CH2303 Chemical Engineering Thermodynamics I

Unit – V

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Compression and Expansion of Fluids

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Compression

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Contents

- Thermodynamic aspects of compression process
- Classification of compression processes
- Work expression for different situations, the effect of clearance volume, multistage compression

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Introduction

- Compressor is a device used for pumping compressible fluids i.e, air, gas & steam

- According to API, a pressure rise above 0.35 bar is compressor, and below is blower

- Classification based on operating principle
  - Positive displacement (Increase pressure by reducing volume)
  - Dynamic or turbo (By imparting kinetic energy to air/gas and then converting it into pressure: $\Delta P \propto \rho u^2$)

- Flow rate: m^3/hr , cfm (cubic foot per minute)

- Pressure ratio: ratio of absolute discharge pressure /absolute inlet pressure

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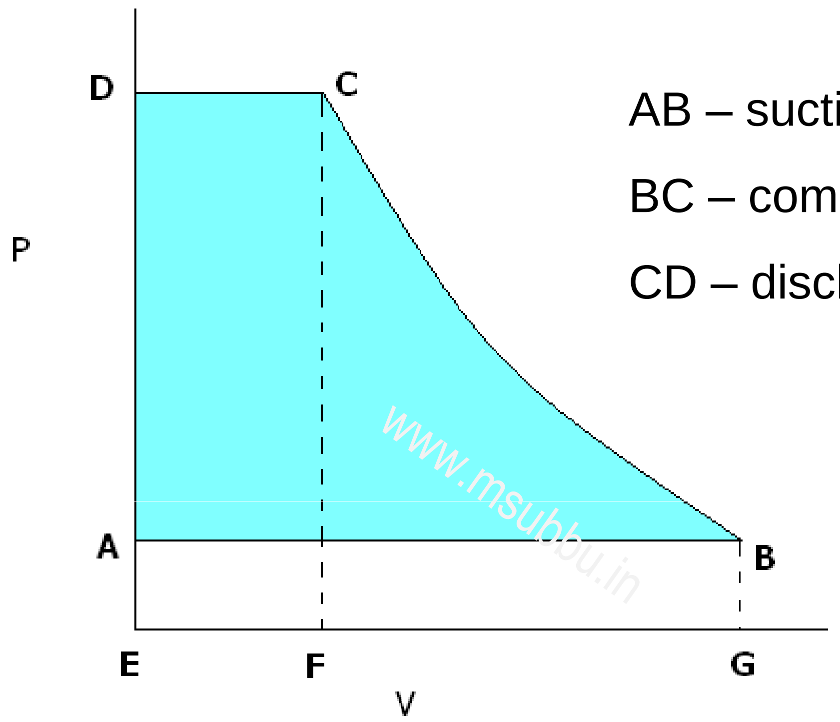


AB – suction

BC – compression

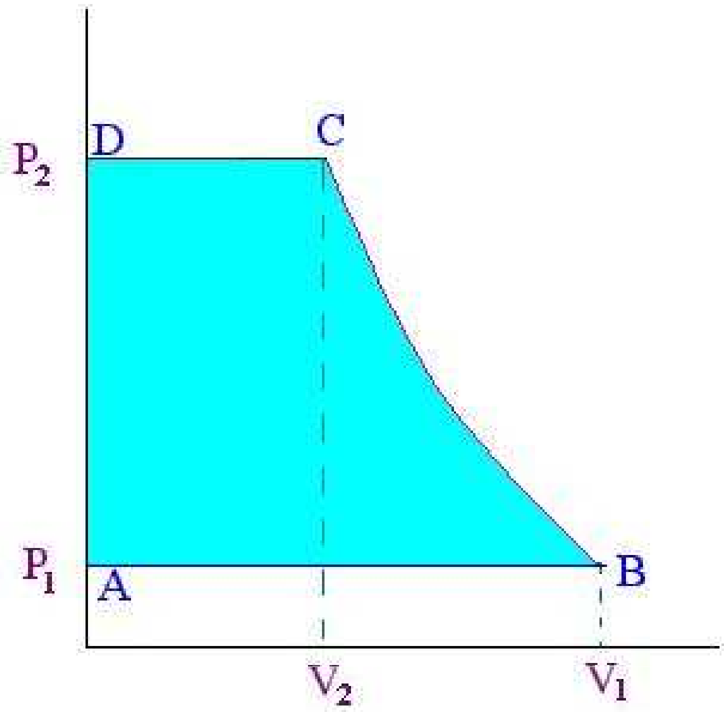
CD – discharge

Area of cycle ABCDA = Area of FGBCF + Area of CDEFC - Area of ABGEA

Work of compression = area of ABCDA = $\int V\,dP$

$$W_{\text{cycle}} = W_{\text{AB}} + W_{\text{BC}} + W_{\text{CD}} + W_{\text{DA}}$$

$$\begin{aligned}
W_{\text{AB}} &= -\int_0^{V_1} PdV = -P_1 \int_0^{V_1} dV = -P_1V_1 \\
W_{\text{BC}} &= -\int_{V_1}^{V_2} PdV \\
W_{\text{CD}} &= -\int_{V_2}^{0} PdV = -P_2 \int_{V_2}^{0} dV = P_2V_2 \\
W_{\text{DA}} &= -\int_0^{0} PdV = 0
\end{aligned}$$



$$d(PV) = PdV + VdP$$

$$\begin{aligned}
W_{\text{cycle}} &= -P_1V_1 - \int_{V_1}^{V_2} PdV + P_2V_2 + 0 \\
&= P_2V_2 - P_1V_1 - \int_{V_1}^{V_2} PdV
\end{aligned}$$

$$\Delta(PV) = \int PdV + \int VdP$$

$$P_2V_2 - P_1V_1 = \int PdV + \int VdP$$

$$P_2V_2 - P_1V_1 - \int PdV = \int VdP$$

$$\boxed{W_{\text{cycle}} = \int_{P_1}^{P_2} VdP}$$

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Isothermal Compression

c = PV = constant

$$W_i = \int_{P_1}^{P_3} V\,dP = c\int_{P_1}^{P_3} \frac{dP}{P} = P_1V_1\log_e\frac{P_3}{P_1} = P_1V_1\log_e r$$

r = compression ratio (isothermal)

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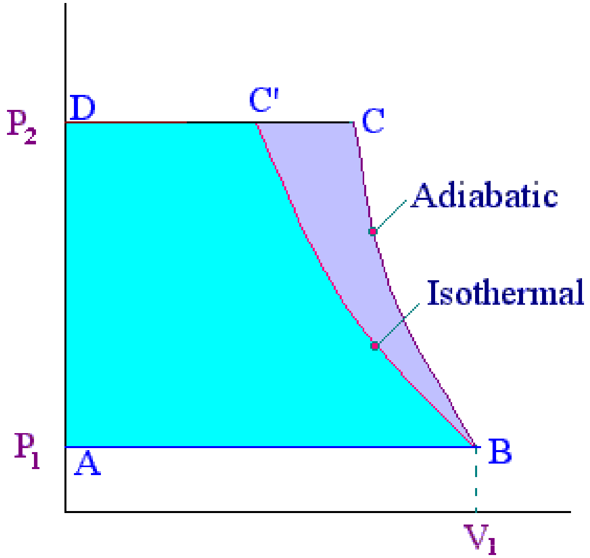
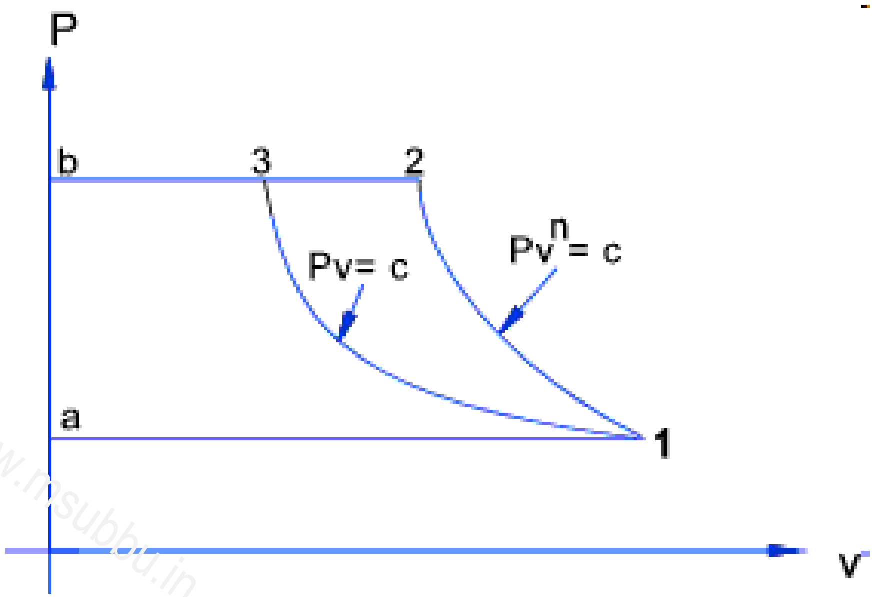
$$\eta_{iso} = \frac{\text{Calculated isothermal power}}{\text{Power imparted to air}}$$

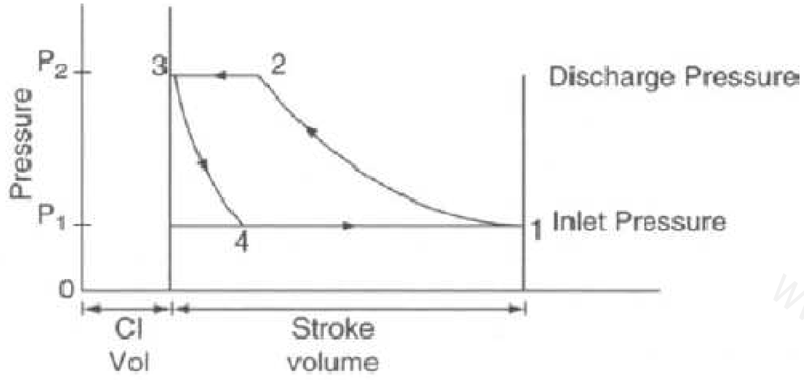


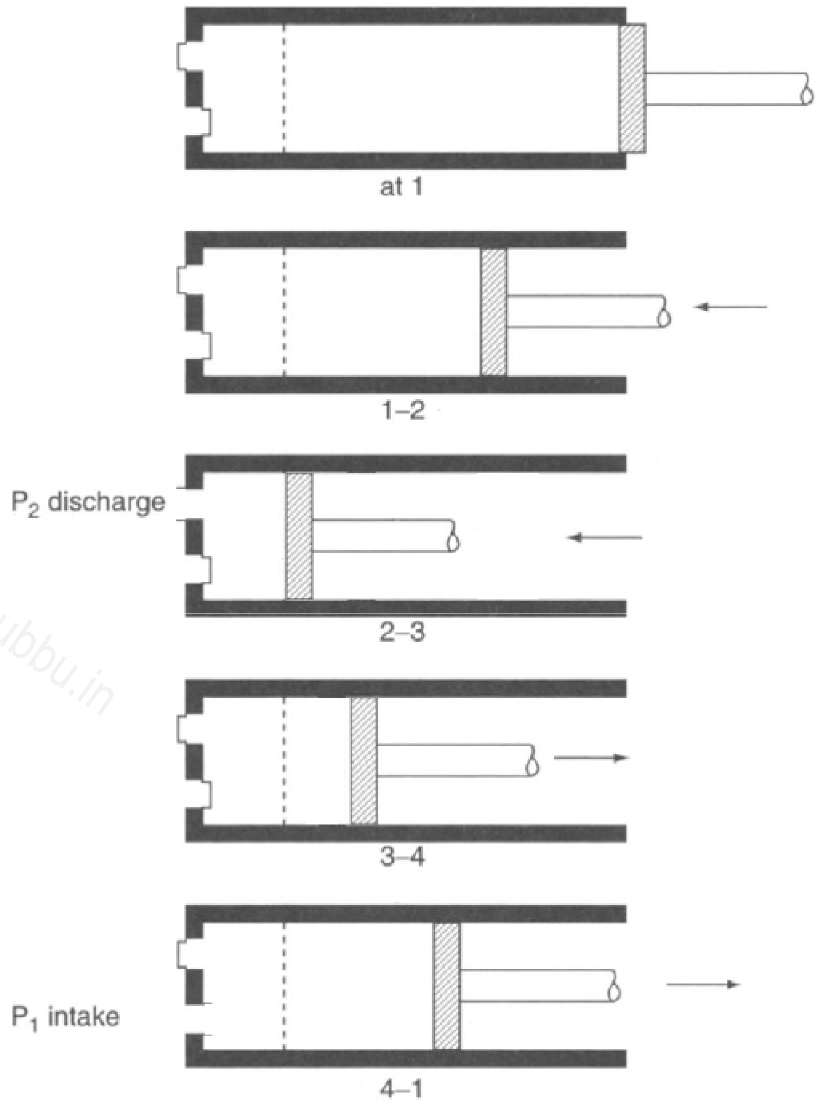


$$W = P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_1V_1$$

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

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





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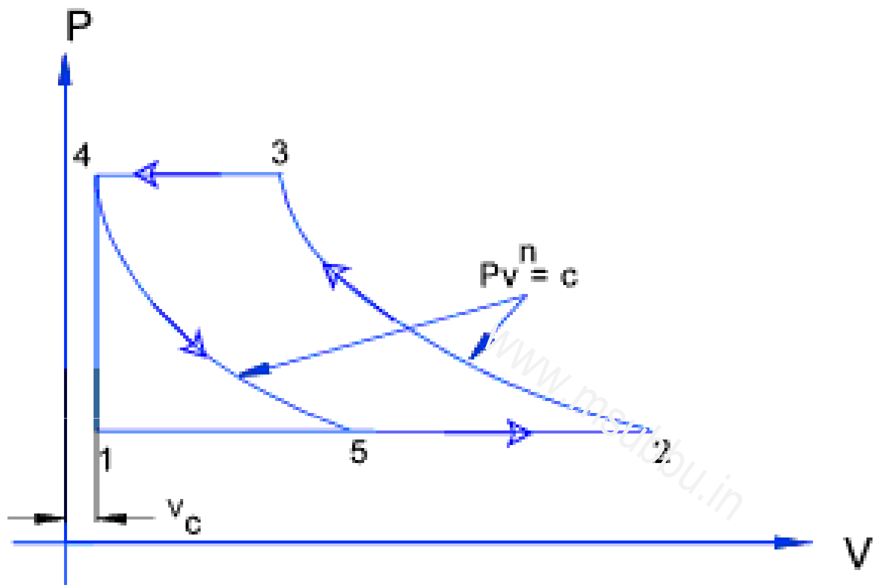
Effect of Clearance Volume



