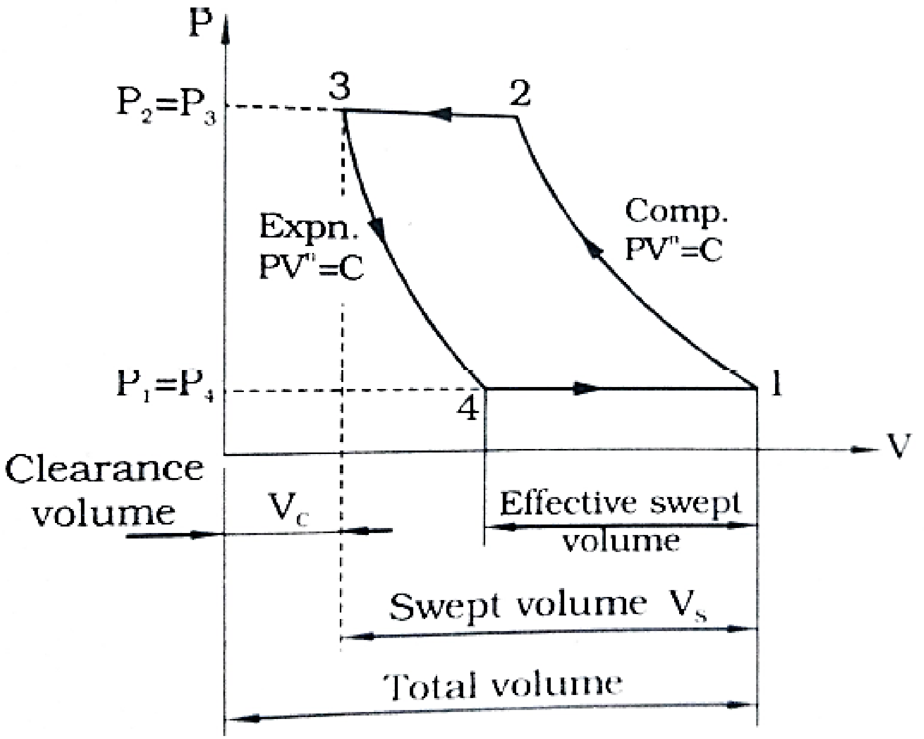
$$\eta_{vol} = 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3}$$

w.k.t. clearance ratio (C or k) $= \dfrac{\text{Clearance volume}(V_C)}{\text{Swept volume}(V_S)}$

From p-v diagram $V_C = V_3$ and $V_S = V_1 - V_3$

$$\therefore \quad C = \frac{V_3}{V_1 - V_3}$$

$$\eta_{vol} = 1 + C - \frac{V_4}{V_1 - V_3}$$



Volumetric efficiency of compressor

For expansion process 3-4, following the law $PV^n = C$,

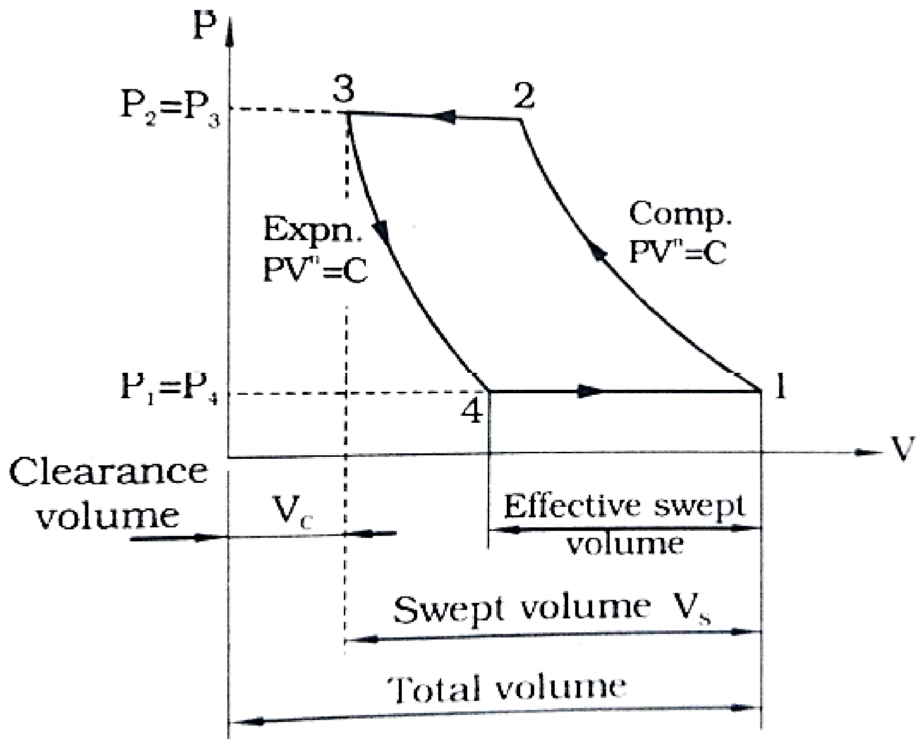
$$\frac{V_4}{V_3} = \left(\frac{P_3}{P_4}\right)^{\frac{1}{n}} \qquad \therefore\ V_4 = V_3\left(\frac{P_3}{P_4}\right)^{\frac{1}{n}}$$

we get $\eta_{vol} = 1 + C - \frac{1}{V_1 - V_3} V_3 \left(\frac{P_3}{P_4}\right)^{\frac{1}{n}}$

$$= 1 + C - C\left(\frac{P_3}{P_4}\right)^{\frac{1}{n}}$$

But $P_3 = P_2$ and $P_1 = P_4$

$$\therefore\ \eta_{vol} = 1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

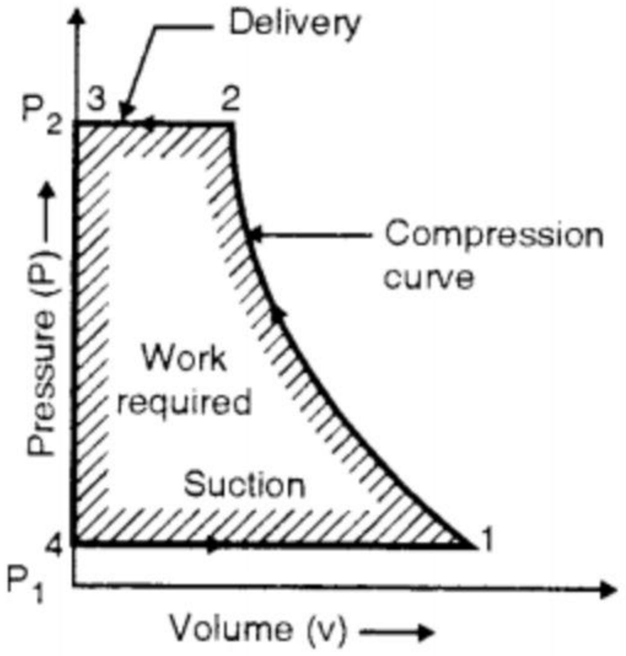


PROBLEMS ON SINGLE STAGE RECIPROCATING COMPRESSOR

1. A single stage reciprocating compressor takes 1 m^3 of air per minute at 1.013 bar and 15^OC and delivers it at 7 bar. Assuming that the law of compression is $Pv^{1.35}$=constant and the clearance is negligible, calculate the indicated power.

- **Data given:** Volume of air taken in, $V_1 = 1\ m^3/min$
- Intake pressure, $P_1 = 1.013$ bar
- Initial temperature, $T_1 = 15 + 273 = 288$ K
- Delivery pressure, $P_2 = 7$ bar
- Law of compression: $PV^{1.35}=C$

Indicated power I.P=?



We have IP $= \frac{n}{n-1} mR(T_2 - T_1) kJ/\min$

Mass of air delivered per min.,

$$m = \frac{p_1 v_1}{RT_1} = \frac{1.013 \times 10^5 \times 1}{287 \times 288} = 1.266 kg/\min$$

Problem (1) contd.

-

$$\text{Delivery temperature, } T_2 = T_1 \left(\frac{p2}{p1}\right)^{(n-1/n)}$$

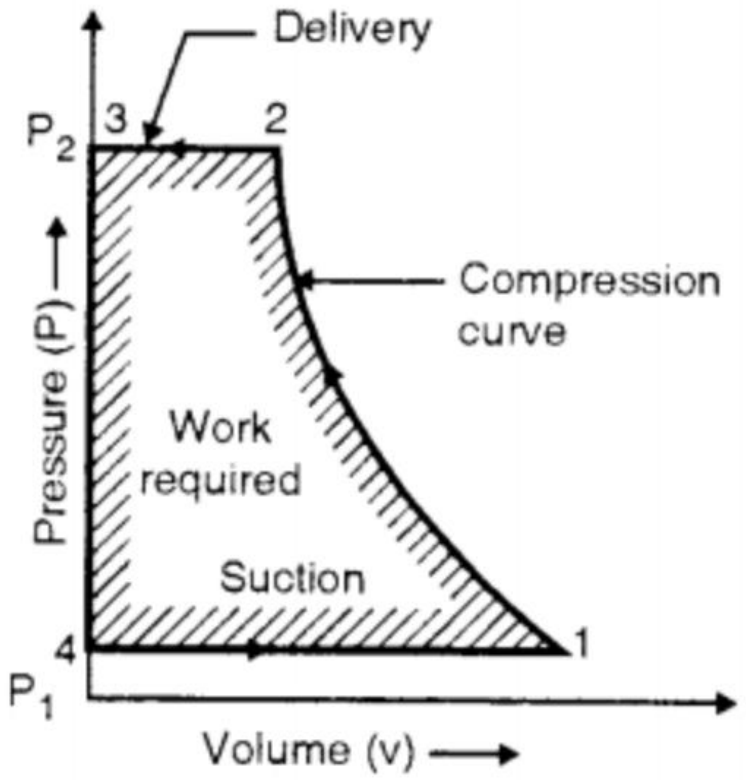
$$= 288\left(\frac{7}{1.013}\right)^{(1.35-1)/1.35}$$

$$= 475.2K$$

$$\text{IP } = \frac{n}{n-1} mR(T_2 - T_1) kJ/\text{min}$$

$$= \frac{1.35}{1.35-1} \times 1.226 \times 0.287(475.2 - 288$$

$$\text{Indicated power I.P} = \frac{254}{60} = 4.23kW.$$



2. A single acting reciprocating air compressor has cylinder diameter and stroke of 200 mm and 300 mm respectively. The compressor sucks air at 1 bar and 27^{O}C and delivers at 8 bar while running at 100 r.p.m Find: 1) Indicated power of the compressor; 2.) Mass of air delivered by the compressor per minute; and 3) temperature of the delivered by the compressor. The compression follows the law $Pv^{1.25} = C$, Take R as 287 J/kg-K.

- **Given:**

D=200 mm = 0.2m, L = 300 mm = 0.3m

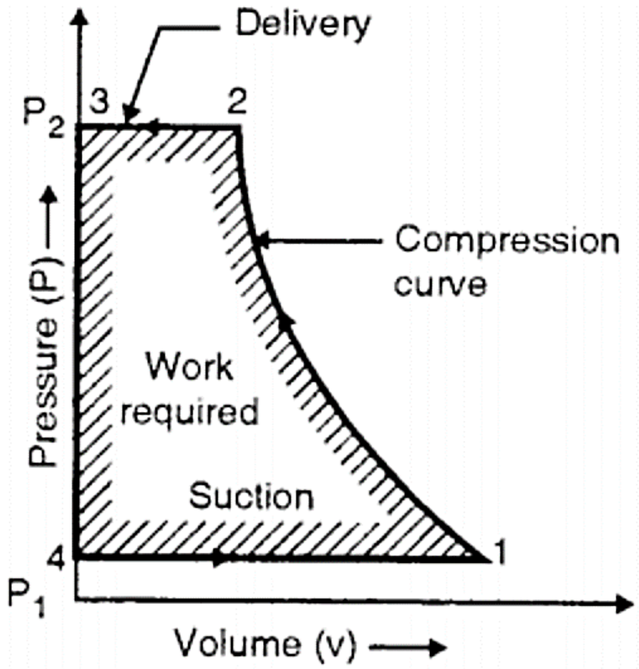
P_1= 1 bar = 1 x 10^5 N/m^2; T_1 = 27^{O}C = 27 + 273 = 300K;

P_2 = 8 bar, N = 100 r.p.m, n=1.25, R= 287 J/kg K

1) Indicated power of the compressor:

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.25}{1.25-1} \times 1 \times 10^5 \times 0.0094 \left[\left(\frac{8}{1} \right)^{\frac{1.25}{1.25-1}} - 1 \right] N-m \quad = 2425 \text{ N-m}$$

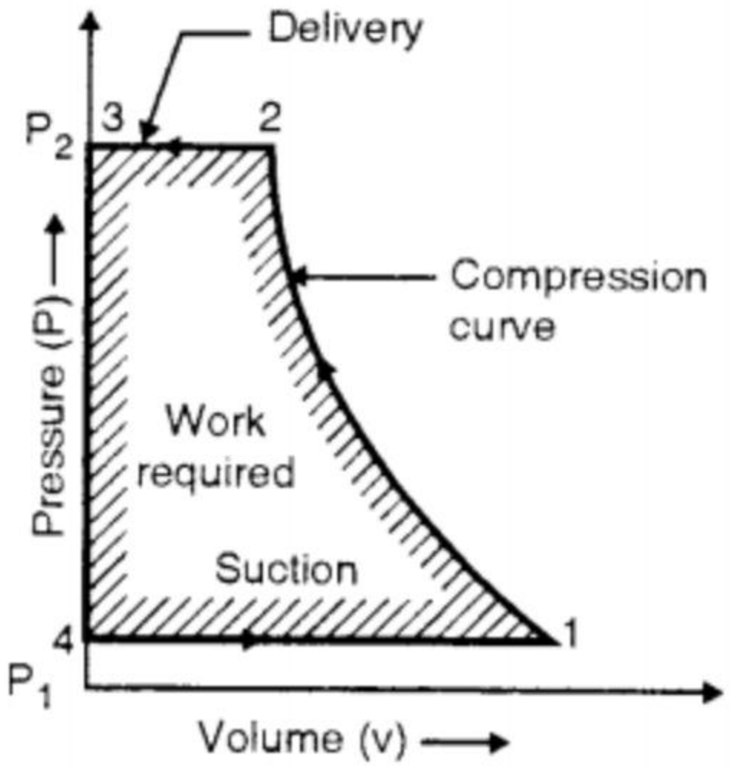


Problem (2) contd.

Since the compressor is single acting, therefore number of working strokes per minute, Nw = N = 100

Indicated power of the compressor

$$= \frac{W \times N_w}{60} = \frac{2425 \times 100}{60} = \mathbf{4042kW}$$



2) Mass of air delivered by the compressor per minute

- Let m= Mass of air delivered by the compressor per stroke.
- We know that $P_1V_1 = mRT_1$

$$m = \frac{P_1V_1}{RT_1} = \frac{1 \times 10^5 \times 0.0094}{287 \times 300} = 0.0109\,kg \text{ per stroke}$$

Mass of air delivered minute = mx N = 0.0109x 100 **= 1.09 kg**

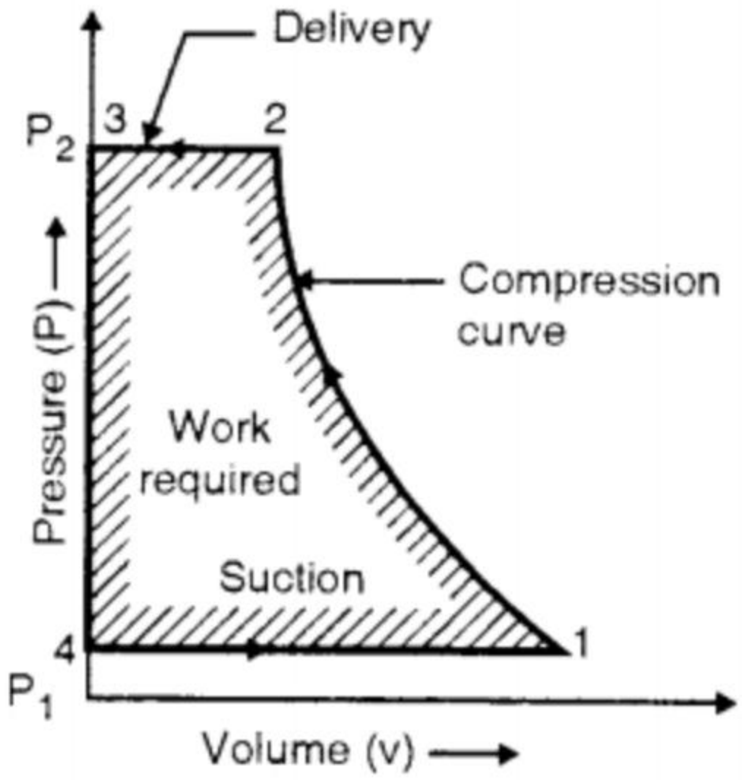
Problem (2) contd.

3) Temperature of air delivery by the compressor:

- Let T_2= Temperature of air delivered by the compressor
- We know that

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\frac{8}{1}\right)^{\frac{1.25-1}{1.25}} = 1.516$$

T_2 = $1.516 x T_1 = 1.516 x 300$ **= 454.8K**



3. An air compressor cylinder has 150mm bore and 150mm stroke and the clearance is 15%. It operates between 1 bar, 27^{O}C and 5 bar. Take polytrophic exponent n=1.3 for compression and expansion processes.

Find i) Cylinder volume at the various salient points of in cycle.

ii.) Flow rate in m3/min at 720 rpm and .

iii. The volumetric efficiency.

- **Given**

$D = 150\times10^{-3}$m　　$P_2 = 5\times10^5$ N/m^2

$L = 150\times10^{-3}$ m　　$T_1 = 27+273 = 300$K

$V_c = 0.15V_s$　　$N = 720$rpm

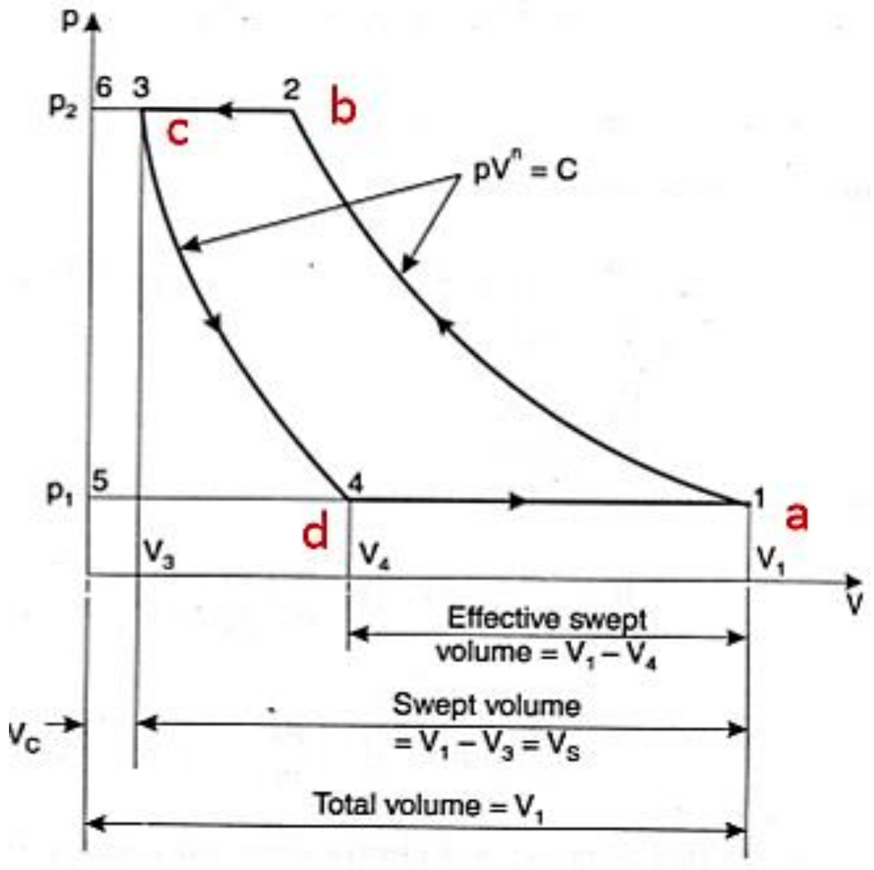
$P_1 = 1\times10^5$ N/m^2　　$pv^n = C$ n=1.3

Find

i. V_1, V_2, V_3, V_4

ii. FAD (V_a)

iii. η_v



$$V_1 = V_c + V_s$$

Problem (3) contd.

$$V_s = \frac{\pi}{4} D^2 L.N = \frac{\pi}{4}(0.15)^2 \times .0.15 \times 720 = 1.9085 m^3 / \text{min}$$

$$V_c = 0.15\ V_s$$

$$= 0.15 \times 1.9085$$

$$V_c = 0.2862\ m^3/\text{min}$$

$$V_1 = V_c + V_s$$

$$= 0.2862 + 1.9085$$

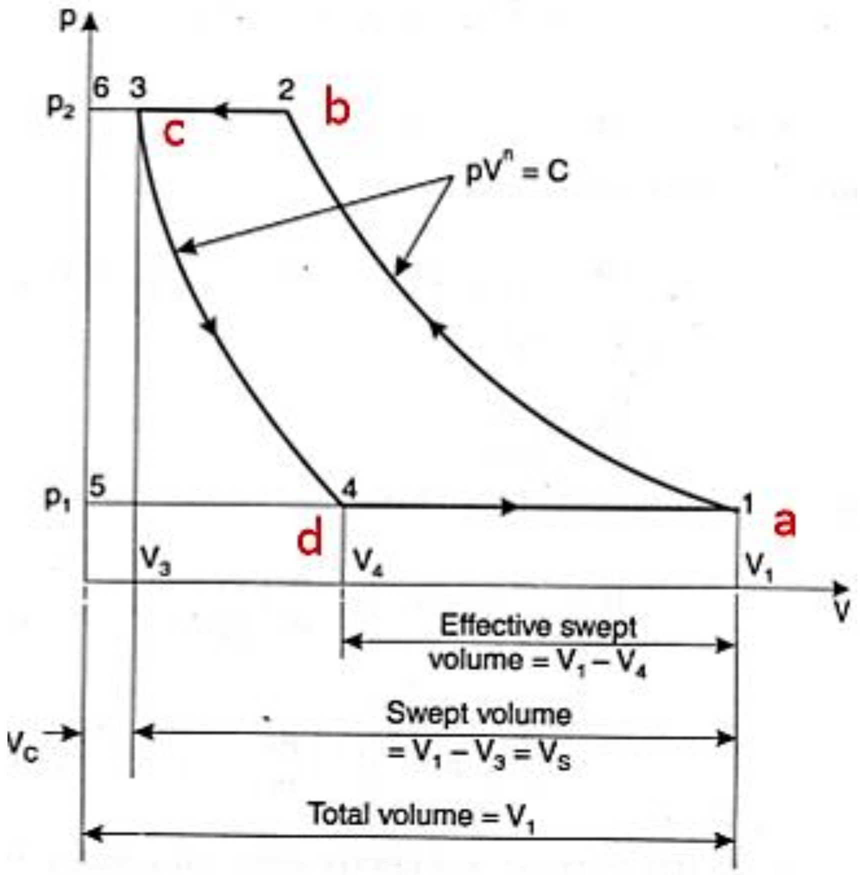
$$\boxed{V_1 = 2.1948}\ m^3/\text{min}$$

$$P_1 V_1^n = P_2 V_2^n$$

$$V_2 = V_1 \left(\frac{P_1}{P_2}\right)^{1/n}$$

$$= 2.1948 \left(\frac{1 \times 10^5}{5 \times 10_5}\right)^{1/1.3}$$

$$\boxed{V_2 = 0.6366\ m^3/\text{min}}$$



Problem (3) contd.

$\boxed{V_3 = 0.2862\ m^3/min}$ $= V_c$

$$P_3V_3^n = P_4V_4^n$$

$$V_4 = V_3\left(\frac{P_3}{P_4}\right)^{1/n}$$

WKT

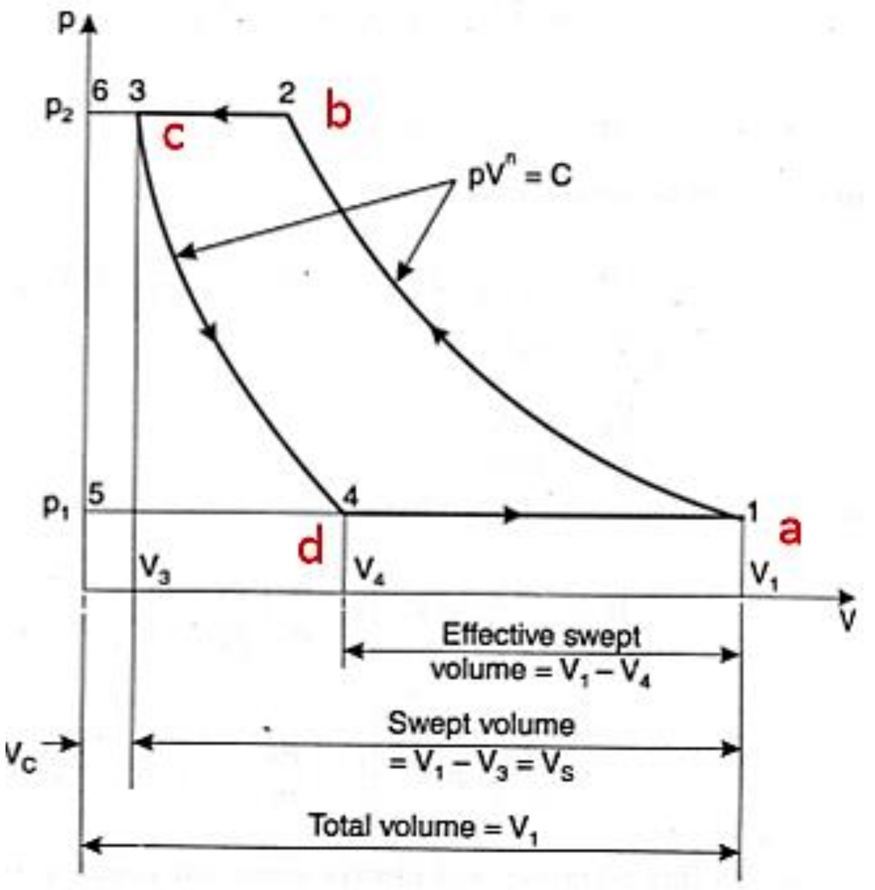
$$P_2 = P_3$$

$$P_1 = P_4$$

$$\therefore V_4 = V_3\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$= 0.2862\left[\frac{5\times10^5}{1\times10^5}\right]^{1/1.3}$$

$\boxed{V_4 = 0.98674\ m^3/min}$



Problem (3) contd.

$$\therefore \text{Volumetric efficiency } (\eta_v) = 1 + k - k\left(\frac{P^2}{P_1}\right)^{1/n}$$

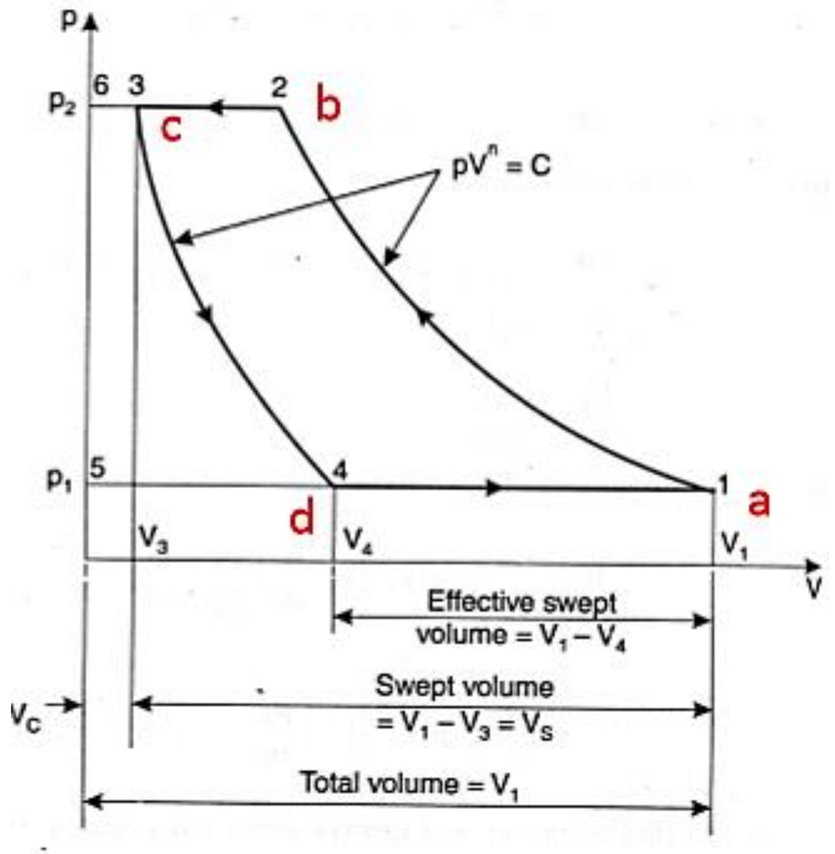
$$k = \text{Clearance Raito} = \frac{V_c}{V_s} = \frac{0.2862}{1.9085}$$

$$\boxed{K = 0.1499}$$

$$\therefore\ \eta_v = 1 + 0.1499 - 0.1499\left[\frac{5}{1}\right]^{1/1.3}$$

$$\eta_v = 0.633 = 63.3\%$$

$$\boxed{\eta_v = 63.3\%}$$



Problem (3) contd.

