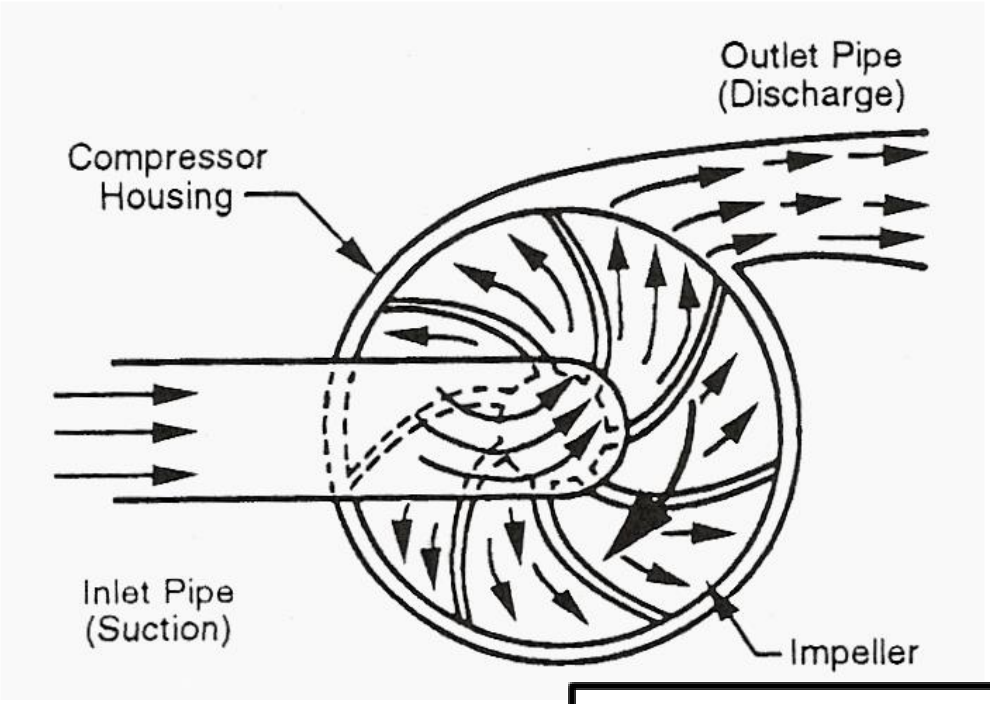
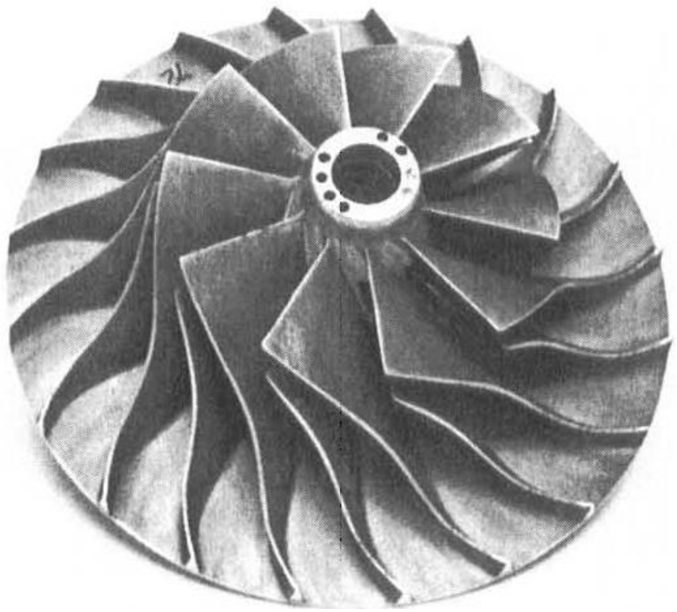
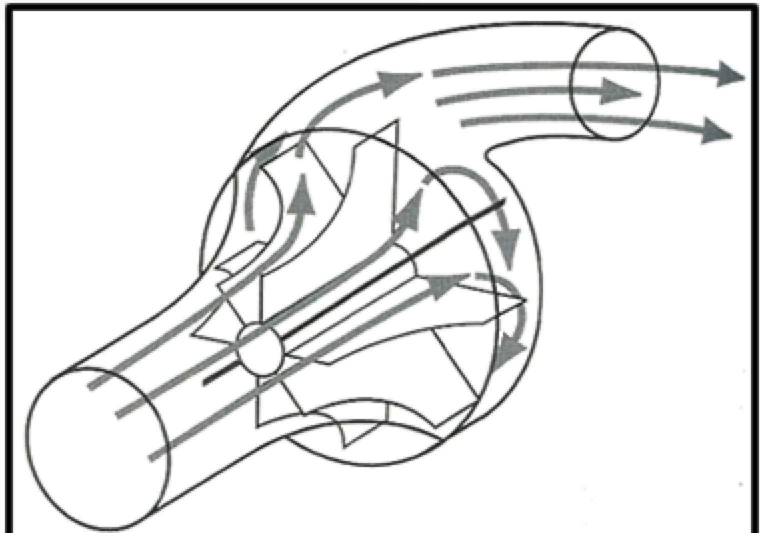
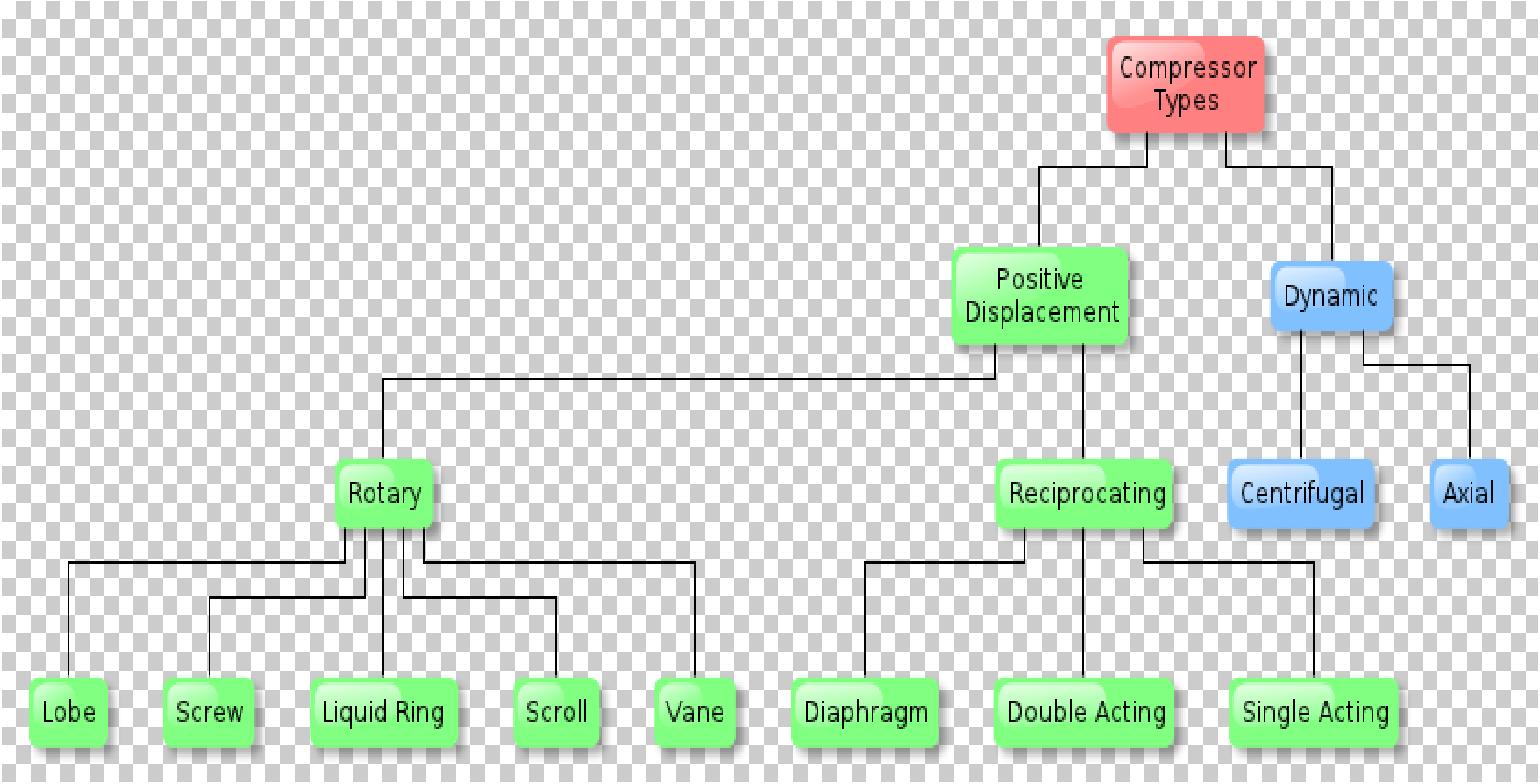
Unit 5. Centrifugal Compressor