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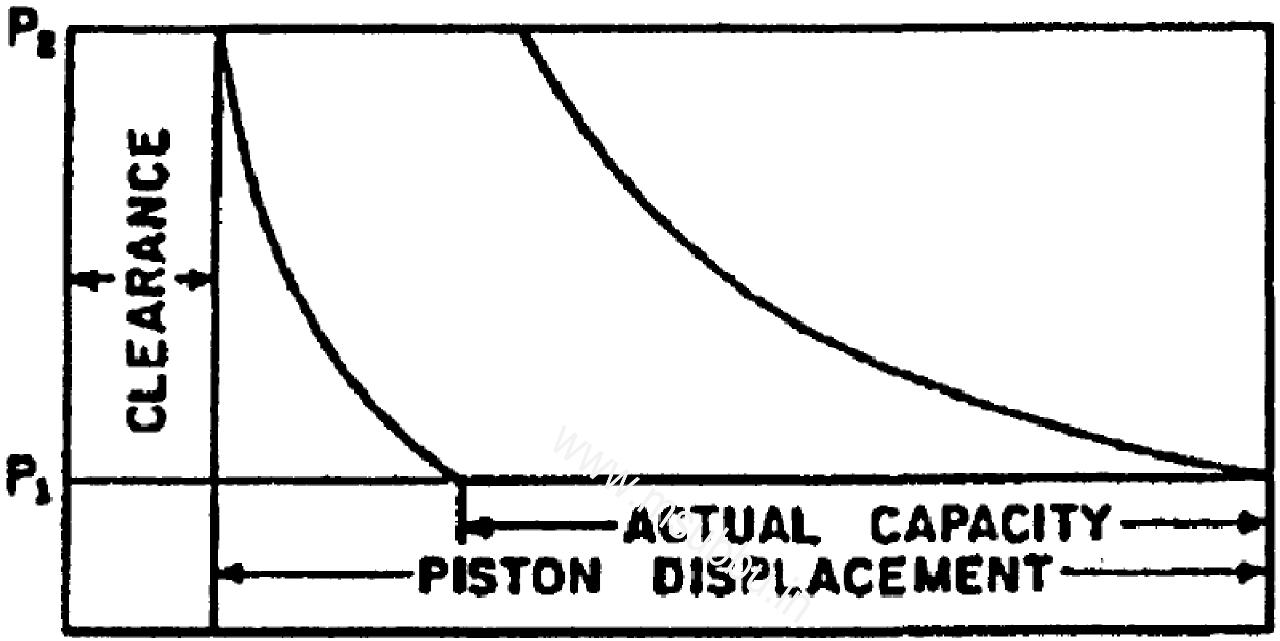
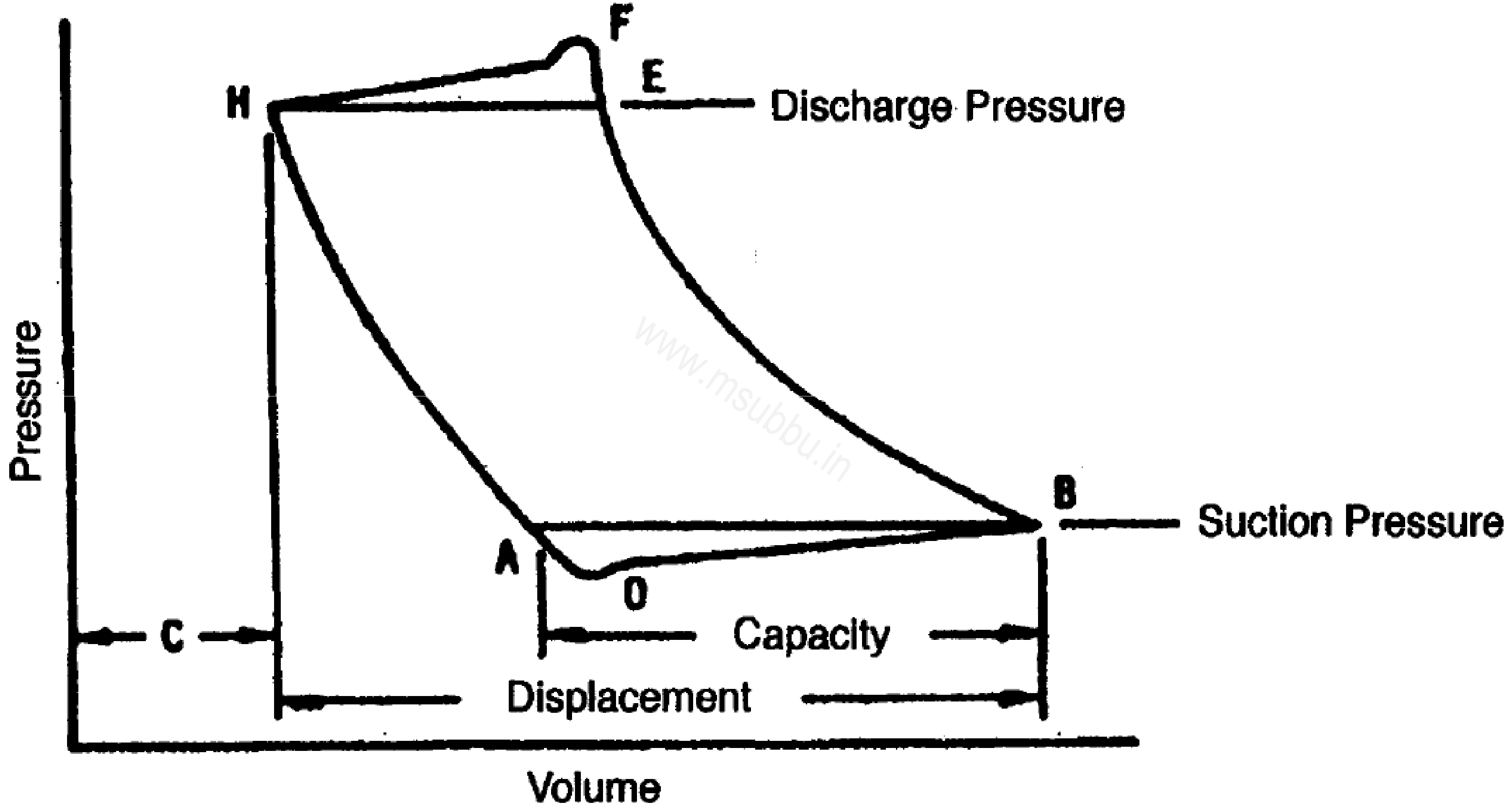
$$\eta_v = \frac{V_2 - V_5}{V_2 - V_1} = \frac{\text{Volume of air pumped /cycle}}{\text{Stroke volume of cylinder}} = \frac{\text{mass of air delivered/cycle}}{\text{mass of air to fill stroke volume}}$$

$$\left(\frac{V_2}{V_3}\right) = \left(\frac{P_3}{P_2}\right)^n = \left(\frac{P_4}{P_5}\right)^n = \left(\frac{V_5}{V_4}\right)$$



Effect of clearance on the capacity of a reciprocating compressor.


F
E
H
Discharge Pressure
Pressure
B
Suction Pressure
A
O
C
Capacity
Displacement
Volume
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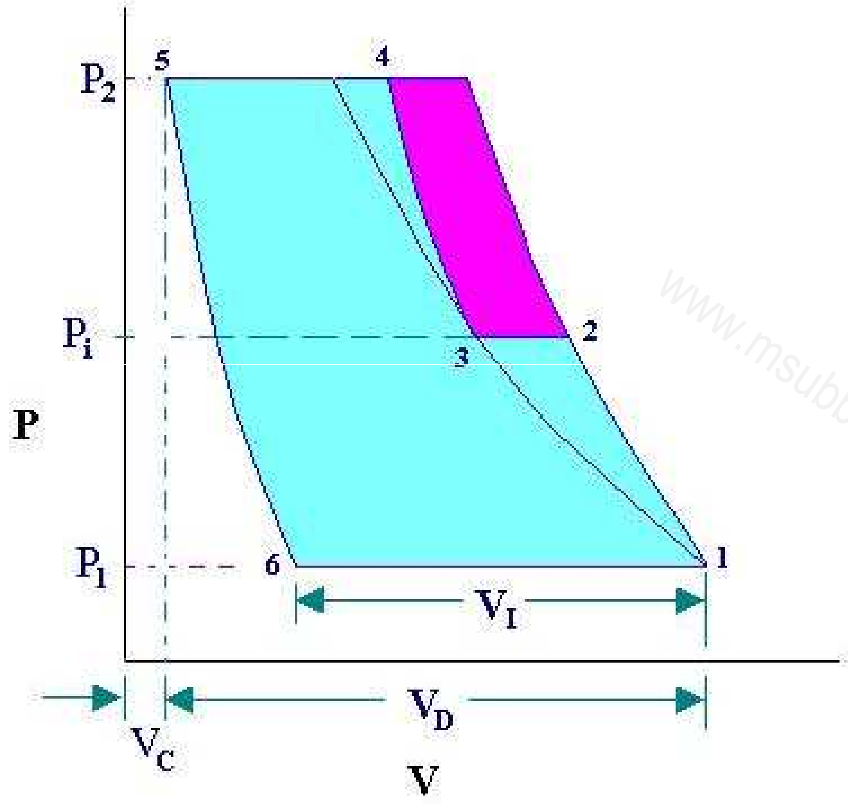
Multistage Compression



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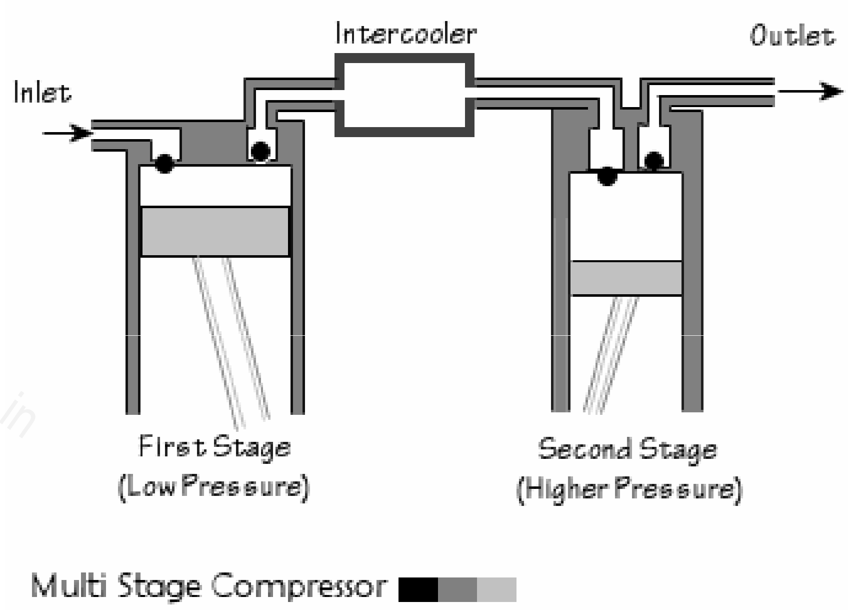

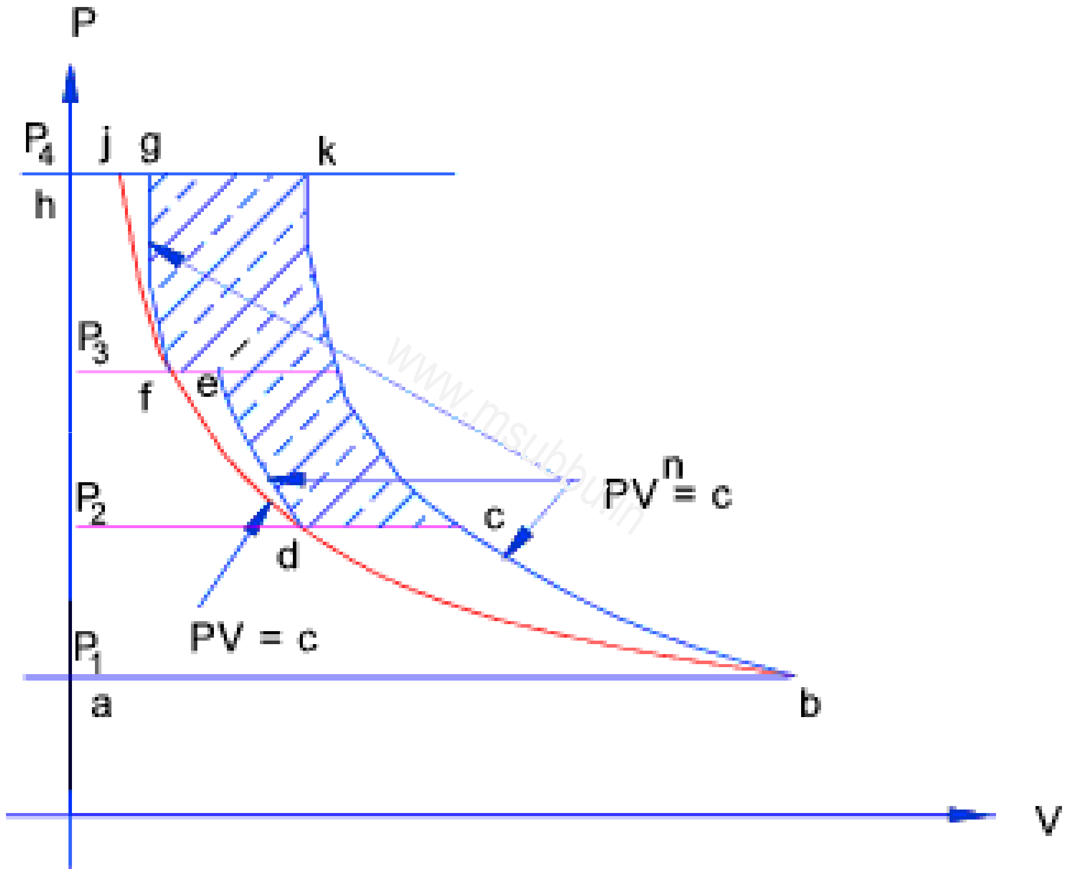

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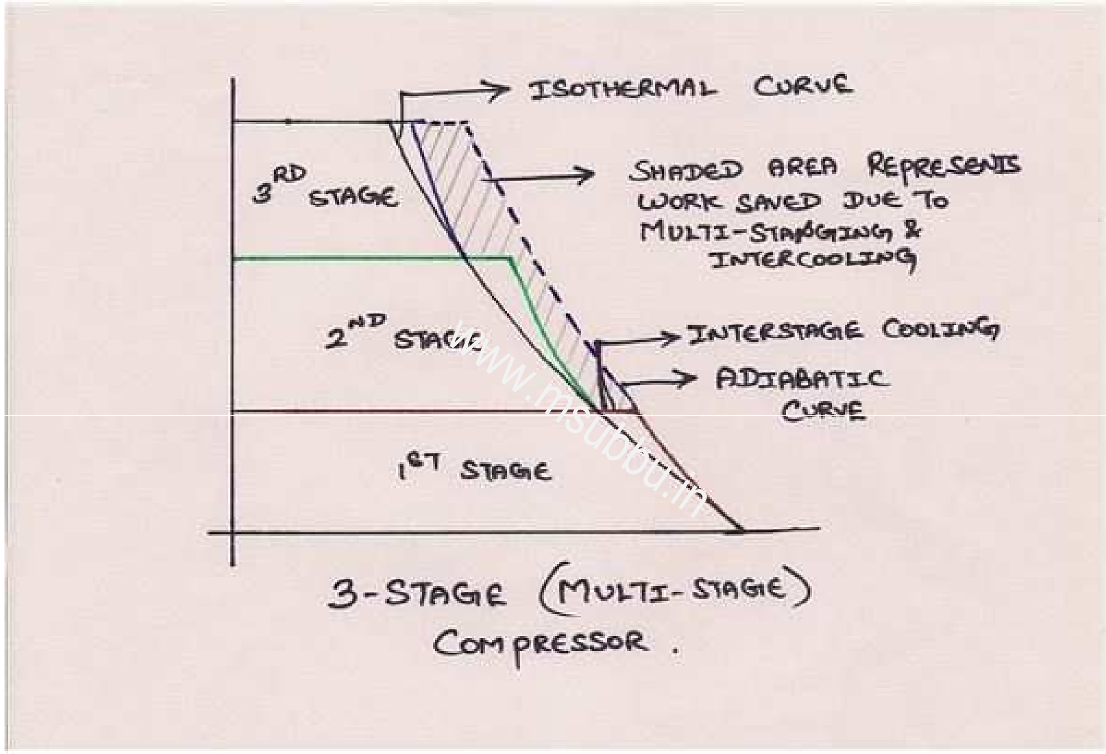

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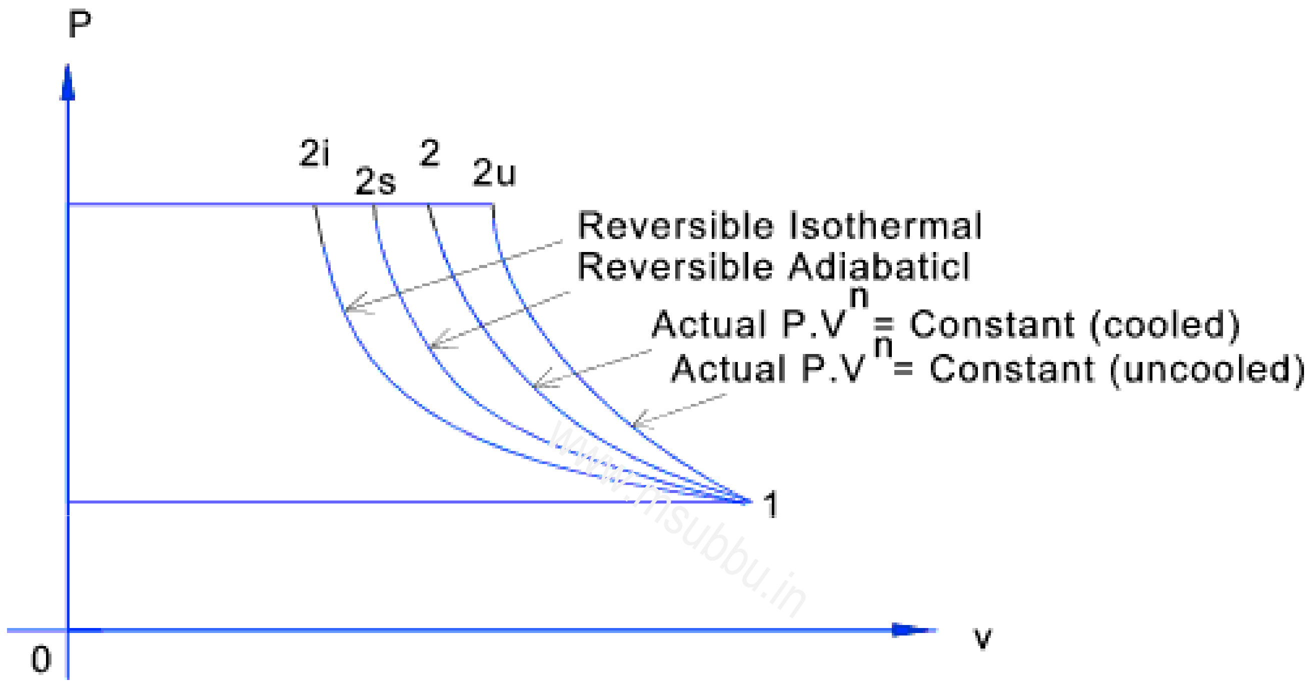
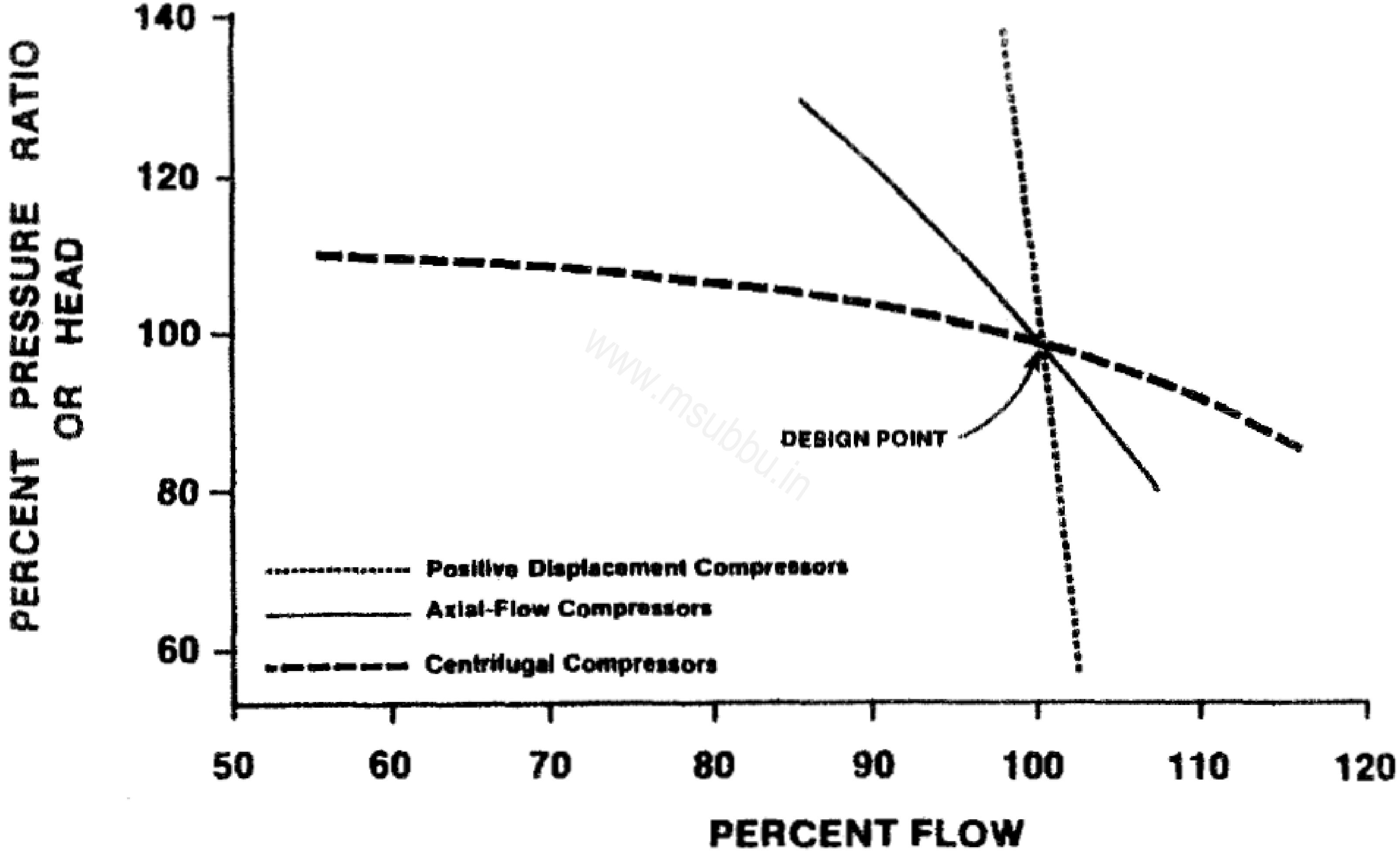