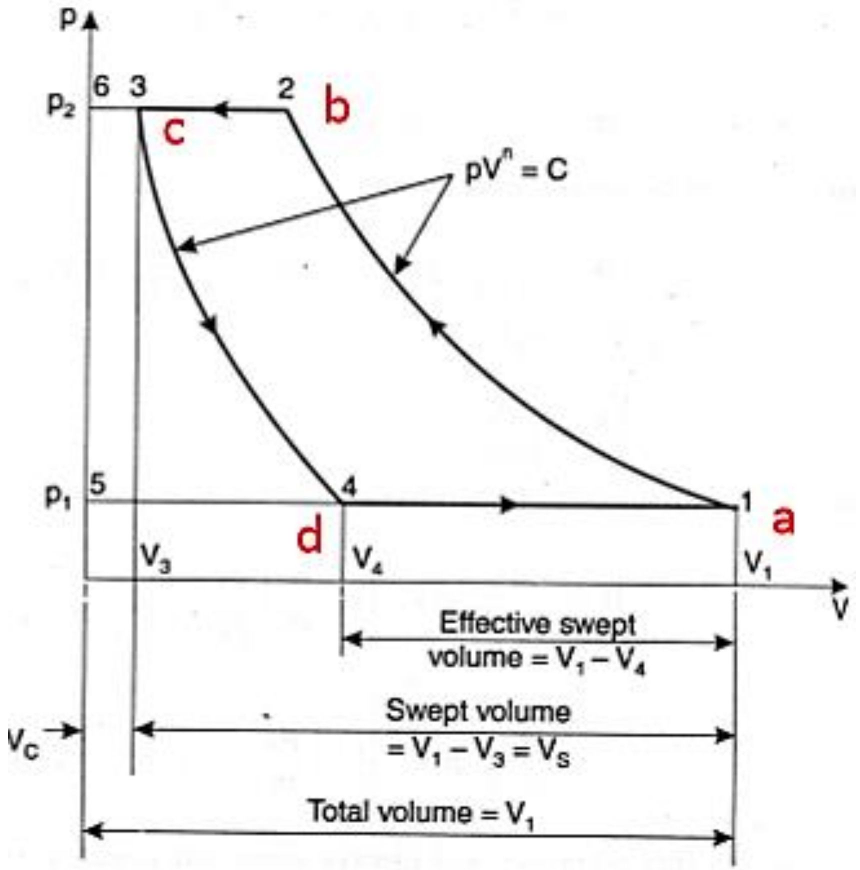
$\therefore$ **WKT**

$$\eta_v = \frac{FAD}{V_s}$$

$$\therefore FAD = \eta_v \times V_s$$

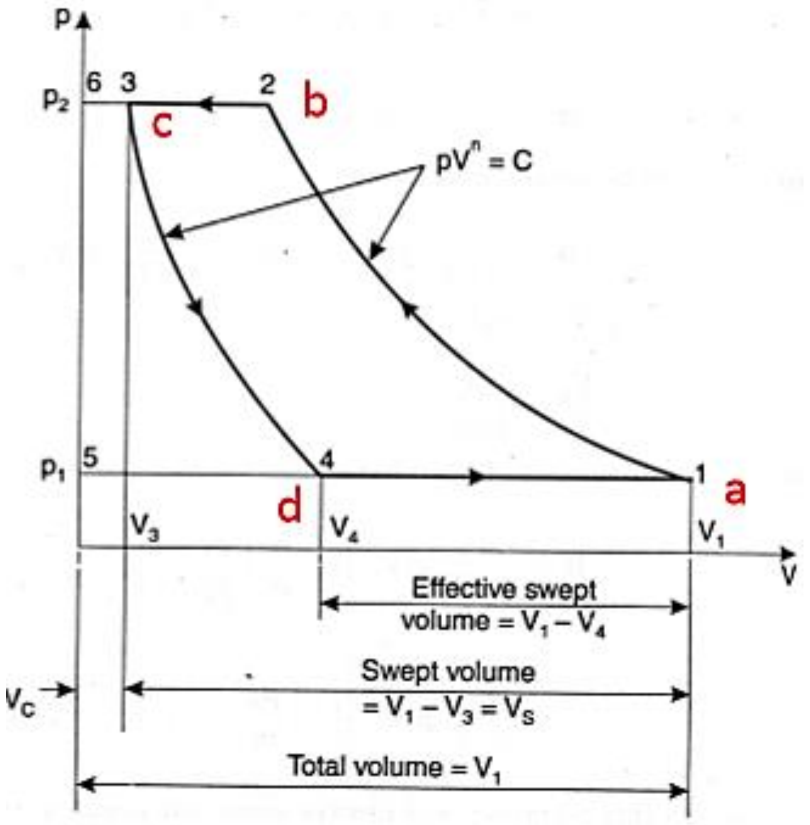
$$= 0.633 \times 1.9085$$

$$\boxed{FAD = 1.2083\ m^3/min}$$



4. A single stage single acting air compressor delivers 0.6 kg of air per minute at 6 bar . The temperature and pressure at the end of suction stroke are $30^{o}C$ and 1 bar. The bore and stroke of the compressor are 100 mm and 150 mm respectively. The clearance is 3% of the swept volume. Assuming the index of compression and expansion to be 1.3. Find: 1) Volumetric efficiency of the compressor 2) Power required if the mechanical efficiency is 85%, and 3) Speed of the compressor

- **Given**
- Mass of air delivered, m=0.6 kg/min
- Delivery Pressure, P_2 =6 bar
- Induction Pressure, P_1 =1 bar
- Induction temperature, T_1 = 30 + 273 = 303 K
- Bore, D= 100mm = 0.1m
- Stroke length , L=150mm = 0.15 m
- Clearance volume, Vc = 0.03 Vs
- Mechanical efficiency =85%



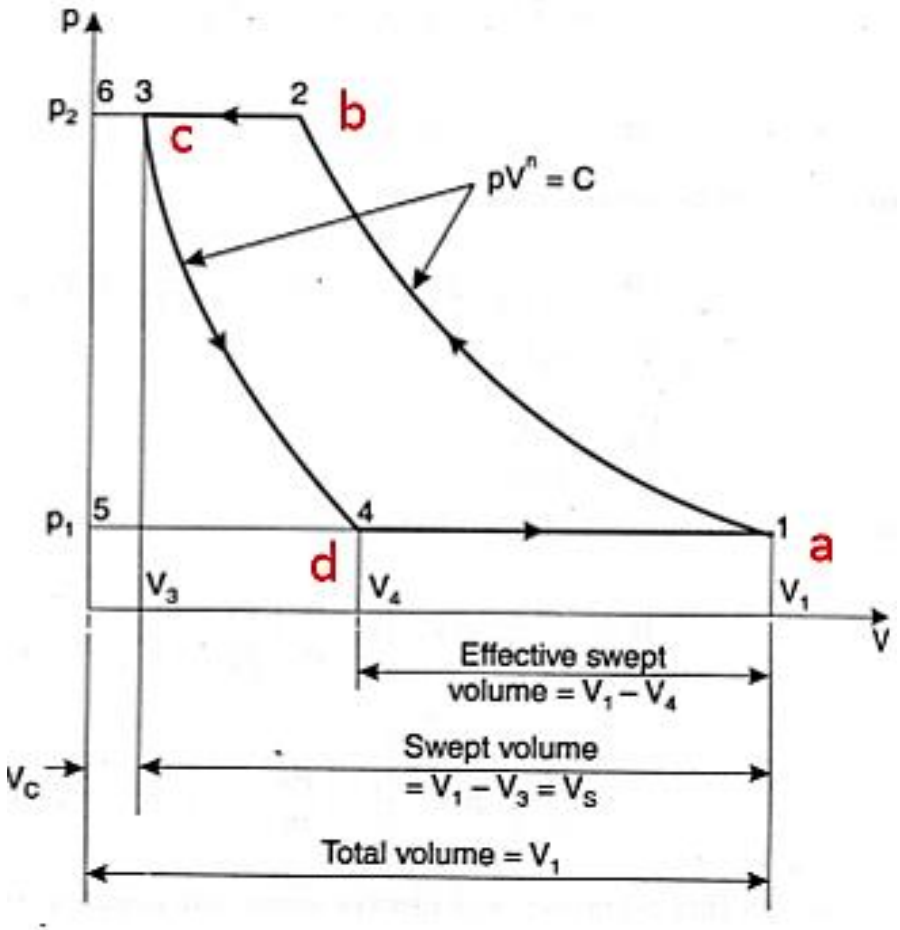
Problem (4) contd.

- **Volumetric efficiency of the compressor**

$$\eta_{vol} = 1 + k - k\left(\frac{P_2}{p_1}\right)^{\frac{1}{n}}$$

Where $k = \frac{V_c}{V_s} = 0.03$

$$\therefore \ \eta_{vol} = 1 + 0.03 - 0.03\left(\frac{6}{1}\right)^{\frac{1}{1.3}} = 0.91096 or 91.096\%$$



Power required if the mechanical efficiency is 85%

$$\text{Indicated power} == \frac{n}{n-1} mRT_1\left[\left(\frac{P_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\frac{1.3}{1.3-1} \times \frac{0.6}{60} \times 0.287 \times 303\left[\left(\frac{6}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right] = 1.93 kW$$

Problem (4) contd.

$\therefore$ Power required to drive the compressor: $= \dfrac{1.93}{\eta_{mech}} = \dfrac{1.93}{0.85} = 2.27kW$

Speed of the compressor (r.p.m)

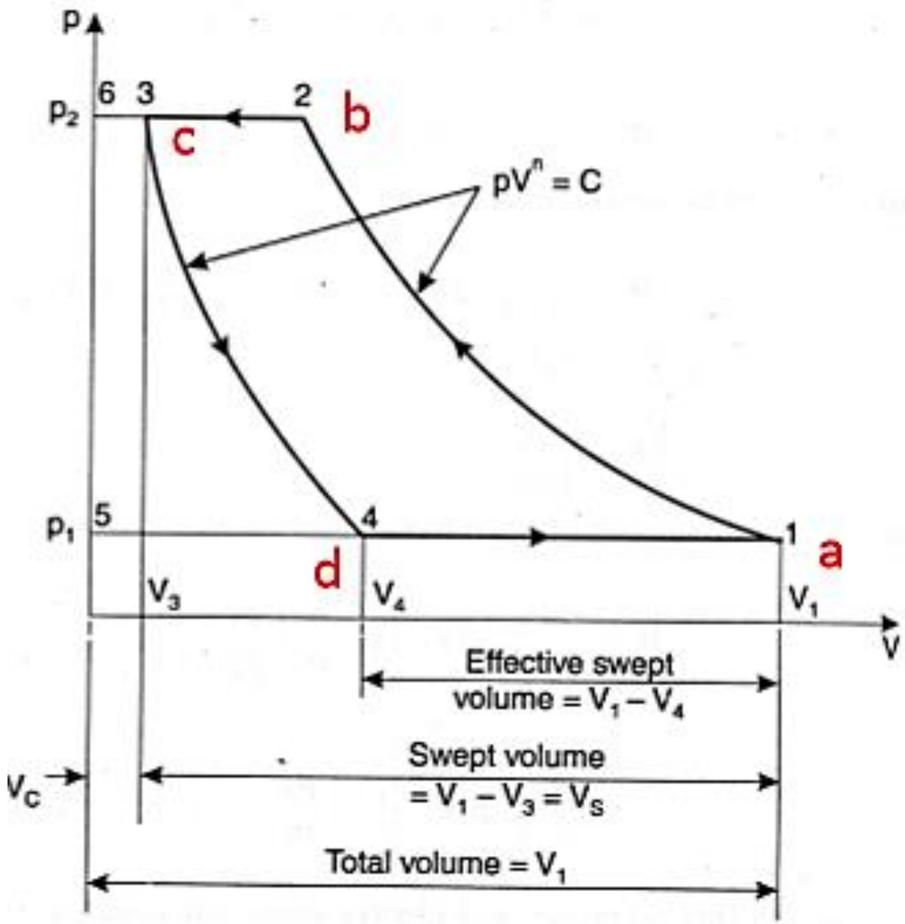
Free air delivery, F.A.D $= \dfrac{mRT_1}{p_1}$:

$$\frac{0.6\times 0.287\times 1000\times 303}{1\times 10^5} = 0.5218 \text{ m}^3/\text{min}$$

Displacement volume $= \dfrac{F.A.D}{\eta_{vol}} = \dfrac{0.5218}{0.91096} = 0.5728\text{m}^3/\text{min}$

$$0.5728 = \frac{\pi}{4} D^2 L \times N \text{ (for single – acting compressor)}$$

$\therefore$ Speed of compressor $N = \dfrac{0.5728\times 4}{\pi\times 0.1^2\times 0.15} =$ **486.2 r.p.m**



5. A single stage , single air compressor running at 1000 r.p.m delivers air at 25 bar . For this purpose the induction and free air conditions can be taken as 1.013 bar and 15^{o}C and the free air delivery as 0.25 m^{3}/min. The clearance volume is 3% of the swept volume and the stroke bore ratio is 2:1. Take the index of compression and expansion as 1.3. Calculate: i)Volumetric efficiency of the compressor ii) Diameter and length of the cylinder, iii) also the indicated power and the isothermal efficiency

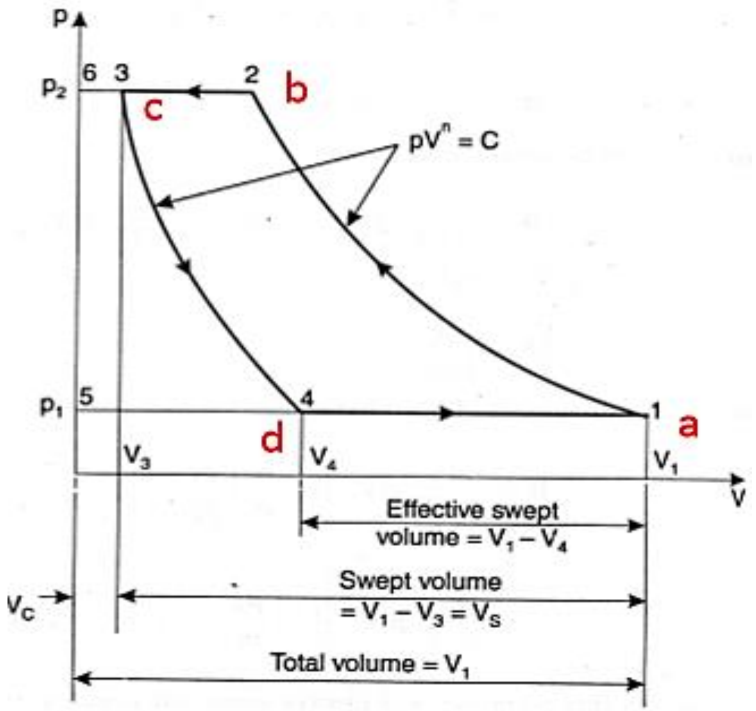
Given: Single stage single acting air compressor

N=1000 rpm, P2= 25 bar = 25x105 N/m2, T1 = 15^{O}C = 273+15 =288K

Vc = 0.03Vs , L = 2D, n = 1.3

To find: 1) Volumetric efficiency , D and L

2) I.P and Isothermal efficiency



1) Volumetric efficiency

i. $\eta_v = 1 + k - k\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$ $\eta_v = 1+0.03-0.03\left(\frac{25\times10^5}{1.013\times10^5}\right)^{\frac{1}{1.3}}$ $\eta_v = 0.6766$

Problem (5) contd.

$$IP = \frac{Wpoly \times N}{60}$$

$$\text{Wpoly} = \frac{n}{n-1} p_1 (V_1 - V_4) \left\{ \left(\frac{P_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\}$$

$$\therefore \eta_v = \frac{FAD / V_a}{V_s}$$

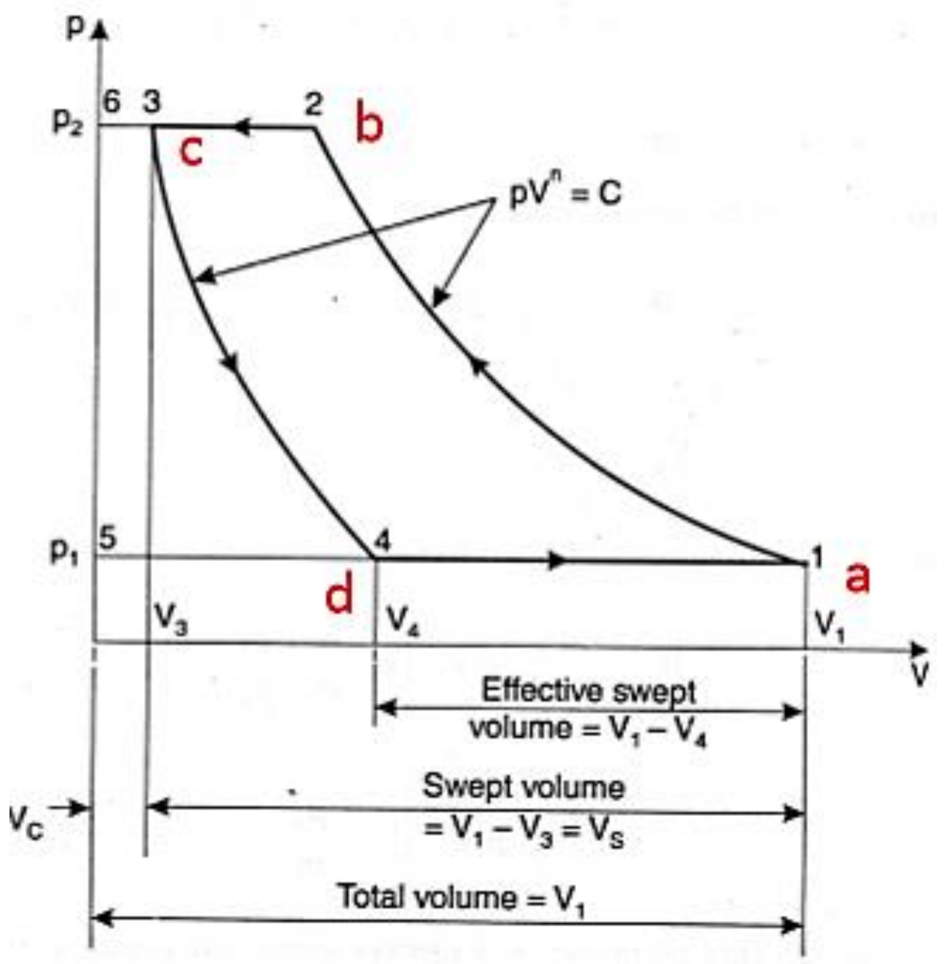
$$V_s = \frac{V_a}{\eta_v} = \frac{0.25}{0.6766}$$

$$V_s = 0.369 \ \ m^3/min$$

$$V_s = \frac{\pi}{4} \times D^2 \times L$$

$$0.369 = \frac{\pi}{4} \times D^2 \times 2D = \frac{\pi D^3}{2}$$

D =0.617 m = 617 mm



Problem (5) contd.

L = 2D = 2 x 0.617, L = 1234 mm

Vc = 0.011 m3/min = V3, V1 = Vc + Vs

V1 = 0.011 + 0.369 =0.38 m3/min

3-4 is expansion process

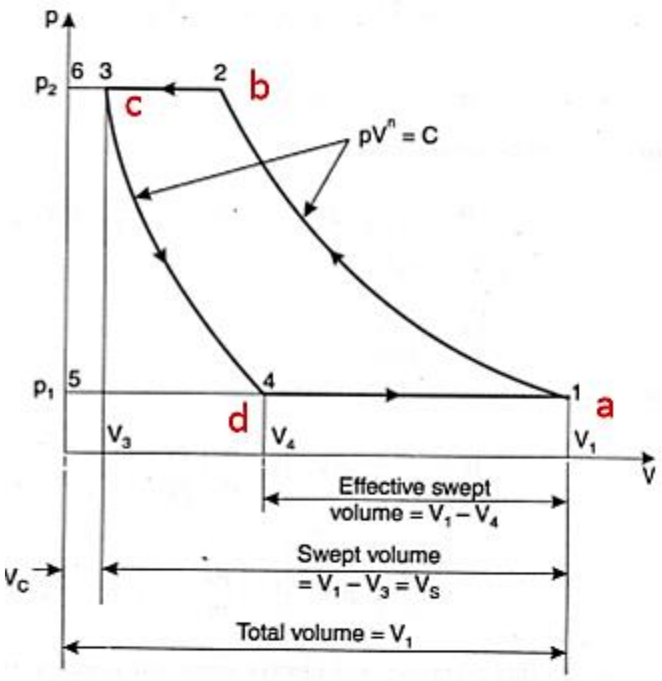
$$pv^n = C$$

$$\frac{P_4}{P_3} = \left(\frac{v_3}{v_4}\right)^n$$

$$\frac{v_4}{v_3} = \left(\frac{P_3}{P_4}\right)^{\frac{1}{n}} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$V_4 = V_3\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

V4 = 0.129 m3/min



Problem (5) contd.

$$=\frac{1.3}{0.3}\times1.013\times10^{5}(0.38-0.129)\left[\left(\frac{2.5}{1.013}\right)^{\frac{0.3}{1.3}}-1\right]$$

W.D poly = 120713.193 N-M

$$\text{I.P}=\frac{W.Dpoly\times N}{60}$$

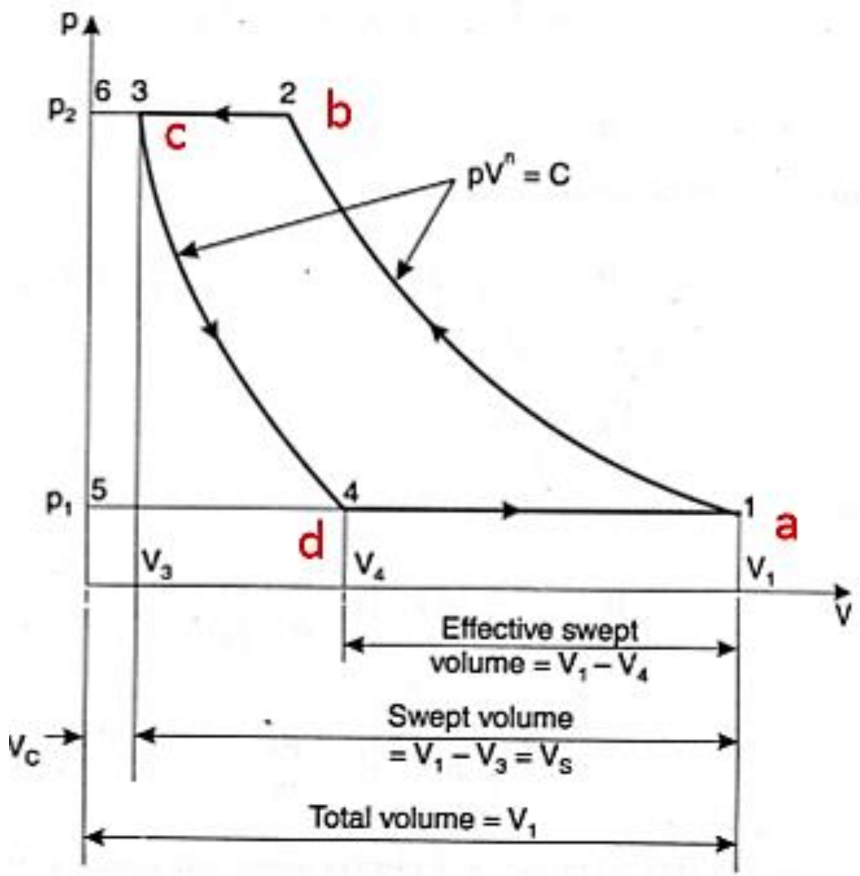
$$\text{I.P}=\frac{12713\times1000}{60}$$

$$\eta_{iso}=\frac{W_{iso}}{W_{poly}}$$

$$W_{iso}\ p_1(V_1-V_2)\ln\left(\frac{p_2}{p_1}\right)\ -\ 1.013\times10^{5}\times(0.0380-0.129)\ln\left(\frac{25}{1.013}\right)$$

$$\eta_{iso}=0.675$$

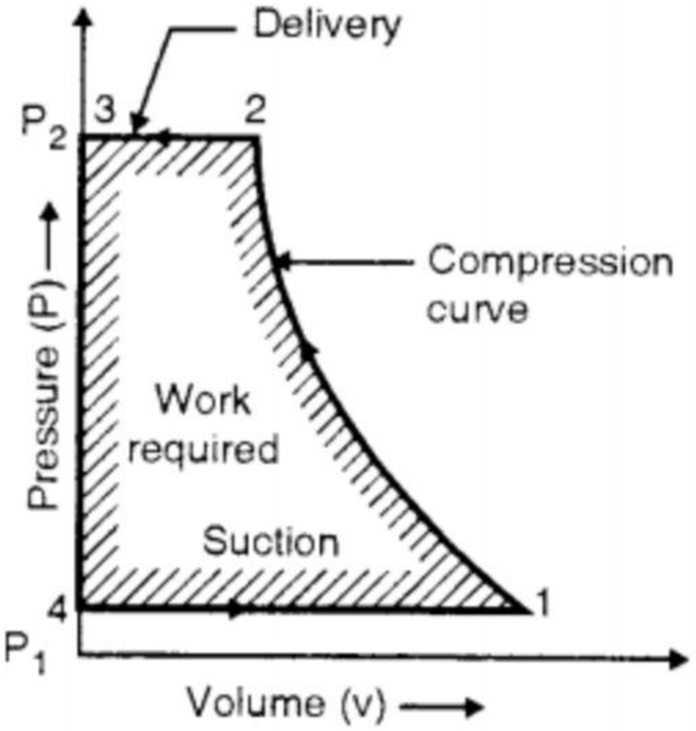
$$\eta_{iso}=67.5\%$$



6. Calculate the diameter and stroke for a double acting single stage reciprocating air compressor of 50 kW having induction pressure 100 kN/m^2 and temperature 15^{O}C. The law of compression is Pv$^{1.2}$ =C and delivery pressure is 500 kN/ m^2. The revolution/sec =1.5 and mean piston speed in 150 m/min. Clearance is neglected.

Given:

- Double acting single stage Compressor
- IP = 50kW, $P_1 = 100 \times 10^3$ N/m^2,
- $T_1 = 15 + 273 = 288$K, $PV^{1.2} = C$, n=1.2
- $P_2 = 500 \times 10^3$ N/m^2, N = 1.5 rps = 1.5x60 rpm
- 2LN = 150m/min (Double acting)



Find: D and L

For double acting compressor average piston speed = 2LN, 2LN = 150 m/min

$$\therefore L = \frac{150}{2 \times 1.5 \times 60} = 0.833m$$

$$\boxed{L = 0.833 \text{ m}}$$

Problem (6) contd.

To Find D

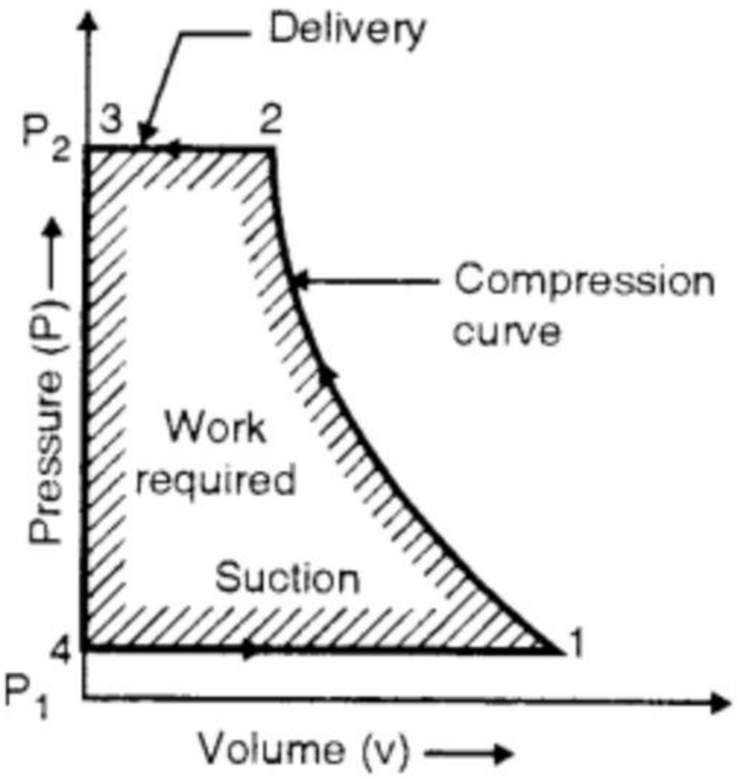
- IP = W.Nw , where Nw = Number of working stroke
- For Double acting Nw = 2N, For single acting Nw = N

Nw = 2x1.5x60 = 180 rpm

$$\therefore \text{W.D/cycle} = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.2}{1.2-1} \times 100 \times 10^3 \left(\frac{\pi}{4} D^2 \times 0.833 \right) \times \left[\left(\frac{(500)}{100} \right)^{\frac{0.2}{1.2}} - 1 \right]$$

$$\boxed{W = 120764.2D^2}$$



Problem (6) contd.

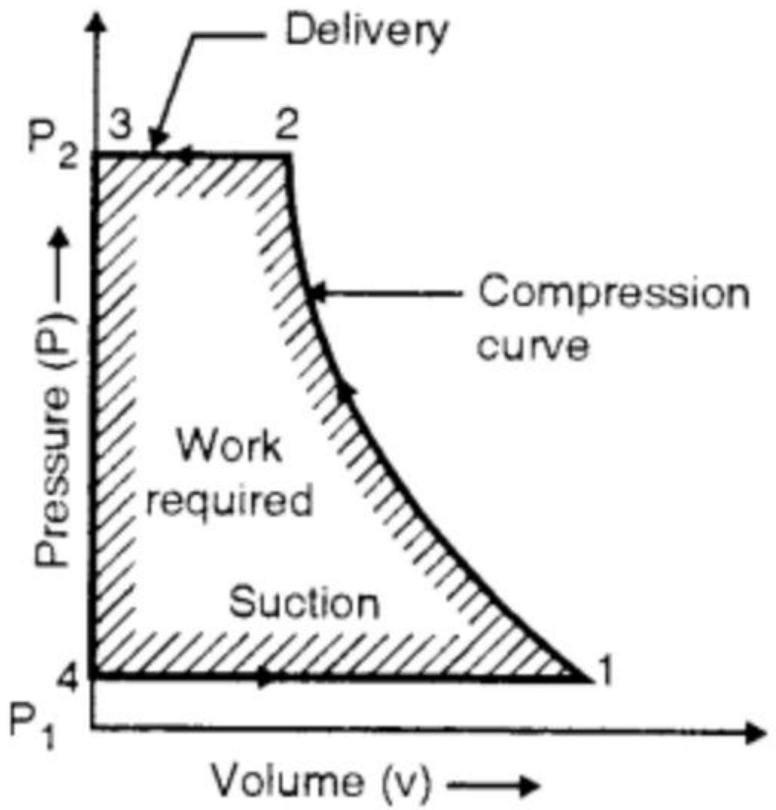
Indicated power,

$$\therefore \text{IP} = \frac{W.N_w}{60}$$

$$50 \times 10^3 = \frac{1207642 D^2 \times 180}{60}$$

$$D^2 = 0.1380$$

$$\boxed{D = 0.371 \text{ m}}$$



7. A single –stage double –acting air compressor is required to deliver 14 m^3 of air per minute measured at 1.013 bar and $15^{O}C$. The delivery pressure is 7 bar and the speed 300 rpm. Take the clearance volume as 5% of the swept volume with the compression and expansion index of 1.3 Calculate: i) Swept volume of the cylinder; ii) The delivery temperature; iii) Indicated power.

Given: Quantity of air to be delivered $= 14\ m^3/min$

Intake pressure and temperature $P_1 = 1.013$ bar,

$T_1 = 15 + 273 = 288$ K, Delivery pressure $P_2 = 7$ bar,

Compressor speed, $N = 300$ r.p.m, Clearance volume, $V_c = 0.05\ V_s$

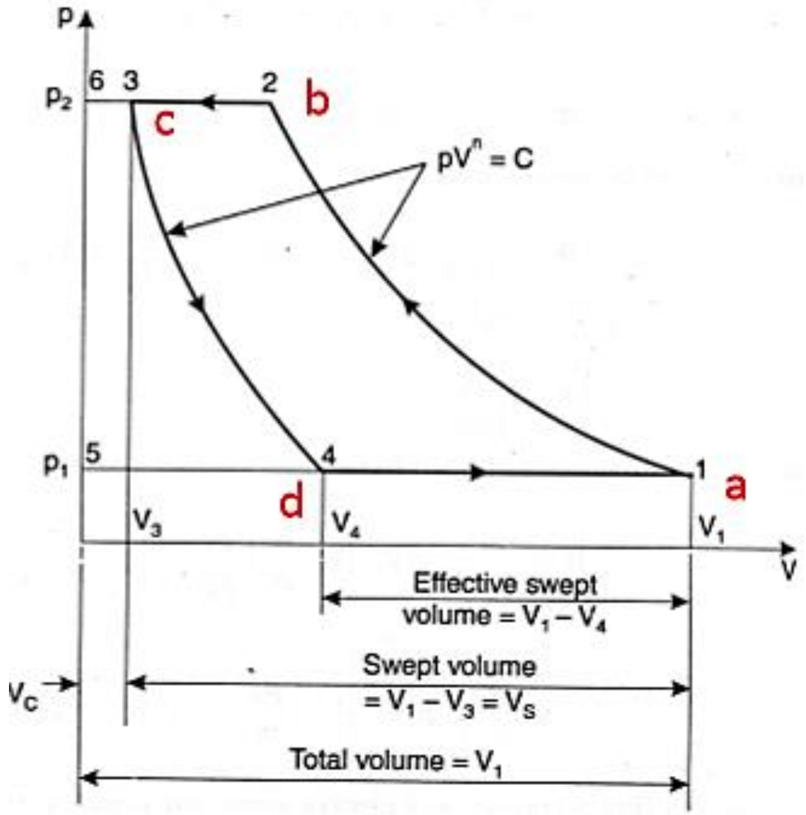
Compression and expansion index $n=1.3$

Swept volume of the cylinder, Vs:

- Swept volume $V_s = V_1 - V_3 = V_1 - V_c = V_1 - 0.05V_s$

$$V_1 - V_4 = \frac{FAD}{N_w}$$

$$V_1 - V_4 = \frac{14}{300 \times 2} = 0.0233\ m^3$$



Problem (7) contd.