Classification of Compressor



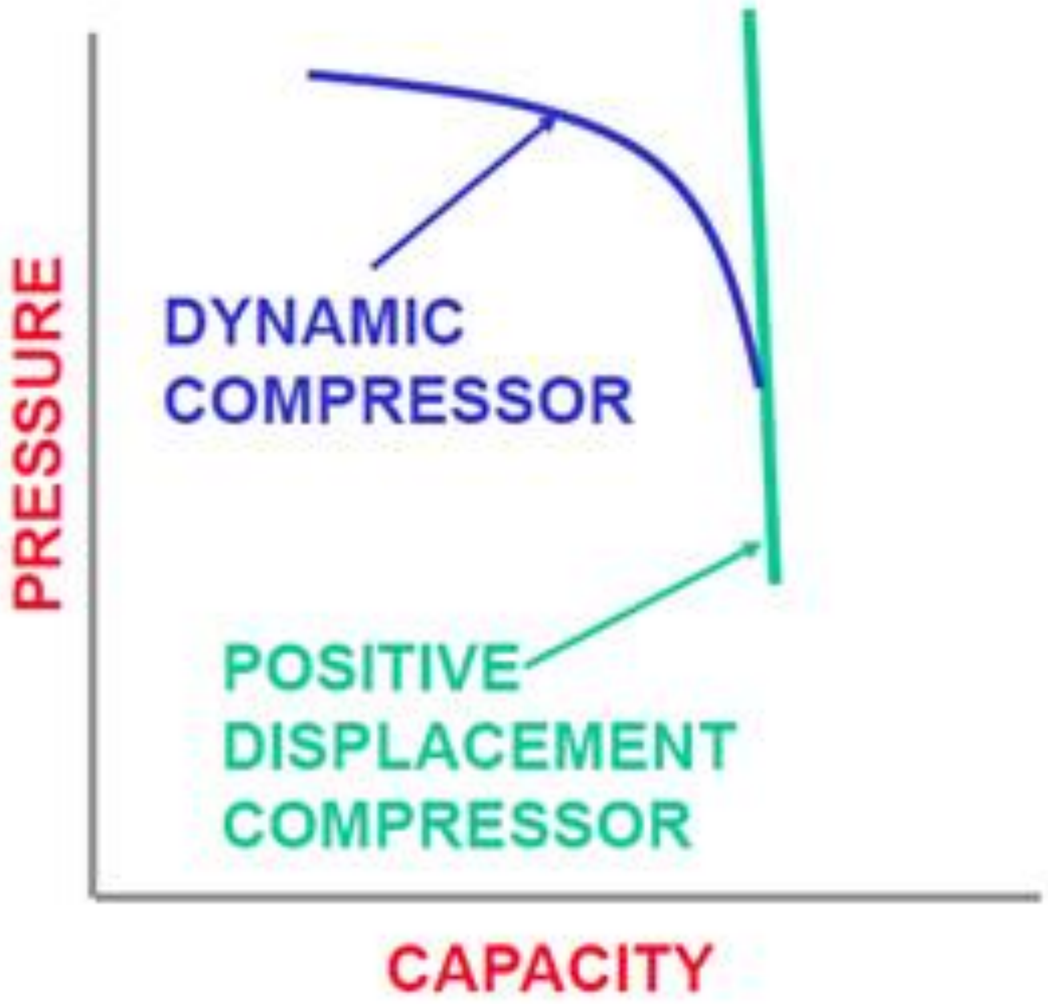
- *Positive Displacement Type:*
- In this type of compressors, *air is physically trapped between to relatively moving components and forced to occupy lower volume,* thereby increasing its pressure.