$$\eta_s = \frac{\text{Calculated power for reversible adiabatic compression}}{\text{indicated power of compressor}}$$

$$\eta_s = \frac{(h_{2s} - h_1)}{(h_{2u} - h_1)} = \frac{(T_{2s} - T_1)}{(T_{2u} - T_1)}$$



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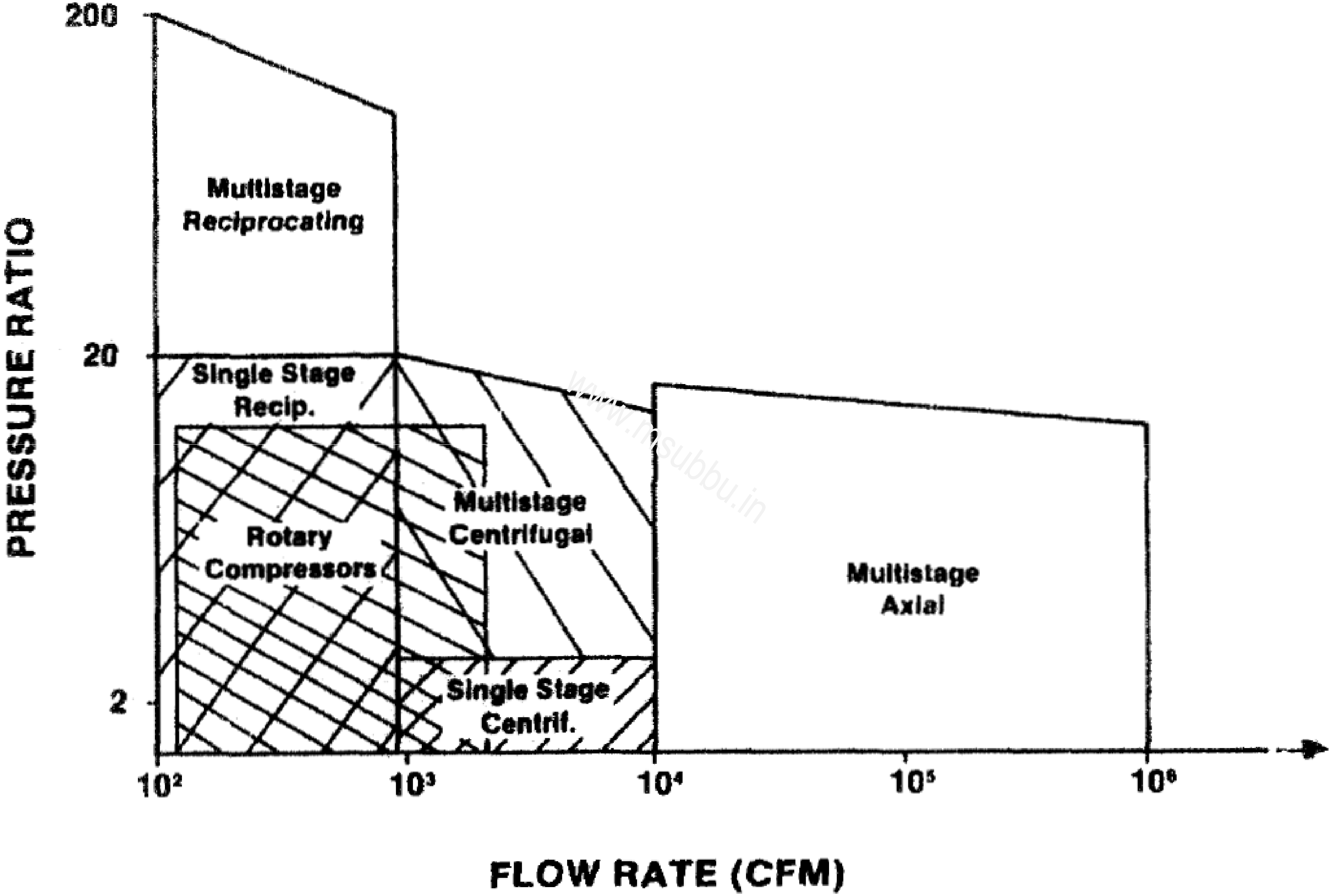

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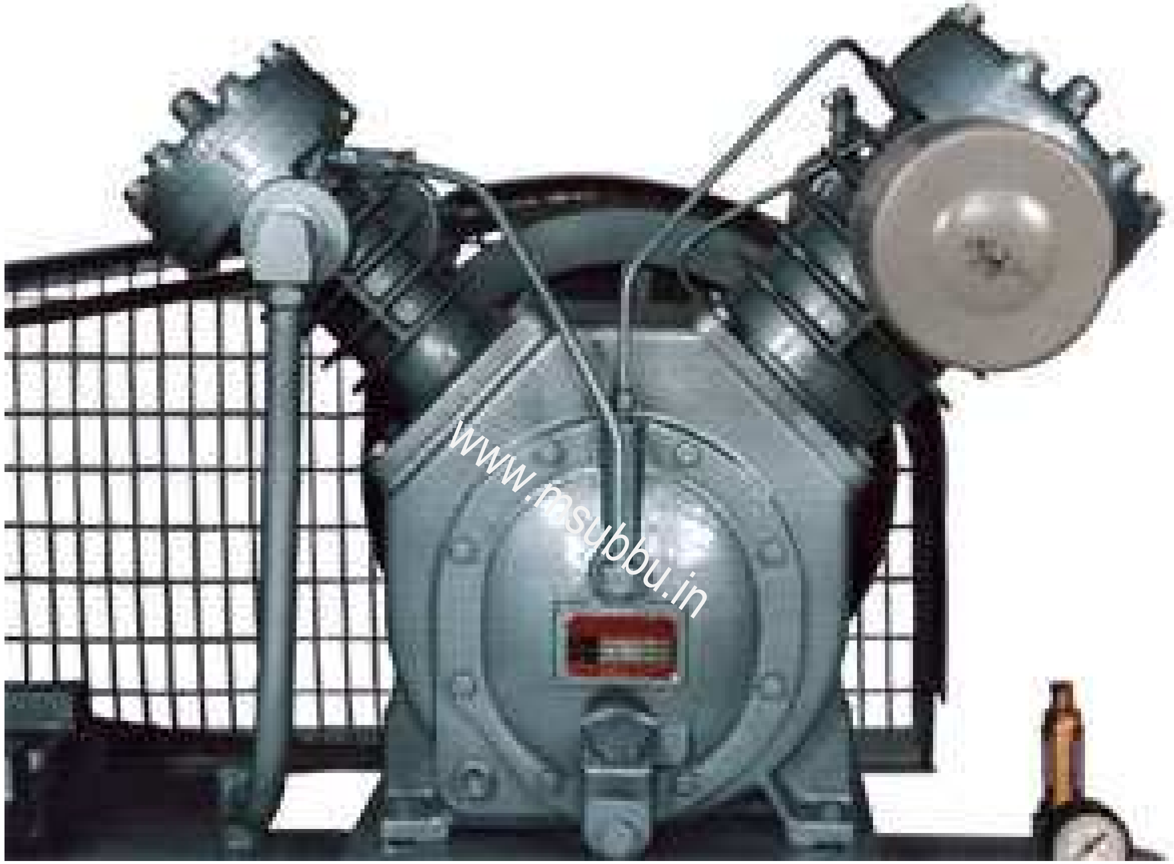
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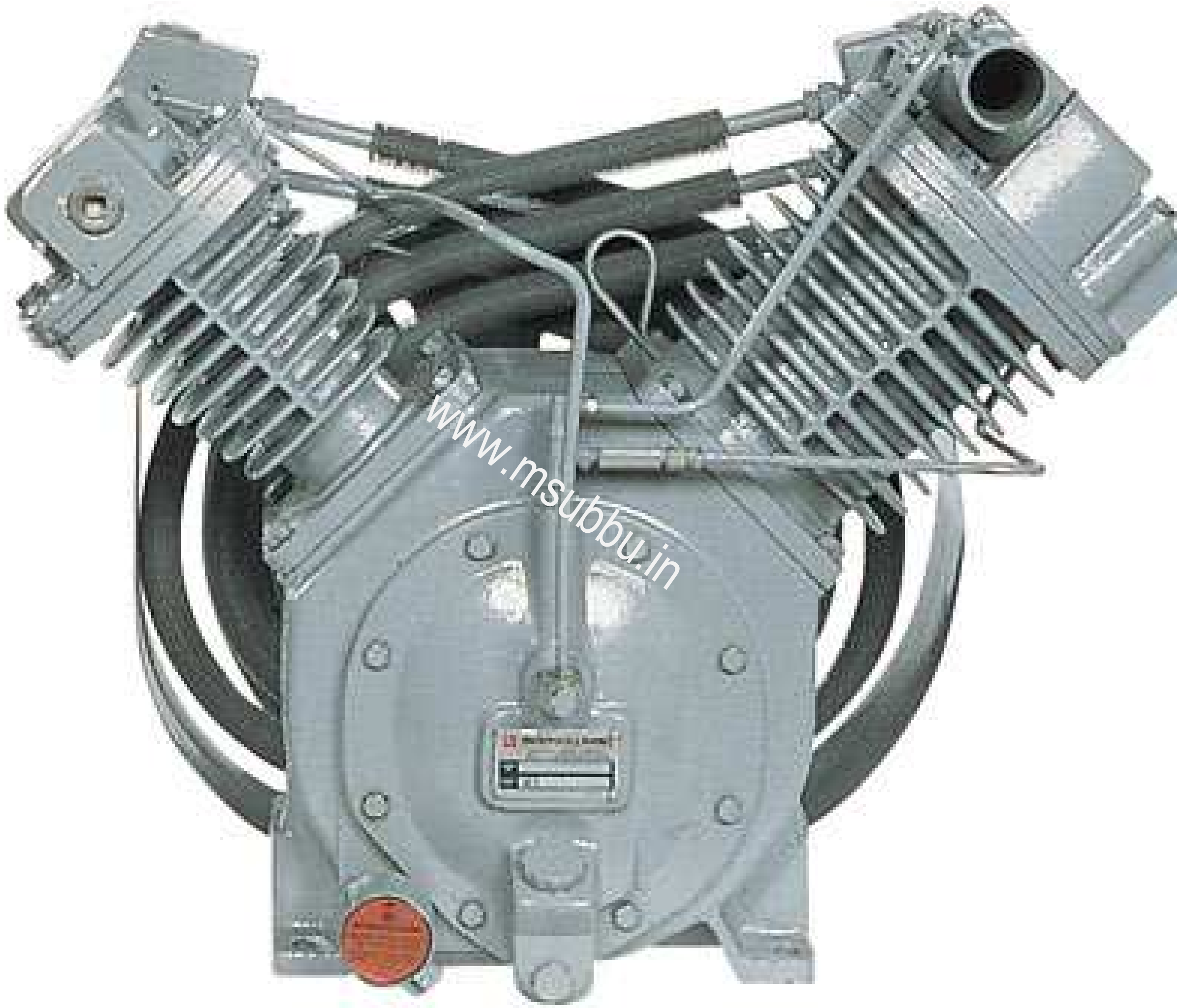
Two stage reciprocating compressor