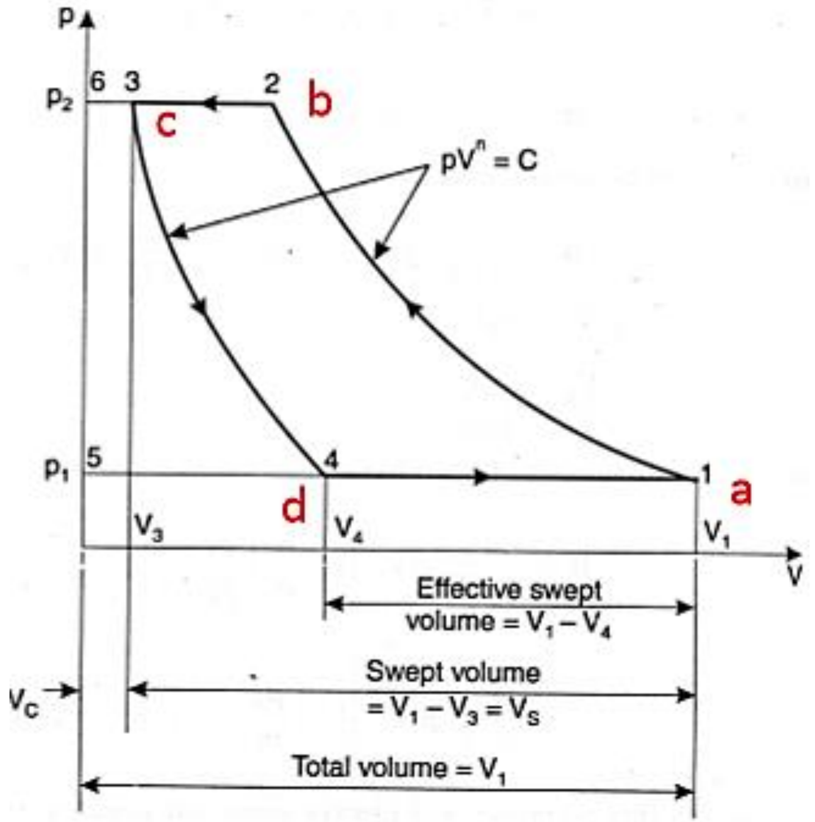
Now, $V_1 = 1.05\ V_s$ and $\dfrac{V_4}{V_3} = \left(\dfrac{p_2}{p_1}\right)^{1/n} = \left(\dfrac{7}{1.013}\right)^{1/1.3}$

$= 4.423$

$V_1 - V_3 = 4.423\ V_3 = 4.423 \times 0.05\ V_s = 0.221\ V_s$

$(V_1 - V_4) = 1.05\ V_s - 0.221 \quad V_s = 0.221\ V_s$

$\therefore \quad V_s \dfrac{0.0233}{1.05 - 0.221} = 0.0281 m^3$



The delivery temperature, T2

Using the relation $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

$\therefore \quad T_2 = T_1 \times \left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 288 \times \left(\dfrac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} = 450K$

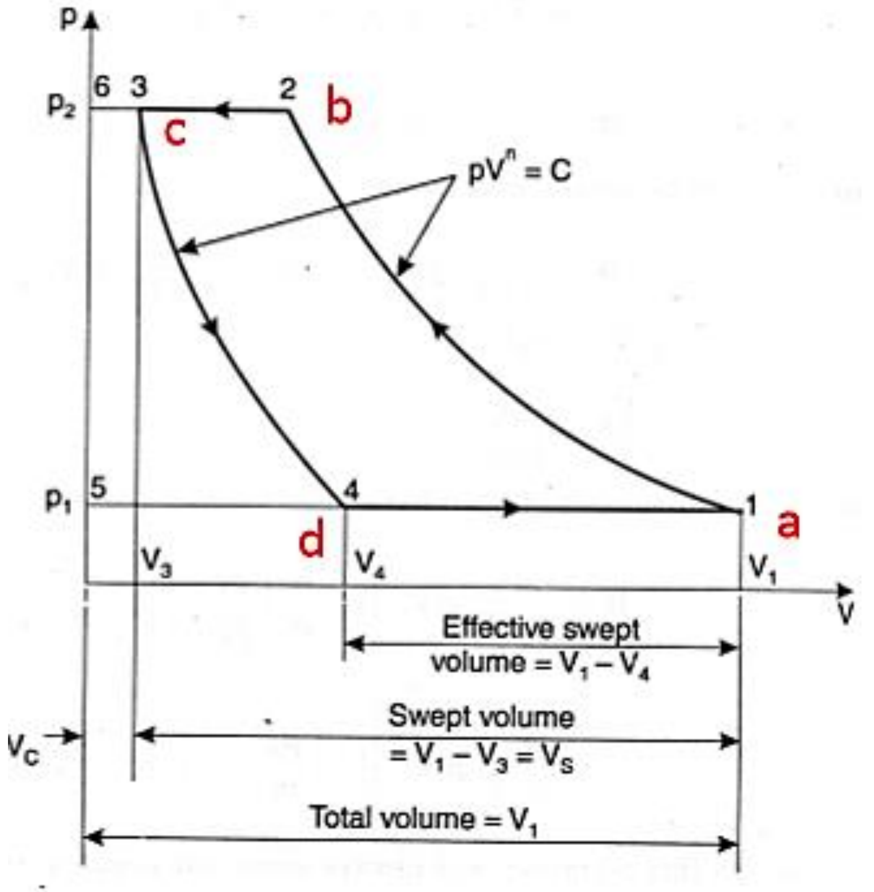
Problem (7) contd.

- **Indicated power:**

$$\text{Indicated power} = \frac{n}{n-1} p_1 (V_1 - V_4) \left\{ \left(\frac{P_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\}$$

$$= \frac{1.3}{1.3-1} \times \frac{1.013 \times 10^5 \times 14}{10^3 \times 6} \left\{ \left(\frac{7}{1.013} \right)^{\frac{1.3-1}{1.3}} - 1 \right\}$$

Indicated power = **57.56 kW**



8. A single acting reciprocating air compressor has a swept volume of 2000 cm^3 and runs at 800 rpm. It operates with pressure ratio of 8 and clearance of 5% of the swept volume. Inlet pressure and temperature are 1.013 bar, and 15^OC respectively. Assume n=1.25 for both compression and expansion. Find: i) Indicated power ii) Volumetric efficiency iii) Mass flow rate iv) FAD v) Isothermal efficiency vi) Actual Power required to drive the compressor if mechanical efficiency =85%

Given:

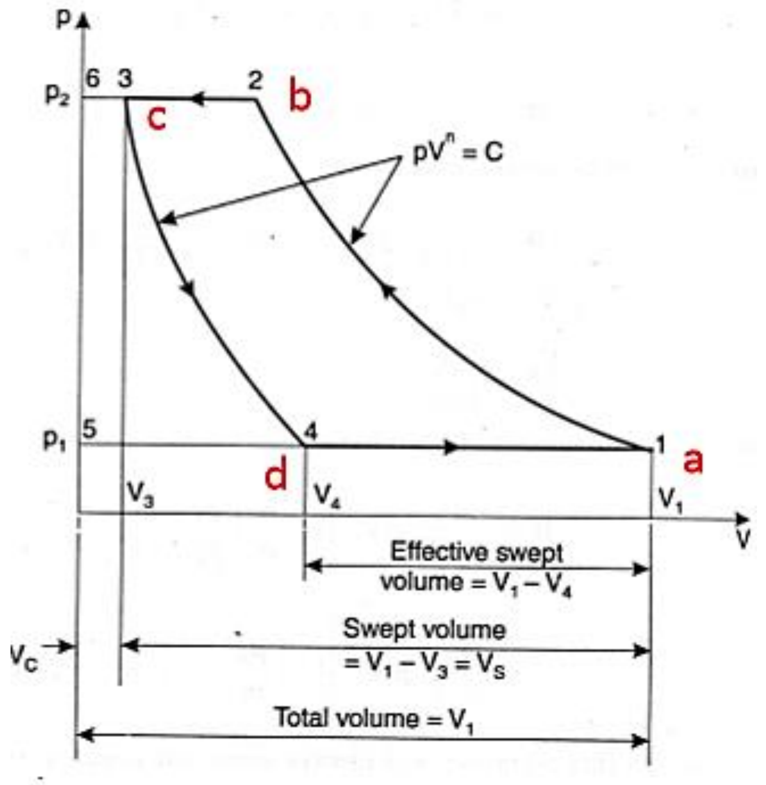
Single acting reciprocating compressor Vs = 2000 cm^3 = 0.002 m^3

N = 800 rpm, Vc = 5% Vs = 0.05x0.002,

Vc = 0.0001 m^3, P1 = 1.013x105 N/m^2

T1 = 15+273 = 288 K. n = 1.25

Find: i)IP ii)η_v iii)m iv) FAD(V_a) v) η_{iso} vi) P_{act} if η_m=85%



WKT

$$\text{Clearance ratio (k)} = \frac{V_c}{V_s} = \frac{0.0001}{0.002}$$

$$\boxed{k \quad = \quad 0.05}$$

Problem (8) contd.

$\therefore$ Volumetric efficiency $(\eta_v) = 1 + k - k\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$

$= 1 + 0.05 - 0.05\,(8)^{1/1.25}$

$\eta_v = 78.61\%$

WKT

$\eta_v = \frac{FAD}{V_s}$

$FAD = \eta_v \times V_s$

$= 0.7861 \times 0.002$

$\frac{FAD}{min} = \frac{FAD}{Stroke} \times Speed$

$= 1.5722 \times 10^{-3} \times 800$

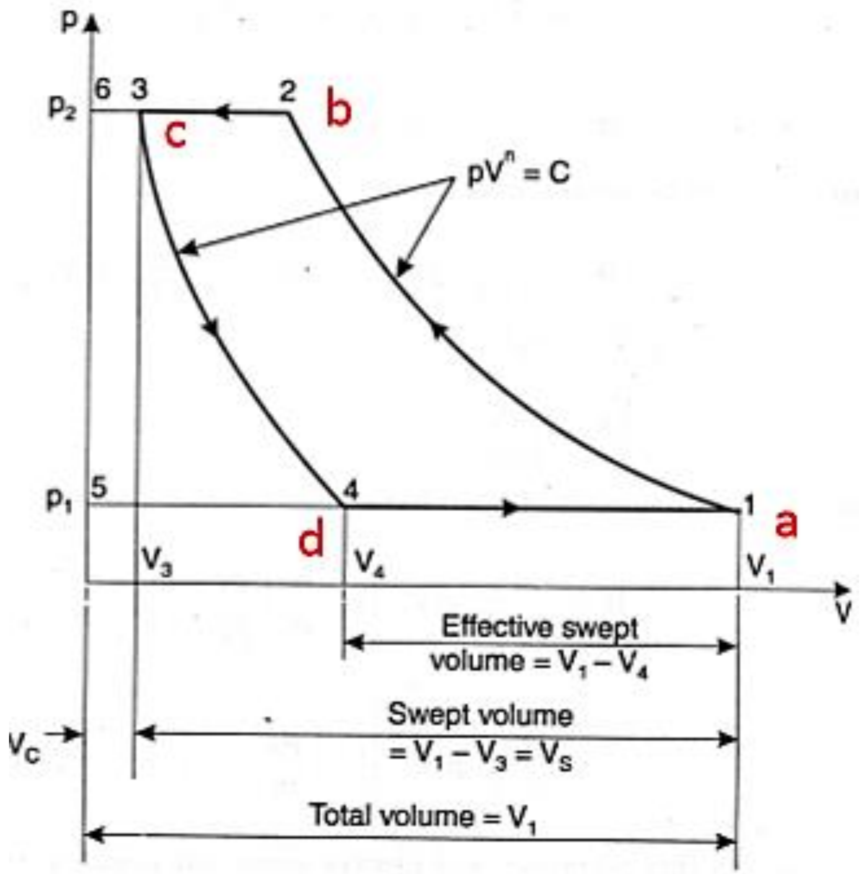
$\therefore FAD = 1.2578\ m^3/min$

To find mass flow rate

$Pv = mRT$

$$m = \frac{pV}{RT} = \frac{1.013 \times 10^5 \times 1.2578}{287 \times 288}$$

$m = 1.542\ kg/min$



Problem (8) contd.

To find – Indicated power

$$IP = \frac{W.D \times N_\omega}{sec}$$

$$N_w = N \text{ (single acting)} = 800 \text{ rpm.}$$

$$W.D = \frac{n}{n-1} mR(T_2 - T_1) \text{ kJ/min.}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

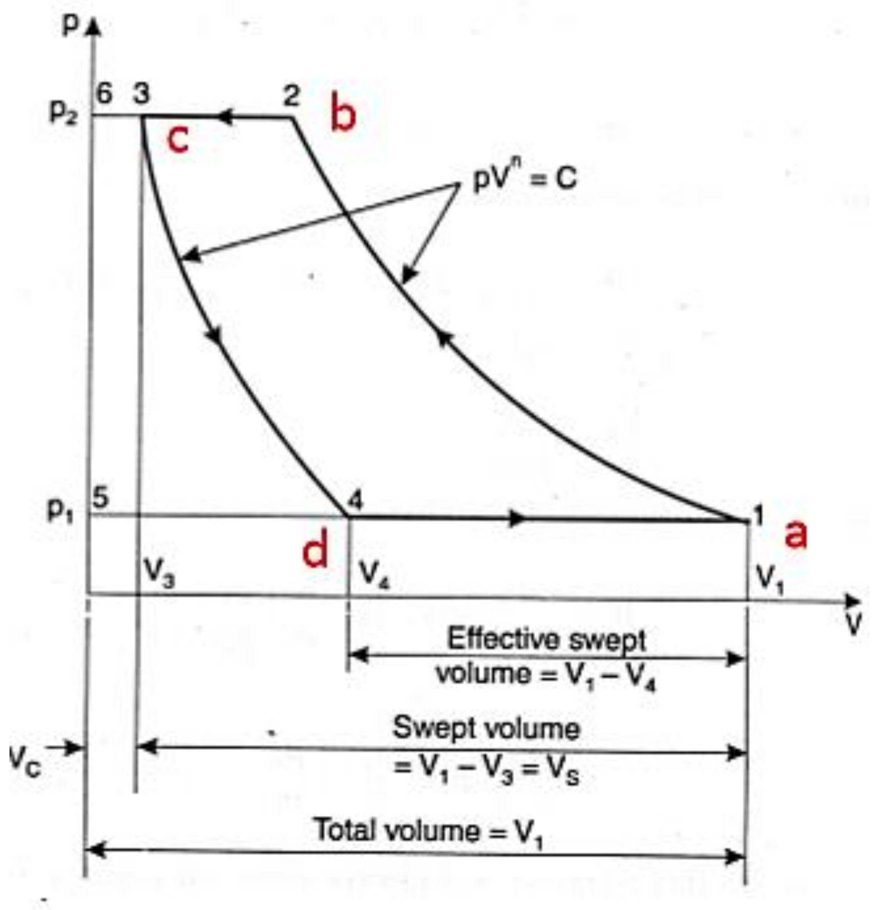
$$= 288(8)^{\frac{1.25-1}{1.25}} \qquad T_2 = 436.53 \text{ K}$$

$$\therefore W.D = \frac{1.25}{0.25} \times 1.542 \times 0.287(436.53 - 2880$$

$$W.D = 328.66 \text{ kJ/min}$$

$$\therefore IP = \frac{W.D}{s} = \frac{328.66}{60}$$

$$IP = 5.48 \text{ kW}$$



Problem (8) contd.

To find isothermal efficiency

$$W_{iso} = P_1V_1 \ln\left(\frac{P_2}{P_1}\right)$$

$$= mRT_1 \ln\left(\frac{P_2}{P_1}\right)$$

$$= 1.542\times0.287\times288\times\ln(8)$$

$$(W.D)_{iso} = 265.04 \text{ kJ/min}$$

$$\eta_{iso} = \frac{(W.D)_{iso}}{(W.D)_{act}}$$

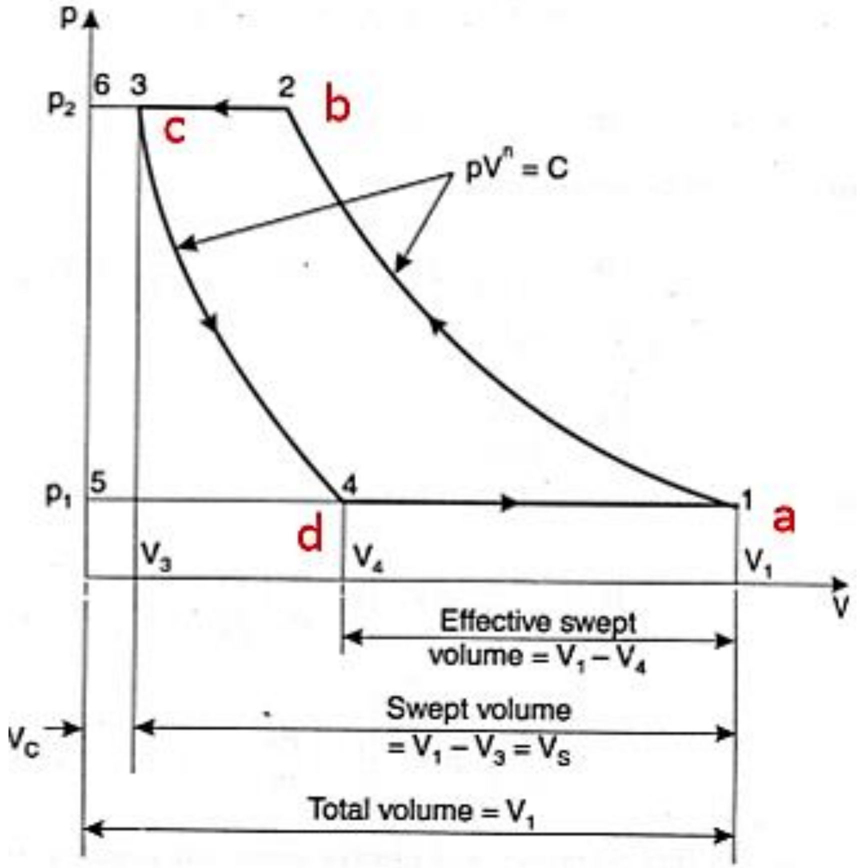
$$\therefore \text{Actual work} = \eta_{mech} \times (W.D)_{theoretical}$$

$$= 0.85\times328.66$$

$$(W.D)_{act} = 279.361 \text{ kJ/min}$$

$$\therefore \eta_{iso} = \frac{265.04}{279.361}\times100$$

$$\boxed{\eta_{iso} = 94.87\%}$$



Limitations of single stage compressor

a) Cylinder wall thickness increases making the compressor heavy.

b) Requires heavy moving/working components to compress air to high pressures.

c) Use of heavy working components increase the balancing problem, and the high torque fluctuation will require a heavier flywheel installation.

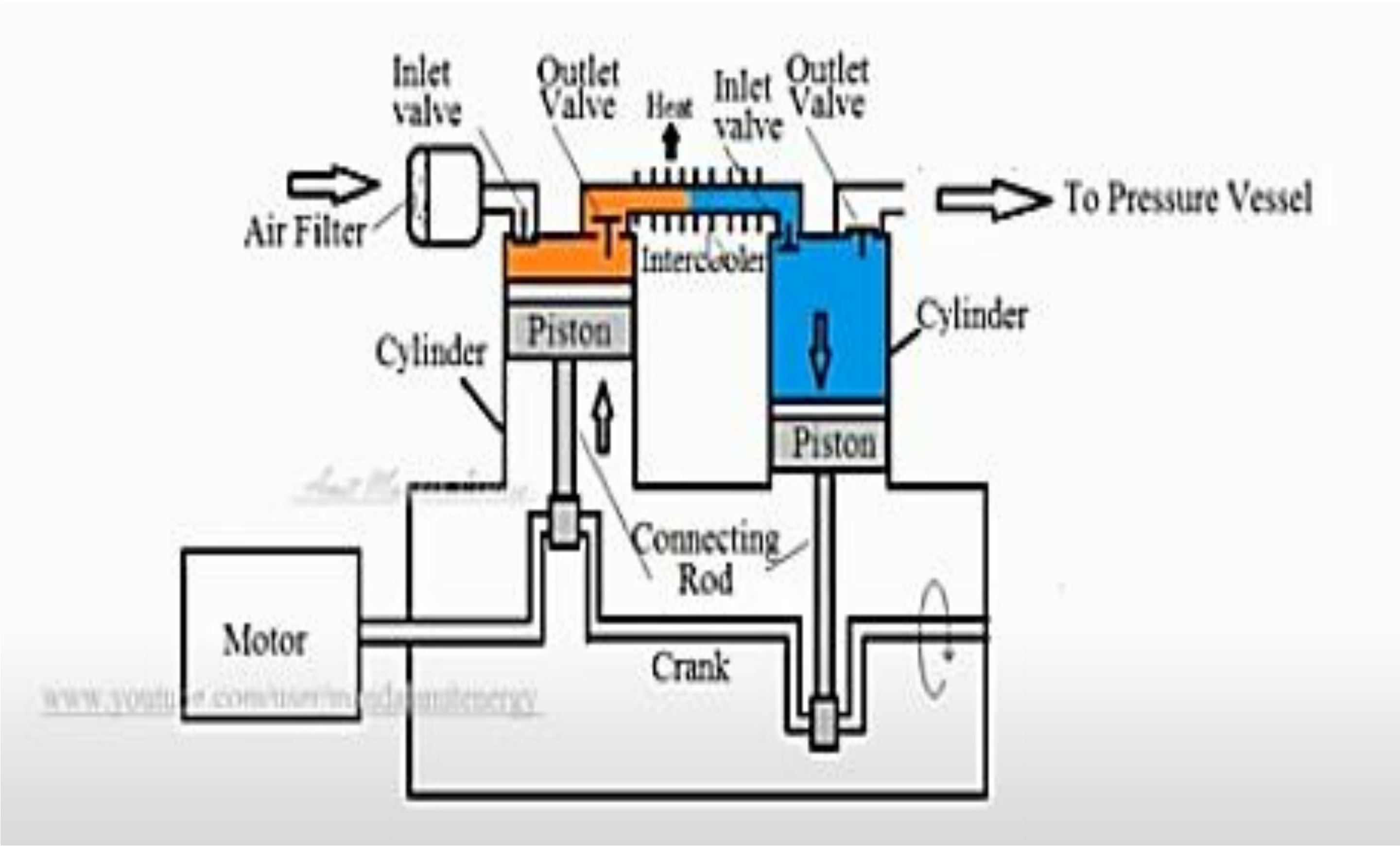
d) Temperature of air at discharge increases making it difficult to reject the heat from the air in small interval.

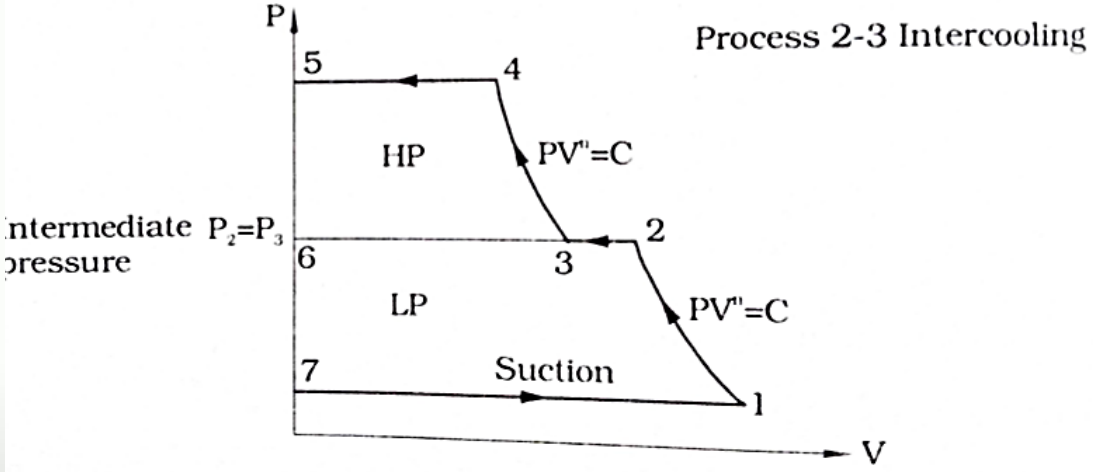
e) High temperatures may heat up the cylinder head and burn the lubricating oil.

f) Work input increases with increase in pressure ratio.

g) Volumetric efficiency decreases with an increase in pressure ratio which affects the air handling capacity of the compressor adversely.

Multistage compressors





(b) p-v diagram of a 2-stage compressor

Advantages of Multistage Compressor

- Less Power Required
- Increased volumetric efficiency
- Better mechanical balance
- Better lubrication
- Reduced size of cylinder
- Reduced leakage loss
- Reduced cost of compressor

Condition for minimum workdone in 2-stage compressor with intercooler

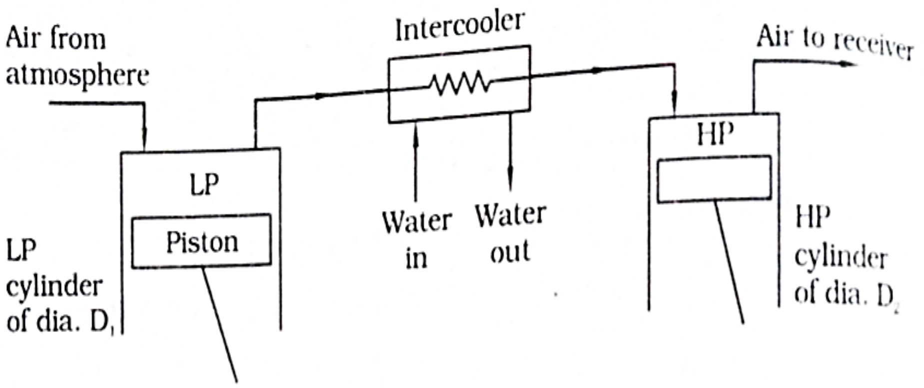
Assuming the compression process in each stage to follow the law $PV^n = C$, and for perfect intercooling, the temperature of air at entry to each stage is the same, i.e., $T_1 = T_3$. Since points 1 and 3 have to fall on the isothermal line, we have

$$P_1 V_1 = P_3 V_3.$$

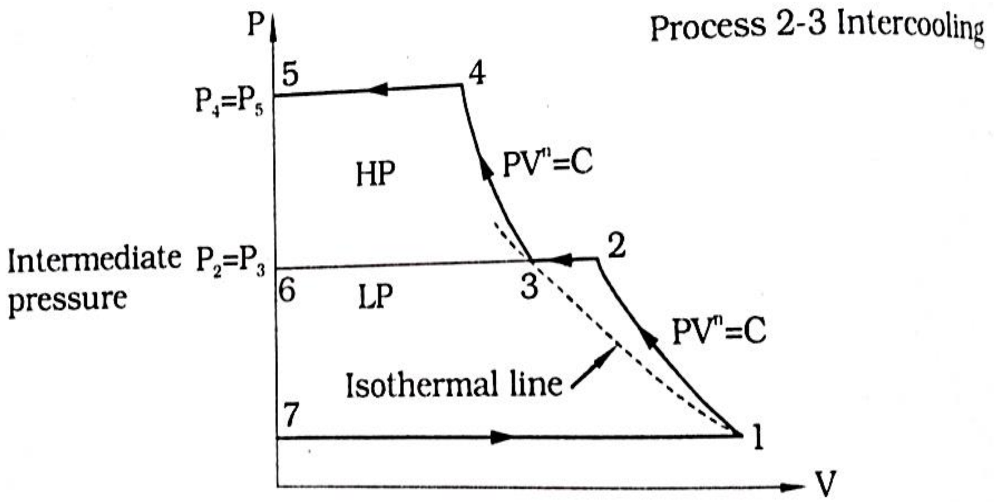
Total work of compression $= W = W_{LP} + W_{HP}$

$$= \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} P_3 V_3 \left[\left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 2 \right] \qquad \because P_1 V_1 = P_3 V_3$$



(a) 2-stage compressor



Condition for minimum workdone (contd.)

For a given value of P_1 and V_1, work of compression (W) is minimum, when $\frac{dW}{dP_3} = 0$

For simplicity, let $= \frac{n-1}{n} = a$

Also $P_2 = P_3$ from p-v diagram

$$\therefore \text{ compressor work reduces to } W = \frac{P_1 V_1}{a}\left[\left(\frac{P_3}{P_1}\right)^a + \left(\frac{P_4}{P_3}\right)^a - 2\right]$$

$$\text{or } \quad W = \frac{P_1 V_1}{a}\left[\frac{1}{P_1^a} P_3^a + P_4^a \frac{1}{P_3^a} - 2\right]$$

$$\therefore \frac{dW}{dP_3} = \frac{P_1 V_1}{a}\left[\frac{1}{P_1^a}\left(a.P_3^{a-1}\right) + P_4^a \frac{\left(-a P_3^{a-1}\right)}{\left(P_3^a\right)^2}\right] = 0$$

Condition for minimum workdone (contd.)

$$P_1^{-a}\left(P_3^{a-1}\right)-P_4^{a}\frac{\left(P_3^{a-1}\right)}{P_3^{2a}}=0$$

$$P_1^{-a}\;P_3^{a-1}=P_4^{a}\left(P_3^{a-1}.P_3^{-2a}\right)$$

$$P_1^{-a}\;P_3^{a-1}=P_4^{a}\;P_3^{a-1-2a}$$

$$\frac{P_1^{-a}}{P_4^{a}}=\frac{P_3^{-1-a}}{P_3^{a-1}}$$

$$\text{or}\quad P_1^{-a}\;P_4^{-a}=P_3^{-1-a}.P_3^{-a+1}$$

$$(P_1\,.\,P_4)^{-a}=(P_3)^{-2a}$$

$$\text{or}\quad (P_1\;P_4)=P_3^{2}$$

$$\therefore\quad P_3=\sqrt{P_1.P_4}$$

Thus for minimum work, the intermediate pressure P_3 ($P_3 = P_2$) is the geometric mean of the suction (P_1) and discharge pressures (P_4).