  Most notable example would be, a reciprocating compressor. In which air is trapped between piston and cylinder volume and then literally pressed to increase its pressure.

- ***Non-Positive Displacement Type:***
- In this type, *a rotating component imparts its kinetic energy to the air which is eventually converted into pressure energy.*

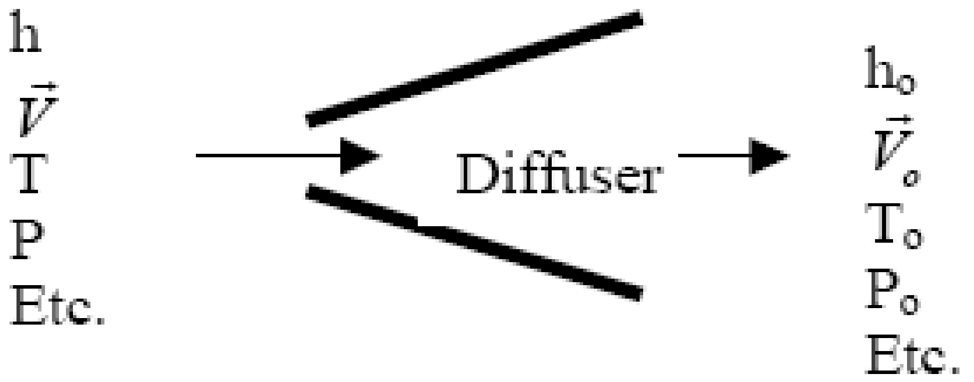
Centrifugal compressors are non-positive displacement type. Rotating impeller imparts KE to the air which is converted to PE as air passes through the diffuser.

The term *'positive displacement'* emphasizes the fact that a volume is physically being displaced, whereas there is no physical displacement in centrifugal compressor.


DYNAMIC
COMPRESSOR
POSITIVE
DISPLACEMENT
COMPRESSOR
PRESSURE
CAPACITY

Stagnation Properties

Consider a fluid flowing into a diffuser at a velocity $\vec{V}$, temperature *T*, pressure *P*, and enthalpy *h*, etc. Here the ordinary properties *T*, *P*, *h*, etc. are called the static properties; that is, they are measured relative to the flow at the flow velocity. The diffuser is sufficiently long and the exit area is sufficiently large that the fluid is brought to rest (zero velocity) at the diffuser exit while no work or heat transfer is done. The resulting state is called the stagnation state.



We apply the first law per unit mass for one entrance, one exit, and neglect the potential energies. Let the inlet state be unsubscripted and the exit or stagnation state have the subscript o.

$$q_{net} + h + \frac{\vec{V}^2}{2} = w_{net} + h_o + \frac{\vec{V}_o^2}{2}$$

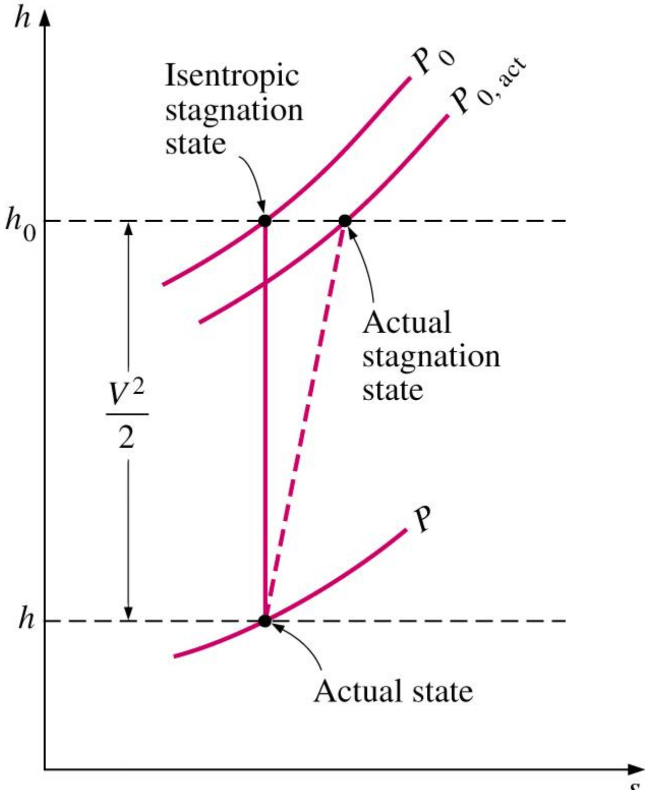
Since the exit velocity, work, and heat transfer are zero,

$$h_o = h + \frac{\vec{V}^2}{2}$$

The term h_o is called the stagnation enthalpy (some authors call this the total enthalpy). It is the enthalpy the fluid attains when brought to rest adiabatically while no work is done.

If, in addition, the process is also reversible, the process is isentropic, and the inlet and exit entropies are equal. $S_o = S$

The stagnation enthalpy and entropy define the stagnation state and the isentropic stagnation pressure, P_o. The actual stagnation pressure for irreversible flows will be somewhat less than the isentropic stagnation pressure as shown below.