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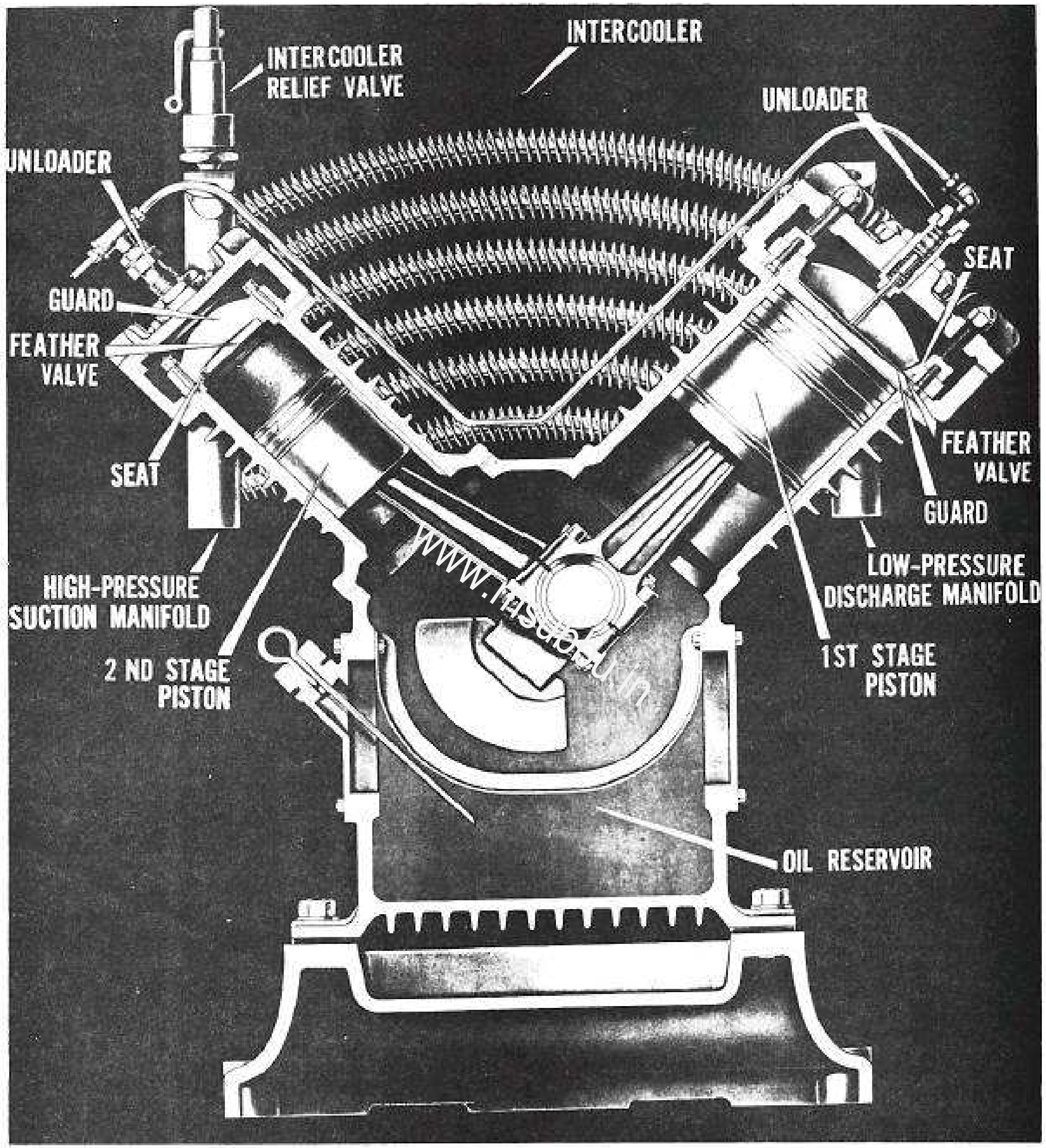
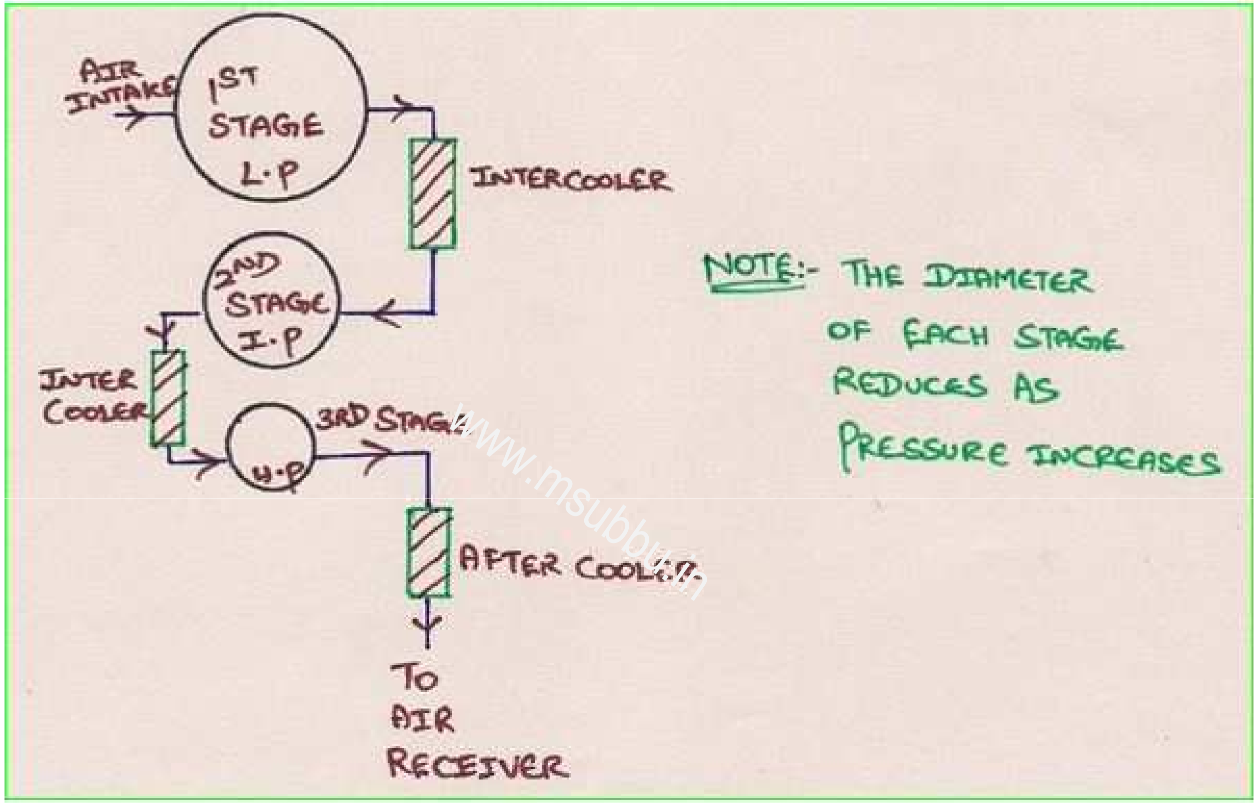

INTERCOOLER
INTERCOOLER RELIEF VALVE
UNLOADER
UNLOADER
SEAT
GUARD
FEATHER VALVE
SEAT
FEATHER VALVE
GUARD
LOW-PRESSURE DISCHARGE MANIFOLD
HIGH-PRESSURE SUCTION MANIFOLD
2 ND STAGE PISTON
1ST STAGE PISTON
OIL RESERVOIR



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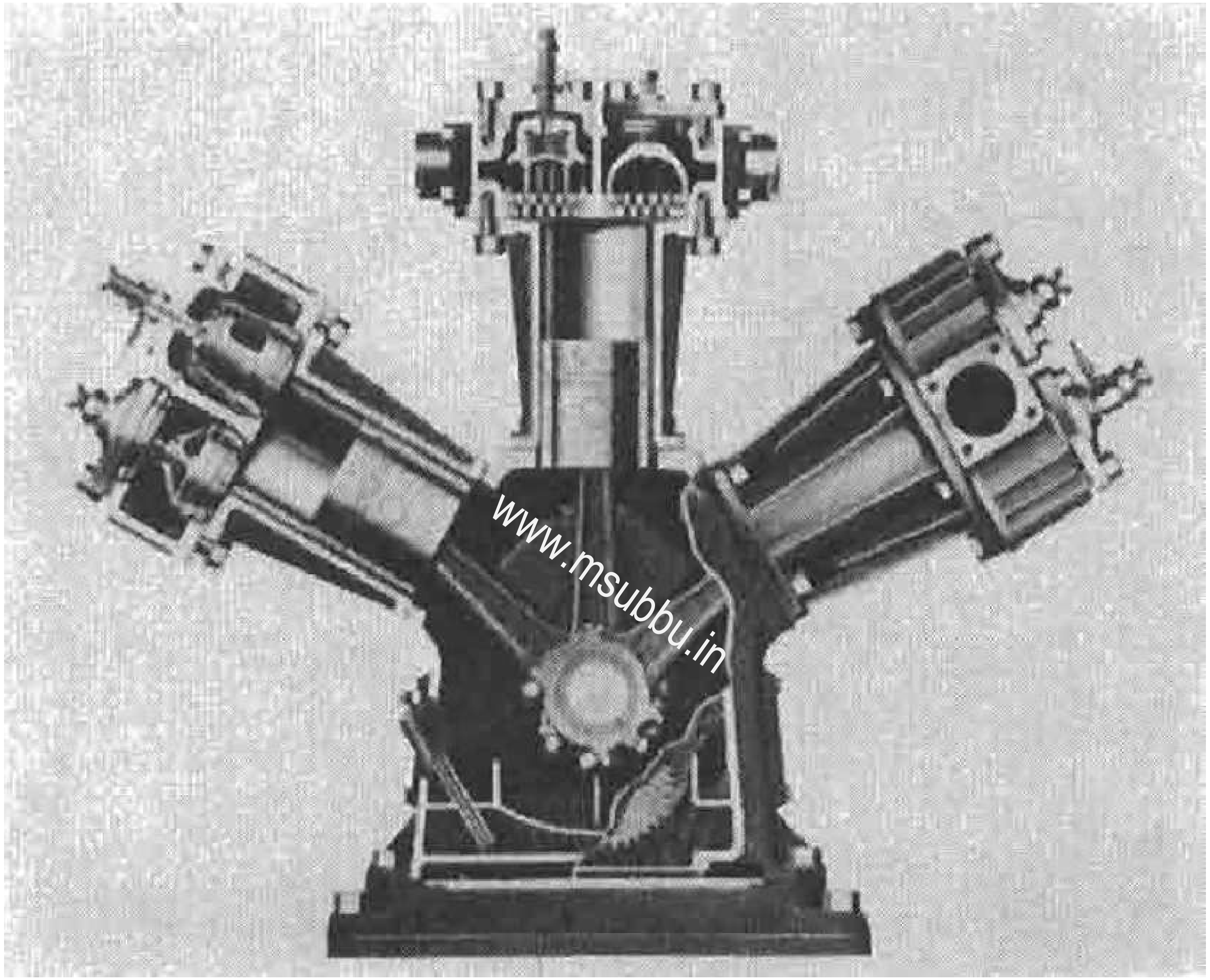
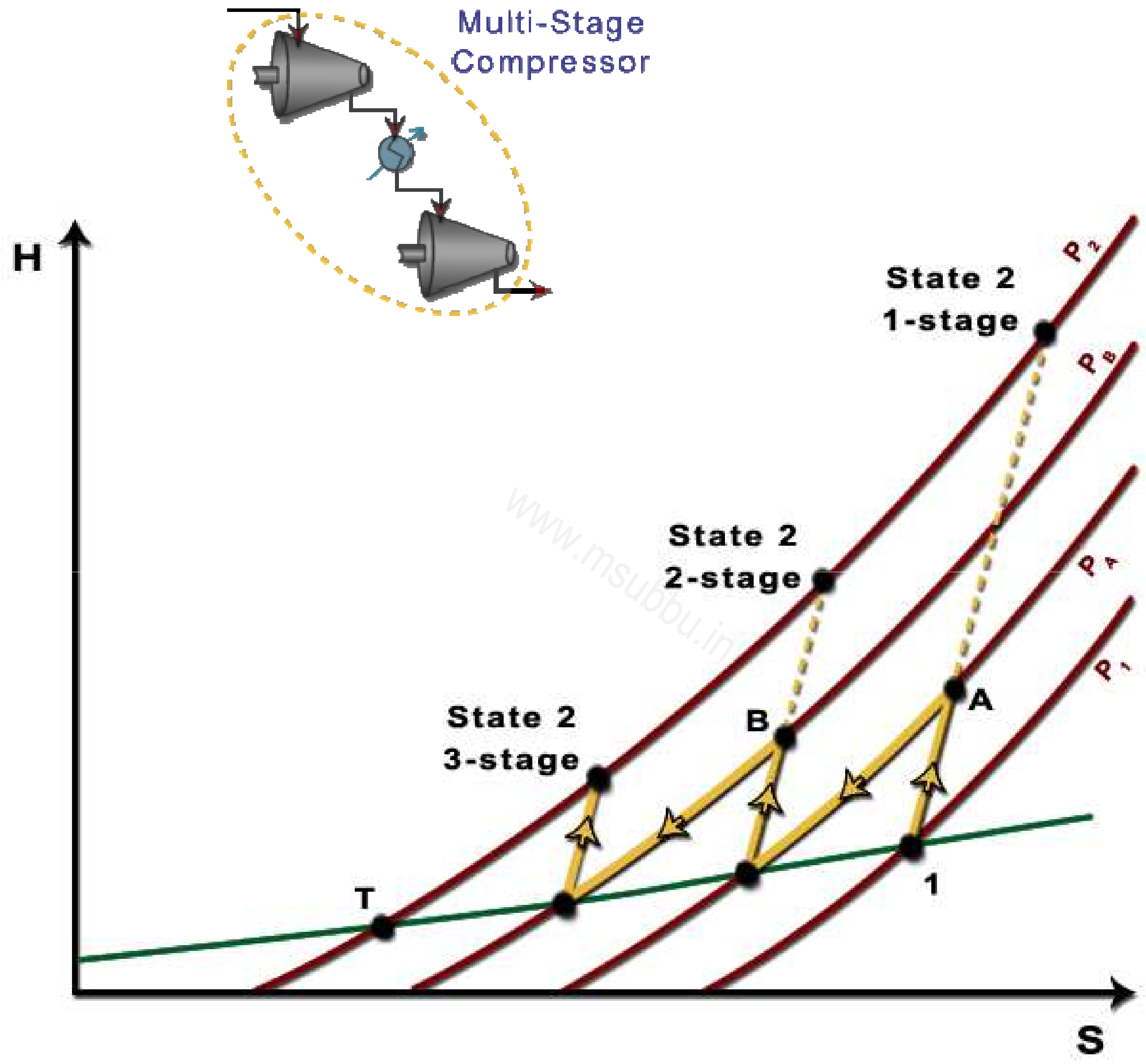


Three stage reciprocating compressor



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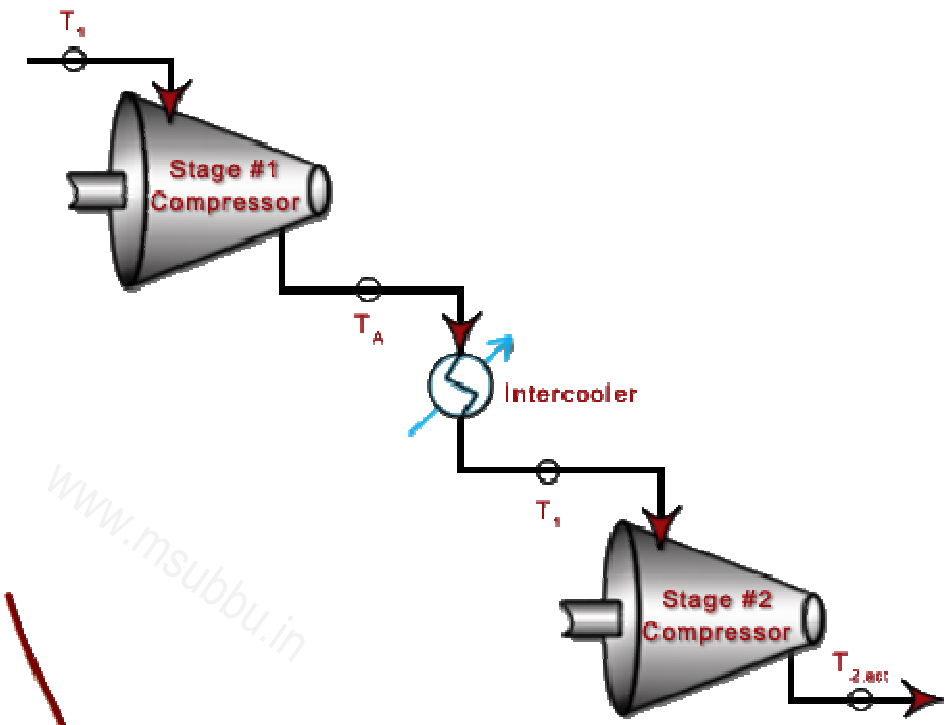

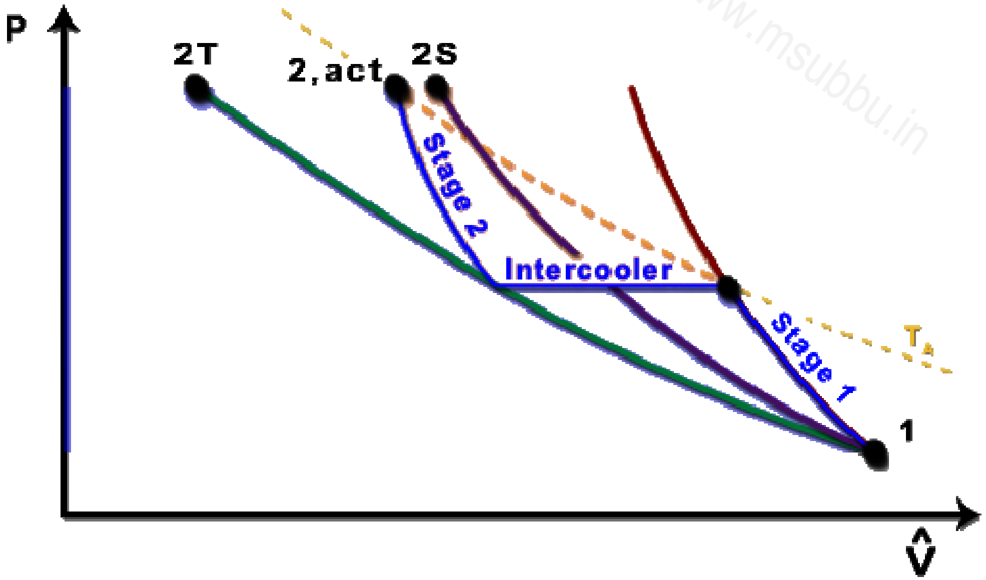

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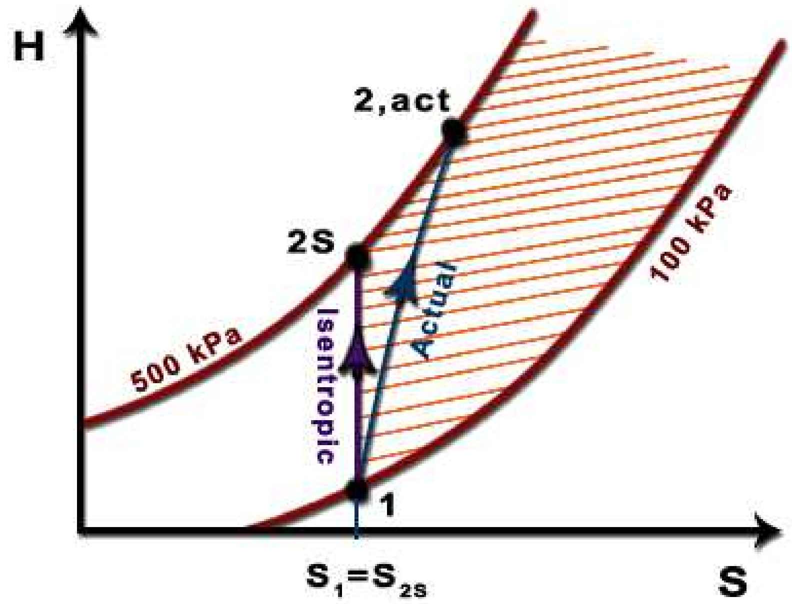