Note Equation (1) can also be written as $\frac{P_4}{P_3}=\frac{P_3}{P_1}$ ----(3)

This also means that the pressure ratio in each stage is the same.

9. A two stage air compressor, compresses the air from 1 bar and 20°C to 42 bar. If the law of compression is $Pv^{1.35}=C$ and the intercooling is complete to 20°C, find per kg of air:1) The work done for compressing; and 2)The mass of water necessary for abstracting the heat in the intercooler, if the temperature rise of the cooling water is 25°C

Given:

P_1= 1 bar = 1x105 N/m2, T_1 = 20°C = 20+273 =293K, P_4= 42 bar = 42 x 105 N/m2, n=1.35,

T3 = 20°C= 20+273 =293 K; m= l kg;

Rise in temperature of cooling water = 25°C;

R=287 J/kg K, Cp= 1 kJ /kg K

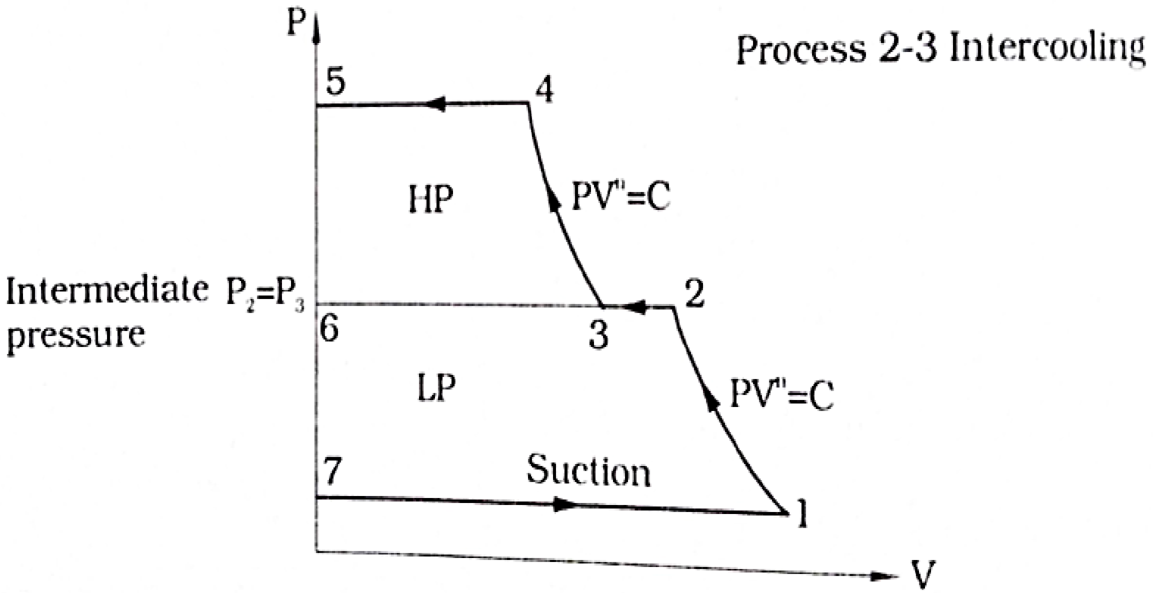
we know that for complete intercooling.

The intermediate pressure

$$p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 42} = 6.48 \text{ bar}$$

Volume of air admitted for compression

$$V_1 \frac{mRT_1}{p_1} = \frac{1 \times 287 \times 293}{1 \times 10^5} = 0.84 m^3/kg \text{ of air}$$



(b) p-v diagram of a 2-stage compressor

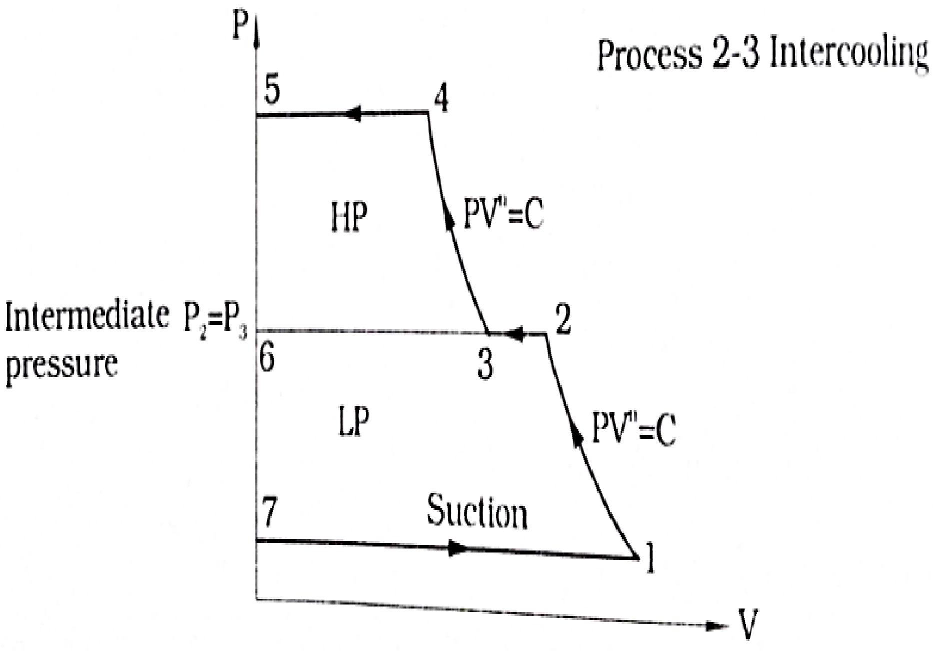
Problem (9) contd.

1. Work done compressing the air

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 2 \right]$$

$$= \frac{1.35}{1.35-1} \times 1 \times 10^5 \times 0.84 \left[\left(\frac{6.48}{1} \right)^{\frac{1.35-1}{1.35}} + \left(\frac{42}{6.48} \right)^{\frac{1.35-1}{1.35}} - 2 \right] N-m$$

$$= 3.24 \times 10^5 (1.62 + 1.62 - 2) = \mathbf{4.017 \times 10^5 \ N\text{-}m}$$



(b) p-v diagram of a 2-stage compressor

2.Mass of water necessary for abstracting the heat in the intercooler.

Let m_w= Mass of water necessary /kg of air ,and

T_2 = Temperature of the air entering the intercooler.

Problem (9) contd.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \left(\frac{6.48}{1}\right)^{\frac{1.35-1}{1.35}} = 1.622$$

$$T_2 = T_1 \times 1.622 = 293 \times 1.622 = 475.6 \text{ K}$$

We know that heat gained by water

= Heat lost by air

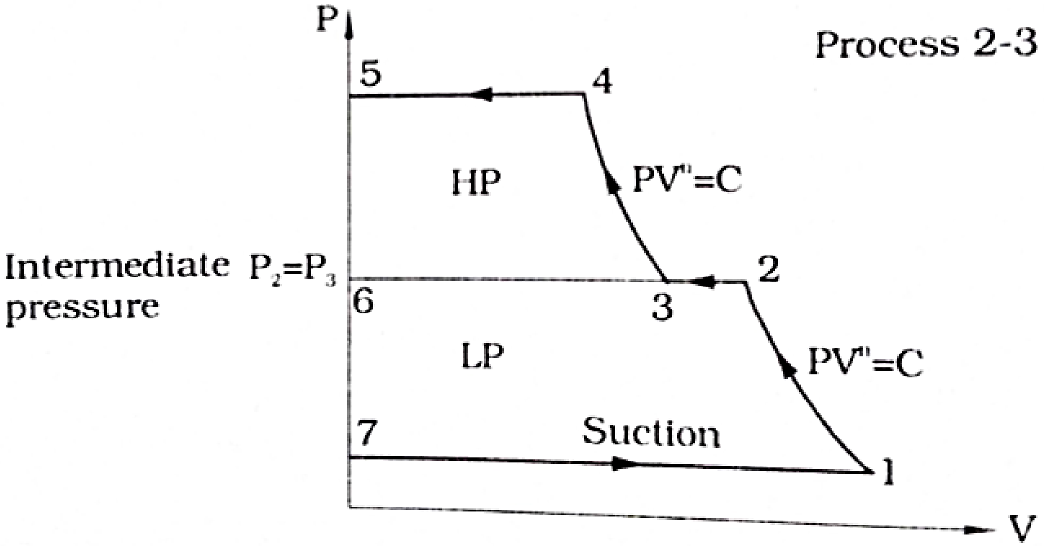
$\therefore m_w \times c_w \times$ Rise in temperature

$= mc_p(T_2 - T_3)$

$$m_w \times 4.2 \times 25 = 1 \times 1(475.6 - 293) = 182.6$$

$$\mathbf{m_w = 1.74kg}$$

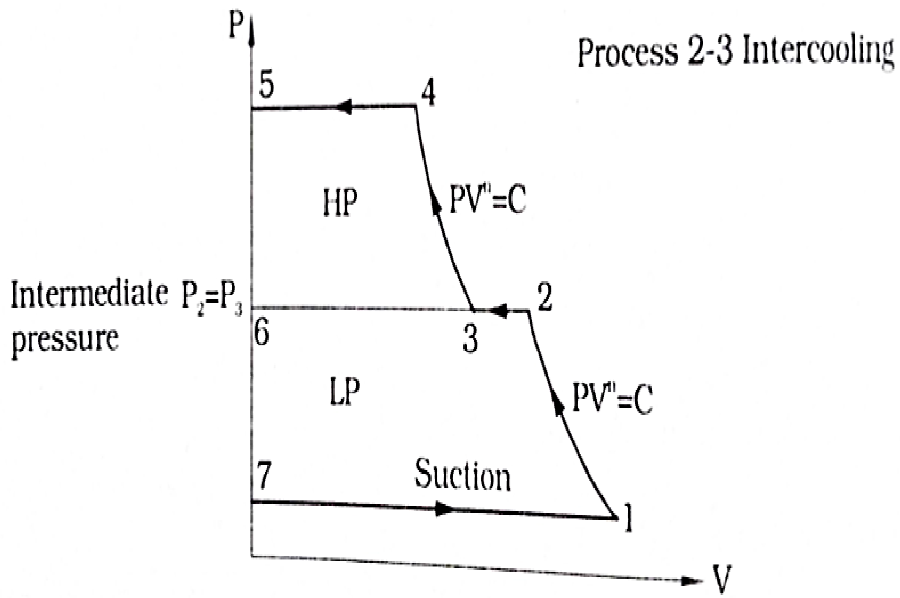
Process 2-3 Intercooling



(b) p-v diagram of a 2-stage compressor

10.A two- stage acting reciprocating compressor takes in air at the rate of 0.2 m^3/s. The intake pressure and temperature of air 0.1 MPa and 16^OC. The air is compressed to a final pressure of 0.7Mpa. The intermediate pressure is ideal and intercooling is perfect. The compression index in both the stages is 1.25 and the compressor runs at 600 r.p.m. Neglecting clearance determine: i) The intermediate pressure ii) The total volume of each cylinder, iii) The power required to drive the compressor and iv. The rate of heat rejection in the intercooler. Take Cp= 1.005 kJ/kg K and R =0.287 kJ/kg K

Intake volume	$V_1 = 0.2\ m^3/s$
Intake pressure	$p_1 = 0.1 MP_a$
Intake temperature	$T_1 = 16 + 273 = 289\ K$
Final pressure	$p_3 = 0.7 MP_a$
Compression index in both stages,	$n_1 = n_2\ n = 1.25$
Speed of the compressor	$N = 600$ r.p.m
	$c_p = 1.005 kJ/kg\ K;\ R = 0.287\ kJ/kg\ K$



(b) p-v diagram of a 2-stage compressor

i. The intermediate pressure

$$p_2 = \sqrt{p_1 p_3} = \sqrt{0.1 \times 0.7} = 0.2646\ MP_a$$

Problem (10) contd.

ii. The total volume of each cylinder ,Vs1,Vs2:

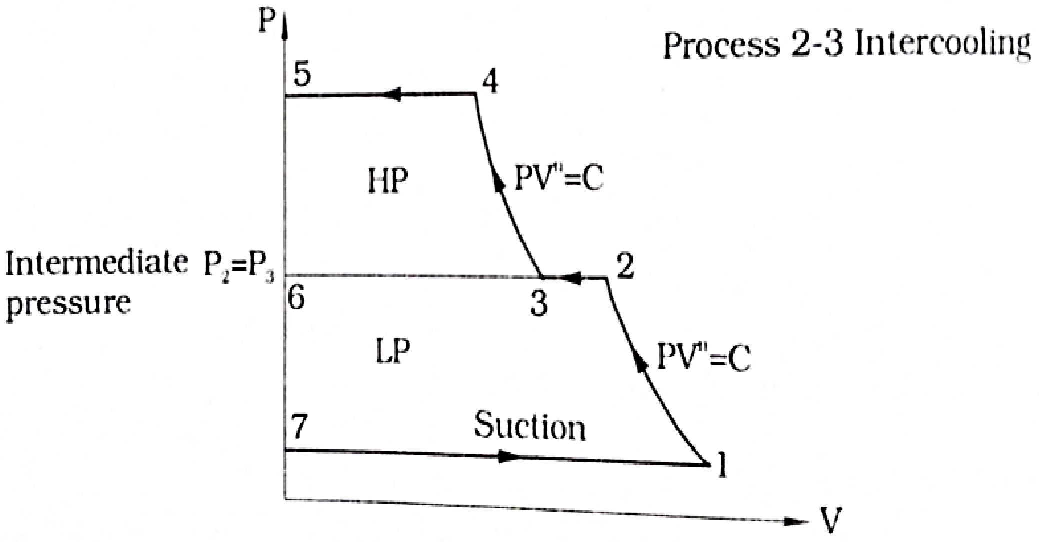
We know that $V_{s1} \times \frac{N}{60} = V_1$ or $V_{s1} \times \frac{600}{60} = 0.2$

$\therefore$ V_{s1} (Volume of L.P cylinder) $= 0.02 m_3$ (Ans).

Also $p_1 V_{s1} = p_1 V_{s2}$ or $V_{s2} = \frac{p_1 V_{s1}}{p_2}$

V_{S2} (Volume of H.P. Cylinder) $= \frac{0.1 \times 0.02}{0.2646}$

$= 0.00756 m^3$ (Ans)



(b) p-v diagram of a 2-stage compressor

Problem (10) contd.

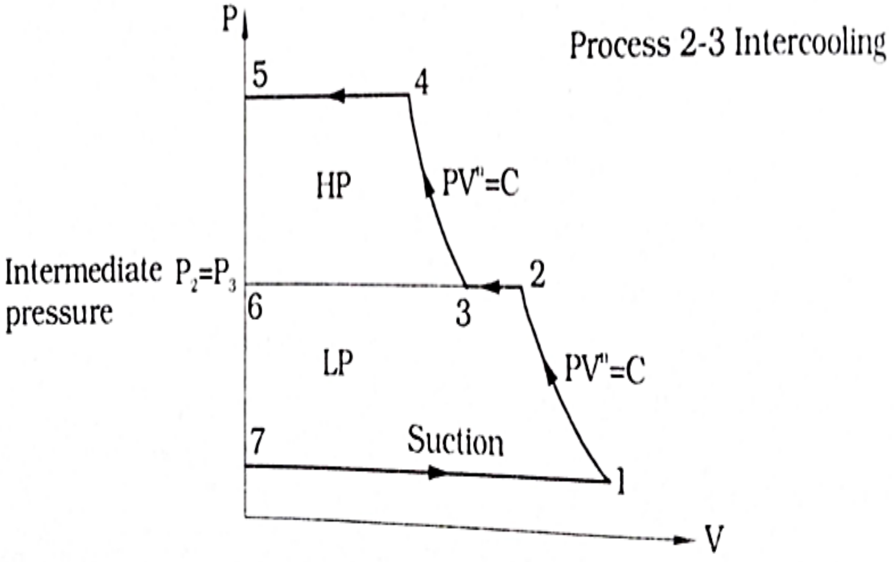
- **iv. The rate of heat rejection in the intercooler:**

Mass of air handled, $m = \frac{p_1 V_1}{R T_1} = \frac{(0.1 \times 10^3) \times 0.2}{0.287 \times 289} = 0.241\, kg/s$

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

$$\frac{T_2}{289} = \left(\frac{0.2646}{0.1}\right)^{\frac{1.25-1}{1.25}}$$

$T_2 = 351.1K$



(b) p-v diagram of a 2-stage compressor

$\therefore$ Heat rejected in the intercooler $= m \times c_p \times (T_2 - T_1)$

$= 0.241 \times 1.005 \times (351.1 - 289)$

= 15.04 kJ/s or 15.04 kW.(Ans)

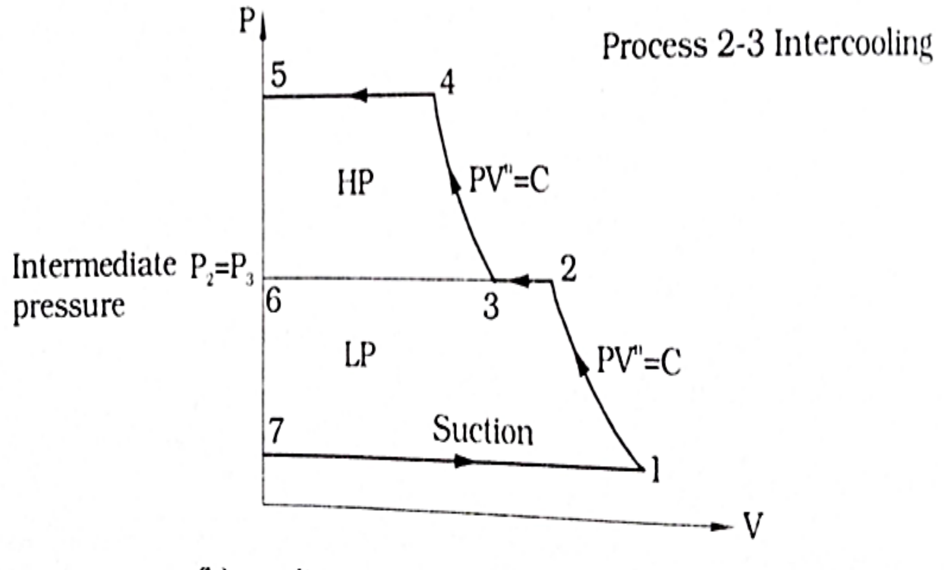
Problem (10) contd.

iii. The power required to drive the compressor

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 2 \right]$$

$$\text{w.k.t. Power} = WD/sec = \frac{2n}{n-1} P_1 V'_1 \left[\left(\frac{P_d}{P_1} \right)^{\frac{n-1}{2.n}} - 1 \right]$$

$$= \frac{2(1.25)}{1.25-1} (1 \times 10^2)(0.2) \left[\left(\frac{7}{1} \right)^{\frac{1.25-1}{2(1.25)}} - 1 \right]$$



(b) p-v diagram of a 2-stage compressor

Power = 42.96 kW

11. A single acting, 2-stage air compressor deals with 4 m^3 /min of air under atmospheric conditions of 1.016 bar and 15^{O}C with a speed of 250rpm. The delivery pressure is 78.65 bar. Assuming complete intercooling, find the maximum power required by the compressor & the bore and stroke of the compressor. Assume a piston speed of 3 m/s, mechanical efficiency of 75%, and volumetric efficiency of 80% per stage. Assume the polytropic index of compression in both stage to be n=1.25 & neglect clearance.

$V'_1 = 4\ m^3/min = 0.0666\ m^3/sec$, $P_1 = 1.016$ Bar, $T_1 = 15°C = 288$ K ; N = 250 rpm

$P_d = P_4 = 78.65$ bar ; Piston speed = 3 m/s, $\eta_{mech} = 75\% = 0.75$;

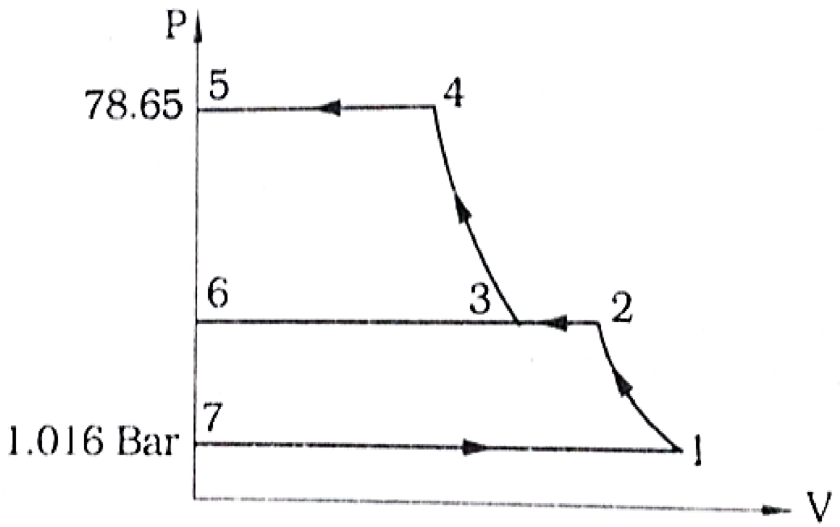
$\eta_{vol} = 80\% = 0.8$; n = 1.25

To find minimum power required by compressor

$$\text{w.k.t. Power} = WD/sec = \frac{2n}{n-1} . P_1 V'_1 \left[\left(\frac{P_d}{P_1} \right)^{\frac{n-1}{2n}} - 1 \right]$$

$$= \frac{2(1.25)}{1.25-1} (1.016 \times 10^2)(0.0666) \left[\left(\frac{78.65}{1.016} \right)^{\frac{1.25-1}{2(1.25)}} - 1 \right]$$

Power = 36.86 kW



Problem (11) contd.

w.k.t. $\eta_{mech} = \dfrac{\text{Indicated power}}{\text{actual power}}$

$\therefore$ actual power $= \dfrac{\text{Indicated power}}{\eta_{mech}} = \dfrac{36.86}{0.75}$

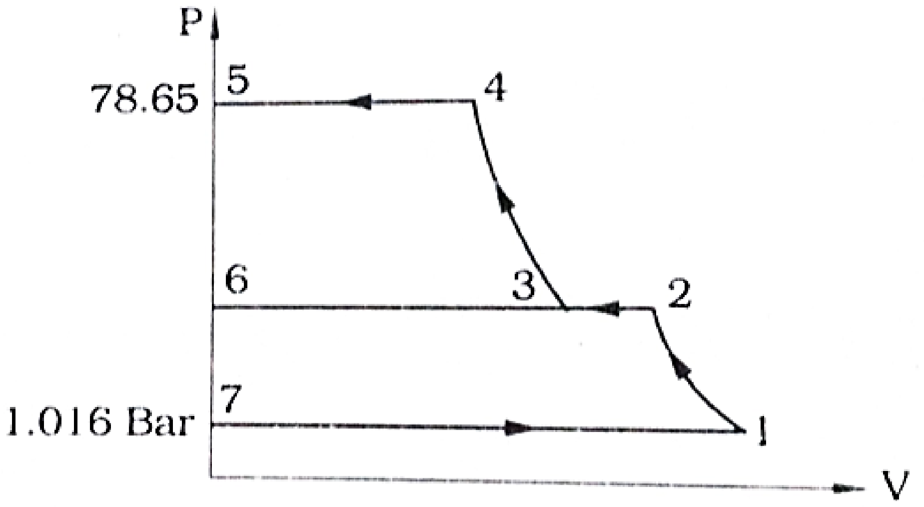
$\therefore$ actual power required to drive the compressor = **49.15 kW**

To find bore (d) & stroke (L) of compressor.

Given piston speed $= 2\,L\,N = 3$ m/sec

or $2 \times L \times 250$ rpm $= 3 \times 60$ m/min

$\therefore$ Stroke L $= 0.36$ m



Problem (11) contd.

Given volume of air handled = $V'_1 = 4\ m^3/min$ and $\eta_{vol} = 0.8$

From p-v diagram, $\eta_{vol} = \dfrac{V_1 - V_7}{V_S}$

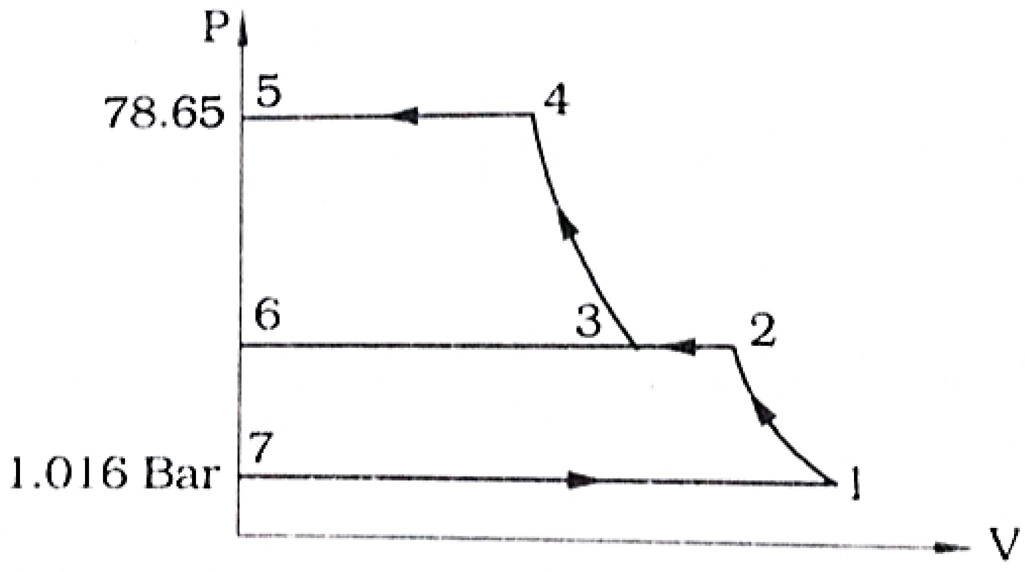
neglecting clearance $V_7 = 0$

Volume of air sucked/min $= V'_1 = (V_1 - V_7).\dfrac{N}{60}\ m^3/sec$

$$0.0666 = V_1\left(\frac{250}{60}\right)$$

$$V_1 = 0.01598\ m^3$$

$$0.8 = \frac{0.01598}{V_S} \qquad \therefore \quad V_S = 0.01998\ m^3$$

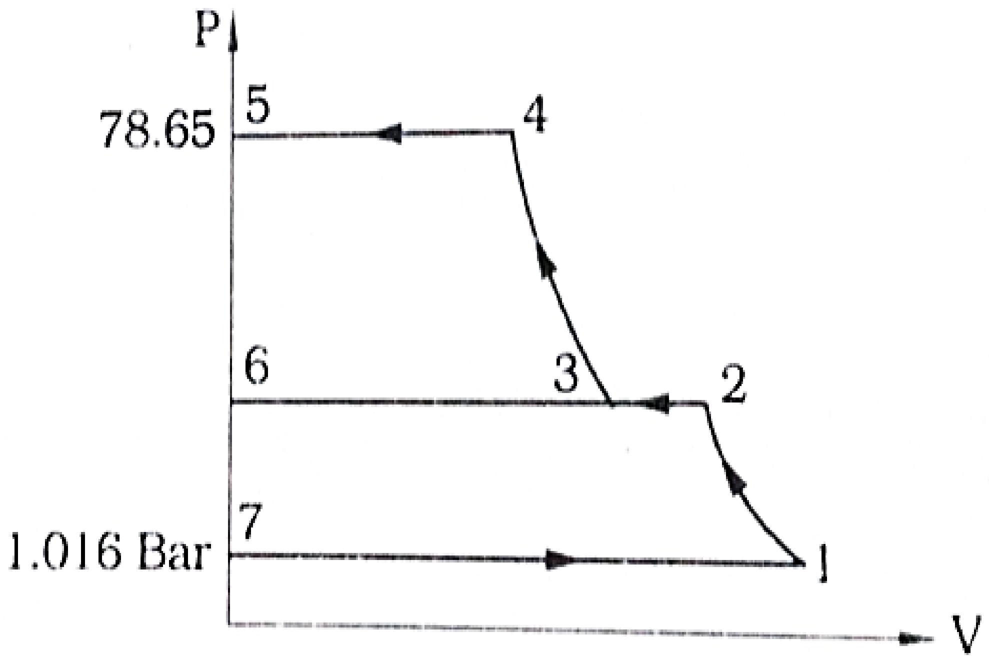


Problem (11) contd.

But stroke volume $= V_s = \frac{\pi}{4} d^2.L$

$$0.01998 = \frac{\pi}{4} d^2(0.36)$$

$d^2 = 0.07066$ or $d = 0.265$ m



12. A 2-stage compressor compresses air from 15^{O}C & 100 kPa to 6000 kPa. The air is cooled in the intercooler to 30^{O}C and the intermediate pressure is steady at 733 kPa. The low pressure cylinder is 10cm in diameter and the stroke for both cylinders is 11.25 cm. Assuming a compression law of $Pv^{1.35}$ =C and that the volume of air at atmospheric conditions drawn in per stroke is equal to the low pressure cylinder swept volume, find the power of the compressor when running at 250 rpm, Also find the diameter of high pressure cylinder.