- For an ideal gas, $h = C_p T$, which allows the h_0 to be rewritten

$$c_p T_0 = c_p T + \frac{V^2}{2} \Longrightarrow T_0 = T + \frac{V^2}{2c_p}$$

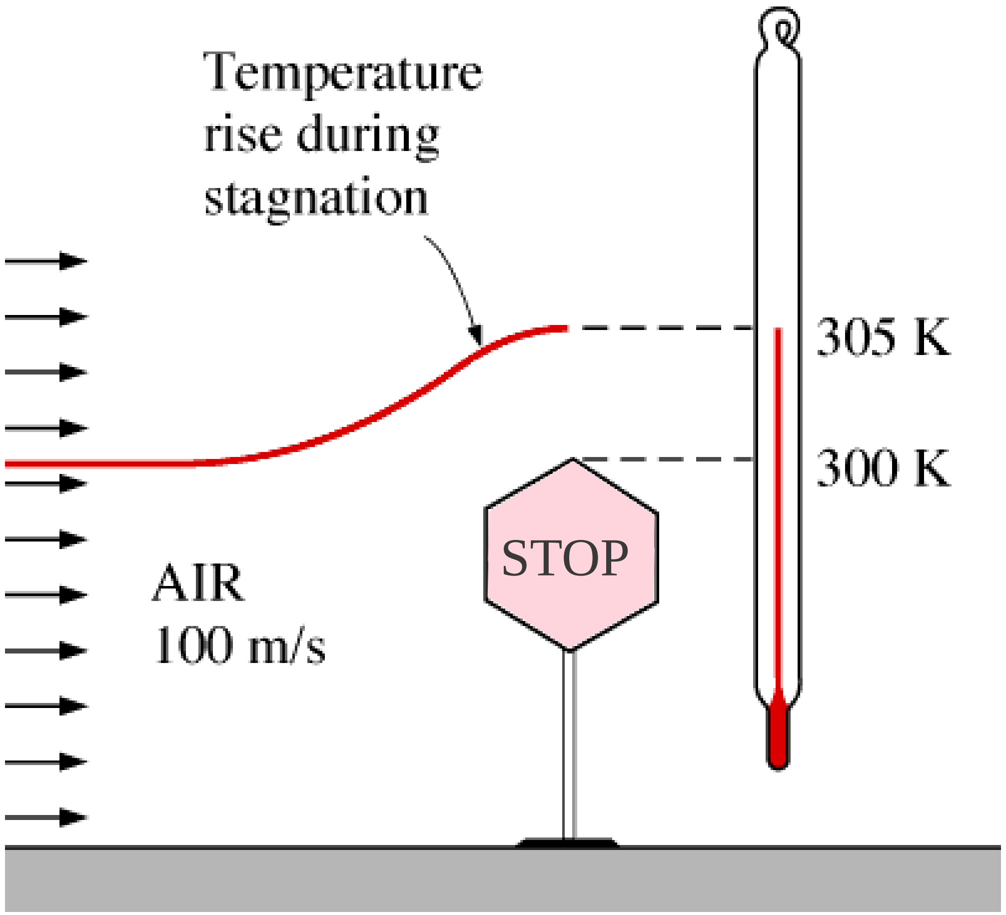
  - T_0 is the stagnation temperature. It represents *the temperature an ideal gas attains when it is brought to rest adiabatically.*
  - $V^2/2C_p$ corresponds to the temperature rise, and is called the **dynamic temperature**

- For ideal gas with constant specific heats, stagnation pressure can be expressed as

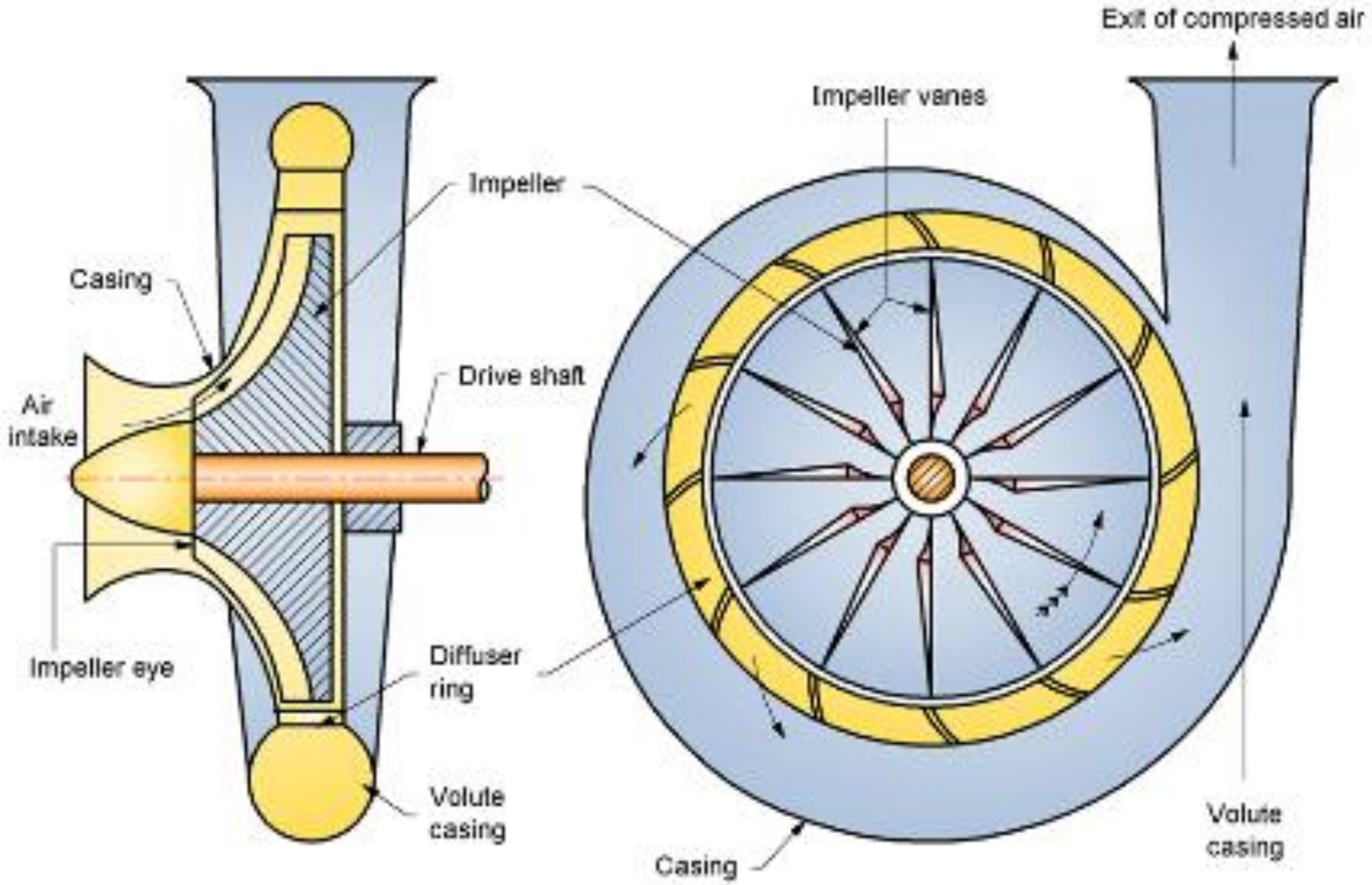
$$\frac{P_0}{P} = \left(\frac{T_0}{T}\right)^{k/(k-1)}$$