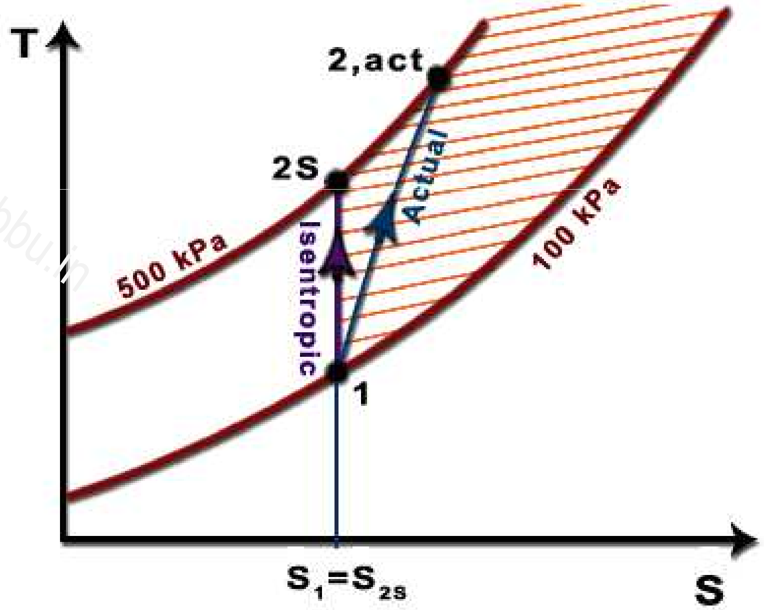

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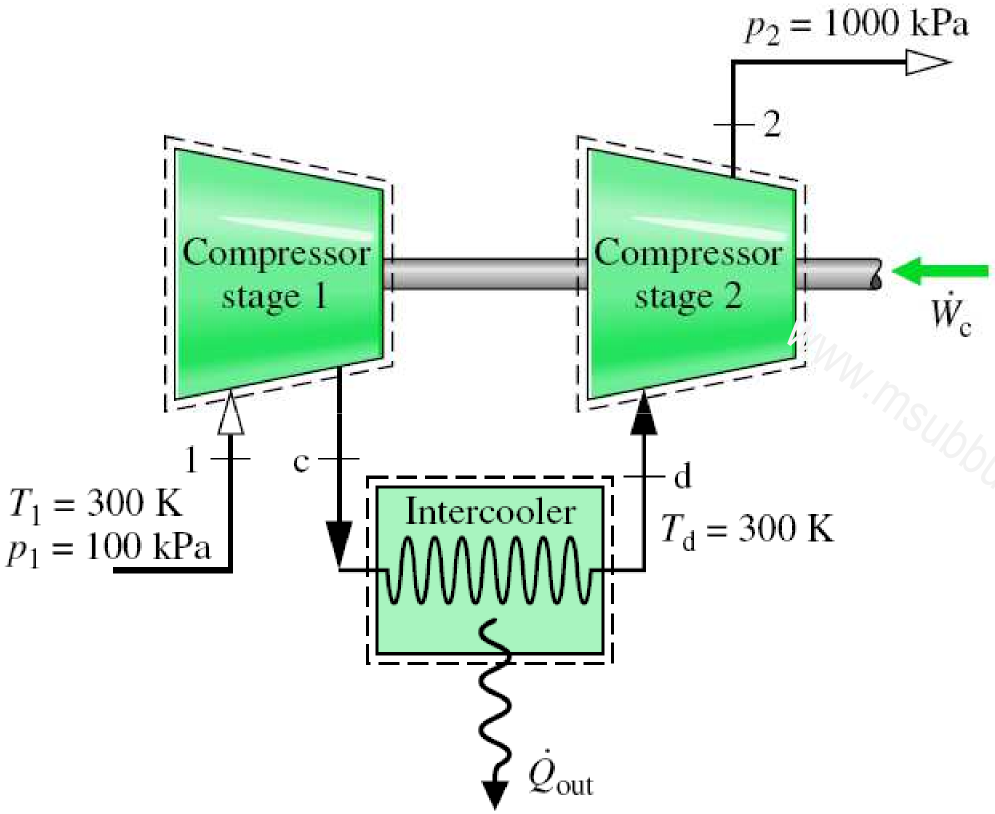

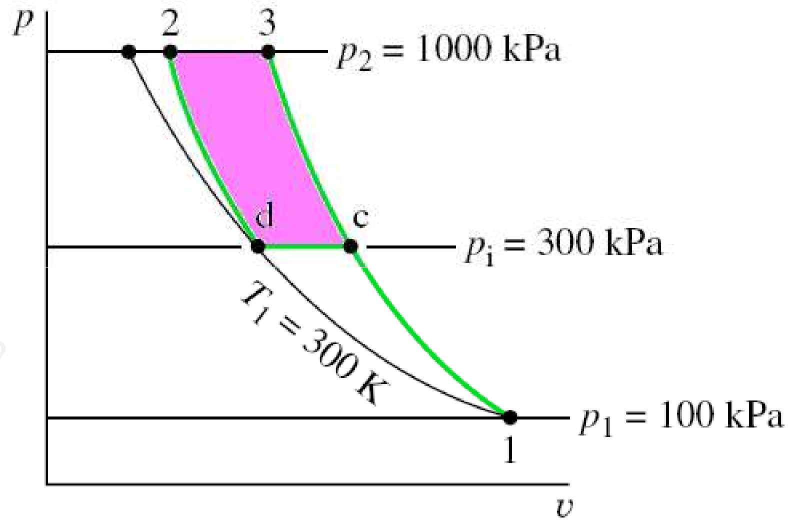

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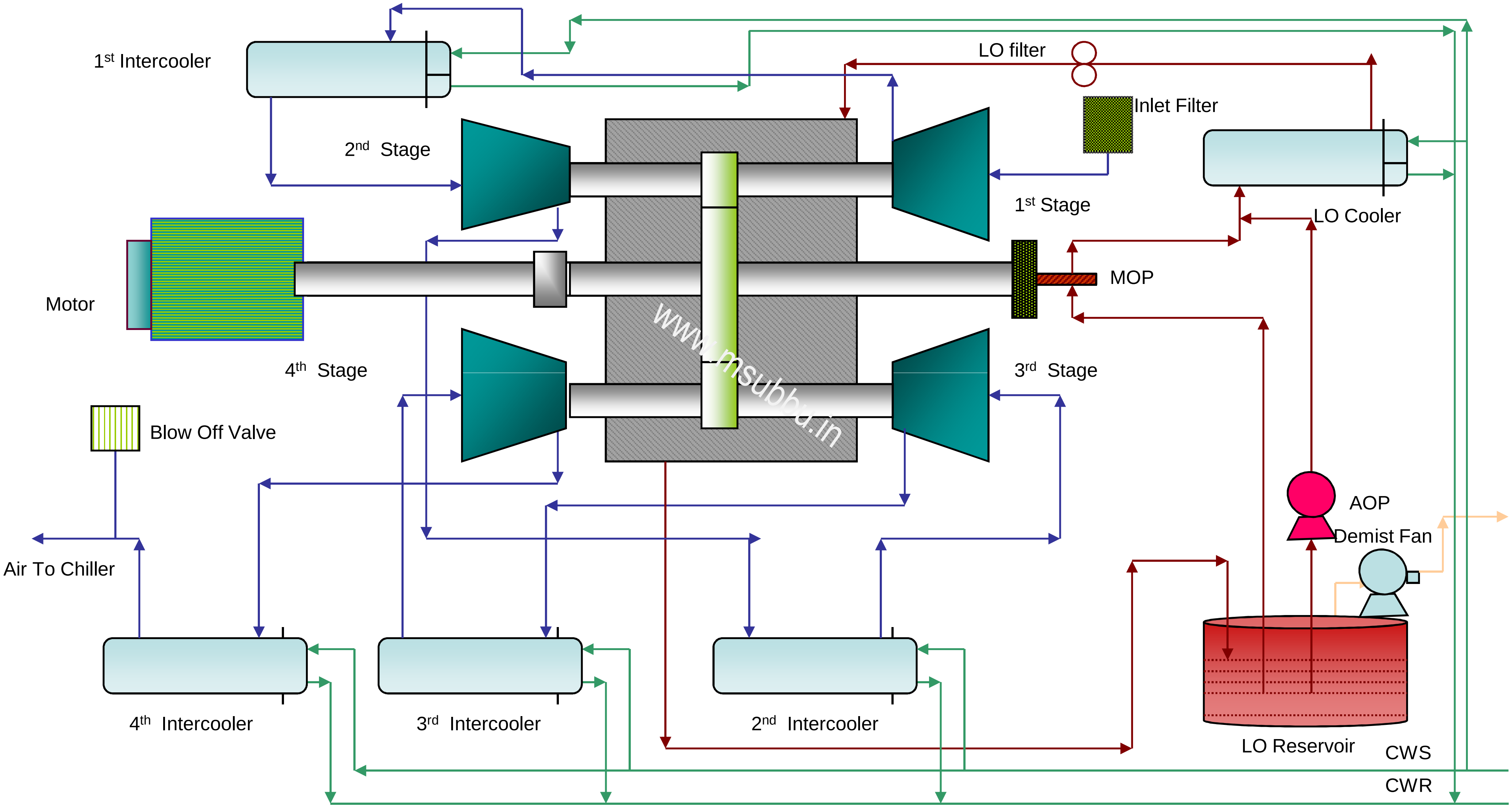
Compressor- Flow Description



Expansion

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Contents

- Velocity of sound
- Relation between area and velocity
- Flow through pipe
- Chocked flow

- Ejectors

- Turbine
- Joule-Thompson expansion

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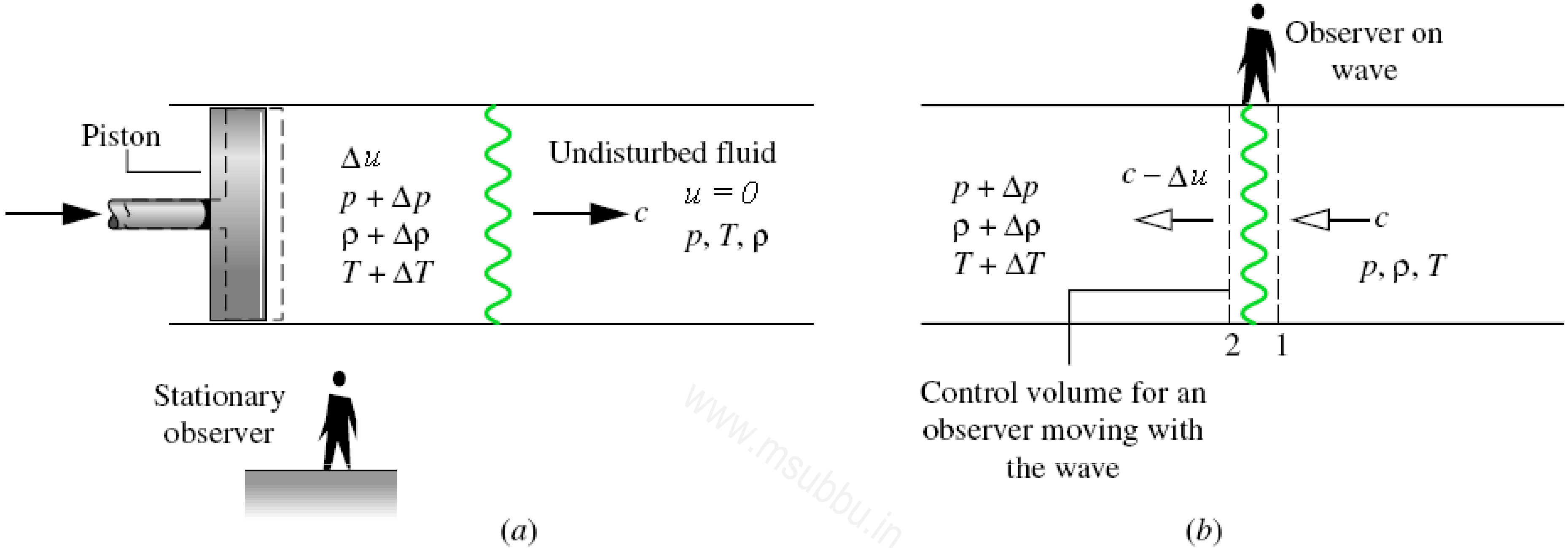
Velocity of Sound



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▲ **Figure 9.27** Illustrations used to analyze the propagation of a sound wave. (*a*) Propagation of a pressure wave through a quiescent fluid, relative to a stationary observer. (*b*) Observer at rest relative to the wave. Moran - Fundamentals of Engineering Thermodynamics 5E,

It is easier to analyze this situation from the point of view of an observer at rest relative to the wave.

Mass balance:

$$\dot{m} = \text{constant}$$

$$\rho A c = (\rho + \Delta\rho)A(c - \Delta u)$$

$$0 = c\Delta\rho - \rho\Delta u - \Delta\rho\Delta u$$

$$\Delta u = (c/\rho)\Delta\rho$$

Momentum Balance:

$$\begin{aligned} PA + \dot{m}c &= \dot{m}(c - \Delta u) + (P + \Delta P)A \\ -\Delta PA &= \dot{m}(-\Delta u) \\ &= (\rho A c)(-\Delta u) \end{aligned}$$

$$\Delta P = \rho c \Delta u$$

$$\Delta u = (c/\rho)\Delta\rho$$

$$c = \sqrt{\frac{\Delta P}{\Delta \rho}}$$

$$c = \sqrt{\left(\frac{\partial P}{\partial \rho}\right)_s}$$

Since $\rho = 1/V$, $d\rho = (-1/V^2)dV$

$$\boxed{c = \sqrt{-V^2 \frac{dP}{dV}}}$$

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Compressible Flow: Relation between Area and Velocity

For steady state flow, $\dot{m}$ = constant. Since $\dot{m} = \rho A u = Au/V$, we have

$$d(uA/V) = 0$$

Expanding the above equation, we get

$$\frac{1}{V}(udA + Adu) - uA\frac{dV}{V^2} = 0$$

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or

$$\frac{udA + Adu}{uA} = \frac{VdV}{V^2}$$

i.e.,

$$\frac{dA}{A} + \frac{du}{u} = \frac{VdV}{V^2} \tag{1.15}$$

From the fundamental property relation for $dH(= VdP + TdS)$ and from steady flow energy equation (i.e., from Eqn.(1.18)) we have

$$-VdP = udu \qquad \text{(const. } S\text{)}$$

i.e., $V = -udu/dP$ at constant S. Therefore

$$\frac{dA}{A} + \frac{du}{u} = \frac{udu}{-V^2(\partial P/\partial V)_S}$$

From the relation for velocity of sound as given in Eqn.(1.14), the Eqn.(1.15) becomes

$$\frac{dA}{A} + \frac{du}{u} = \frac{udu}{c^2}$$

Therefore

$$\frac{dA}{A} = \frac{udu}{c^2} - \frac{du}{u} = \left(\frac{u^2}{c^2} - 1\right)\frac{du}{u}$$

The ratio of actual velocity (u) to the velocity of sound (c) is called the *Mach Number* **M**.

$$\frac{dA}{A} = (\mathbf{M}^2 - 1)\frac{du}{u} \qquad (1.16)$$

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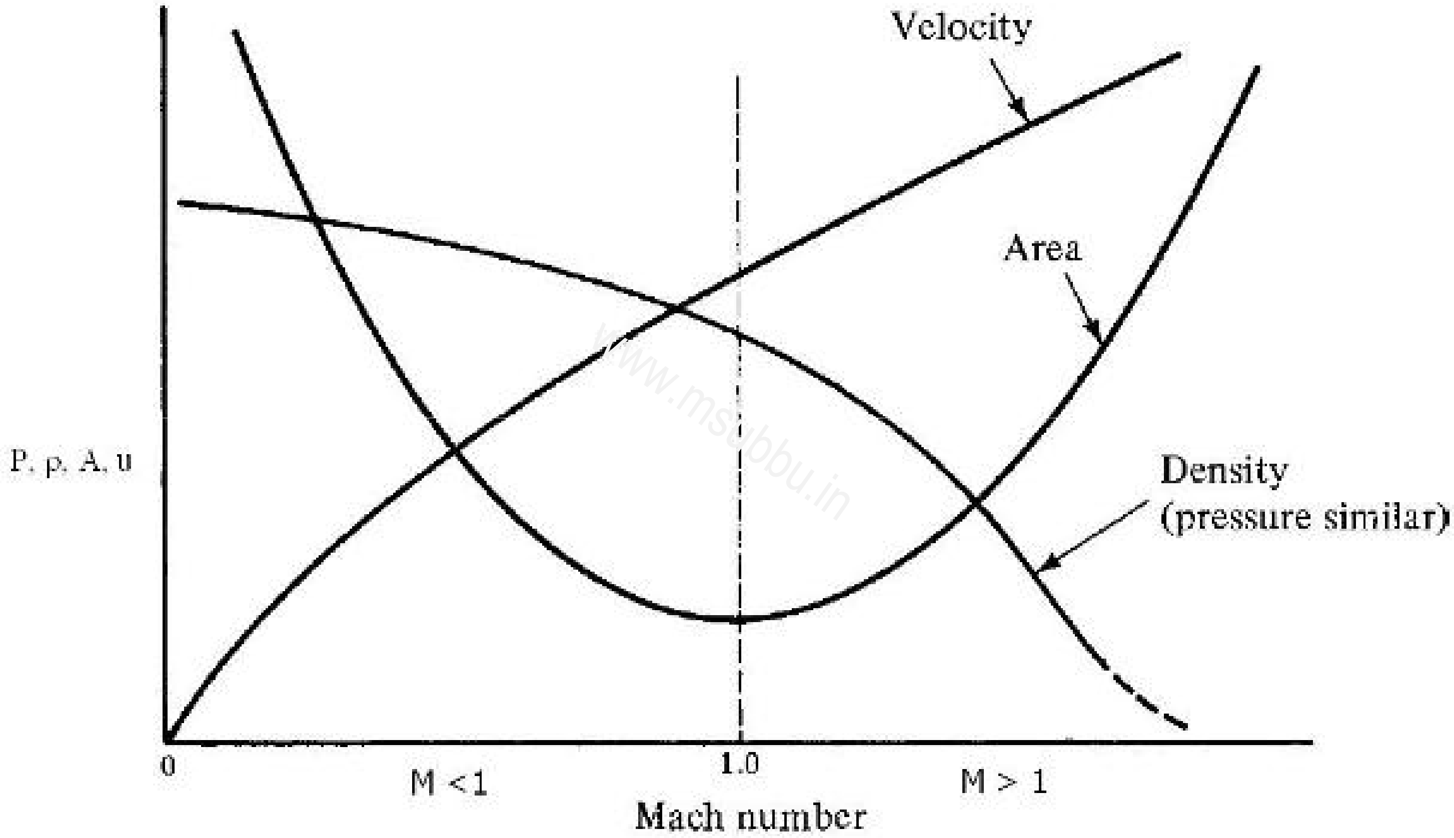
Nozzle, Diffuser sizing

$$\frac{dA}{A} = (\mathrm{M}^2 - 1)\frac{du}{u}$$

Inlet Velocity	**Nozzle (du > 0, dP < 0)**	**Diffuser (du < 0, dP > 0)**
Subsonic (M < 1)	A decreases	A increases
Supersonic (M < 1)	A increases	A decreases

Mach number (M): M = u/c

Depending on whether M is greater than unity (supersonic) or less than unity (subsonic), the cross sectional area increases or decreases with velocity increase.