2-stage = $i = 2$, $T_1 = 15°C = 288$ K, $P_1 = 100$ kPa = 1 Bar, $P_d = P_4 = 6000$ kPa = 60 Bar,
$T_3 = 30°$ C = 303 K, $P_2 = P_3 = 733$ kPa = 7.33 Bar, $d_{LP} = 10$ cm = 0.1 m

$$L_{LP} = L_{HP} = 11.25 \text{ cm} = 0.1125 \text{ m}$$
$$PV^n = PV^{1.35} = \text{const}, \; V_1 = (V_S)_{LP}, \; \text{Speed } N = 250 \text{ rpm}$$

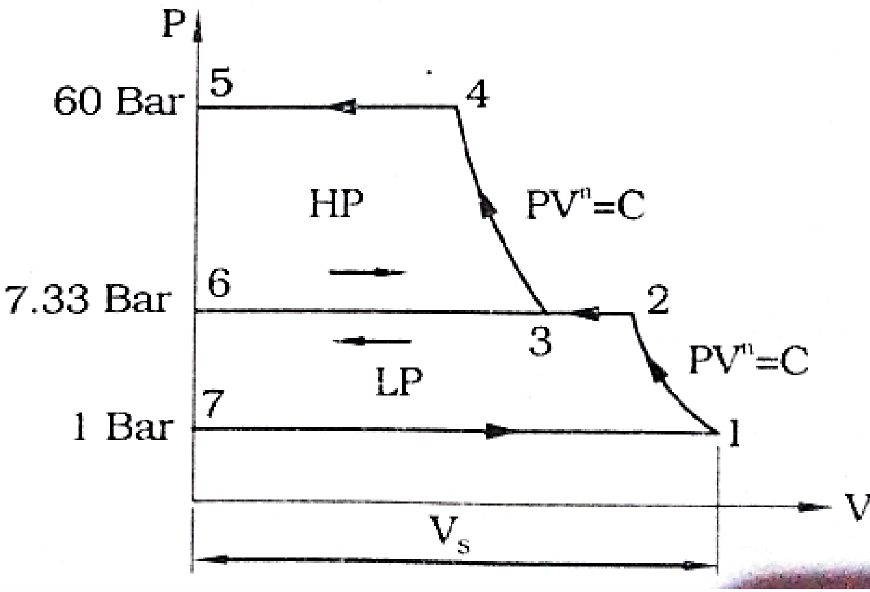
Step 1 To find power required by the compressor

w.k.t. Indicated power $= IP = W.D/sec = (WD/sec)_{LP} + (WD/sec)_{HP}$ ------(1)

$$= \left[\frac{n}{n-1} . m' R (T_2 - T_1)\right]_{LP} + \left[\frac{n}{n-1} . m' R (T_4 - T_3)\right]$$

mass m' will remain same for both the cylinders.

$$\therefore \quad IP = \frac{n}{n-1} . m' R [(T_2 - T_1) + (T_4 - T_3)] \quad \text{------(2)}$$



Problem (12) contd.

w.k.t. volume of air sucked/sec = $V'_1 = (V_1 - V_7)\dfrac{N}{60}$ m³/sec

$$\therefore V'_1 = \left(\frac{\pi}{4}d^2L\right)_{LP} \cdot \frac{250}{60}$$

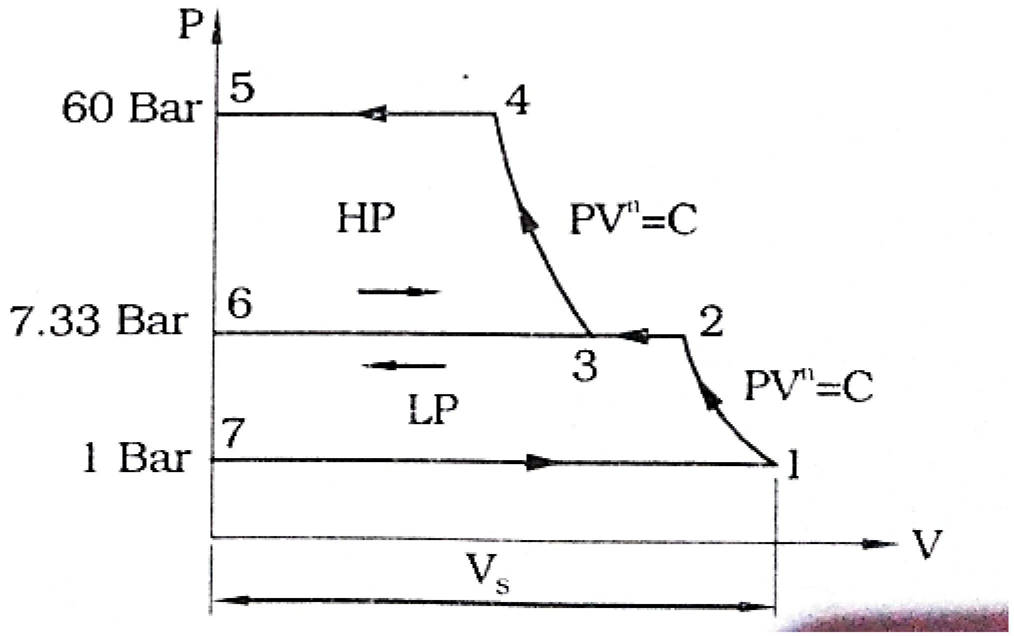
$$V'_1 = \left(\frac{\pi}{4} \times 0.1^2 \times 0.1125\right)\frac{250}{60} = 3.681 \times 10^{-3}$$

$$V'_1 = 3.681 \times 10^{-3} \text{ m}^3/\text{sec}$$

At condition (1), we have $P_1V'_1 = m' R T_1$

$$\therefore m' = \frac{P_1V'_1}{RT_1} = \frac{(1\times10^2)(3.681\times10^{-3})}{0.287(288)}$$

$$m' = 4.45 \times 10^{-3} \text{ kg/sec}$$



Problem (12) contd.

To find T_2 For polytropic process 1-2, we have $\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$

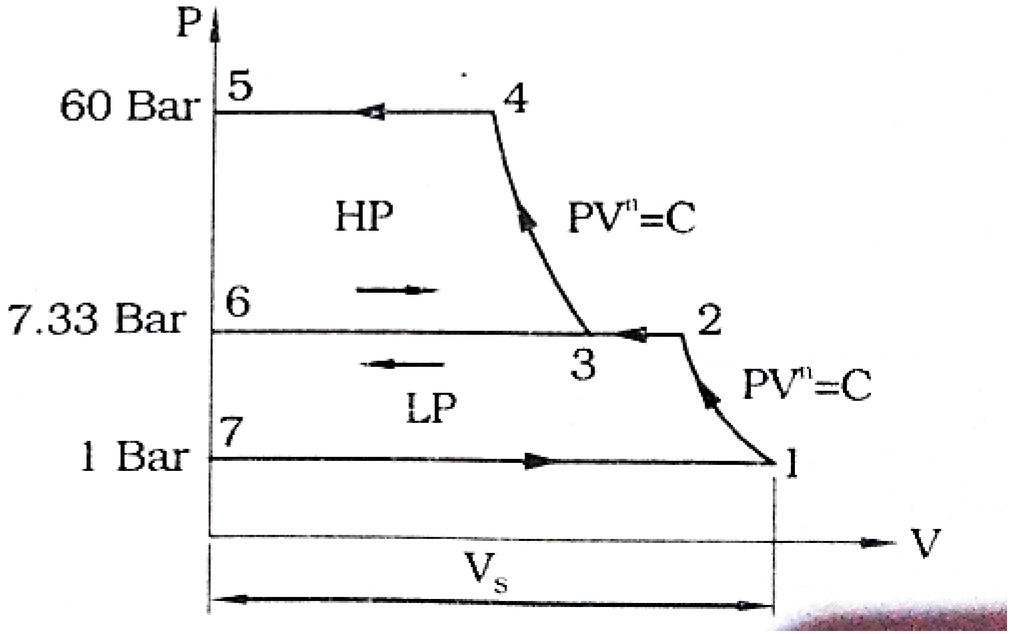
$$\frac{288}{T_2} = \left(\frac{1}{7.33}\right)^{\frac{1.35-1}{1.35}}$$

$T_2 = 482.7$ K

process 3-4, we have $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}}$

$$\frac{303}{T_4} = \left(\frac{7.33}{60}\right)^{\frac{1.35-1}{1.35}}$$

$T_4 = 522.58$ K



Problem (12) contd.

$$IP = \frac{1.35}{1.35-1}(4.45 \times 10^{-3})(0.287)[(482.7 - 288) + (522.58 - 303)]$$

$$\mathbf{IP = 2.04\ kW}$$

Step 2 To find d_{HP}

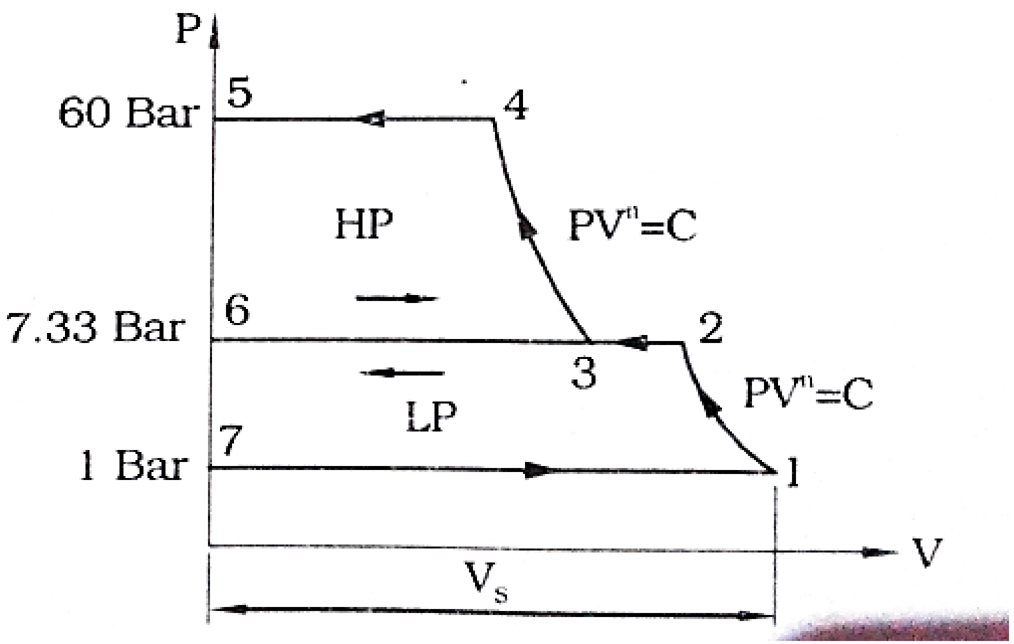
Since mass of air (m') will remain same for both the cylinders for steady state flow condition, we have $(m')_{LP} = (m')_{HP}$

$$\frac{P_1 (V_S)_{LP}}{RT_1} = \frac{P_3 (V_S)_{HP}}{RT_3}$$

$$\frac{1}{288}\left(\frac{\pi}{4}d^2L\right)_{LP} = \frac{7.33}{303}\left(\frac{\pi}{4}d^2L\right)_{HP}$$

$$\frac{\pi}{4}(0.1)^2(0.1125) = 6.967 \times \frac{\pi}{4}\, d^2_{HP}\,(0.1125)$$

$$d_{HP} = 0.0378\ m$$



13. A 3-stage compressor is used to compress air from 1 bar to 36 bar. The compression in all stage follows the law $Pv^{1.25}$ =C. The temperature of air at the inlet of compressor is 300K. Neglecting the clearance and assuming perfect intercooling, find the indicated power required in kW to deliver 15 m^3 of air per minute measured at inlet conditions and intermediate pressure also.

3–stage compressor = i = 3, P_1 = 1 Bar, $P_d = P_6$ = 36 Bar, $PV^n = PV^{1.25}$, T_1 = 300 K

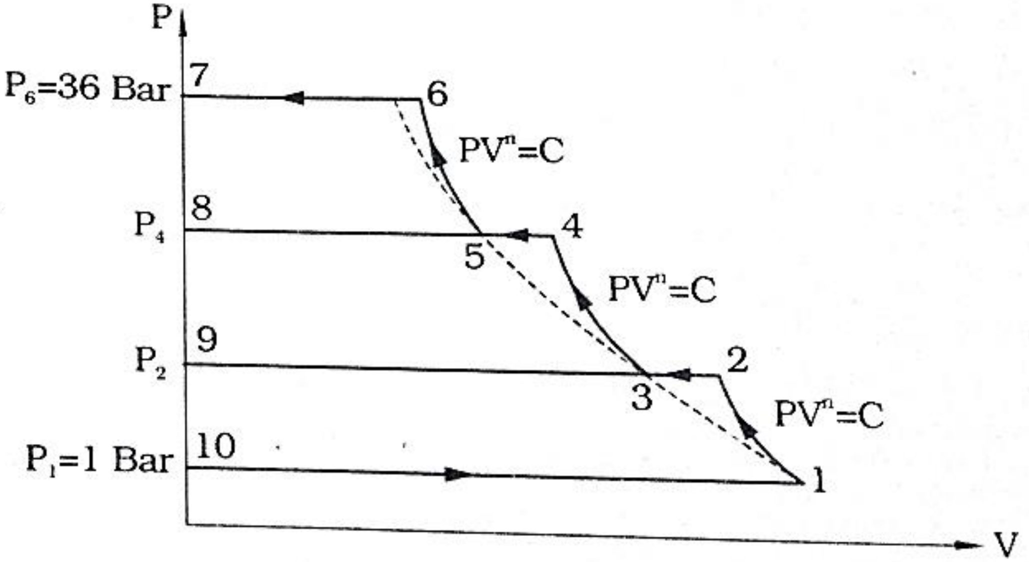
V'_1 = 15 m³/min = 0.25 m³/sec

w.k.t. $\text{Power} = \text{WD/sec} = \left[\frac{n}{n-1}.m'R(T_2 - T_1)\right] i$

or $\text{Power} = \text{WD/sec} = \frac{3n}{n-1} P_1 V'_1 \left[\left(\frac{P_d}{P_1}\right)^{\frac{n-1}{3n}} - 1\right]$

$\therefore P = \frac{3(1.25)}{1.25-1}(1\times10^2)(0.25)\left[\left(\frac{36}{1}\right)^{\frac{1.25-1}{3(1.25)}} - 1\right]$

$\therefore$ **Indicated power = P = 101.2 kW**



Problem (13) contd.

Step 2 To find intermediate pressure (P_2 & P_4)

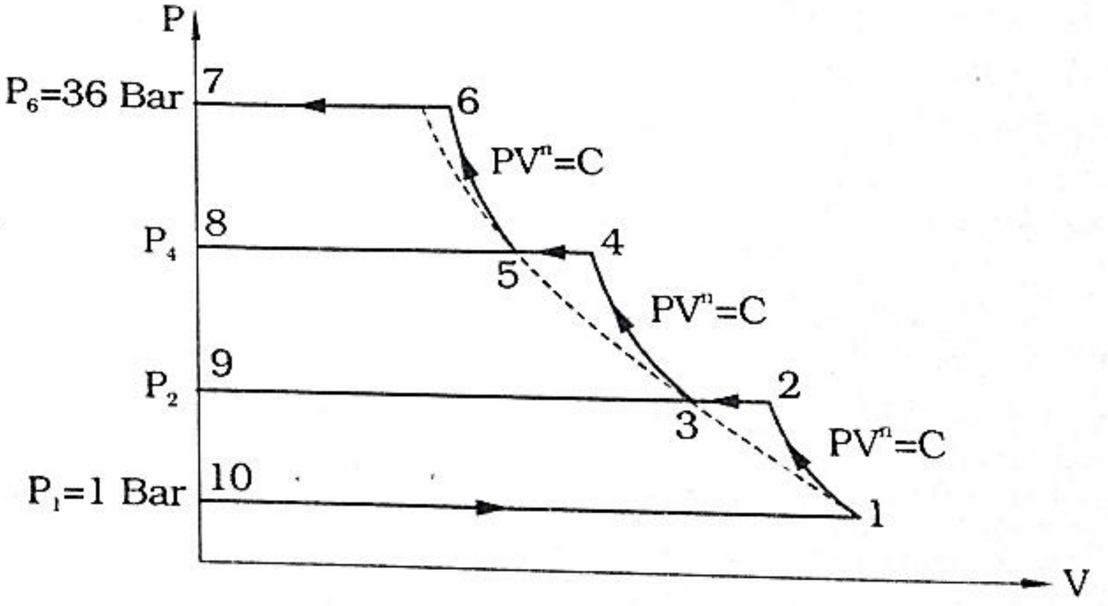
For perfect intercooling, we have pressure ratio

$$\frac{P_2}{P_1} = \frac{P_4}{P_2} = \frac{P_6}{P_4} = \left(\frac{P_6}{P_1}\right)^{\frac{1}{i}}$$ where i = no. of stage

Taking $$\frac{P_2}{P_1} = \left(\frac{P_6}{P_1}\right)^{\frac{1}{i}}$$

$$\frac{P_2}{1} = \left(\frac{36}{1}\right)^{\frac{1}{3}}$$

$P_2 = 3.302$ Bar $= P_3$

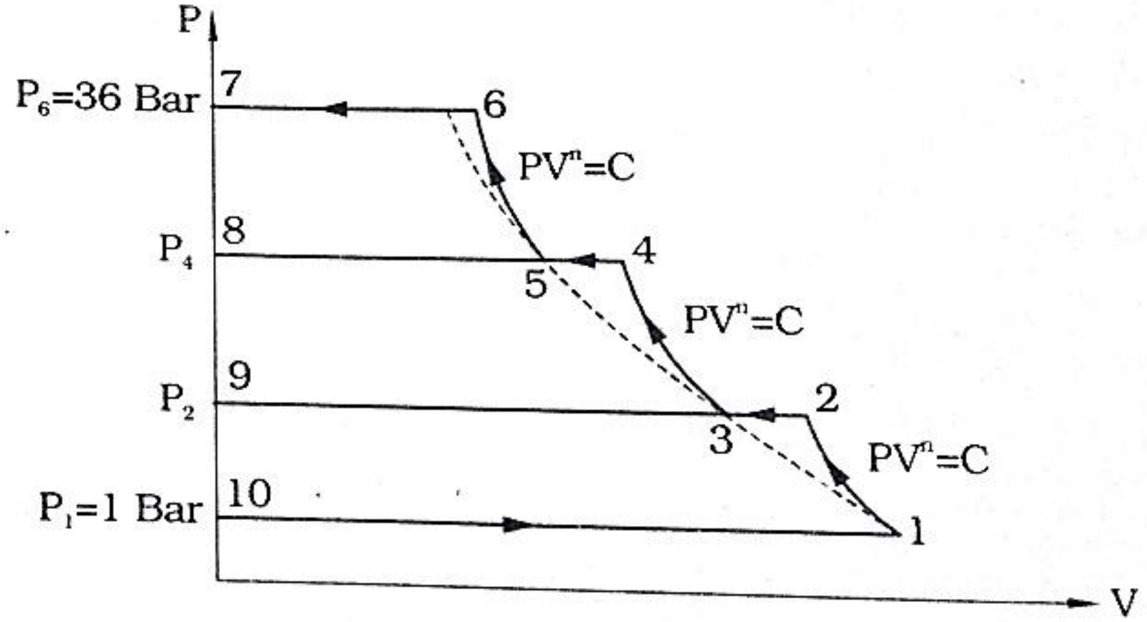


Problem (13) contd.

Taking $\frac{P_4}{P_2} = \left(\frac{P_6}{P_1}\right)^{\frac{1}{i}}$, ,

$$\frac{P_4}{3.302} = \left(\frac{36}{1}\right)^{\frac{1}{3}}$$

$P_4 = 10.9 \text{ Bar} = P_5$



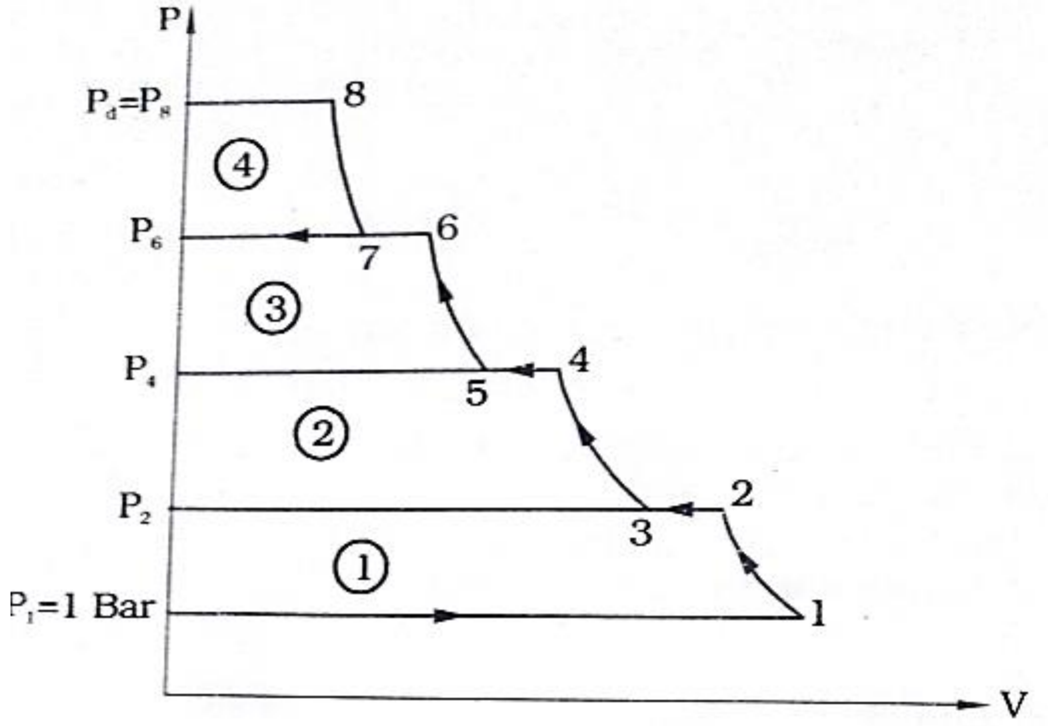
14. A multistage compressor is to be designed to elevate the pressure from 1 Bar to 120 Bar such that the stage pressure ratio will not exceed 4. Determine (a) number of stage (b) exact stage pressure ratio (c) intermediate pressure (d) the minimum power required to compress 15 m^3 /min of free air. Take n=1.2

inlet pressure = P_1 = 1 Bar and delivery pressure P_d = 120 Bar

stage pressure ratio $= \frac{P_2}{P_1} = \frac{P_4}{P_2} = \frac{P_6}{P_4} = \left(\frac{P_d}{P_1}\right)^{\frac{1}{i}} < 4$

we have $\left(\frac{P_d}{P_1}\right)^{\frac{1}{i}} = 4$ $\quad \left(\frac{120}{1}\right)^{\frac{1}{i}} = 4$

$i = 3.45 \cong 4$ $\quad$ ∴ **no. of stages = 4**



Step 2 To find exact stage pressure ratio

From equation (1), we have $\left(\frac{P_d}{P_1}\right)^{\frac{1}{i}} = \left(\frac{120}{1}\right)^{\frac{1}{4}}$

Problem (14) contd.

To find intermediate pressures P_6 P_4 & P_2

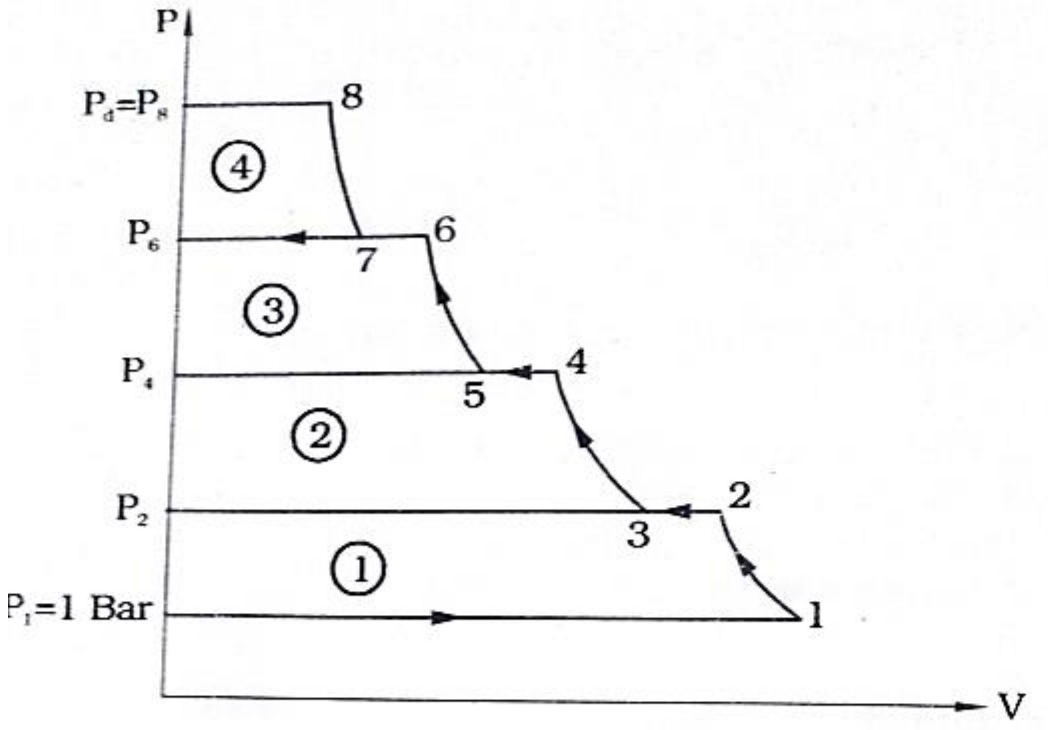
we have $\dfrac{P_2}{P_1} = \left(\dfrac{P_d}{P_1}\right)^{\frac{1}{i}}$

$$\frac{P_2}{1} = \left(\frac{120}{1}\right)^{\frac{1}{4}}$$

$$P_2 = 3.31 \text{ Bar} = P_3$$

$$\frac{P_4}{P_2} = \left(\frac{P_d}{P_1}\right)^{\frac{1}{i}}$$

$$\therefore \quad P_4 = 10.95 \text{ Bar} = P_5$$



Problem (14) contd.

$$\frac{P_6}{P_4} = \left(\frac{P_d}{P_1}\right)^{\frac{1}{4}}$$

$$\mathbf{P_6 = 36.26\ Bar = P_7}$$

To find power required to compress 15m³/min of air

$$\text{w.k.t. Power} = \text{WD/sec} = \frac{4n}{n-1}.P_1V'_1\left[\left(\frac{P_d}{P_1}\right)^{\frac{n-1}{4n}} - 1\right]$$

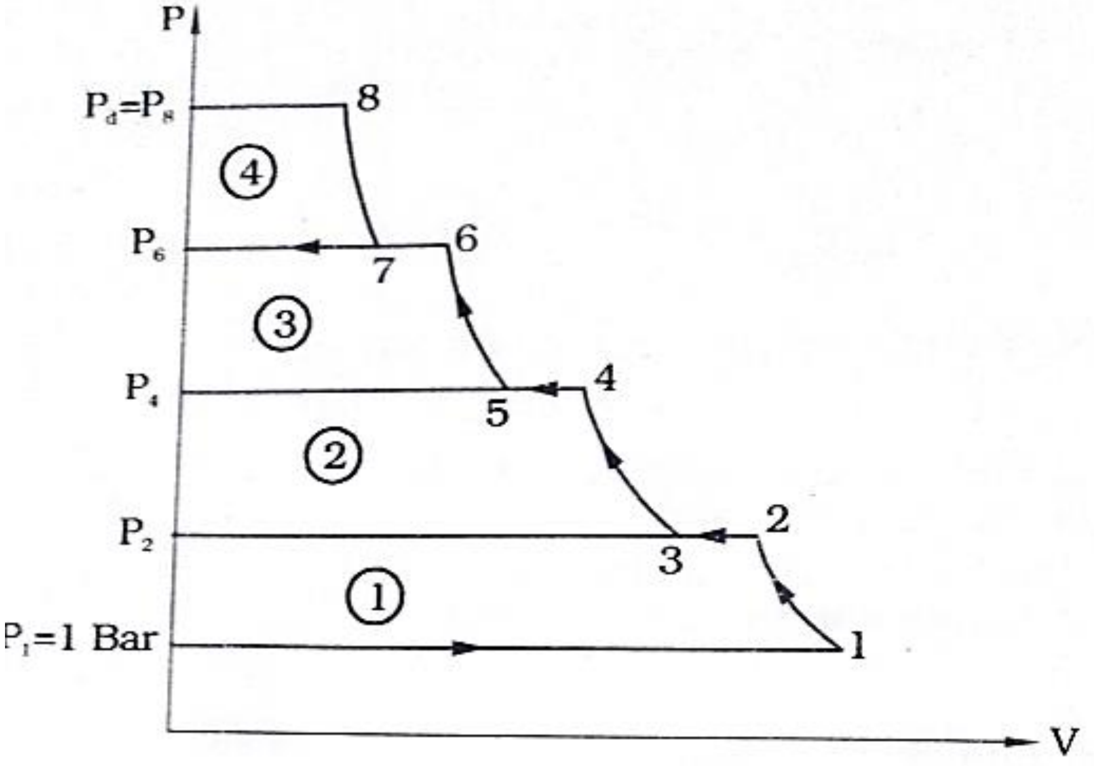
$$= \frac{4(1.2)}{1.2-1}(1 \times 10^2)(0.25)\left[\left(\frac{120}{1}\right)^{\frac{1.2-1}{4(1.2)}} - 1\right]$$

∴ **Power = P = 132.46 kW**

$\mathbf{P_2 = P_3 = 3.31\ Bar}$

$\mathbf{P_4 = P_5 = 10.95}$ and

$\mathbf{P_6 = P_7 = 36.26\ Bar}$



Thermo-Fluids Engineering 21ME52

ATME College of Engineering

Module-2

PSCYCHROMETRY AND AIR-CONDITIONING SYSTEMS

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PSCYCHROMETRY AND AIR-CONDITIONING SYSTEMS

- Psychrometric properties of Air
- Psychrometric Chart
- Analyzing Air-conditioning Processes;
  - Heating and Cooling
  - Dehumidification and Humidification,
  - Evaporative Cooling.
- Adiabatic mixing of two moist air streams.
- Cooling towers.

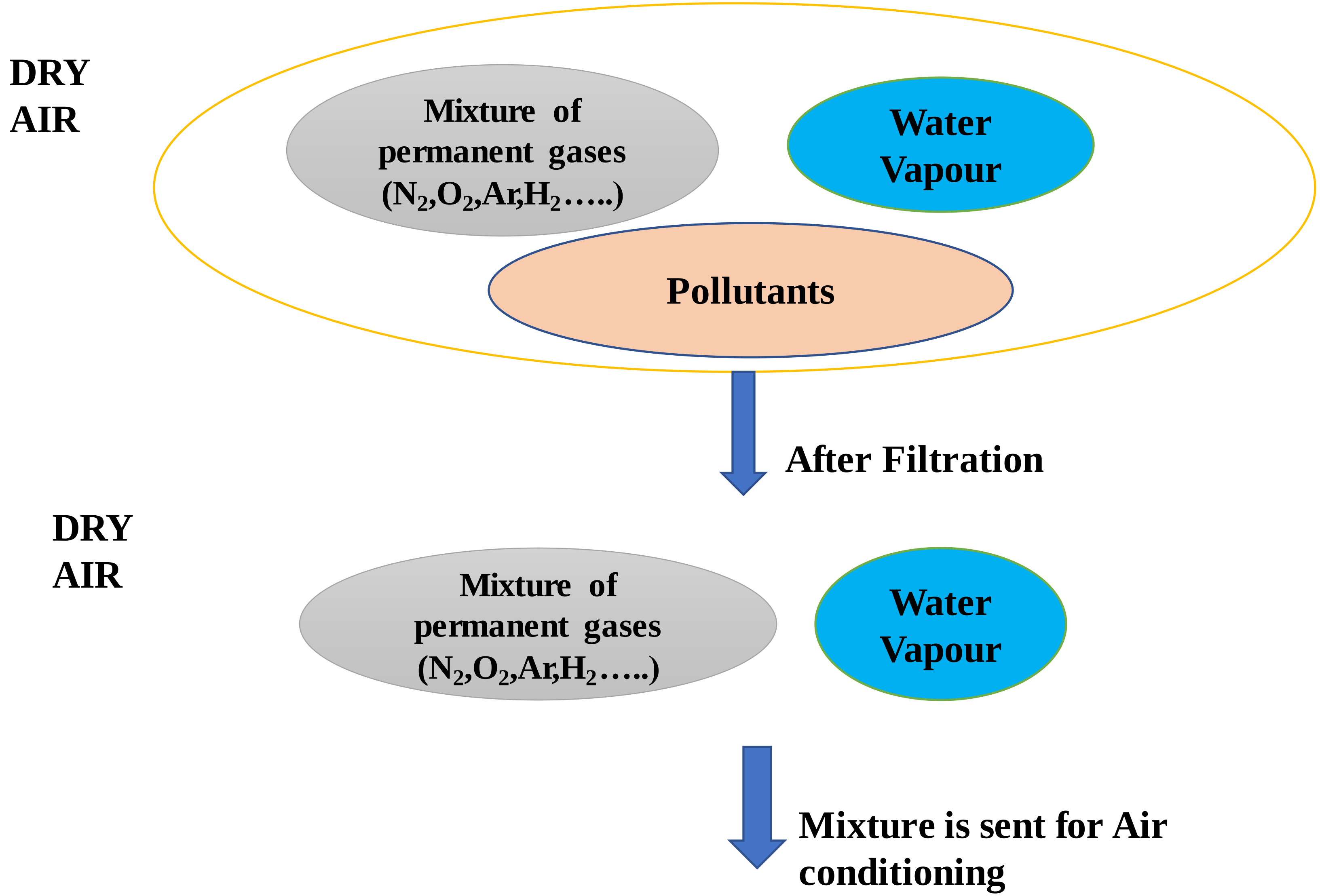
PSCYCHROMETRY

Psychrometry?