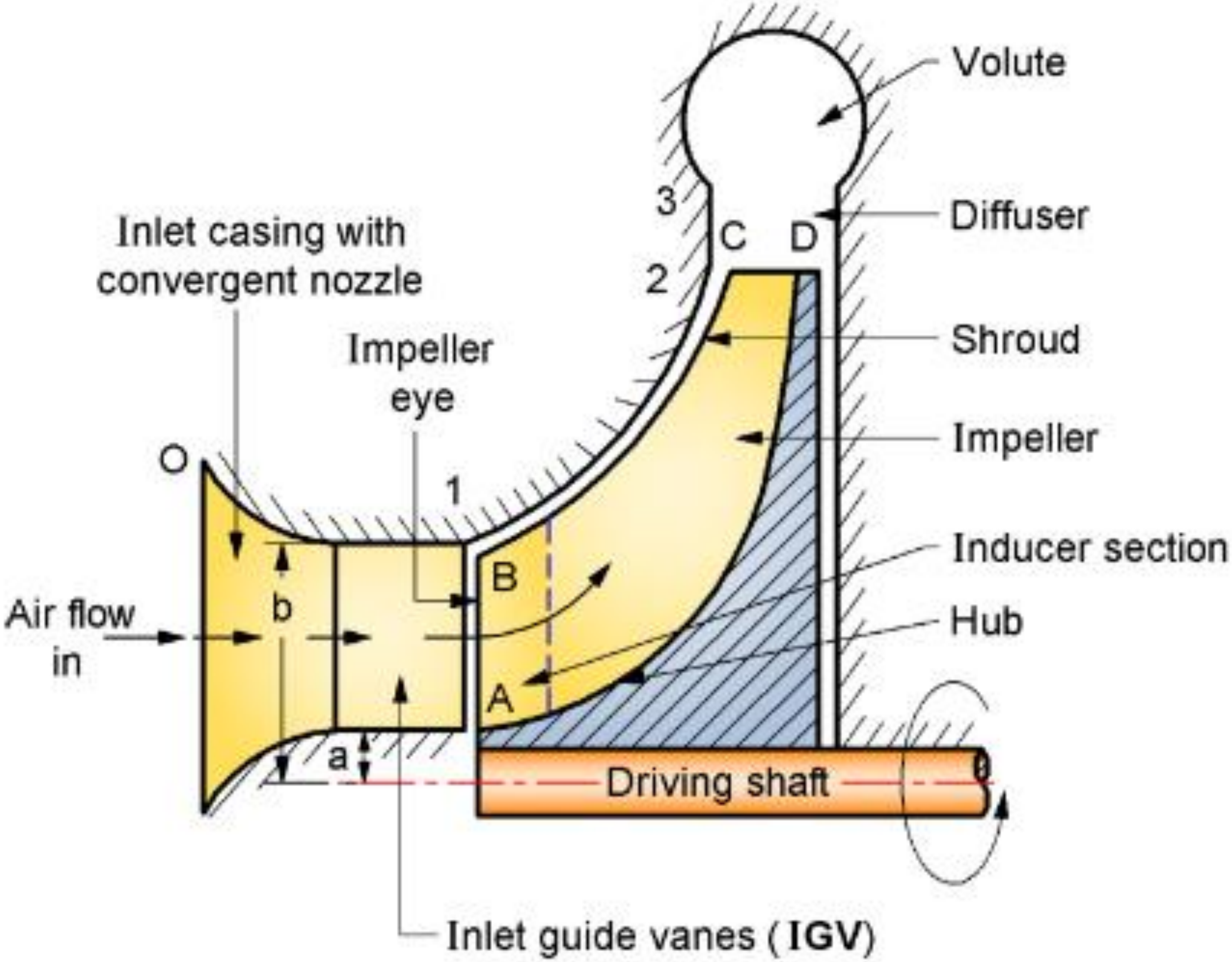
Completely Arresting the Flow of an Ideal Gas can Raise its Temperature

The temperature of an ideal gas flowing at a velocity V rises by $V^2/(2C_P)$ when it is brought to a complete stop