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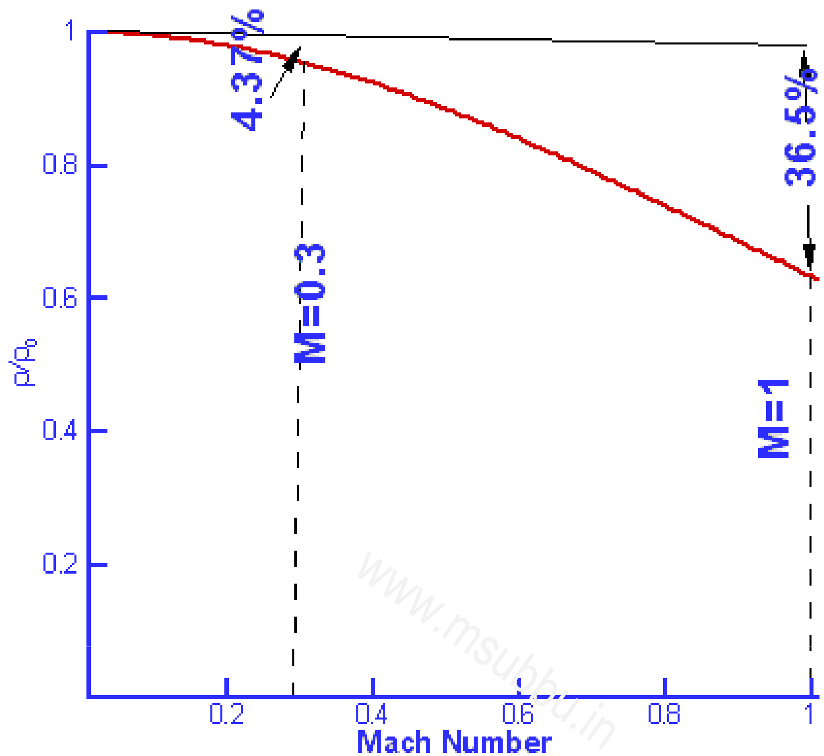
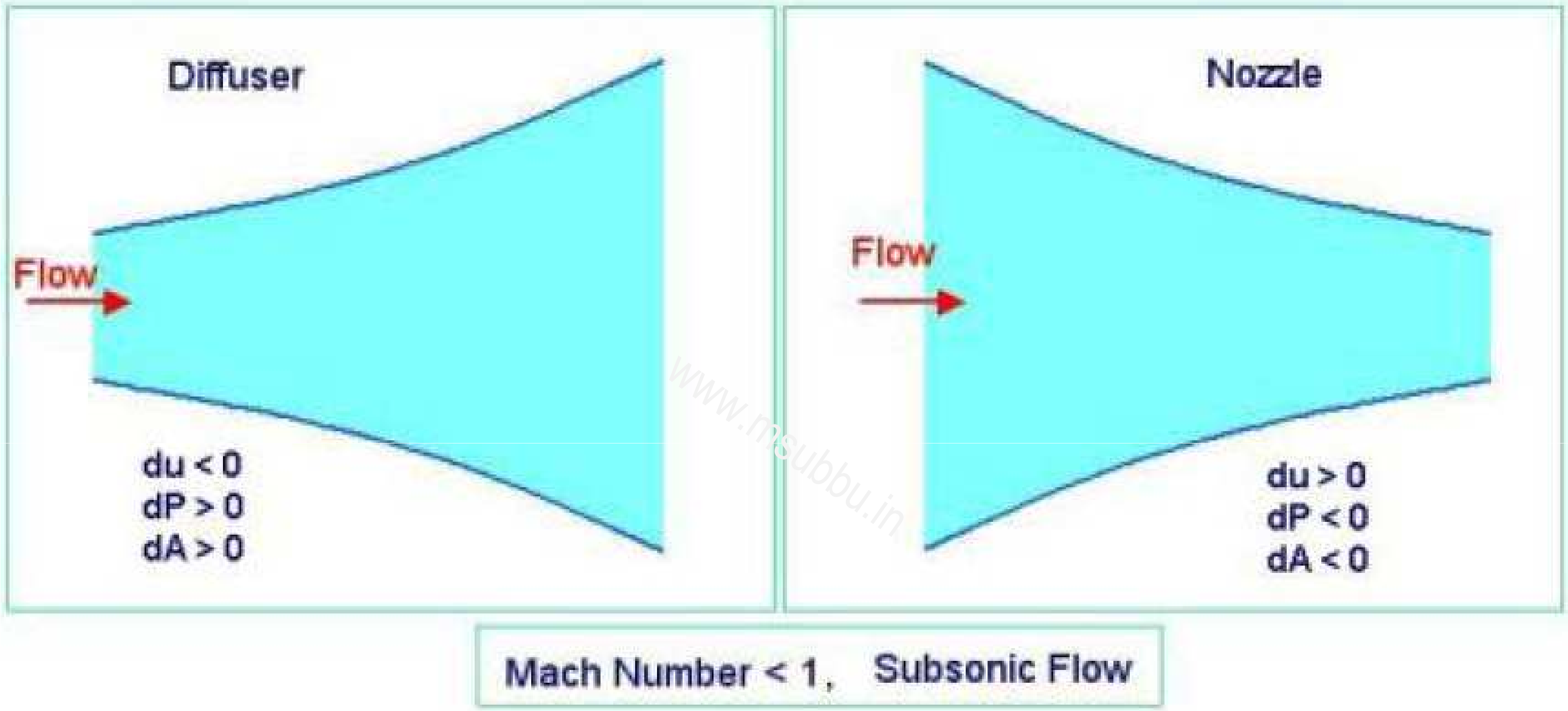
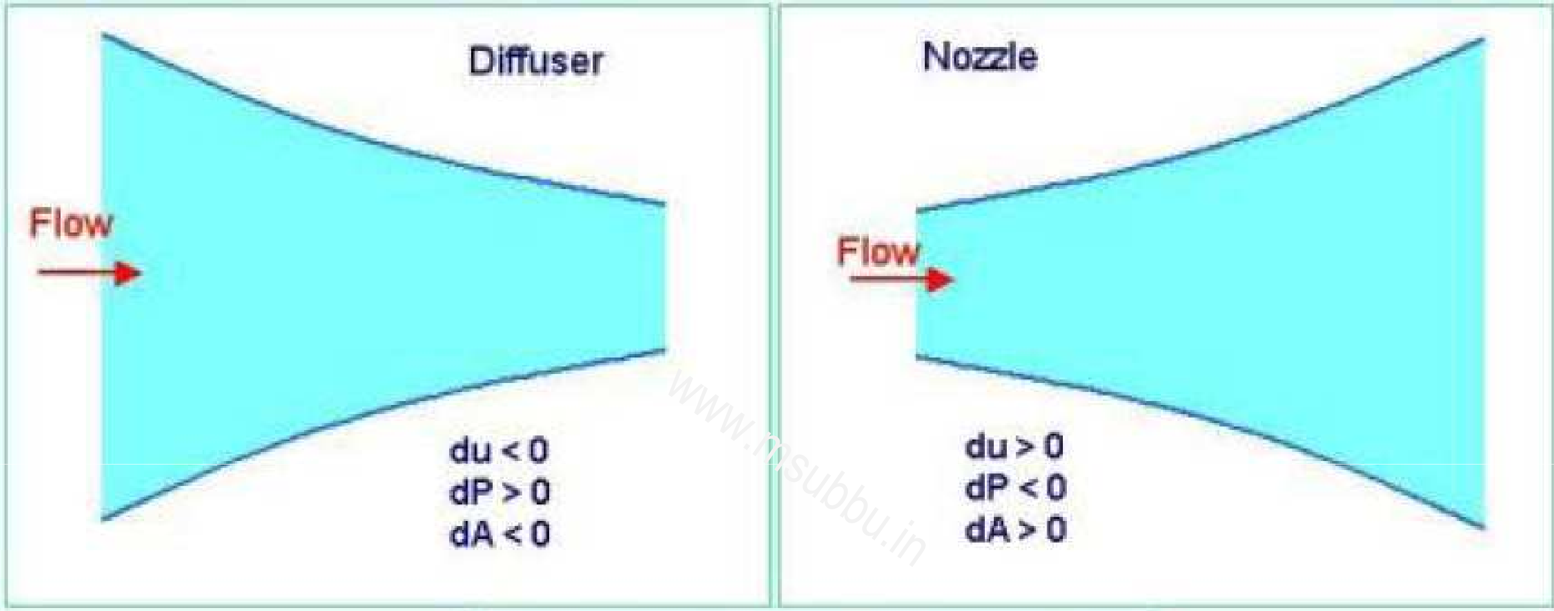


Though gases are compressible, the density changes they undergo at low speeds may not be considerable. Take air for instance. Figure shows the density changes plotted as a function of Mach Number.

We observe that for Mach numbers up to 0.3, density changes are within about 5% of . So for all practical purposes one can ignore density changes in this region. But as the Mach Number increases beyond 0.3, changes do become appreciable and at a Mach Number of 1, it is 36.5% . it is interesting to note that at a Mach Number of 2, the density changes are as high as 77%. **It follows that air flow can be considered incompressible for Mach Numbers below 0.3.**


Diffuser
Flow
du < 0
dP > 0
dA > 0
Nozzle
Flow
du > 0
dP < 0
dA < 0
Mach Number < 1, Subsonic Flow



Mach Number > 1, Supersonic Flow

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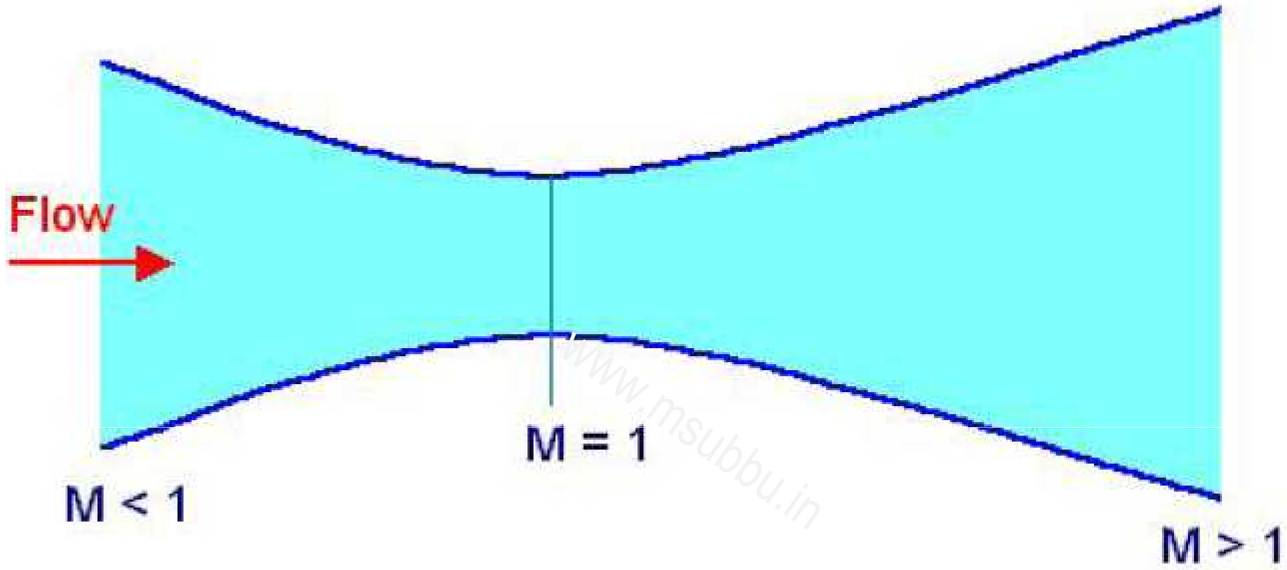

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Converging-diverging Supersonic Nozzle