It is the study of properties of the **moist air** or **mixture of dry air and water vapour**

- Atmospheric air contains 78.09% Nitrogen, 20.95% oxygen, 0.93% argon, 0.04% carbon dioxide, and small amounts of other gases. Air also contains a variable amount of water vapour, on average around 1% at sea level, and 0.4% over the entire atmosphere

COMPOSITION OF ATMOSPHERIC AIR



Psychrometric properties of air

Dry Bulb Temperature (DBT) (t_{db})

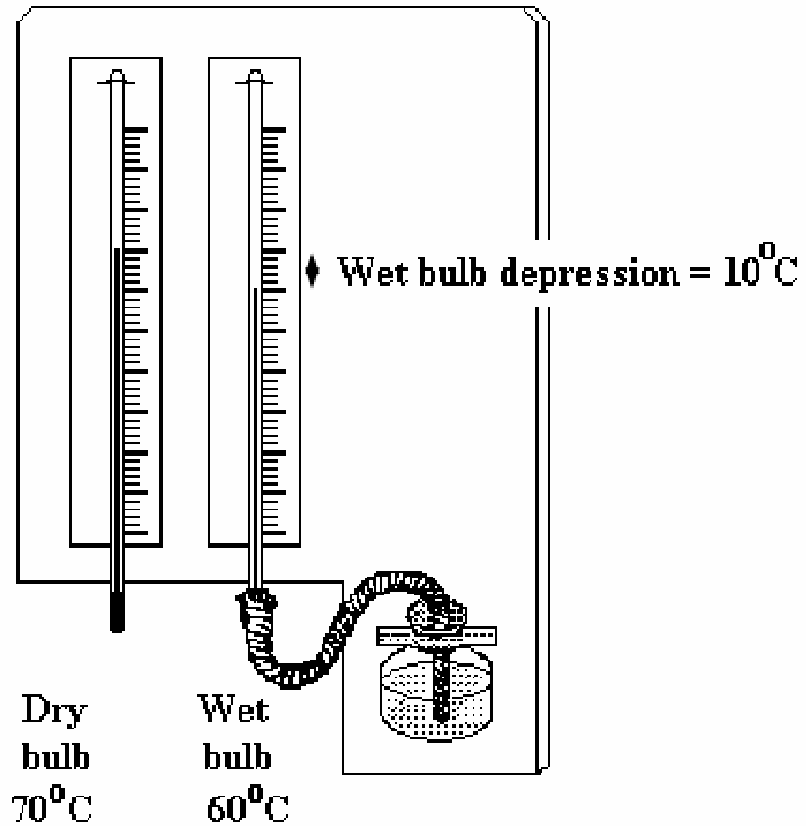
The dry-bulb temperature of air is measured by a thermometer which is freely exposed to the air.

Wet Bulb Temperature(WBT) (t_{wb})

The Wet-bulb Temperature of air is measured by a thermometer whose bulb is covered by a wetted wick which is kept moist with distilled and clean water.

❖**The difference between the wet and dry bulb temperatures is called the Wet Bulb Depression.**

Note : $t_{db} > t_{wb}$



Psychrometric properties of air

- **Saturated air:** When air contains the maximum amount of water vapour, which it can hold at a particular temperature and pressure then it is called as **saturated air.**
- **Unsaturated air:** When air contains water vapour less than its maximum capacity or saturated condition at a particular temperature and pressure then it is said to be **Unsaturated air**
- **Super saturated air:** In some cases air contains water vapour more than the amount necessary for saturation at a particular temperature. Then the air is said to be **super saturated air.**

Psychrometric properties of air

✓ Dew point Temperature (DPT) (t_{dp})

- Dew point temperature is the temperature at which moist air becomes saturated (100% relative humidity) with water vapour when cooled at constant pressure, i.e. temperature at which condensation of water vapour begins when the moist air is cooled.
- **ABSOLUTE HUMIDITY OR SPECIFIC HUMIDITY OR HUMIDITY RATIO**:
- The ratio of mass of water vapour per unit mass of dry air is called absolute humidity
- **Mass of moist air/air (m)= Mass of dry air (m_a) + mass of water vapour(m_v)**
- **Absolute Humidity (ω) = $\frac{m_v}{m_a}$**

Psychrometric properties of air

- **Pressure of moist air(P) = Partial pressure of dry air(P_a) + partial pressure of vapour(P_v)**
- **RELATIVE HUMIDITY:**
- It is the ratio of amount water vapour actually present in the air to the amount of water vapour necessary to saturate it at that temperature. It is expressed as percentage.
- The ratio of the vapour pressure of the mixture at a given temperature to the saturation vapour pressure of the mixture at the same temperature.
- **Relative Humidity = $\phi = \left[\frac{p_v}{p_{vsat}}\right]_T$**

Psychrometric properties of air

DEGREE OF SATURATION OR SATURATION RATIO:

The ratio of absolute humidity of the air to the absolute humidity of the saturated air at the same temperature.

$$\mu = \left[\frac{\omega}{\omega_{sat}}\right]_T$$

ENTHALPY OF MOIST AIR:

The enthalpy of moist air is defined as the sum of its internal energy and the product of its pressure and volume. Specific enthalpy h (kJ/kg) of moist air is defined as the total enthalpy of the dry air and water vapour mixture per kg of moist air.

$$\mathbf{H = H_a + H_v = m_a h_a + m_v h_v}$$

SPECIFIC HUMIDITY

$$\text{specific humidity as } \omega = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}} = \frac{m_v}{m_a} \quad \text{-----(1)}$$

Assuming dry air to behave like a perfect gas, we have

$$P_a V = m_a R_a T$$

where P_a = Partial pressure of dry air
V = Volume of mixture (both masses occupy volume V)
R_a = Gas constant for dry air
T = Temperature of the mixture

$$\therefore \quad m_a = \frac{P_a V}{R_a T} \quad \text{-----(2)}$$

SPECIFIC HUMIDITY

Similarly $m_v = \dfrac{P_V V}{R_V T}$ -----(3)

where R_a = Gas constant for air = 0.287 kJ/kg K

R_v = Gas constant for vapour = 0.461 kJ/kg K

Substituting equations (2) & (3) in (1),

we have $$\omega = \dfrac{\dfrac{P_V V}{R_V T}}{\dfrac{P_a V}{R_a T}} = \dfrac{P_V R_a}{P_a R_V} = \dfrac{P_V (0.287)}{P_a (0.461)}$$

w.k.t. total pressure $P = P_a + P_V$

$\therefore P_a = P - P_V$

specific humidity $$\omega = 0.622 \dfrac{P_V}{P - P_V}$$

Degree of saturation

$$\text{degree of saturation } \mu = \frac{\text{Actual specific humidity}}{\text{Saturated specific humidity}} = \frac{\omega}{\omega_S}$$

where $\omega = 0.622 \dfrac{P_V}{P - P_V}$ $\omega_s = 0.622 \dfrac{P_{VS}}{P - P_{VS}}$

where P = total pressure

P_V = Partial pressure of water vapour

P_{VS} = Partial pressure of water vapour when air is fully saturated.

Degree of saturation

$$\therefore \ \mu = \frac{0.622\dfrac{P_V}{P-P_V}}{0.622\dfrac{P_{VS}}{P-P_{VS}}} = \frac{P_V(P-P_{VS})}{P_{VS}(P-P_V)} = \phi\,\frac{\left[1-\dfrac{P_{VS}}{P}\right]}{\left[1-\dfrac{P_V}{P}\right]}$$

where ϕ = Relative humidity = $\dfrac{P_V}{P_{VS}}$

$$\text{Thus } \mu = \phi\,\frac{\left[1-\dfrac{P_{VS}}{P}\right]}{\left[1-\dfrac{P_V}{P}\right]} \qquad \text{-----(1)}$$

'v_s in

$$\mu = \phi\,\frac{\left[1-\dfrac{P_{VS}}{P}\right]}{\left[1-\dfrac{P_V}{P_{VS}}\dfrac{P_{VS}}{P}\right]} = \phi\,\frac{\left[1-\dfrac{P_{VS}}{P}\right]}{\left[1-\phi\dfrac{P_{VS}}{P}\right]}$$

Degree of saturation

$$\mu = \frac{\phi - \phi \frac{P_{VS}}{P}}{1 - \phi \frac{P_{VS}}{P}}$$

$$\mu - \mu \phi \frac{P_{VS}}{P} = \phi - \phi \frac{P_{VS}}{P}$$

$$\mu = \phi - \phi \frac{P_{VS}}{P} + \mu \phi \frac{P_{VS}}{P} = \phi \left[1 - \frac{P_{VS}}{P} + \mu \frac{P_{VS}}{P} \right]$$

$$\mu = \phi \left[1 - \frac{P_{VS}}{P} (1 - \mu) \right]$$

Enthalpy of moist air

enthalpy of moist air = (Enthalpy of dry air) + (Enthalpy of water vapour)

$$m\,h = m_a\,h_a + m_v\,h_v \qquad \text{-----(1)}$$

$$\therefore\; h = \frac{m_a h_a + m_v h_v}{m} \text{ per unit mass of moist air.}$$

Then, enthalpy of moist air per unit mass of dry air (m_a) is written as

$$h = \frac{m_a h_a + m_v h_v}{m_a}$$

$$h = h_a + \frac{m_v}{m_a} h_v$$

Enthalpy of moist air

w.k.t. specific humidity $= \omega = \dfrac{m_V}{m_a}$

$$\therefore \text{ enthalpy } h = h_a + \omega h_V$$

$$= (C_{Pa} t_{db}) + \omega h_V$$

but C_{Pa} = Specific heat of dry air = 1.005 kJ/kg K

$$\therefore h = 1.005\, t_{db} + \omega h_V \quad \text{-----(1)}$$

w.k.t. enthalpy of vapour (superheated) $= h_V = h_g + C_{PS}(t_{db} - t_{dP})$

where C_{PS} = Specific heat of superheated vapour = 1.88 kJ/kg K

$$\therefore h_V = h_g + 1.88\,(t_{db} - t_{dP}) \quad \text{-----(2)}$$

Substituting equation (2) in (1),

$$h = 1.005\, t_{db} + \omega\,[h_g + 1.88\,(t_{db} - t_{dP})]$$

Problems without using the psychrometric chart

1. Moist air at 30˚C, 1.01325 bar has a relative humidity of 80%. Determine without using the psychrometric chart (a) partial pressure of water vapour () specific humidity (c) specific volume (d) dew point temperature

Data Given:

Temperature of moist air = DBT = t_{db} = 30°C

Atmospheric pressure or Total pressure = P = 1.01325 Bar = 101.325 kPa

Relative Humidity RH = ϕ = 80% = 0.8

To find partial pressure of water vapour (P_v) and air (P_a)

w.k.t. Relative humidity $\phi = \dfrac{P_v}{P_{vs}}$

where P_v = Partial pressure of water vapour (kPa)

P_{vs} = Partial pressure of water vapour when air is fully saturated (kPa)

From steam tables, at DBT = 30°C, we have P_{vs} = 0.042415 Bar = 4.2415 kPa

Problem (1)

$$0.8 = \frac{P_v}{4.2415}$$

$$P_v = 3.39 \text{ kPa}$$

w.k.t. total pressure $P = P_a + P_v$

Partial pressure of air $= P_a = P - P_v$

$$= 101.325 - 3.39 = 97.93 \text{ kPa}$$

w.k.t. Specific humidity $\omega = \dfrac{0.622 P_v}{P_a} = \dfrac{0.622(3.39)}{97.93} = 0.0215$

ω = **0.0215 kg vap/kg of dry air**

Problem (1)

From ideal gas equation, we have PV = mRT

$$\text{i.e.,} \quad P_a V_a = m_a R T_a$$

$$V_a = \frac{m_a R T_a}{P_a} = \frac{1 \times 0.287 \times (30 + 273)}{97.93\ \text{kPa}}$$

$$\mathbf{V_a = 0.888\ m^3/kg\ of\ dry\ air}$$

Hence, from steam tables, corresponding to pressure P_v = 3.39kPa (0.0339 Bar), we have by interpolation $\mathbf{t_{sat} = DPT = 26°C}$

2. Moist air at 35°C has a dew point of 15°C. Calculate its relative humidity, specific humidity and enthalpy. Take Cpv= 1.88 kJ/kg K.

Data Given:

Temperature of moist air $= DBT = t_{db} = 35°C$

Dew point temperature $= DPT = t_{dp} = 15°C$

To find relative humidity (ϕ)

$$\text{w.k.t. relative humidity} = \phi = \frac{P_v}{P_{vs}}$$

From steam tables, corresponding to DBT = 35°C, we have

$$P_{vs} = 0.056216 \text{ Bar} = 5.6216 \text{ kPa}$$

corresponding to DPT = 15°C, we have $P_v = 0.017039 \text{ Bar} = 1.7039 \text{ kPa}$

Problem (2)

$$\phi = \frac{1.7039}{5.6216} = 0.303$$

relative humidity = ϕ = 30.3 %

$$\text{w.k.t.}\quad \omega = \frac{0.622P_v}{P_a} = \frac{0.622P_v}{P - P_v}$$

Assume Total Pressure of air = P = 1.01325 Bar or 101.325 kPa

$$\omega = \frac{0.622(1.7039)}{101.325 - 1.7039} = 0.01063$$

ω = 0.01063 kg vapour/kg of dry air

Problem (2)

To find enthalpy (h)

w.k.t. enthalpy $h = 1.005\, t_{db} + \omega(2500 + 1.88\, t_{db})$

$= 1.005\,(35) + 0.01063\,[2500 + 1.88\,(35)]$

$h = 62.45$ kJ/kg of air

3. The dry and the wet bulb temperatures of atmosphere air at 1 atm (101.325 kPa) pressure are measured with a sling psychrometer and determined to be 25°C and 15°C respectively. Determine - i) Specific humidity ii) relative humidity iii) The enthalpy of air. Use properties of table only, without using psychrometric chart

Data Given:

$DBT = t_{db} = 25°C$; $WBT = t_{wb} = 15°C$ Total pressure of air $= P = 101.325$ kPa

w.k.t. Specific humidity $= \omega = \dfrac{0.622\,P_V}{P_a} = \dfrac{0.622\,P_V}{P - P_V}$ -----(1)

when DBT & WBT are given. P_v is calculated using

$$P_V = (P_V)_{Wb} - \frac{\left[P - (P_{Vs})_{\omega b}\right](t_{db} - t_{\omega b})}{1547 - 1.44\,t_{\omega b}}$$

where t_{db} & t_{wb} are in Kelvin & Pressure in kPa

Problem (3)

From steam tables, corresponding to WBT = 15°C, we have

$$(P_{vs})_{wb} = 0.017039 \text{ Bav} = 1.7039 \text{ kPa}$$

equation (2) becomes, $P_v = 1.7039 - \dfrac{[101.325 - 1.7039](25 - 15)}{1547 - 1.44(15)}$

$$= 1.0508 \text{ kPa}$$

Now equation (1) becomes, $\omega = \dfrac{0.622(1.0508)}{101.325 - 1.0508} = 0.00651$

$$\omega = \mathbf{0.00651 \text{ kg vap/kg of dry air}}$$

Problem (3)

To find relative humidity (ϕ)

$$\text{w.k.t. } \phi = \frac{P_V}{P_{Vs}} \quad \text{-----(3)}$$

From steam tables, corresponding to DBT = 25°C, we have

$$P_{vs} = 0.03166 \text{ Bar} = 3.166 \text{ kPa}$$

$$\text{equation (3) becomes, } \phi = \frac{1.0508}{3.166} = 0.3319$$

$$\text{relative humidity} = \phi = \mathbf{33.2\%}$$

To find enthalpy of air (h)

$$\text{w.k.t. } h = 1.005\, t_{db} + \omega(2500 + 1.88\, t_{db})$$

$$= 1.005\,(25) + 0.00651[2500 + 1.88\,(25)] \;\; \mathbf{= 41.7\ kJ/kg\ of\ air}$$

Problem (3)

To find DPT (t_{dp})

From steam tables, corresponding to pressure P_V= 1.0508 kP_a (0.010508 Bar),

DTP = t_{dP} = 7.5°C

4. A room measures 5m x 5m x 3m. It contains atmospheric air at 100kPa, DBT =30°C and relative humidity = 30%. Find the mass of dry air and the mass of associated water vapour in the room. Solve the problem without the use of psychrometric chart and using the properties of water vapour from the steam tables.

Data Given:

Volume of air in the room $= 5 \times 5 \times 3 = 75\ m^3$

$DBT = t_{db} = 30°C$

Atmospheric air pressure or Total Pressure $= P = 100$ kPa (or 1Bar); Relative humidity $= \phi = 30\% = 0.3$

To find mass of dry air & mass of vapour

we have $P_v V_v = m_v R_v T_v$

$$m_v = \left(\frac{PV}{RT}\right)_v \quad \text{where} \quad R_v = \frac{\bar{R}}{M_v}$$

$\bar{R}$ = universal gas constant = 8.314

M_v = molar mass of water (H_2O) = 18

Problem (4)

$$R = \frac{8.314}{18} = 0.462 \text{ kJ/kg K}$$

w.k.t. relative humidity $\phi = \frac{P_V}{P_{VS}}$

$$\therefore \quad P_V = \phi P_{VS}$$

From steam tables, at DBT = 30°C, we have P_{VS} = 0.042415 Bar = 4.2415 kPa

$$P_V = 0.3\,(4.2415) = 1.272 \text{ kPa}$$

$$m_V = \frac{(1.272)(75)}{(0.462)(30+273)}$$

mass of vapour = $\mathbf{m_V = 0.681\ kg}$

Problem (4)

To find mass of dry air (m_a)

From ideal gas equation, $P_a V_a = m_a R_a T_a$

$$\therefore \quad m_a = \left(\frac{PV}{RT}\right)_a$$

where R_v = Gas constant for air = 0.287 kJ/kg K

w.k.t. $P = P_a + P_v$

$$P_a = P - P_v = 100 - 1.272 = 98.728 \text{ kPa}$$

$$m_a = \frac{(98.728)(75)}{(0.287)(30+273)}$$

$$\mathbf{m_a = 85.14\ kg}$$

5. Atmospheric air at 101.325 kPa has 30°C DBT and 15° C DPT. Without using the psychrometric chart, using the property values from the tables, calculate: (i) Partial pressures of air and water vapour (ii) Specific humidity (ii) Relative humidity (iv) Vapour density (v) Enthalpy of moist air.

Data Given: Atmospheric pressure P = 101.325 kPa (or 1.01325 Bar)

$$DBT = t_{db} = 30° C; \; DPT = t_{dp} = 15° C$$

To find P_a and P_v

Corresponding to DBT = 30° C, we have $P_{vs} = 0.042415$ Bar or 4.2415 kPa

Corresponding to DPT = 15°C, we have $P_v = 0.017039$ Bar or 1.7039 kPa

w.k.t. total presure $P = P_a + P_v$

$$P_a = P - P_v = 101.325 - 1.7039 = 99.6211 \text{ kPa}$$

Thus $\mathbf{P_a = 99.6211}$ **kPa** & $\mathbf{P_v = 1.7039}$ **kPa**

Problem (5)

To find specific humidity (ω)

w.k.t. Specific humidity $\omega = \dfrac{0.622P_V}{P_a} = \dfrac{0.622(1.7039)}{99.6211}$

ω = **0.01068 kg vap/kg of dry air.**

To find relative humidity (ϕ)

w.k.t. $\phi = \dfrac{P_V}{P_{VS}} = \dfrac{1.7039}{4.2415} = 0.40$

ϕ = **40%**

Problem (5)

To find vapour density (ρ)

w.k.t. vapour density $(\rho_v) = \dfrac{\text{mass of vapour } (m_V)}{\text{volume of vapour } (V_V)}$ kg/m³

$$\rho_V = \frac{m_V}{V_V}$$

Treating vapour as an ideal gas, we have $P_vV_v = m_vR_vT_v$

$$\text{where } R_v = \frac{\overline{R}}{M_V}$$

$\overline{R}$ = universal gas constant = 8.314 kJ/kg mol-K

M_v = molar mass of water = 18

$$R = \frac{8.314}{18} = 0.4618 \text{ kJ/kg K}$$

Problem (5)

$$P_vV_v = m_v(0.4618)\,T_v$$

$$\frac{m_v}{V_v} = \frac{P_v}{0.4618\,T_v}$$

Using equation(1),

we have $\rho_v = \dfrac{P_v}{0.4618\,T_v}$

$$\rho_v = \frac{1.7039}{0.4618(30+273)}$$

$$\rho_v = \mathbf{0.01217\ kg/m^3}$$

w.k.t. $h = 1.005\,t_{db} + \omega(2500 + 1.88\,t_{db})$

$= \mathbf{57.45\ kJ/kg\ of\ dry\ air}$

6. Atmospheric air has a DBT of 32° C and a WBT of 26° C. Determine the following properties for the given condition without using psychrometric chart. (a) Partial pressure of water vapour (b) Specific humidity (c) dew point temperature (d) relative humidity (e) degree of saturation (f) density of air in the mixture (g) density of vapour in the mixture and (h) Enthalpy of the mixture.

Data Given:

Assuming pressure of air $= P = 101.325$ kPa

DBT $= t_{db} = 32°C$; WBT $= t_{wb} = 26°C$

To find partial pressure of water vapour (P_v)

When DBT & WBT are given, P_v is calculated using

$$P_v = (P_{vs})_{wb} - \frac{\left[P - (P_{vs})_{wb}\right](t_{db} - t_{wb})}{1547 - 1.44\, t_{wb}}$$

Problem (6)

From steam tables, at WBT = $t_{wb} = 26°C$, we have $(P_{vs})_{wb} = 0.0336$ Bar = 3.36 kPa

$$P_v = 3.36 - \frac{[101.325 - 3.36](32 - 26)}{1547 - 1.44(26)}$$

$$\mathbf{P_v = 2.97\ kPa}$$

To find specific humidity (ω)

$$\text{w.k.t.}\ \omega = \frac{0.622 P_V}{P_a} = \frac{0.622 P_V}{P - P_V} = \frac{0.622(2.97)}{101.325 - 2.97}$$

specific humidity $= \omega =$ **0.0187 kg vap/kg of dry air**

Problem (6)

To find DPT (t_{dP})

From steam tables, corresponding to P_v = 2.97 kPa (0.0297 Bar),

$$DPT = t_{dp} = 23.9°C$$

To find relative humidity (ϕ)

$$\text{w.k.t.} \quad \phi = \frac{P_V}{P_{VS}}$$

From steam tables, corresponding to DBT = 32°C, we have

$$P_{vs} = 0.0476 \text{ Bar or } 4.76 \text{ kPa}$$

$$\phi = \frac{2.97}{4.76} = 0.624$$

Problem (6)

To find degree of saturation

w.k.t. degree of saturation $= \mu = \frac{P_V}{P_{VS}}\left(\frac{P - P_{VS}}{P - P_V}\right)$

$$= \frac{2.97}{4.76}\left(\frac{101.325 - 4.76}{101.325 - 2.97}\right)$$

μ = **0.618**

Problem (6)

To find Density of air in the mixture

Under ideal gas condition, we have $P_aV_a = m_aRT_a$

$$\therefore \quad \frac{m_a}{V_a} = \frac{P_a}{RT_a}$$

$$\rho_a = \frac{P_a}{RT_a} = \frac{(P - P_V)}{0.287(32+273)} = \frac{(101.325-2.97)}{87.535}$$

density of air $= \rho_a =$ **1.12 kg/m³ dry air**

To find density of vapour (ρ_V)

we have $P_VV_V = m_VR_VT_V$

$$\frac{m_V}{V_V} = \frac{P_V}{R_VT_V} \text{ or } \rho_V = \frac{P_V}{R_VT_V}$$

$$\rho_V = \frac{2.97}{0.462(305)}$$

ρ_V = **0.021 kg vapour/m³ dry air**

Problem (6)

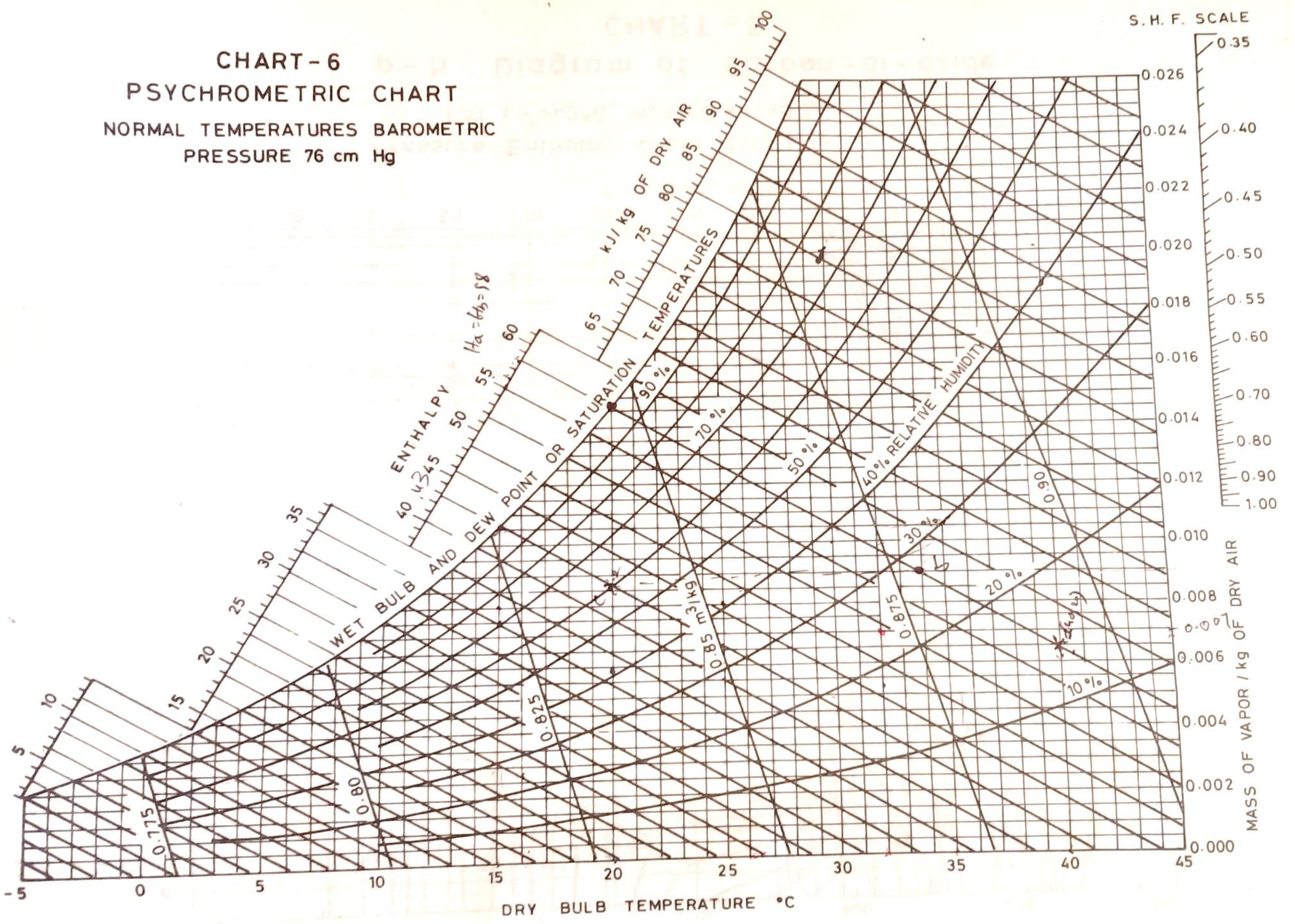
To find enthalpy (h) of the mixture

$$\text{w.k.t. enthalpy } h = 1.005\, t_{db} + \omega(2500 + 1.88\, t_{db})$$

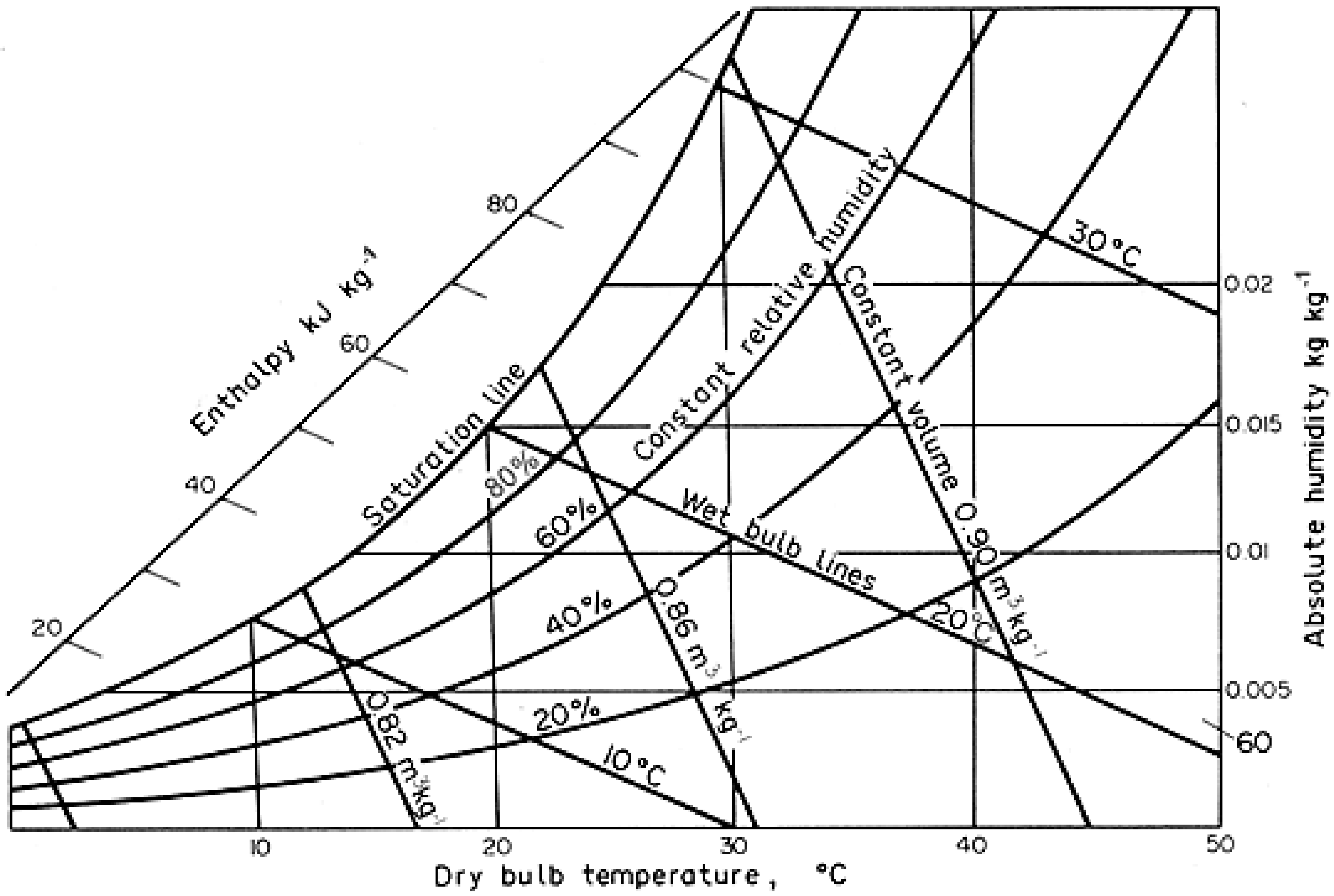
$$= 1.005\,(32) + 0.0187\,(2500 + 1.88\,(32))$$