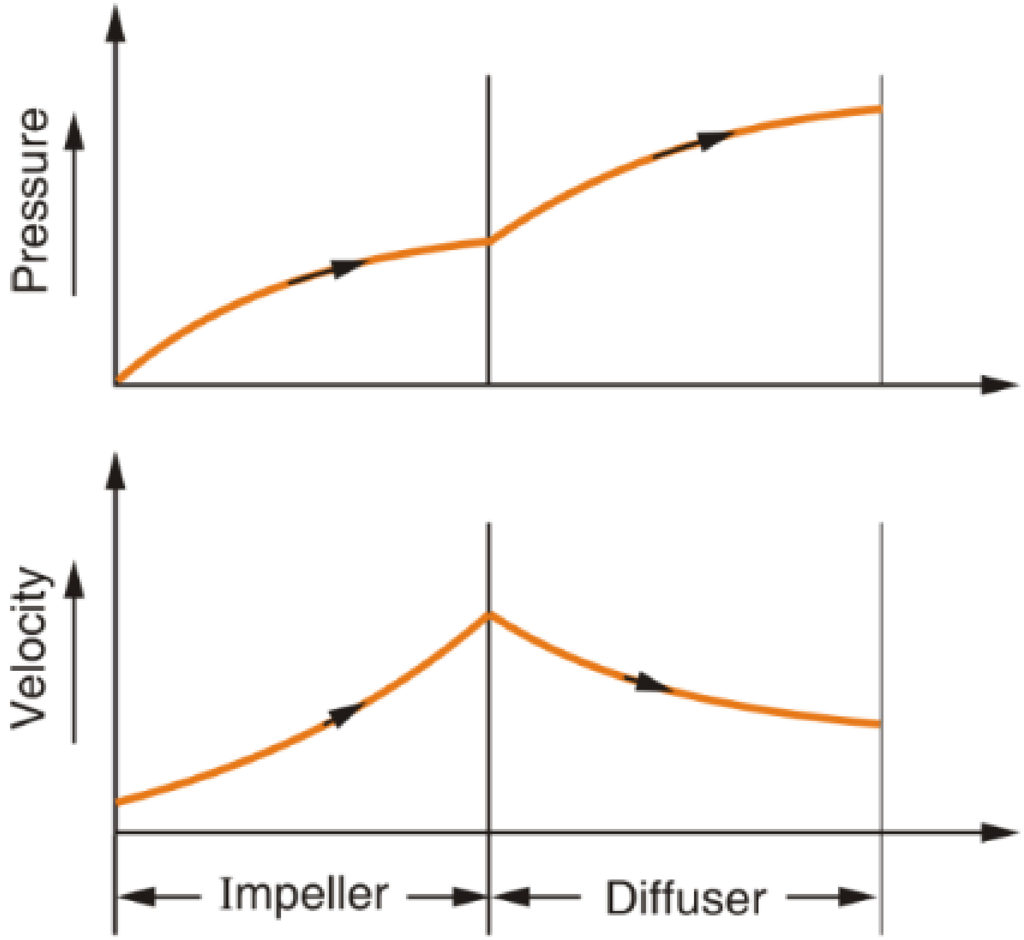
Centrifugal Compressor





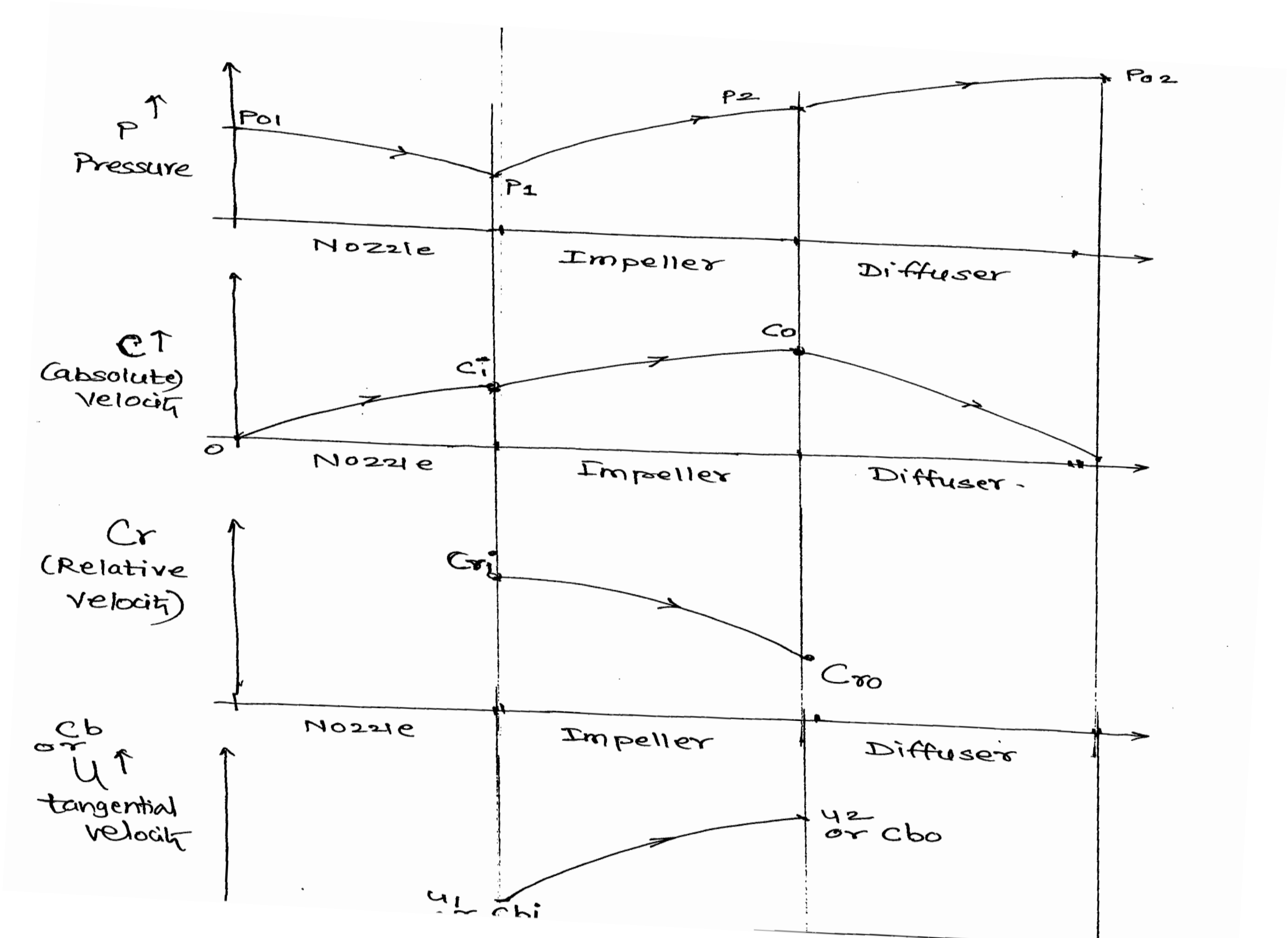
Volute
Diffuser
3
C
D
2
Inlet casing with
convergent nozzle
Shroud
Impeller
eye
Impeller
O
1
Inducer section
B
b
Air flow
in
Hub
A
a
Driving shaft
Inlet guide vanes (IGV)