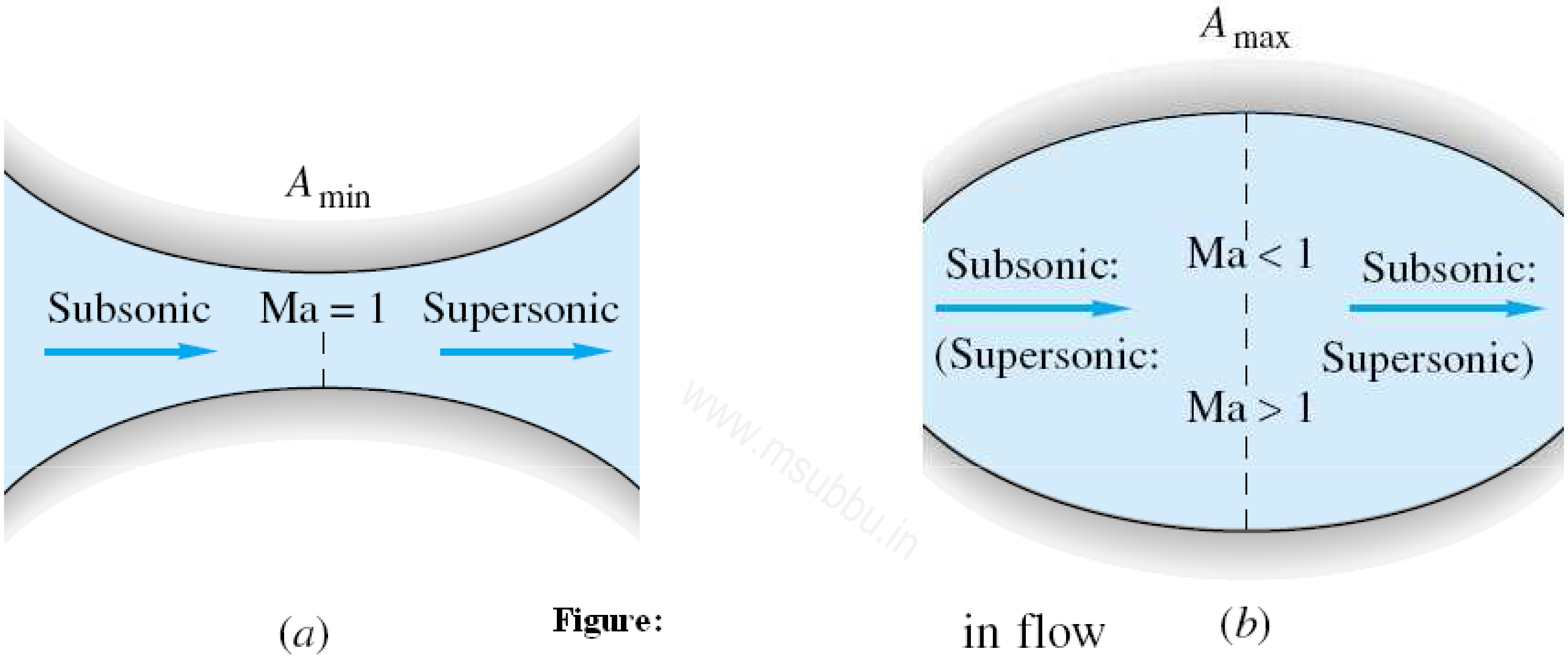

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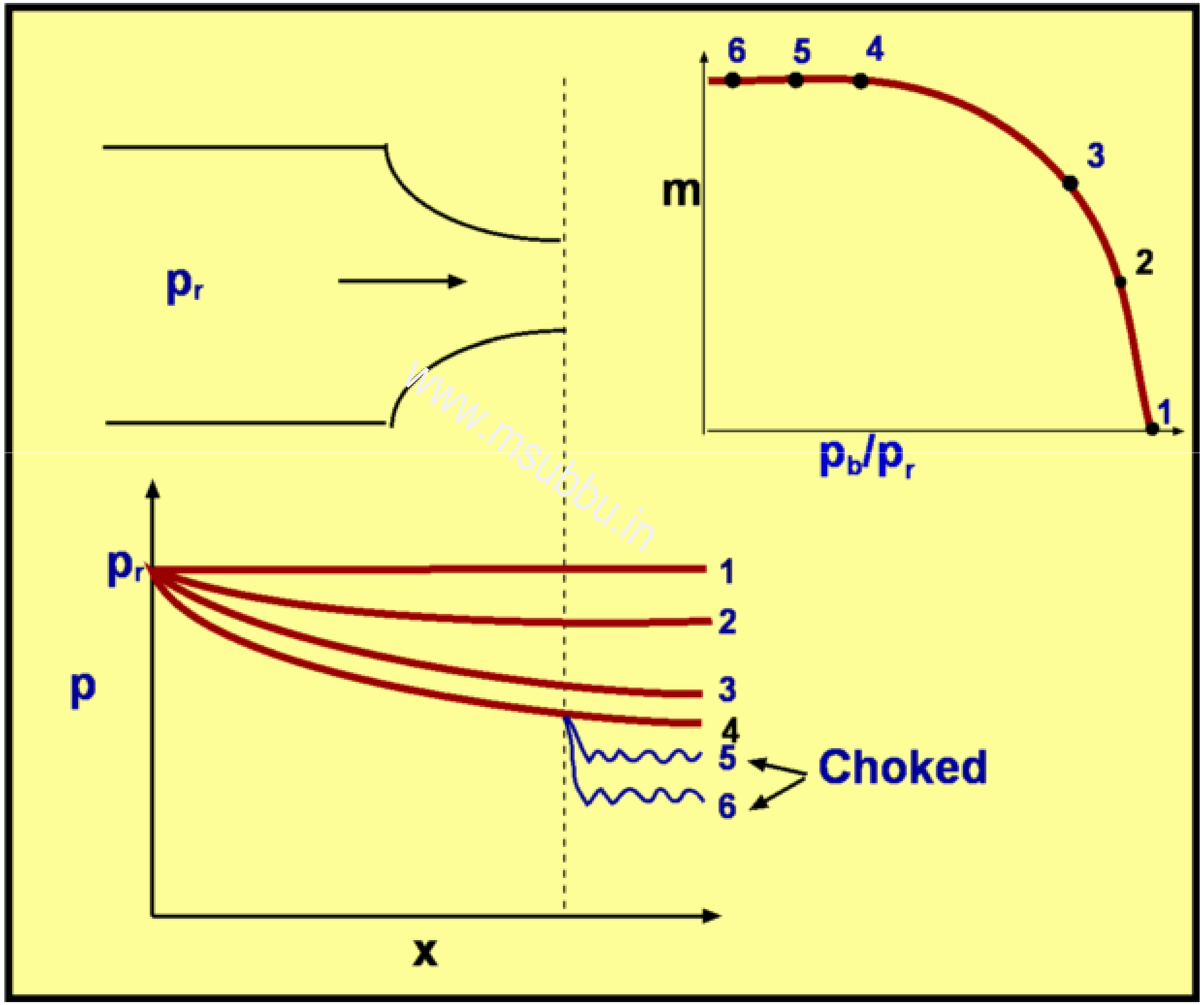
Figure: in flow through a throat (*a*) the fluid can accelerate smoothly through sonic and supersonic flow. In flow through the bulge (*b*) the flow at the bulge cannot be sonic on physical grounds.

Flow Through Pipes of Constant Cross Section



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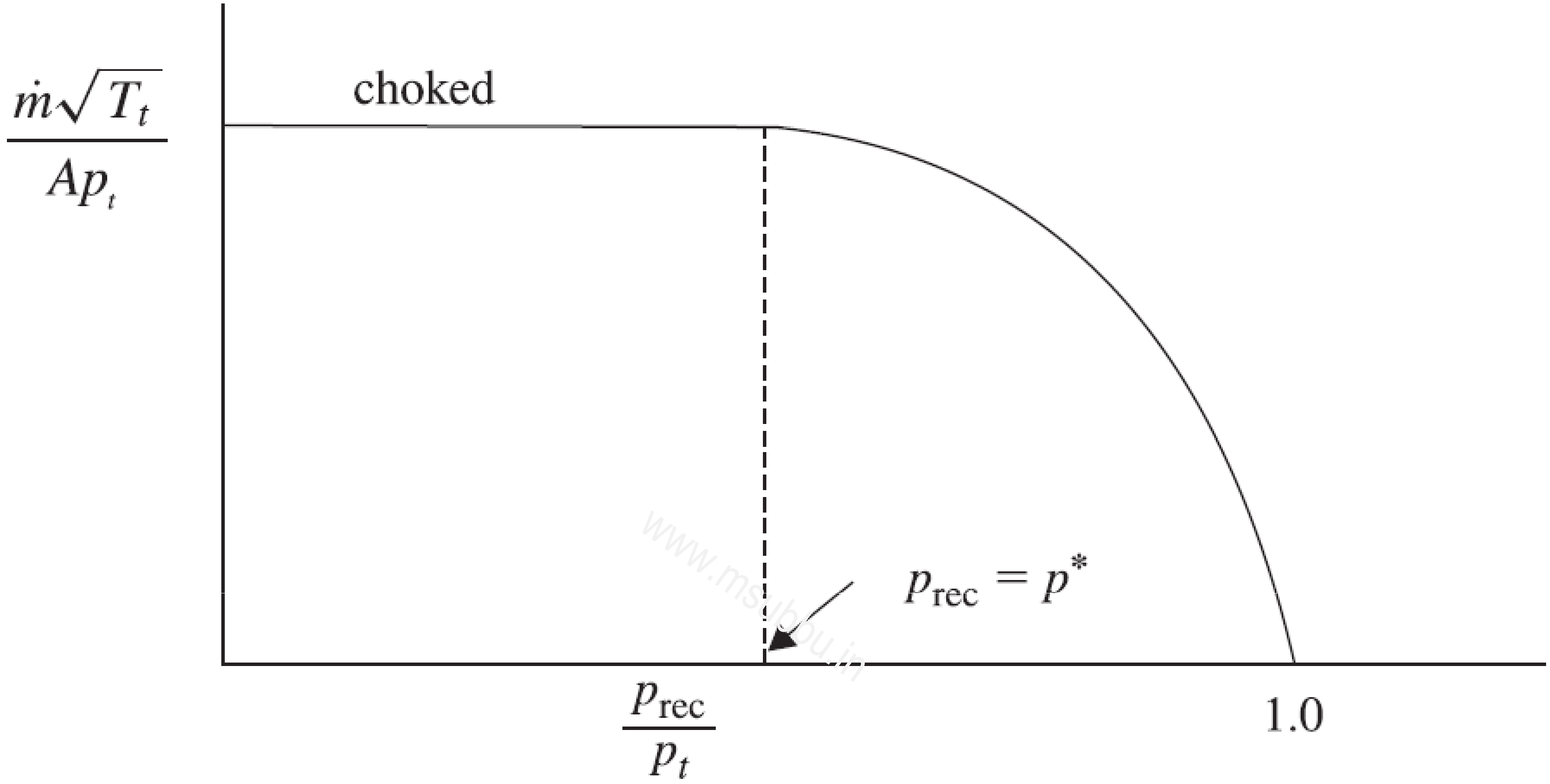
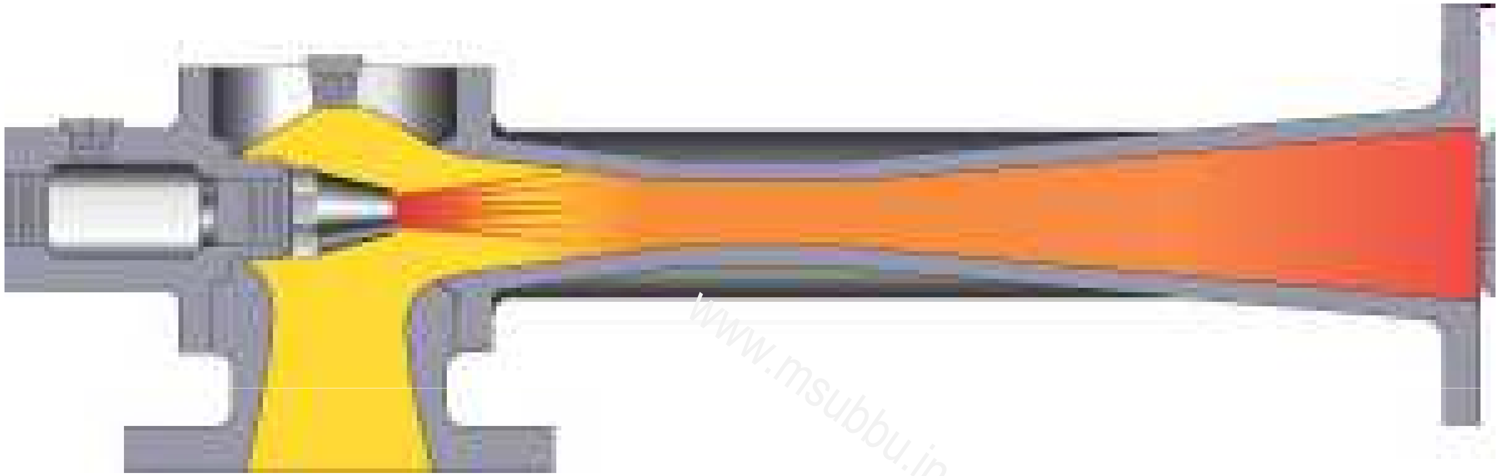


Choked flow: the maximum possible flow in pipe of constant cross section, also in discharge through a nozzle with inlet at subsonic conditions

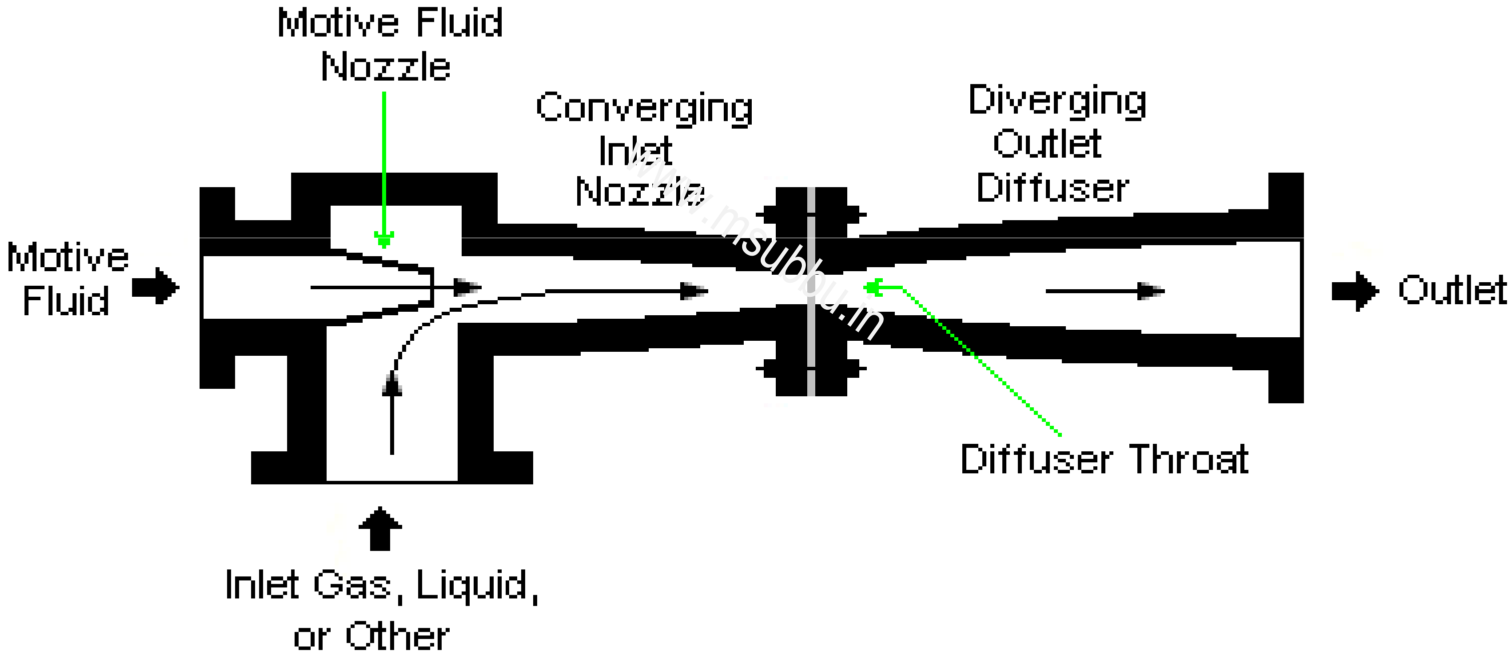
Ejectors

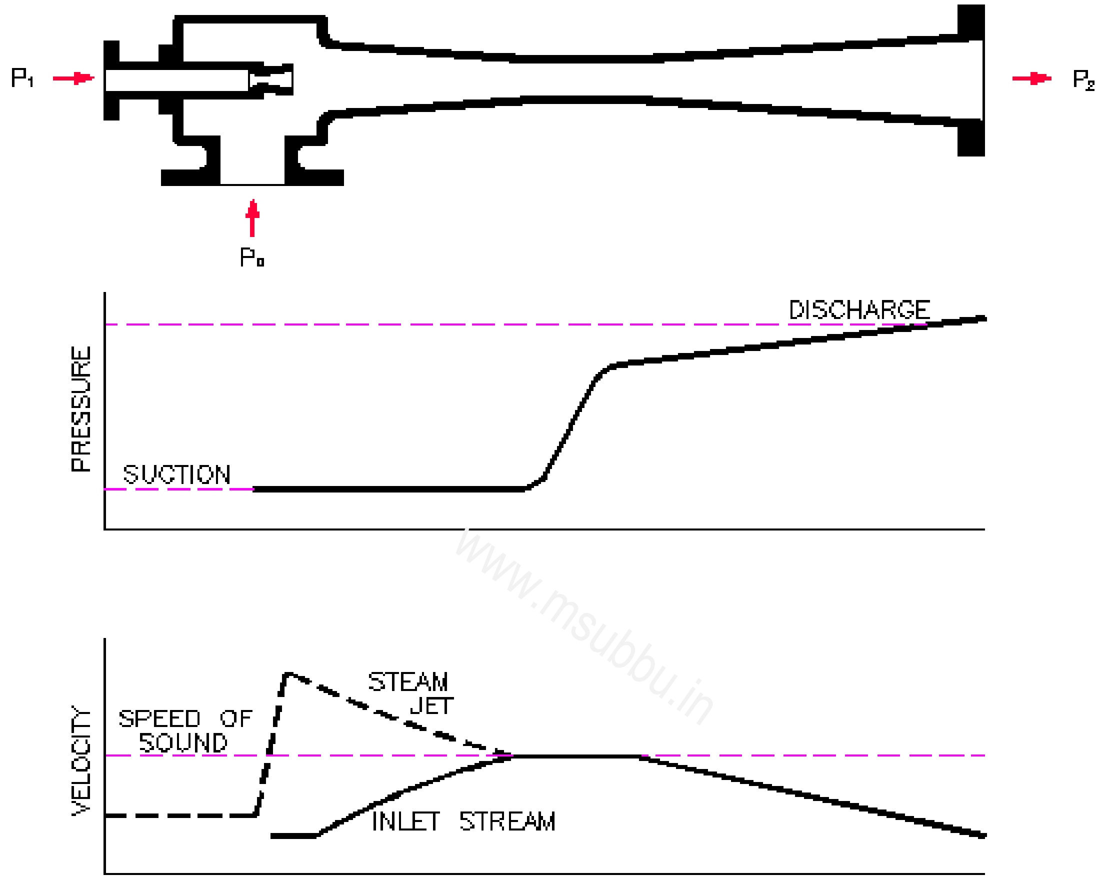


An ejector is a device in which the momentum and the kinetic energy of a high-velocity fluid stream are employed to entrain and compress a second fluid stream.

Working of Steam-Jet Ejector

- A steam-jet ejector consists of an inner converging-diverging (or converging alone) nozzle through which the driving fluid (steam) is fed, and an outer, larger nozzle through which both the extracted gases or vapors and the driving fluid pass.