enthalpy **h** = **80 kJ/kg**

PSYCHROMETRY CHART



PSYCHROMETRY CHART

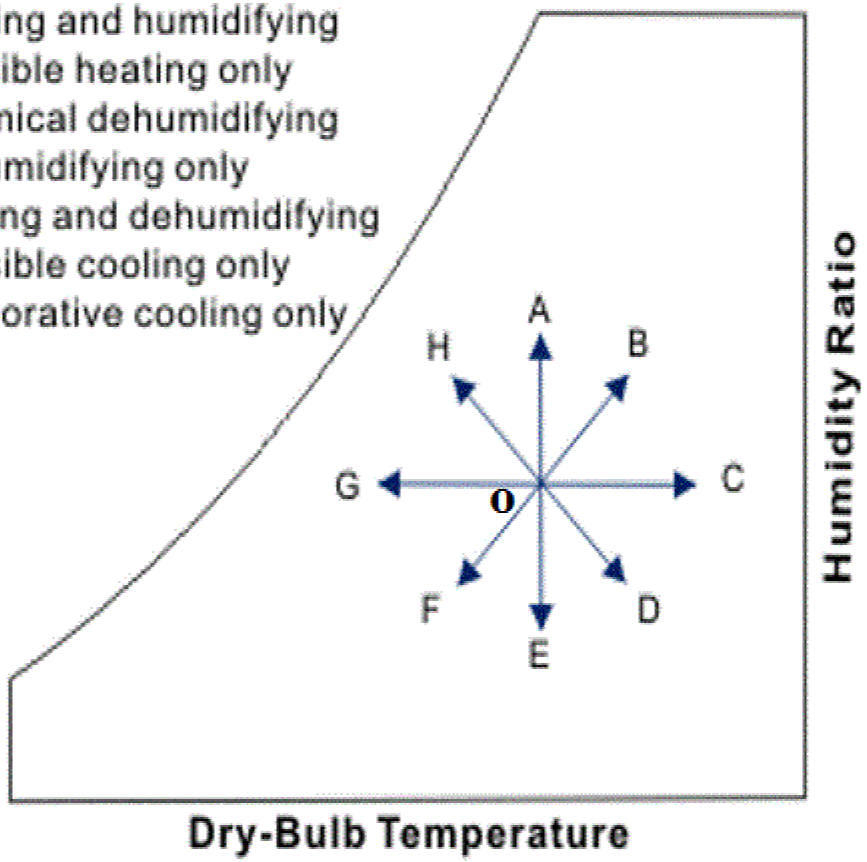


PSYCHROMETRIC PROCESSES

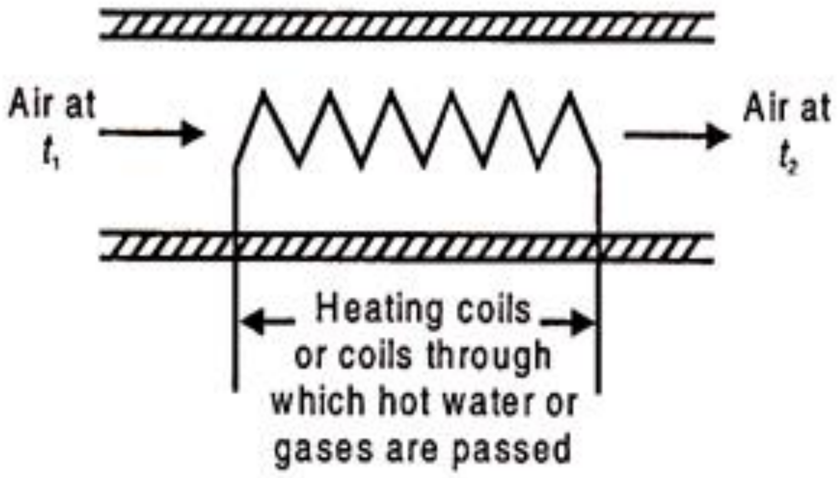
1. Sensible heating
2. Sensible cooling
3. Humidification and dehumidification
4. Cooling and humidification
5. Cooling and dehumidification
6. Heating and humidification
7. Heating and dehumidification
8. Adiabatic mixing of air streams

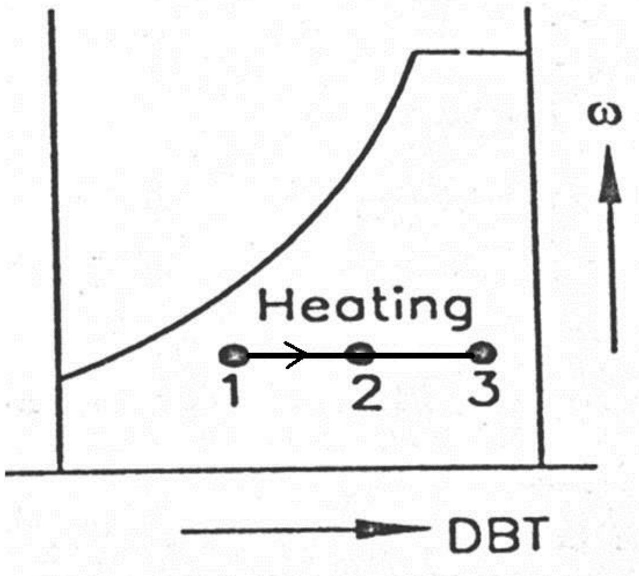
Air Conditioning Process

OA= Humidifying only
OB= Heating and humidifying
OC= Sensible heating only
OD= Chemical dehumidifying
OE= Dehumidifying only
OF= Cooling and dehumidifying
OG= Sensible cooling only
OH= Evaporative cooling only



SENSIBLE HEATING



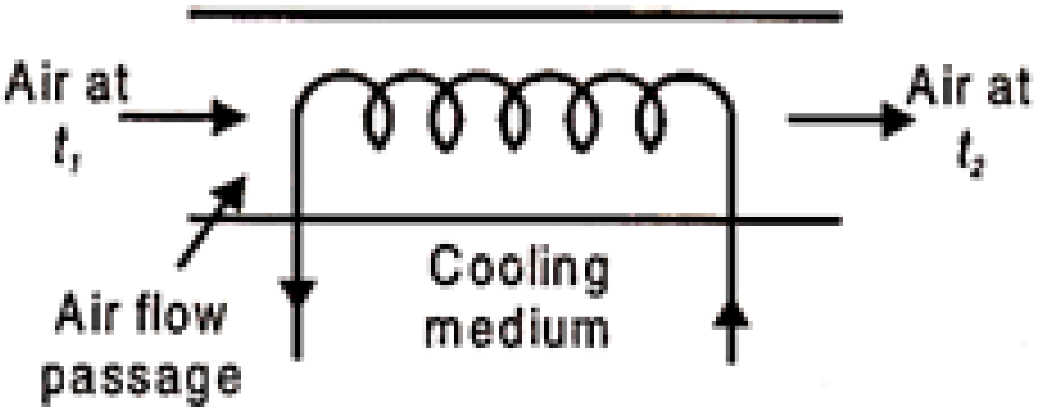


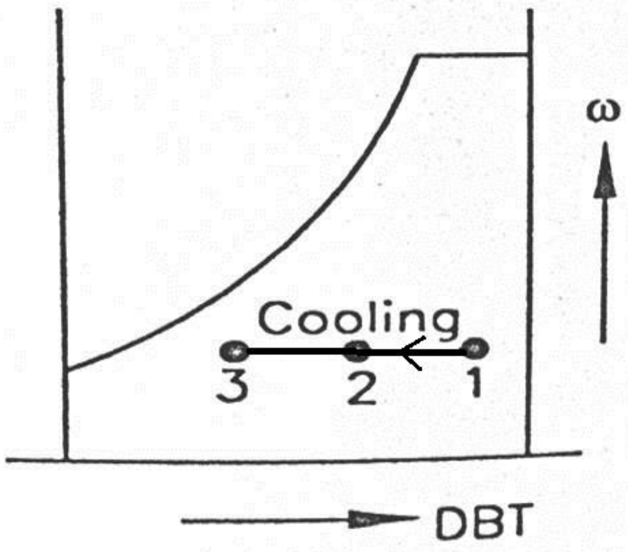
$$\text{By pass factor} = \frac{\text{Temperature difference between Heating coil and air at outlet}}{\text{Temperature difference between Heating coil and air at inlet}}$$

$$\text{By pass factor} = \frac{t_{db3} - t_{db2}}{t_{db3} - t_{db1}}$$

$$\textit{Heat Supplied} = (h_2 - h_1)\text{kJ/kg dry air}$$

SENSIBLE COOLING





$$\text{By pass factor} = \frac{\text{Temperature difference between air at oulet and cooling coil}}{\text{Temperature difference between air at inlet and cooling coil}}$$

$$\text{By pass factor} = \frac{t_{db2} - t_{db3}}{t_{db1} - t_{db3}}$$

$$\text{Heat Transfer} = h_1 - h_2 \ \text{kJ/kg}$$

HUMIDIFICATION AND DEHUMIDIFICATION :

In humidification

Relative humidity increases from ϕ_1 to ϕ_2

Specific humidity increases from ω_1 to ω_2

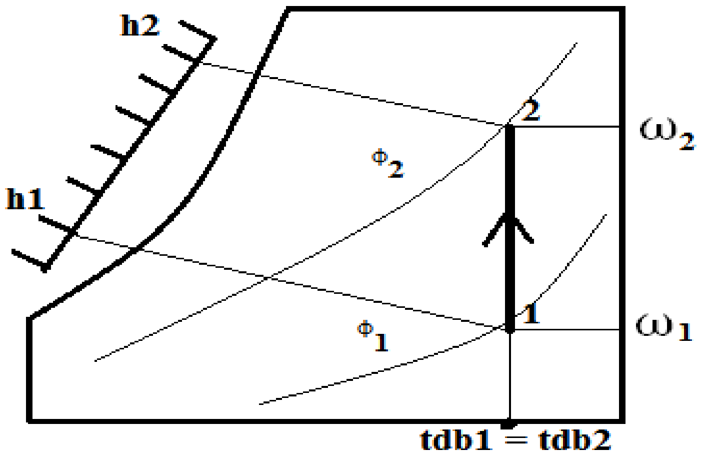
Heat transfer $=$ **$(h_2 - h_1)$ kJ/kg dry air**

In dehumidification

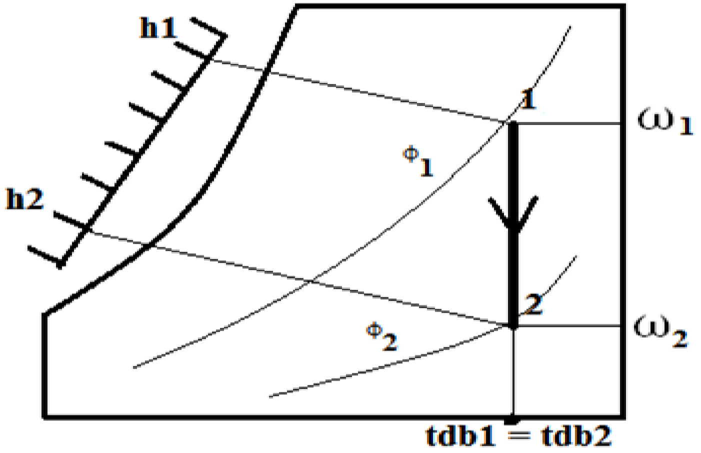
Relative humidity decreases from ϕ_1 to ϕ_2

Specific humidity decreases from ω_1 to ω_2

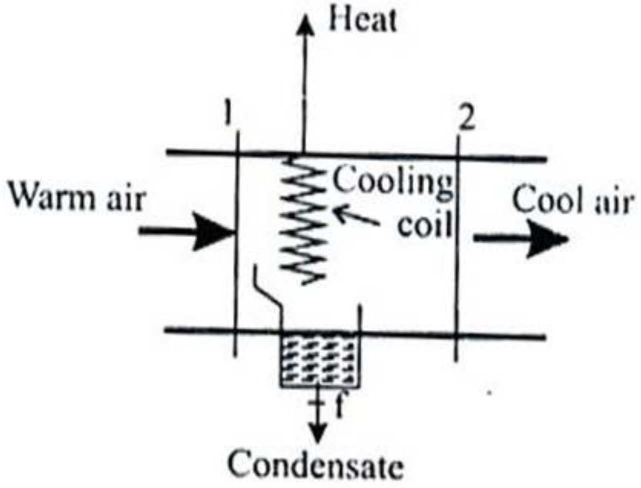
Heat transfer $=$ **$(h_1 - h_2)$ kJ/kg dry air**



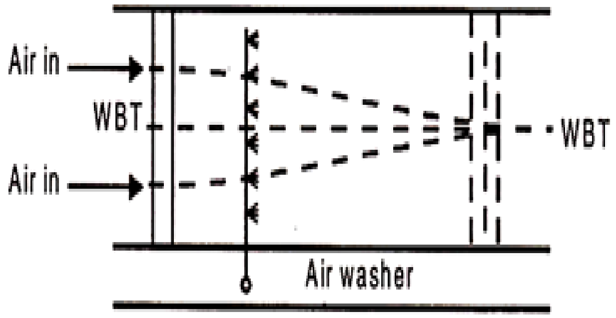
Humidification



Dehumidification



COOLING WITH HUMIDIFICATION OF AIR:

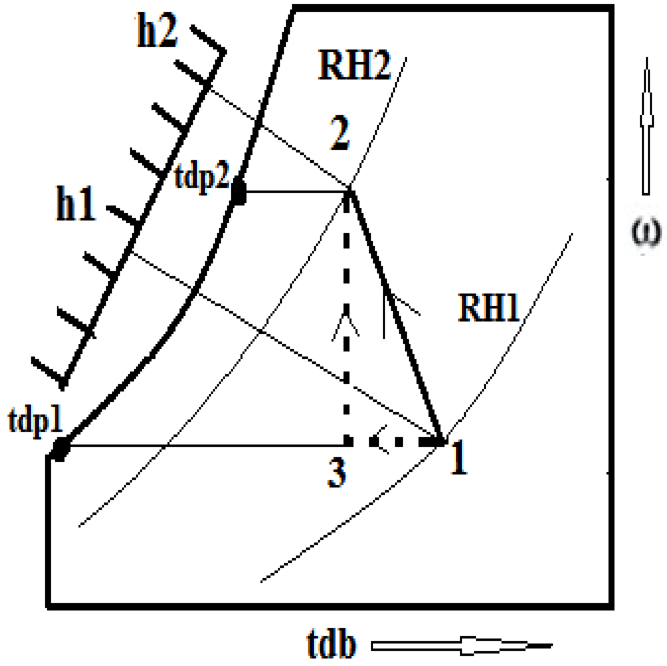


Here process 1-2 is Cooling and Humidification process Which can be divided into two process i.e

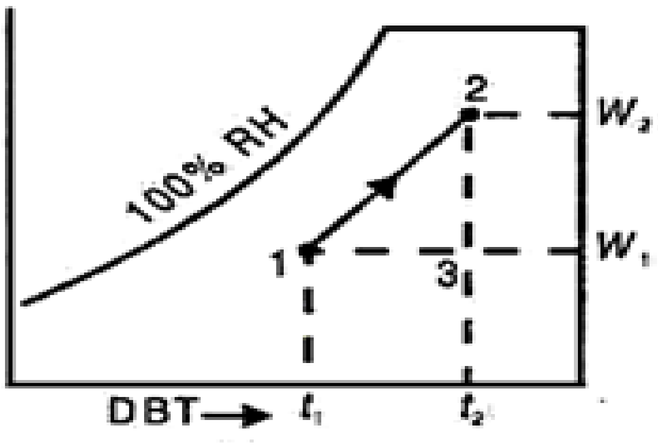
Process 1-3 Sensible cooling(Amount of heat added during this process is called SENSIBLE HEAT)

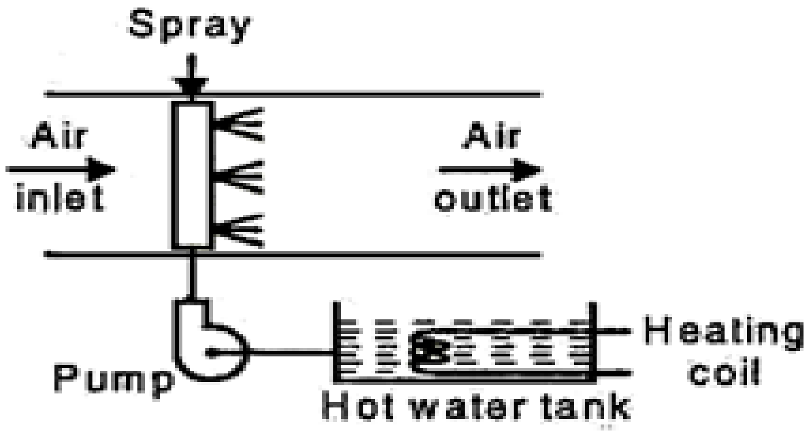
Process 3-2 Humidification process (Amount of heat added during this process is called LATENT HEAT)

Total Heat added = SH + LH



HEATING AND HUMIDIFICATION



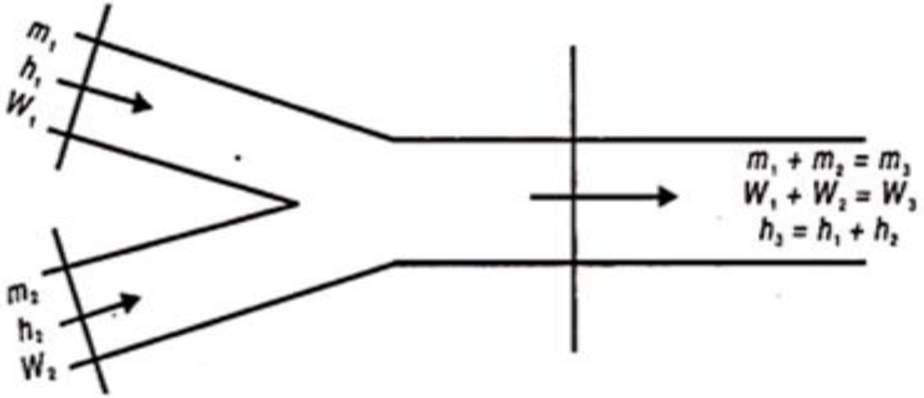


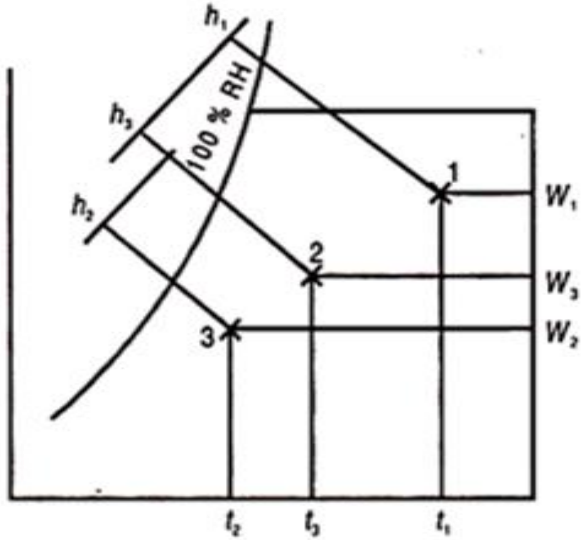
The final state 2 of the air may also be achieved by a combination of sensible heating 1-3 and then by sensible humidification process 3-2.

$$\text{SHF} = \frac{h_3 - h_1}{(h_3 - h_1) + (h_2 - h_3)} = \frac{h_3 - h_1}{h_2 - h_1}$$

ADIABATIC MIXING OF AIR STREAMS:

$$m_3 = m_1 + m_2$$





$$m_3 w_3 = m_1 w_1 + m_2 w_2$$
$$m_3 h_3 = m_1 h_1 + m_2 h_2$$

Eliminating m_3 and rearranging the above equations, we get

$$\frac{m_1}{m_2} = \frac{w_3 - w_2}{w_1 - w_3} = \frac{h_3 - h_2}{h_1 - h_3}$$

Problems using Psychrometry chart

7. The dry and wet bulb temperatures of atmospheric air at 1 atm (101.325 kPa) pressure are measured with a sling psychrometer and determined to be 25°C and 15°C respectively. With the help of psychrometric chart, determine a) specific humidity b) relative humidity c) enthalpy of air and d) DPT.

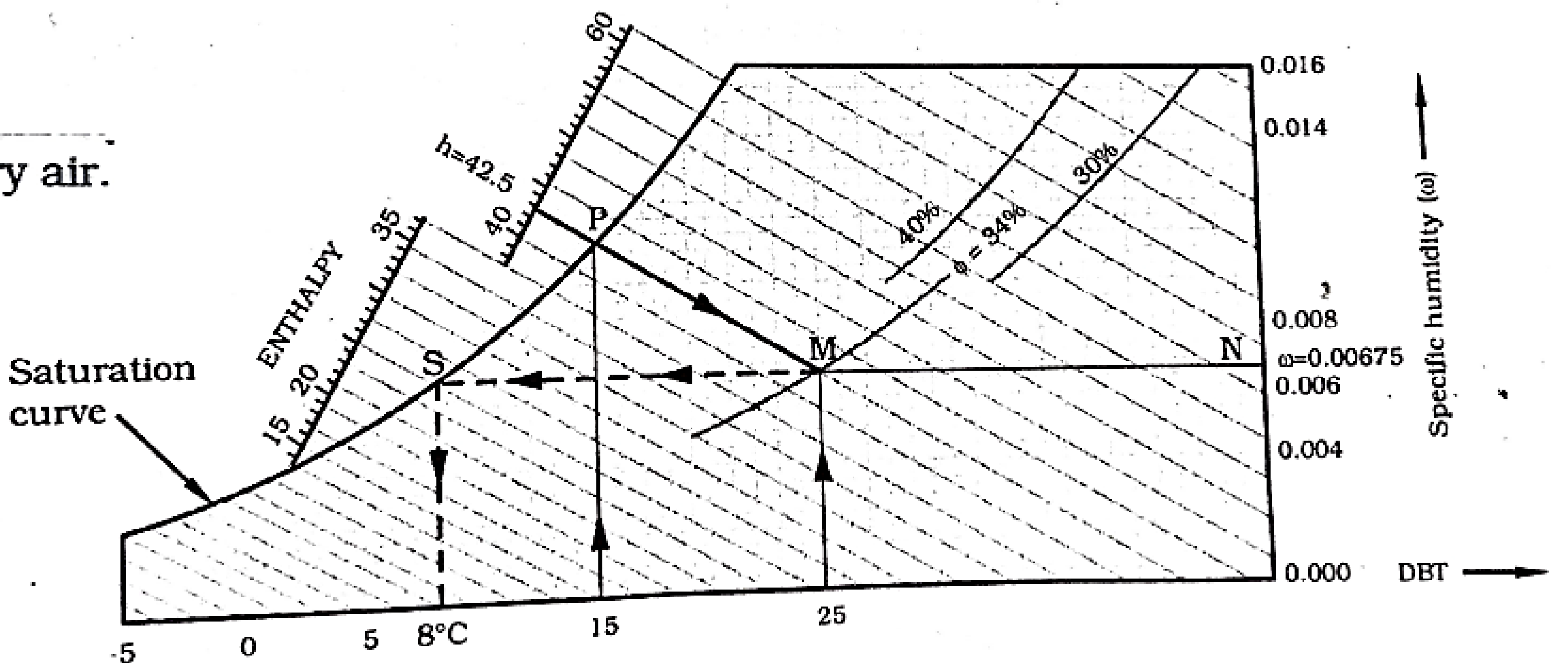
Data Given: DBT = 25°C; WBT = 15°C; Total pressure P = 101.325 kPa

specific humidity $\omega = 0.00675$ kg vap/kg of dry air.

RH $\phi = 34\%$

enthalpy h = 42.5 kJ/kg of air

DPT = 8°C



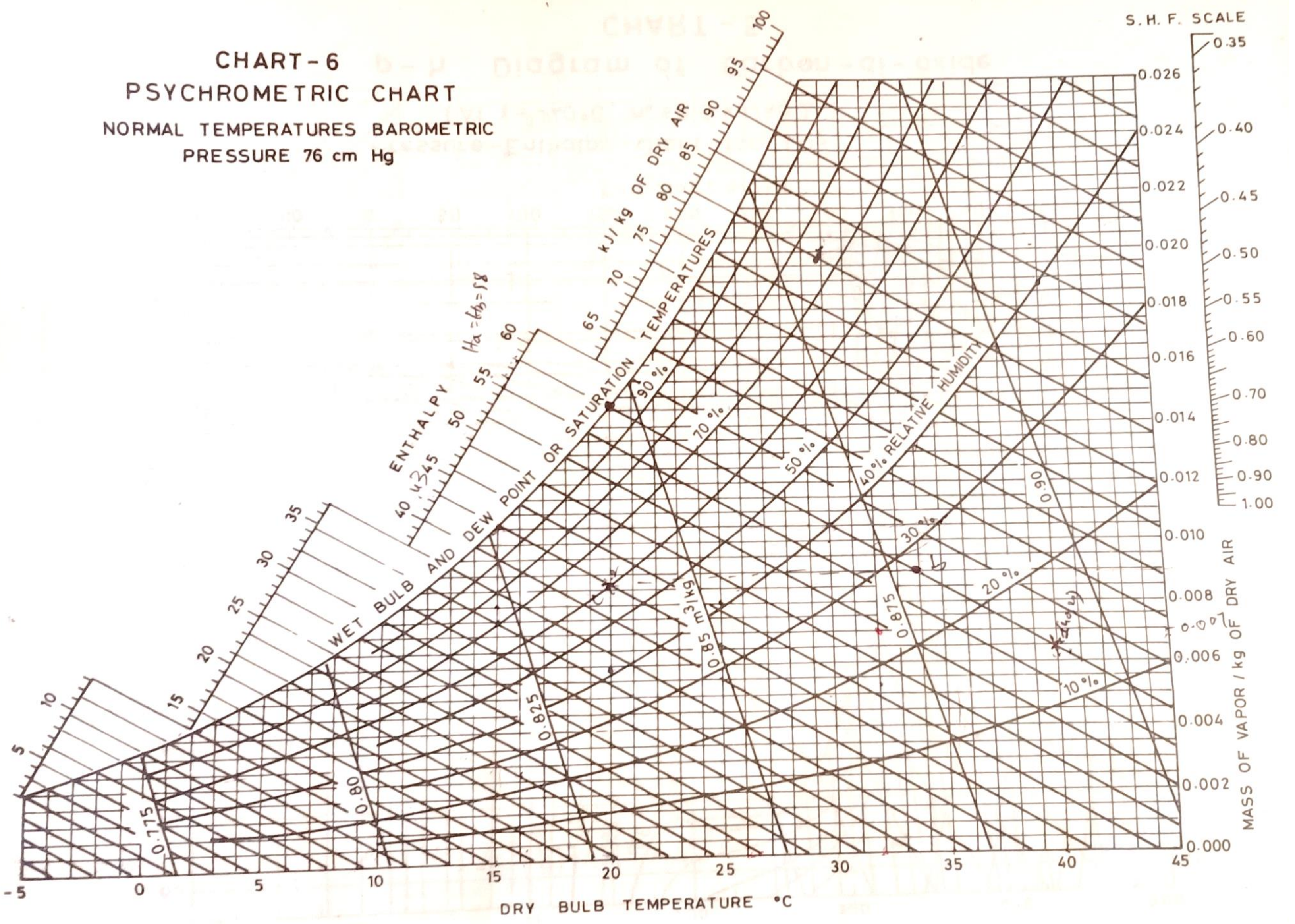


CHART - 6
PSYCHROMETRIC CHART
NORMAL TEMPERATURES BAROMETRIC PRESSURE 76 cm Hg
ENTHALPY kJ/kg OF DRY AIR
WET BULB AND DEW POINT OR SATURATION TEMPERATURES
RELATIVE HUMIDITY
S. H. F. SCALE
DRY BULB TEMPERATURE °C
MASS OF VAPOR / kg OF DRY AIR
0.85 m³/kg

8. Atmospheric air having DBT = 16°C and RH = 25% is passed through a furnace and then through a humidifier to maintain a final DBT of 30°C and 50% RH. Find the heat and moisture added to the air during the process. Also calculate the sensible heat factor of the process

Data Given:

Initial condition of air: DBT = 16°C; RH = 25%

Final condition of air: DBT = 30°C ; RH = 50%

To find heat added to air

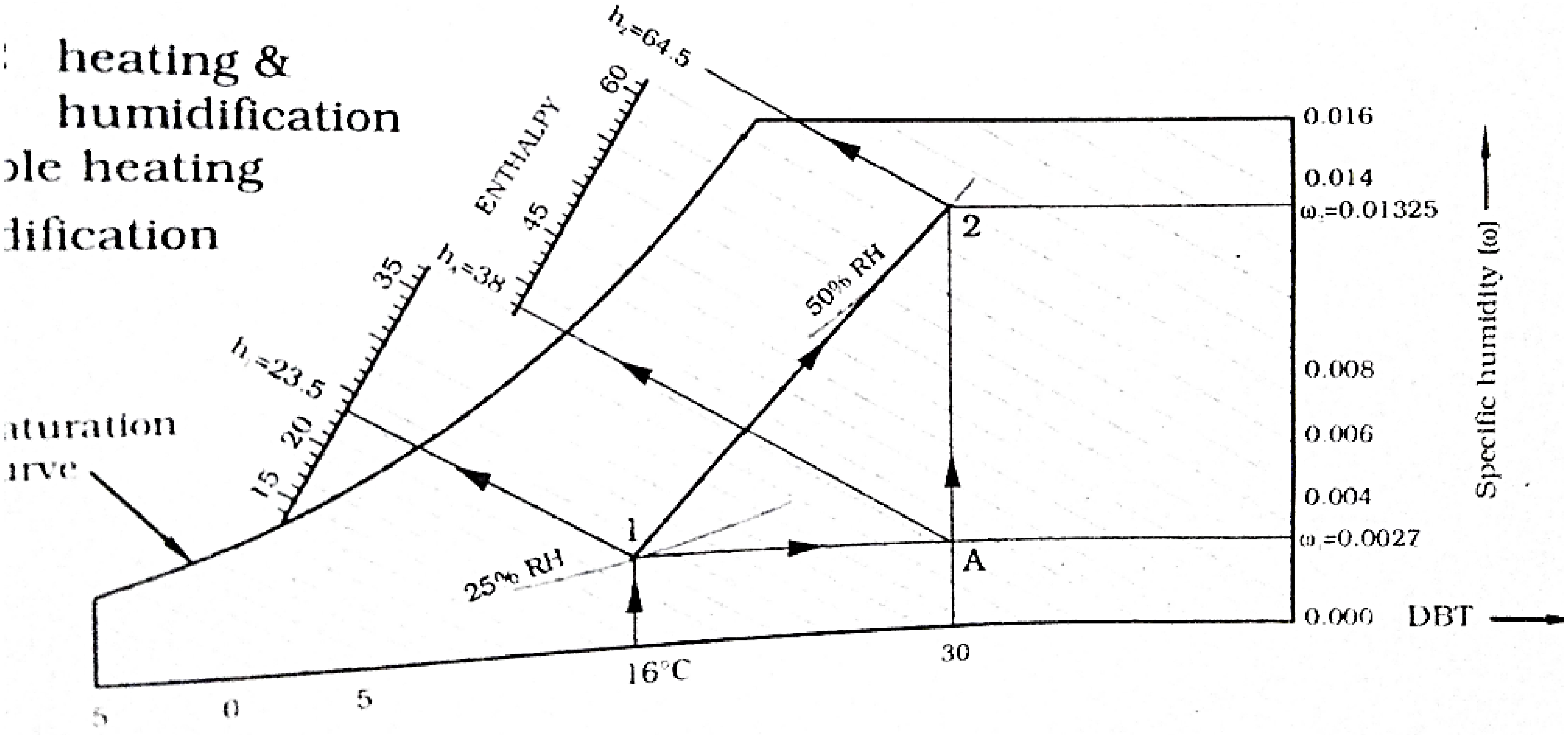
w.k.t. heat added to air = $m(h_2 - h_1)$ Assume m = 1 kg

heat added to air = $(h_2 - h_1)$ kJ/kg of air -----(1)

we have h_1 = 23.5 kJ & h_2 = 64.5 kJ

∴ Equation (1) becomes heat added to air = (64.5 – 23.5) = 41

heat added to air = 41 kJ/kg of air


heating &
humidification
ble heating
dification
ENTHALPY
h_2=64.5
60
45
h_A=38
35
h_1=23.5
20
15
aturation
urve
0.016
0.014
ω_2=0.01325
2
50% RH
0.008
0.006
0.004
ω_1=0.0027
1
A
25% RH
0.000
DBT
Specific humidity (ω)
16°C
30
5
0
5