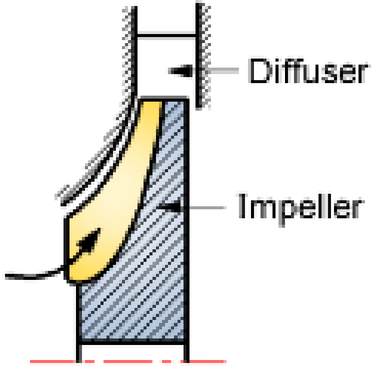
Pressure and Velocity variation

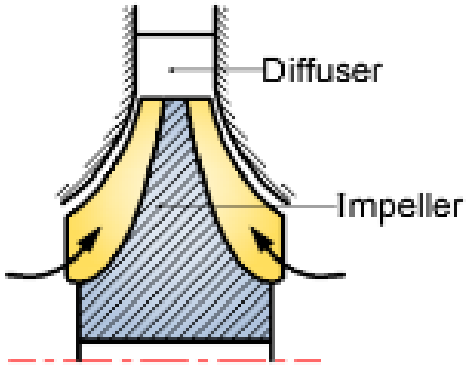


Pressure and velocity variation

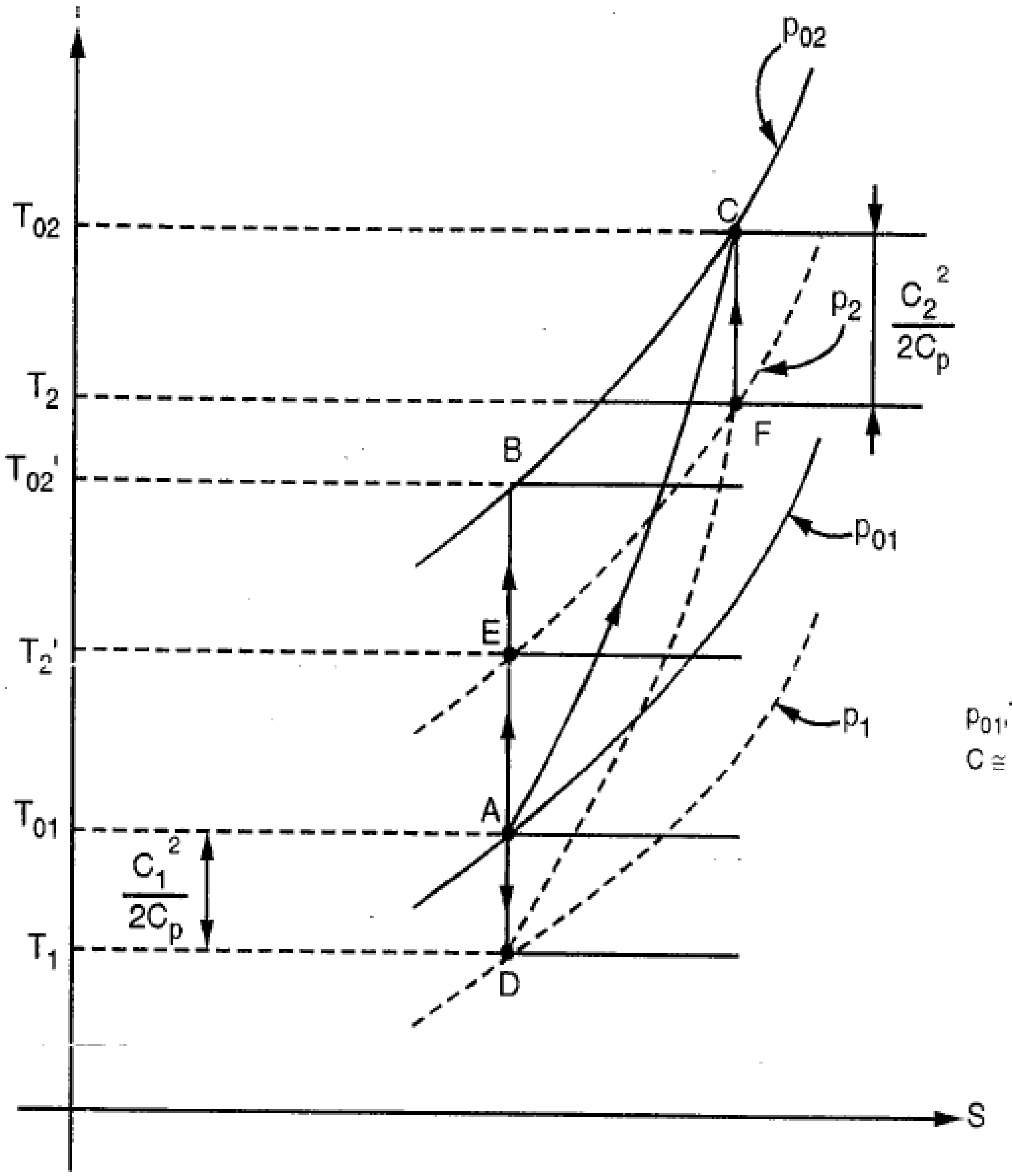
PRESSURE , VELOCITY VARIATIONS IN NOZZLE,IMPELLER,DIFFUSER.



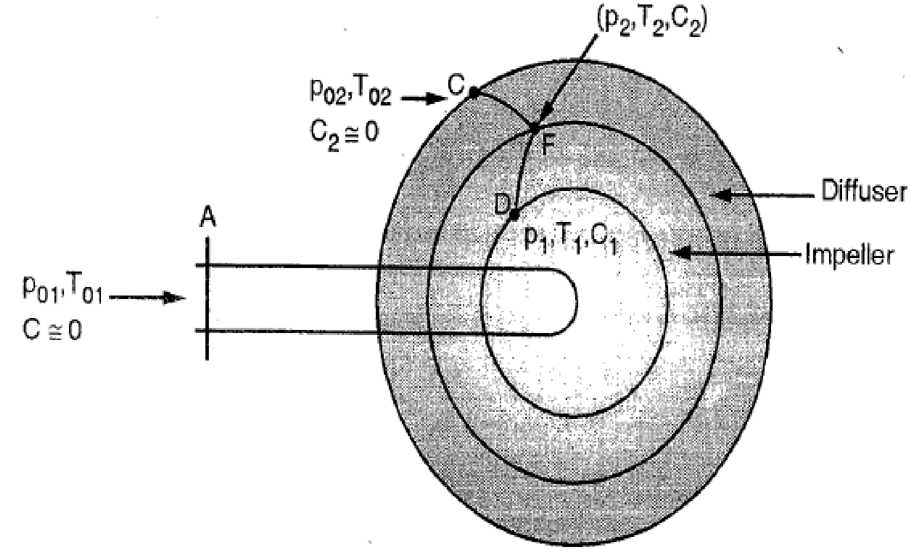




(a) Single sided (b) Double sided Types of impellers



Stagnation and static values on (T – S) diagram for a rotary compressor



Representation of points on rotary compressor

Actual and isentropic work done

Applying S.F.E.E across impeller,

$$q - w = \Delta h + \Delta K.E + [\underbrace{\Delta P.E = 0}_{(z_1 = z_2)}]$$

Assuming adibatic process $q = 0$,

W is –ve for compn. work.