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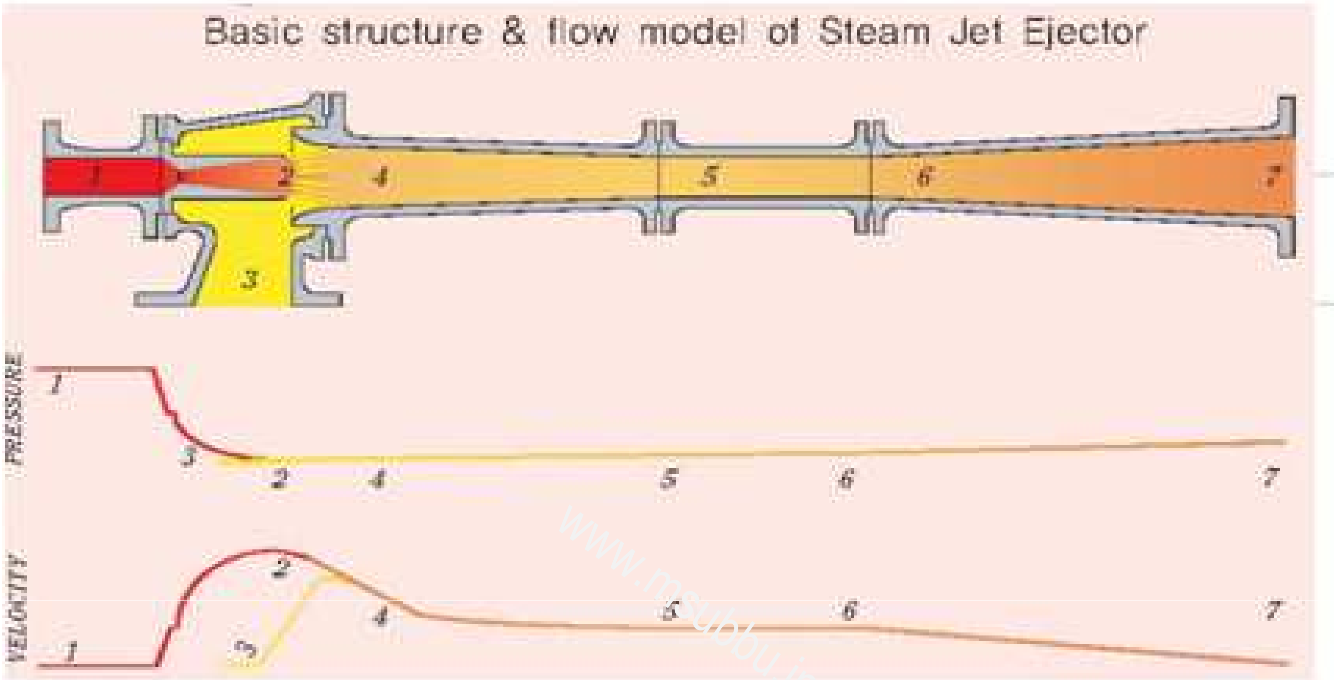
At the present day, Steam jet ejector is the preferred vacuum producing devices for many applications in the petrochemical, food processing, refining, chemical-processing and power generation industries.



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The momentum of the high-speed fluid leaving the driving nozzle is partly transferred to the extracted gases or vapors, and the mixture velocity is therefore less than that of the driving fluid leaving the smaller nozzle. It is nevertheless higher than the speed of sound, and the nozzle therefore acts as a converging-diverging diffuser in which the pressure rises and the velocity decreases, passing at speed of sound through the throat.


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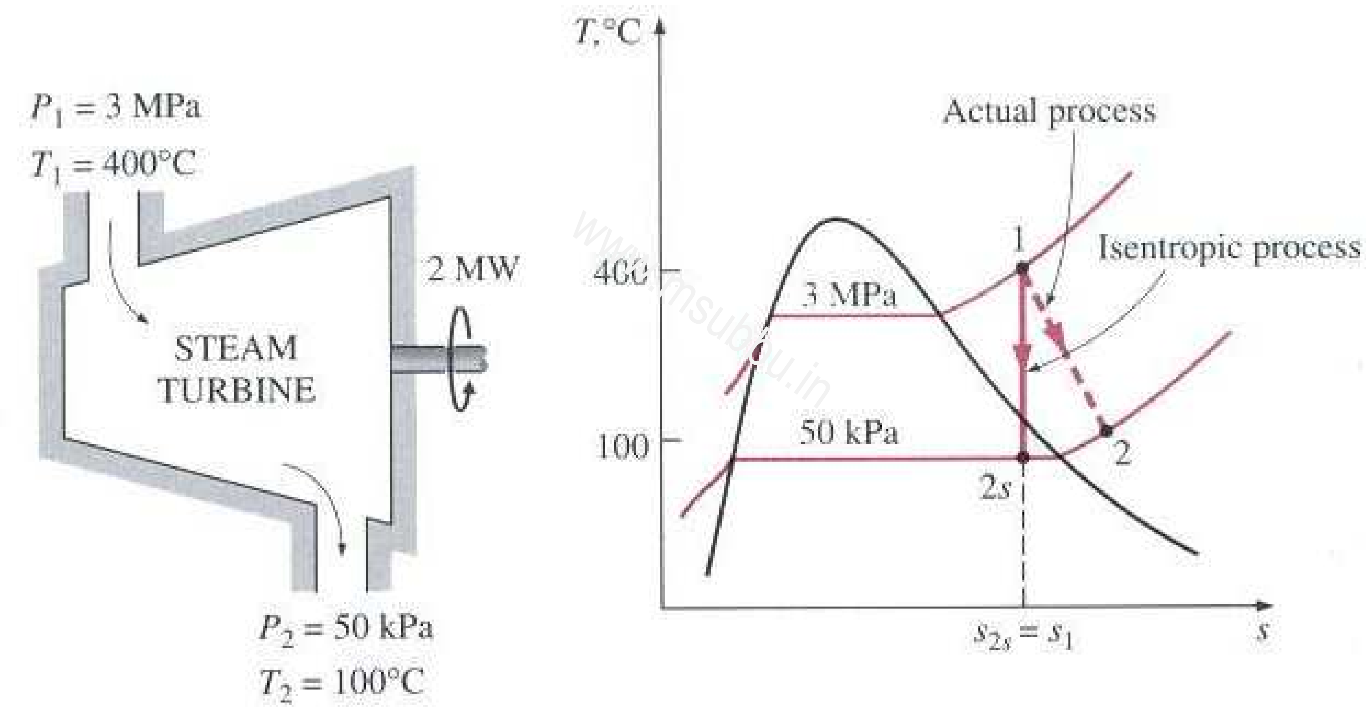
Turbine



P_1 = 3 MPa
T_1 = 400°C
2 MW
STEAM TURBINE
P_2 = 50 kPa
T_2 = 100°C
T,°C
Actual process
Isentropic process
400
100
3 MPa
50 kPa
1
2
2s
$s_{2s} = s_1$
s

Joule-Thompson Expansion

Throttling Valves

- A throttling valve is a steady-flow engineering device used to produce a significant pressure drop usually along with a large drop in temperature
- In a throttling valve, **enthalpy remains constant**
- No work device -mechanical or other forms
- Heat transfer almost always negligible
  –small area; less time available
- PE and KE changes usually negligible

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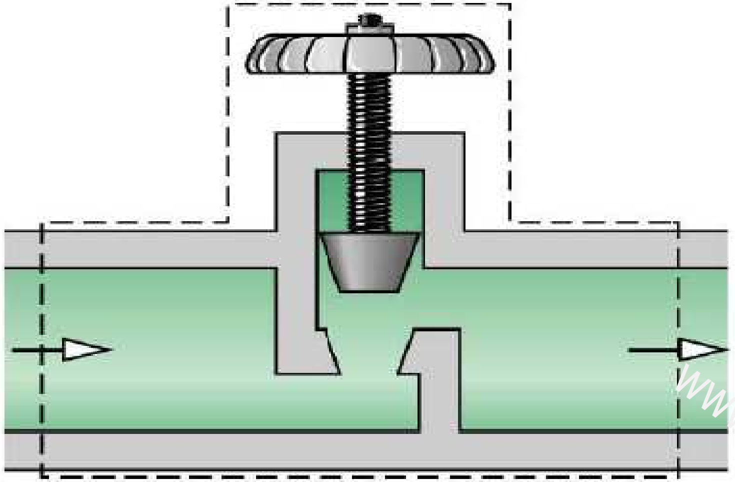
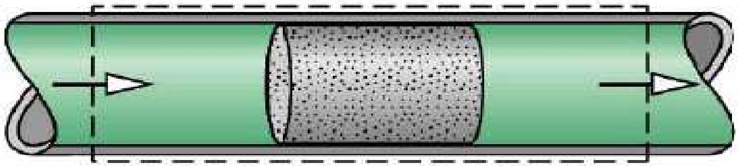
Throttling Devices





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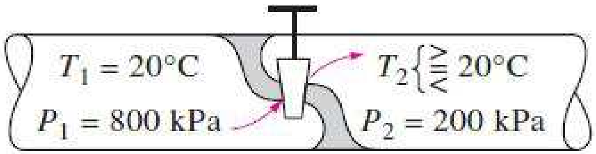

The temperature of a fluid may increase, decrease, or remain constant during a throttling process.

Joule-Thompson Coefficient

- A throttling process produces no change in enthalpy; hence for an ideal gas the temperature remains constant. For real gases, however, the throttling process will cause the temperature to increase or decrease. The Joule-Thomson coefficient, μ_J, relates this change and is defined as:

$$\mu_J = \left(\frac{\partial T}{\partial P}\right)_H$$

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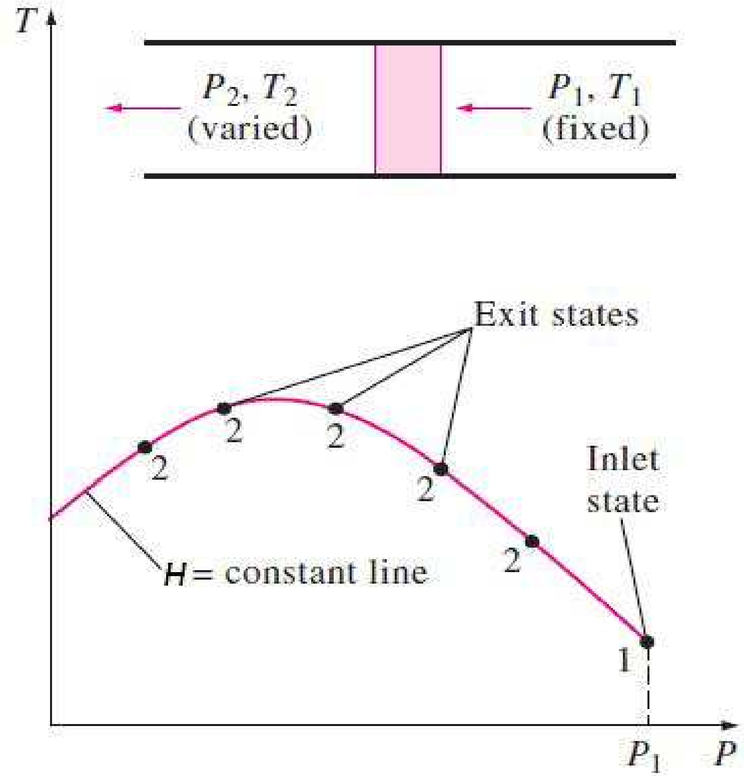
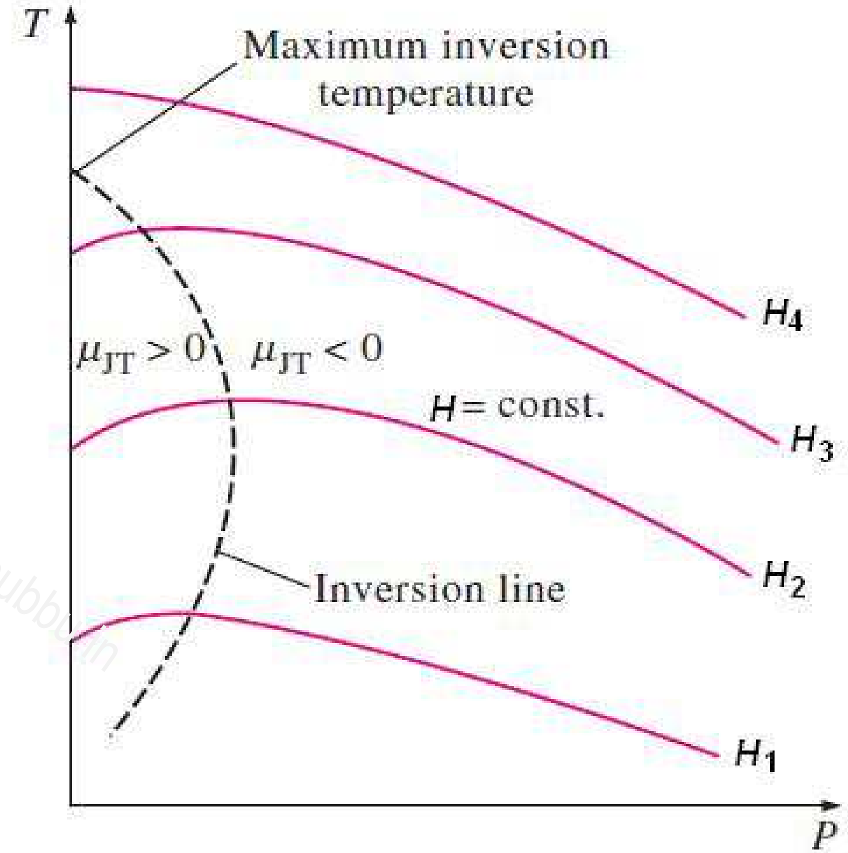
- A positive value of μ_J indicates that the temperature decreases as the pressure decreases; a cooling effect is thus observed. This is true for almost all gases at ordinary pressures and temperatures.

- The exceptions are ***hydrogen, neon, and helium***, which have a temperature increase with a pressure decrease, hence $\mu_J < 0$. Even for these gases there is a temperature above which the Joule-Thompson coefficient changes from negative to positive. At this ***inversion temperature***, $\mu_J = 0$.





$$\mu_{JT} \begin{cases} < 0 & \text{temperature increases} \\ = 0 & \text{temperature remains constant} \\ > 0 & \text{temperature decreases} \end{cases}$$

It can be shown that that

$$\mu_J = \left(\frac{\partial T}{\partial P}\right)_H = \frac{T(\partial V/\partial T)_P - V}{C_p}$$

The derivation for which is given below:
By considering H as a function of T and P,

$$\begin{aligned} H &= H(T,P) \\ dH &= \left(\frac{\partial H}{\partial T}\right)_P dT + \left(\frac{\partial H}{\partial P}\right)_T dP \end{aligned}$$

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At constant H, $dH = 0$. Therefore,

$$\begin{aligned} 0 &= \left(\frac{\partial H}{\partial T}\right)_P dT + \left(\frac{\partial H}{\partial P}\right)_T dP \\ &= C_p dT + \left(\frac{\partial H}{\partial P}\right)_T dP \qquad (1.33) \end{aligned}$$

From fundamental property relation

$$
\begin{aligned}
dH &= VdP + TdS \\
\left(\frac{\partial H}{\partial P}\right)_T &= V + T\left(\frac{\partial S}{\partial P}\right)_T \\
&= V - T\left(\frac{\partial V}{\partial T}\right)_P
\end{aligned}
$$

Substituting this in Eqn.(1.33),

$$
\begin{aligned}
C_p dT + \left[V - T\left(\frac{\partial V}{\partial T}\right)_P\right] dP &= 0 \\
\text{i.e.,}\quad C_p dT &= \left[T\left(\frac{\partial V}{\partial T}\right)_P - V\right] dP
\end{aligned}
$$

At constant H, the above equation becomes

$$
\left(\frac{\partial T}{\partial P}\right)_H = \frac{\left(\frac{\partial V}{\partial T}\right)_P - V}{C_p}
$$

Therefore,

$$
\mu_J = \left(\frac{\partial T}{\partial P}\right)_H = \frac{T(\partial V/\partial T)_P - V}{C_p}
$$

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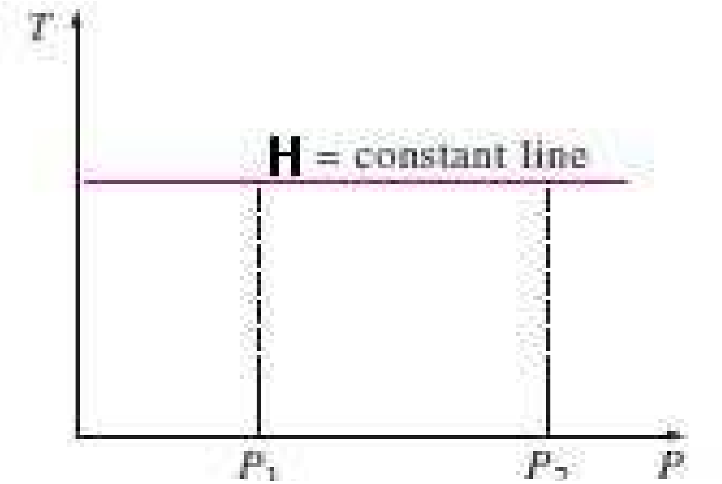
For ideal gases,

$$\begin{aligned} PV &= RT \\ \left(\frac{\partial V}{\partial T}\right)_P &= \frac{R}{P} \end{aligned}$$



Therefore,

$$\mu_J = \left(\frac{\partial T}{\partial P}\right)_H = \frac{T(\partial V/\partial T)_P - V}{C_p} = \frac{(RT/P) - V}{C_p} = \frac{V - V}{C_p} = 0$$

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The fact that $\mu_J = 0$ does not necessarily imply that the gas is ideal or even closely approaching it. $\mu_J = 0$ merely led to V/T = constant = $\phi(P)$. In other words, any gas for which the volume is linear with temperature along an isobar will have zero Joule-Thomson coefficient.

For example, if $V = RT/P + BT$

$$\begin{aligned} \left(\frac{\partial V}{\partial T}\right)_P &= R/P + B \\ \mu_J &= \frac{T(R/P + B) - (RT/P + BT)}{C_p} = 0 \end{aligned}$$