Problem (8)

To find moisture added to air

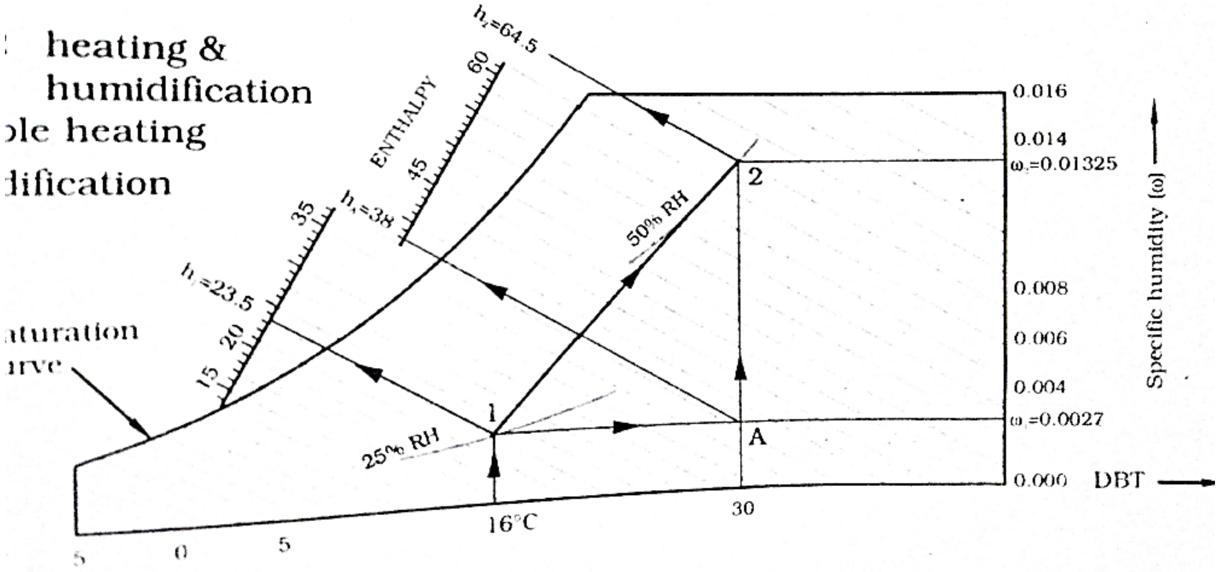
w.k.t. moisture added to air $= (\omega_2 - \omega_1)$ kg vap/ kg of air -----(2)

$\omega_1 = 0.0027$ kg vap/kg of dry air &

$\omega_2 = 0.01325$ kg vap/kg of dry air

moisture added to air $= (0.01325 - 0.0027)$:

$= \mathbf{0.01055}$ **kg vap/kg dry air**



Problem (8)

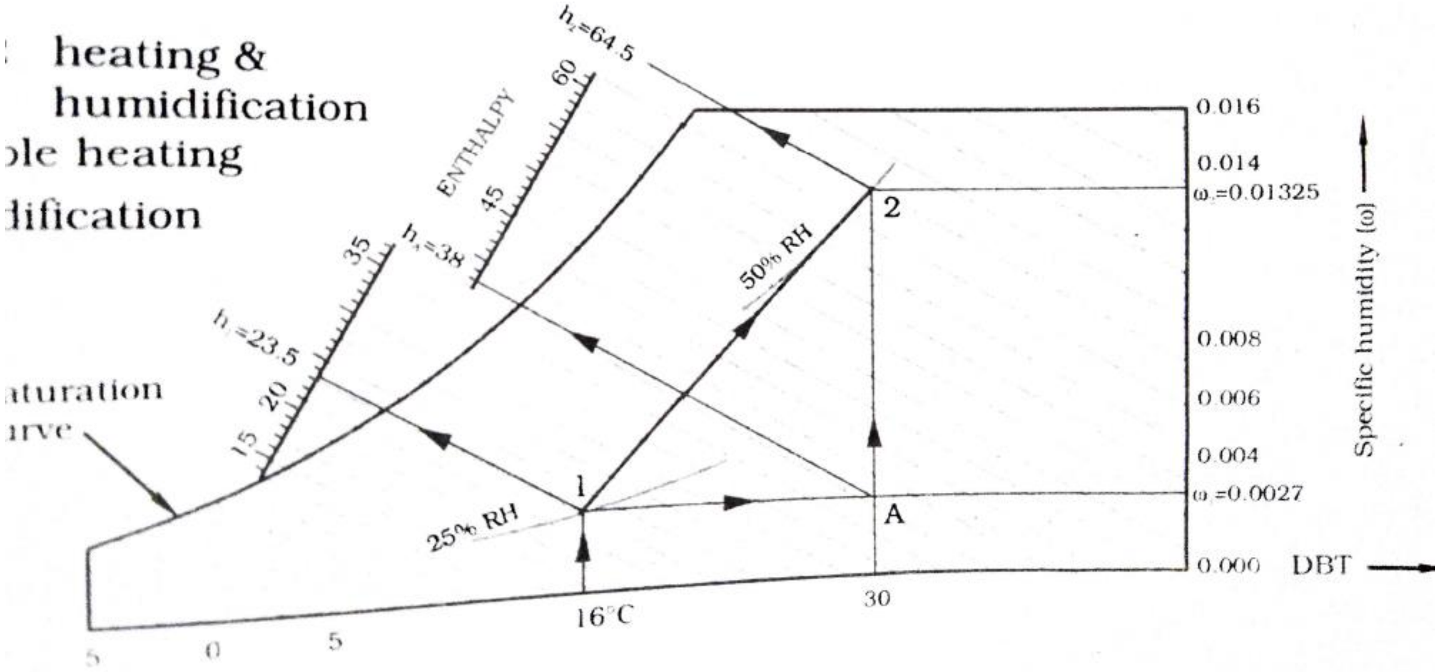
To find sensible heat factor (SHF)

$$\text{w.k.t. SHF} = \frac{h_A - h_1}{h_2 - h_1} \quad \text{-----(3)}$$

$$h_A = 38 \text{ kJ/kg}$$

$$\text{SHF} = \frac{(38 - 23.5)}{(64.5 - 23.5)} = 0.3536$$

SHF = 0.3536



9. It is required to design an air conditioning plant for a office room with the following conditions: Outdoor conditions 14°C DBT and 10°C WBT: Required conditions 20°C DBT and 60% R.H; Amount of Air circulation 0.30 m3/min/person : Seating capacity of office 60. The required condition is achieved first by heating and then by adiabatic humidifying. Determine the following: i) Heating capacity of the coil in kW and the surface temperature required if the by-pass factor of coil is 0.4. ii) The capacity of the humidifier.

Data Given:

Initial (Outdoor) condition : 14°C DBT ; 10°C WBT

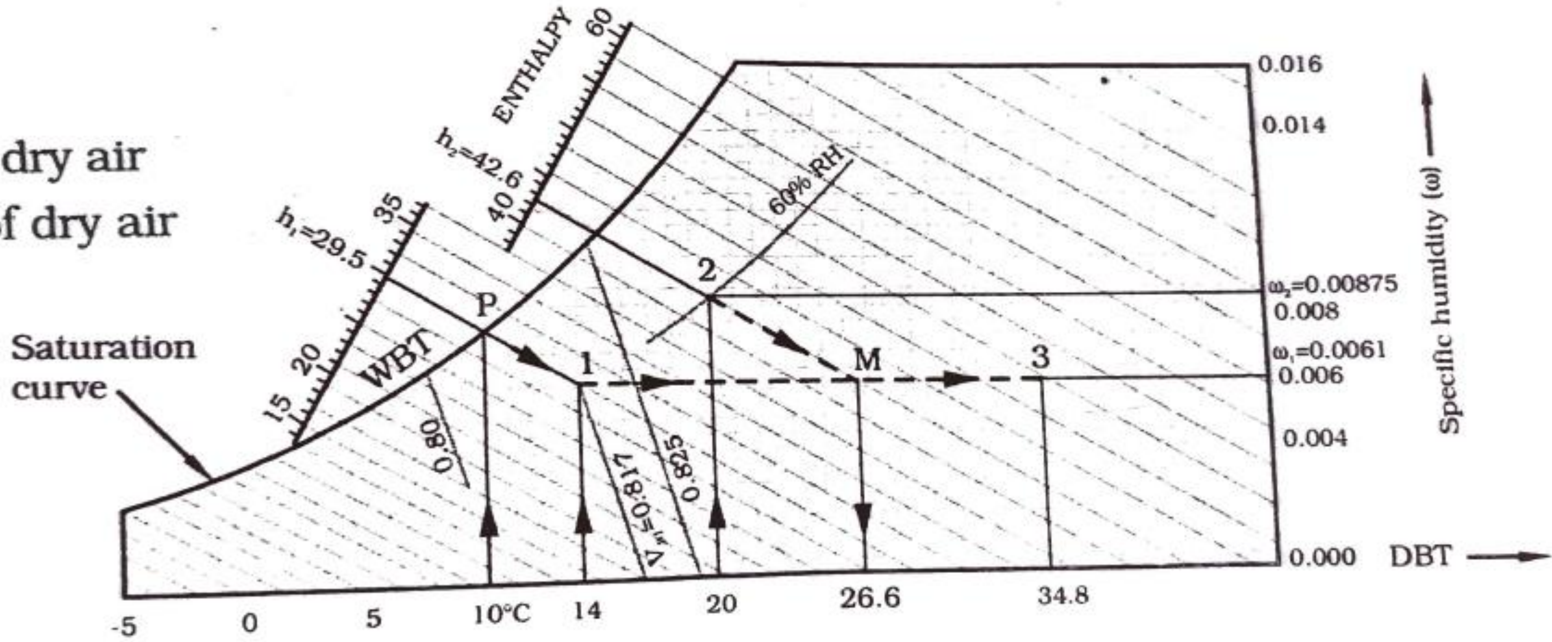
Final (Required) condition : 20°C DBT ; 60% RH

h_1 = 29.5 kJ/kg; ω_1 = 0.0061 kg vap/kg of dry air

h_2 = 42.6 kJ/kg; ω_2 = 0.00875 kg vap/kg of dry air

Specific volume V_{S1} = 0.817 m³/kg

$(t_{db})_M$ = 26.6°C



Problem (9)

To find heating capacity of coil in kW

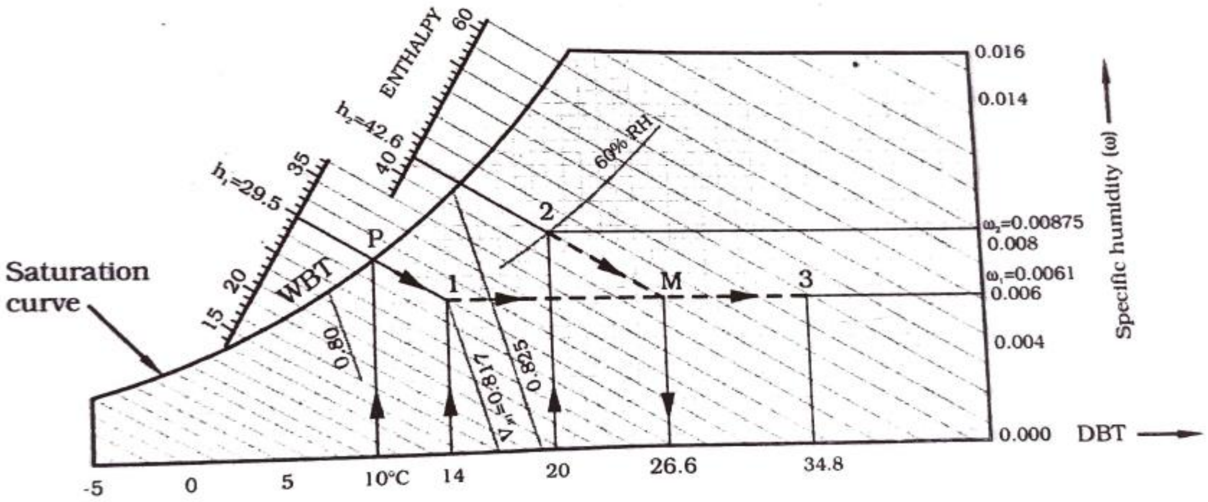
w.k.t. heating capacity $= m_a(h_2 - h_1)$ -----(1)

But $m_a = ?$

w.k.t. mass of air circulated/min $= m_a = \frac{V_a}{V_{S1}}$

$$= \frac{0.30\ m^3/\ min/\ person}{0.817\ m^3/\ kg}$$

$= 0.3672$ kg/min. per person



$\therefore$ for 60 persons, $m_a = 0.3672 \times 60 = 22.03$ kg/min

Problem (9)

Now equation (1) becomes, heating capacity = 22.03 (42.6 – 29.5)

= 288.59 kJ/min

∴ **Heating capacity of coil = 4.8 kW**

To find surface temperature of coil

Let surface temperature of coil be $(t_{db})_3$

Given By-Pass factor (BPF) = 0.4

$$\text{w.k.t. BPF} = \frac{(t_{db})_3 - (t_{db})_M}{(t_{db})_3 - (t_{db})_1} \qquad 0.4 = \frac{(t_{db})_3 - 26.5}{(t_{db})_3 - 14}$$

coil surface temperature = $(t_{db})_3$ = 34.8°C

Problem (9)

To find capacity of humidifier

w.k.t. Capacity of humidifier $= m_a(\omega_2 - \omega_m)$

But $\omega_m = \omega_1 = 0.0061$ kg vap/kg of dry air

$= 22.03\ (0.00875 - 0.0061)$

$= 0.0583 \dfrac{kg}{min} \cdot \dfrac{kg\ vap}{kg\ of\ dry\ air}$

$= 0.0583$ kg/min

$= 3.5$ kg/hr

capacity of humidifier = 3.5 kg/hr

10. A summer air conditioning system for hot and humid weather (DBT = 32°C and RH = 70%) consists in passing the atmospheric air over a cooling coil where the air is cooled and dehumidified. The air leaving the cooling coil is saturated at the coil temperature. It is then sensibly heated to the required comfort condition of 24°C and 50% RH by passing it over an electric heater and then delivered to the room.

Data Given:

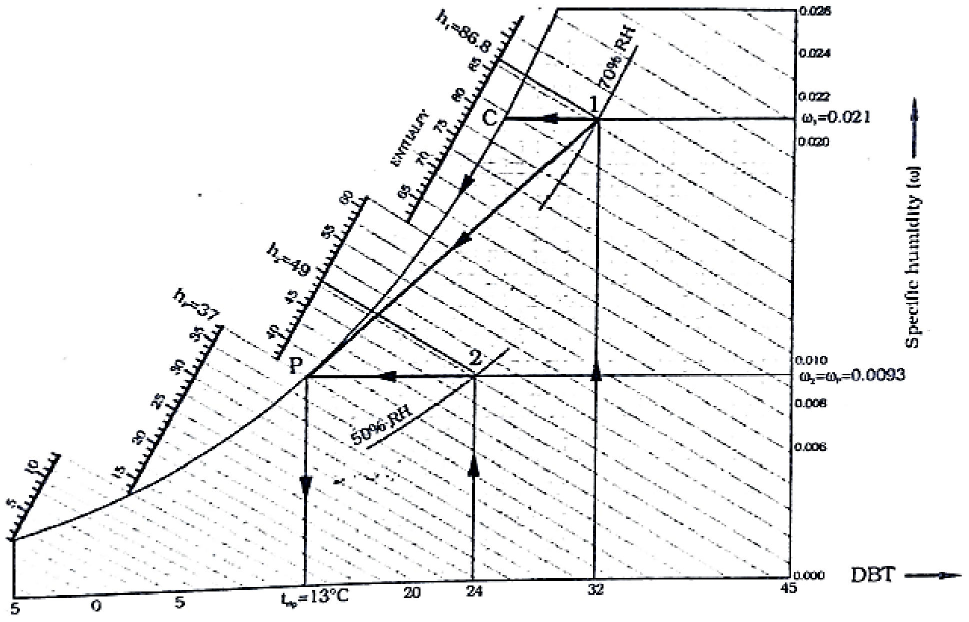
Initial condition of air : 32°C DBT and 70% RH

Final condition of air : 24°C DBT and 50%

Path 1-C : Constant pressure cooling of air

Path C-P : Dehumidification to saturated state *P* at DPT (t_{dp})

Path P-2 : Constant pressure heating of air



we have

h_1 = 86.8 kJ/kg

h_2 = 49 kJ/kg

h_p = 37 kJ/kg

ω_1 = 0.021 kg vap/kg of air

t_{dp} = 13°C

$\omega_2 = \omega_p$ = 0.0093 kg vap/kg of air

Problem (10)

temperature of cooling coil = t_{dP} = 13°C

w.k.t. amount of moisture removed = $m_a(\omega_1 - \omega_p)$ = 1(0.021 – 0.0093)

amount of moisture removed = **0.0117 kg vap/kg of air**

w.k.t. heat removed in the cooling coil = $m_a(h_1 - h_p)$ = 1(86.8 – 37)

Heat removed in the cooling coil = **49.8 kJ/kg of air**

heat added in the heating coil = $m(h_2 - h_p)$ = 1 (49 – 37)

heat added in the heating coil = **12 kJ/kg of air**

11. An air conditioning system is designed under the following conditions: Outdoor conditions : 30°C DBT, 75% RH, Required indoor conditions : 22°C DBT, 70% RH. Amount of free air circulated 3.33 m3/s, Coil dew point temperature (DPT) 14°C. The required condition is achieved first by cooling and dehumidification and then by heating. Estimate: i) The capacity of the cooling coil in Tonnes of refrigeration. ii) The capacity of the heating coil in kW. iii) The amount of water vapour removed in kg/hr.

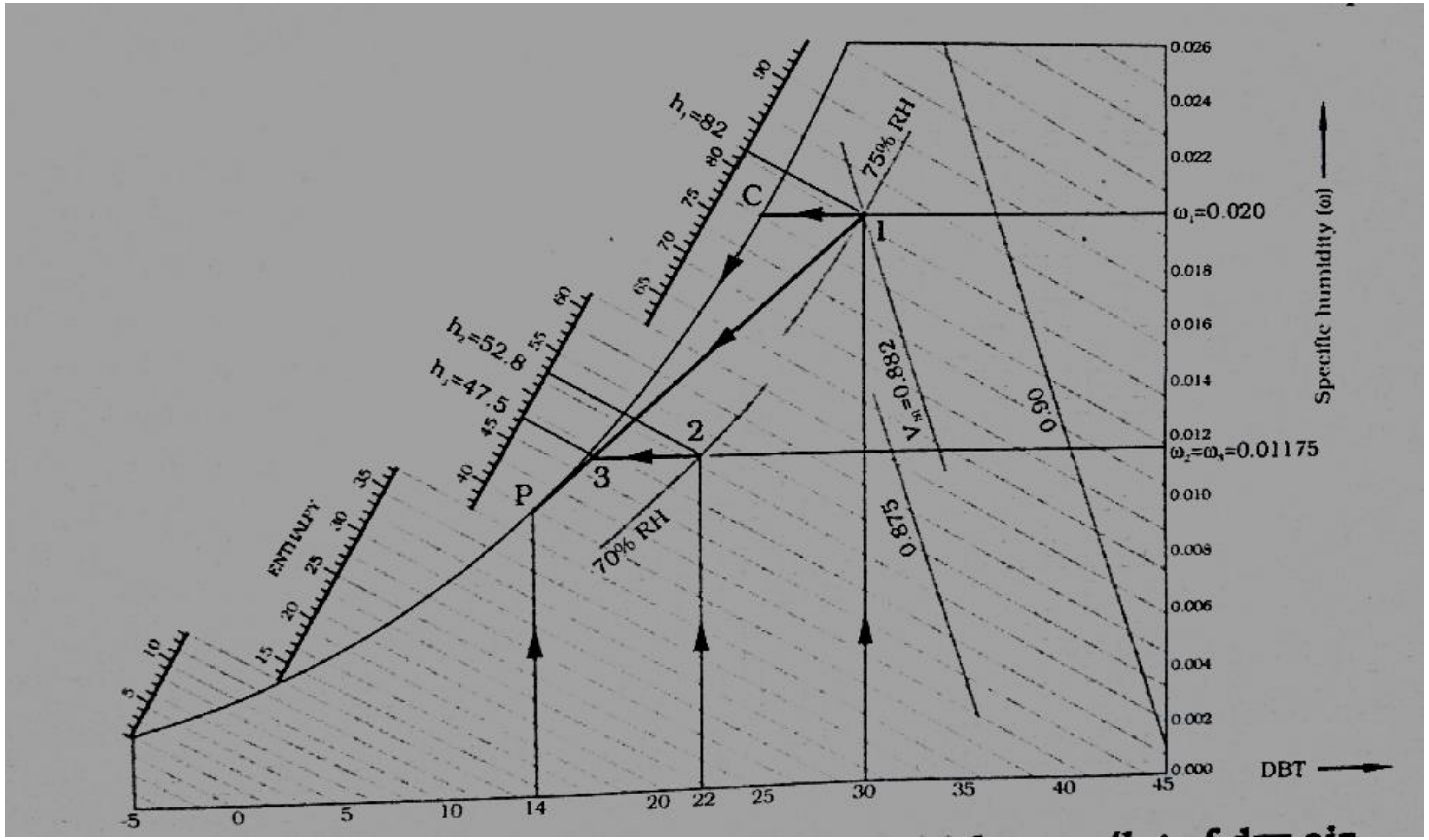
Initial (outdoor) condition : 30° DBT and 75% RH

Final (required) condition : 22°C DBT and 70% RH.

Cooling coil dew point temperature : DPT = 14°C

Final condition is achieved by cooling & dehumidification and then by heating.

Problem (11)



we have $h_1 = 82$ kJ/kg $\quad \omega_1 = 0.02$ kg vap/kg of dry air

$h_2 = 52.8$ kJ/kg $\quad \omega_2 = \omega_3 = 0.01175$ kg vap/kg of dry air

$h_3 = 47.5$ kJ/kg $\quad V_{S1} = 0.882\ m^3/kg$

Problem (11)

Capacity of cooling coil in TOR

cooling coil capacity = $m_a(h_1 - h_3)$

where $m_a = \frac{V_a}{V_{S1}} = \frac{3.33\ m^3/sec}{0.882\ m^3/kg} = 3.77\ kg/sec$

Cooling coil capacity = $3.77(82 - 47.5)$

$= 130.06\ kJ/sec$

1 TOR = 210 kJ/min = 3.5 kJ/s

Cooling coil capacity in TOR = **37.16 Tonnes**

Problem (11)

To find capacity of heating coil in kW

capacity of heating coil $= m_a(h_2 - h_3)$

$= 3.77(52.8 - 47.5)$

$= 19.98$ kJ/scc

capacity of heating coil = **20 kW**

To find amount of water vapour removed

Amount of water vapour removed $= m(\omega_1 - \omega_3)$

$= 3.77(0.02 - 0.01175) = 0.0311$ kg/s

amount of water vapour removed = **111.9 kg/hr**

12. It is required to design an A/C for the following condition: Outdoor condition : 32°C DBT and 65% RH; Indoor conditions : 25°C DBT and 60% RH Amount of air circulated : 250m3/min ; Coil dew temperature : 13°C. If the required condition is achieved first by cooling and dehumidifying and then by heating, calculate (i) Cooling coil capacity and its by-pass factor (ii) Heating coil capacity and its surface temperature if its by-pass factor is 0.3 (iii) Mass of water vapour removed per hour.

Initial (outdoor) condition : 32°C DBT and 65% RH

Final (indoor) condition : 25°C DBT and 60% RH

Cooling coil DPT = 13°C

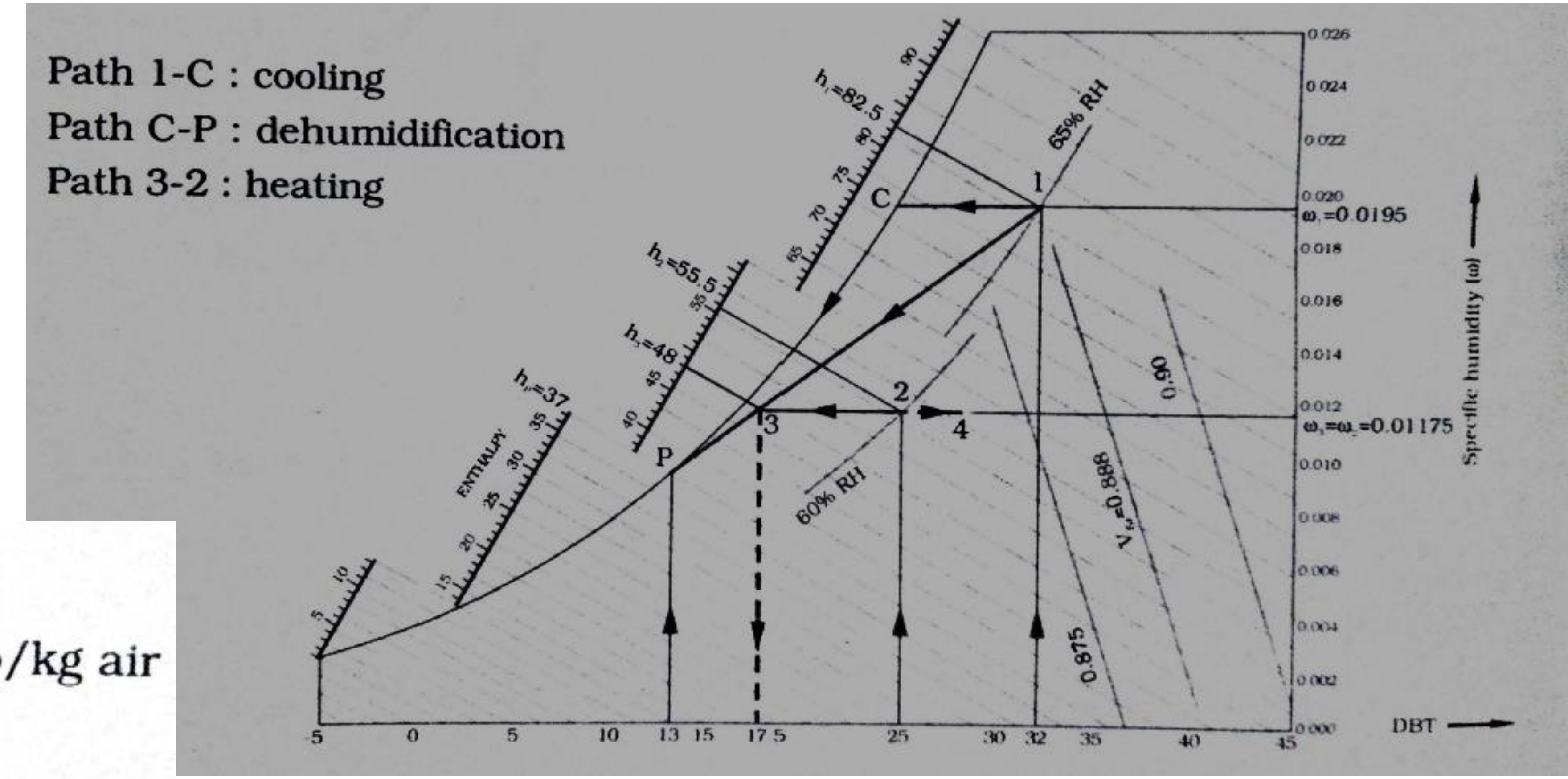
Problem (12)

h_1 = 82.5 kJ/kg

h_2 = 55.5 kJ/kg

h_3 = 48 kJ/kg

h_p = 37 kJ/kg



ω_1 = 0.0195 kg vap/kg air

ω_3 = ω_2= ω_4= 0.01175 kg vap/kg air

$(t_{db})_3$ = 17.5°C

V_{S1} = 0.888 m³/kg

Problem (12)

$$\text{cooling coil capacity} = m_a (h_1 - h_3)$$

$$m_a = \frac{V_a}{V_{S1}} = \frac{250\, m^3/\min}{0.888\, m^3/kg}$$

$$m_a = 281.53 \text{ kg/min}$$

$$\text{cooling coil capacity} = 281.53\,(82.5 - 48)$$

$$= 9712.78\ \frac{kJ}{\min}$$

cooling coil capacity = **46.2 TOR**

Problem (12)

To find By–Pass factor (BPF) of cooling coil

$$(BPF)_{Cooling\ Coil} = \frac{h_3 - h_P}{h_1 - h_P} = \frac{(48 - 37)}{(82.5 - 37)}$$

BPF of cooling coil = 0.241

$$\text{heating coil capacity} = m_a(h_2 - h_3)$$
$$= 281.53(55.5 - 48)$$
$$= 2111.47 \text{ kJ/min}$$

heating coil capacity = **35.2 kJ/sec** (or **kW**)

Problem (12)

$$\text{w.k.t. } (BPF)_{\text{heating coil}} = \frac{(t_{db})_4 - (t_{db})_2}{(t_{db})_4 - (t_{db})_3}$$

$$0.3 = \frac{(t_{db})_4 - 25}{(t_{db})_4 - 17.5}$$

Surface temperature of heating coil = $(t_{db})_4$ = **28.21°C**

$$\text{mass of water vapour removed} = m_a(\omega_1 - \omega_3)$$

$$= 281.53(0.0195 - 0.0117)$$

$$= 2.196 \text{ kg/min}$$

mass of water vapour removed/hour = **131.75 kg/hr**