$$\therefore \quad -(-w) = (h_2 - h_1) + \frac{c_2^2 - c_1^2}{2} + \underbrace{g(z_2 - z_1)}_{0}$$

$$\therefore \quad W = \left[h_2 + \frac{c_2^2}{2}\right] - \left[h_1 + \frac{c_1^2}{2}\right]$$

$$\therefore \quad \boxed{W_{actual} = h_{02} - h_{01}} = C_p\,(T_{02} - T_{01})$$

$$\frac{T_{02}}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\frac{n-1}{n}}$$

$$\therefore \quad W_{actual} = C_p \cdot (T_{01}) \left[\left(\frac{P_{02}}{P_{01}}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W_{isentropic} = C_p (T_{01}) \left[\left(\frac{P_{02}}{P_{01}}\right)^{r-1/r} - 1\right]$$

$$\text{Isentropic efficiency} = \frac{\text{Isentropic Work}}{\text{Actual Work}}$$

$$\boxed{\underset{\text{(stagnation)}}{\text{Isentropic } \eta} = \frac{C_p\,(T_{02}' - T_{01})}{C_p\,(T_{02} - T_{01})}}$$

$$\boxed{\underset{\text{(static)}}{\text{Isentropic } \eta} = \frac{(T_2' - T_1)}{(T_2 - T_1)}}$$

Pb: A gas compressor compresses the gas at the rate of 2 Kg/sec from inlet static pressure of 1 bar to a static pressure of 4 bar. The power input to the comp. is 400 kW. The velocity of air at entry to impeller blades is 100 m/sec. and exit of impeller blades is 160 m/sec. Determine the stagnation pressures and temperatures at inlet and exit of the compressor, diameter of suction pipe required and isentropic efficiency based on static and total values.

Assume $\gamma = 1.4$, $C_p = 1.05$ KJ/kg K

$R = 300$ Nm/kg K. Temp. at inlet to impeller blades is 280 K.

Data: $\dot{m} = 2$ kg/sec, $P_1 = 1$ bar, $P_2 = 4$ bar

$\dot{W} = 400$ KW, $C_1 = 100$ m/sec, $C_2 = 160$ m/sec

What is asked?

i) stagnation Pr. at inlet = P_{01}?

ii) Stagnation Temp. at inlet = T_{01}?

iii) Stagnation Pr. at outlet = P_{02}?

iv) Stagnation Temp. at outlet = T_{02}?

v) A_i or d_i?

vi) Isentropic η based on static and stagnation values?

Applying SFEE for impeller,

$$\dot{Q} - \dot{W} = \dot{m}\left[\Delta h + \Delta K.E + \Delta P.E.\right]$$

$\dot{Q} = 0$ (adibatic) , $Z_1 = Z_2 \Rightarrow \Delta P.E. = 0$

$W = -ve$ for comp. work.

$$0 - (-400) = 2\left[C_p(T_2 - T_1) + \frac{[C_2^2 - C_1^2]}{2 \times 1000} + 0\right]$$

$$\therefore \quad 400 = 2\left[1.05\,(T_2 - 280) + \frac{(160)^2 - (100)^2}{2 \times 1000}\right]$$

$\therefore$ $\boxed{T_2 = 466.8\ K}$, $T_1 = 280\ K$

$$\therefore T_{01} = T_1 + \frac{C_1^2}{2 \cdot C_P}$$

$$= (280) + \frac{(100)^2}{2 \times (1.05) \times 1000}$$

$$\boxed{T_{01} = 284.76\,K}$$

$$T_{02} = T_2 + \frac{C_2^2}{2 \cdot C_P} = (466.8) + \frac{(160)^2}{2 \times (1.05) \times 1000}$$

$$\therefore \boxed{T_{02} = 479\,K}$$

Now $\frac{T_{01}}{T_1} = \left(\frac{P_{01}}{P_1}\right)^{\gamma-1/\gamma}$

$\therefore \frac{P_{01}}{P_1} = \left(\frac{T_{01}}{T_1}\right)^{\gamma/\gamma-1}$

$\therefore P_{01} = 1\left(\frac{284.76}{280}\right)^{1.4/1.4-1} = 1.0094 \text{ bar}.$

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

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

$$\therefore \quad \frac{P_{02}}{4} = \left(\frac{479}{466.8}\right)^{1.4/1.4-1}$$

$$\therefore \quad \boxed{P_{02} = 4.38 \text{ bar}}$$

$$P_1 v_1 = R T_1$$

$$v_1 = \frac{(300)(280)}{(1 \times 10^5)} = 0.84 \, m^3/kg$$

$$\dot{m} = \rho_1 A_1 C_1$$

$$\dot{m} = \frac{A_1 C_1}{v_1} \quad \therefore \quad 2 = \frac{A_1 \times 100}{0.84}$$

$$\therefore \; A_1 = 1.68 \times 10^{-2} \, m^2 = \frac{\pi}{4} d_i^2$$

$$\therefore \; d_i = 0.1463 \, m$$

$$\frac{T_2'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\gamma-1/\gamma} \quad \therefore \quad T_2' = (280)\left(\frac{4}{1}\right)^{1.4-1/1.4}$$

$$\therefore \quad T_2' = 416\,K$$

$$\eta_{\substack{\text{isentropic}\\ \text{(static)}}} = \frac{T_2' - T_1}{T_2 - T_1}$$

$$= \frac{416 - 280}{466.8 - 280}$$

$$= \frac{136}{186.8}$$

$$= 0.7280$$

$$\eta_{isentropic\,(stagnation)} = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}}$$

$$\frac{T_{02}'}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}}$$

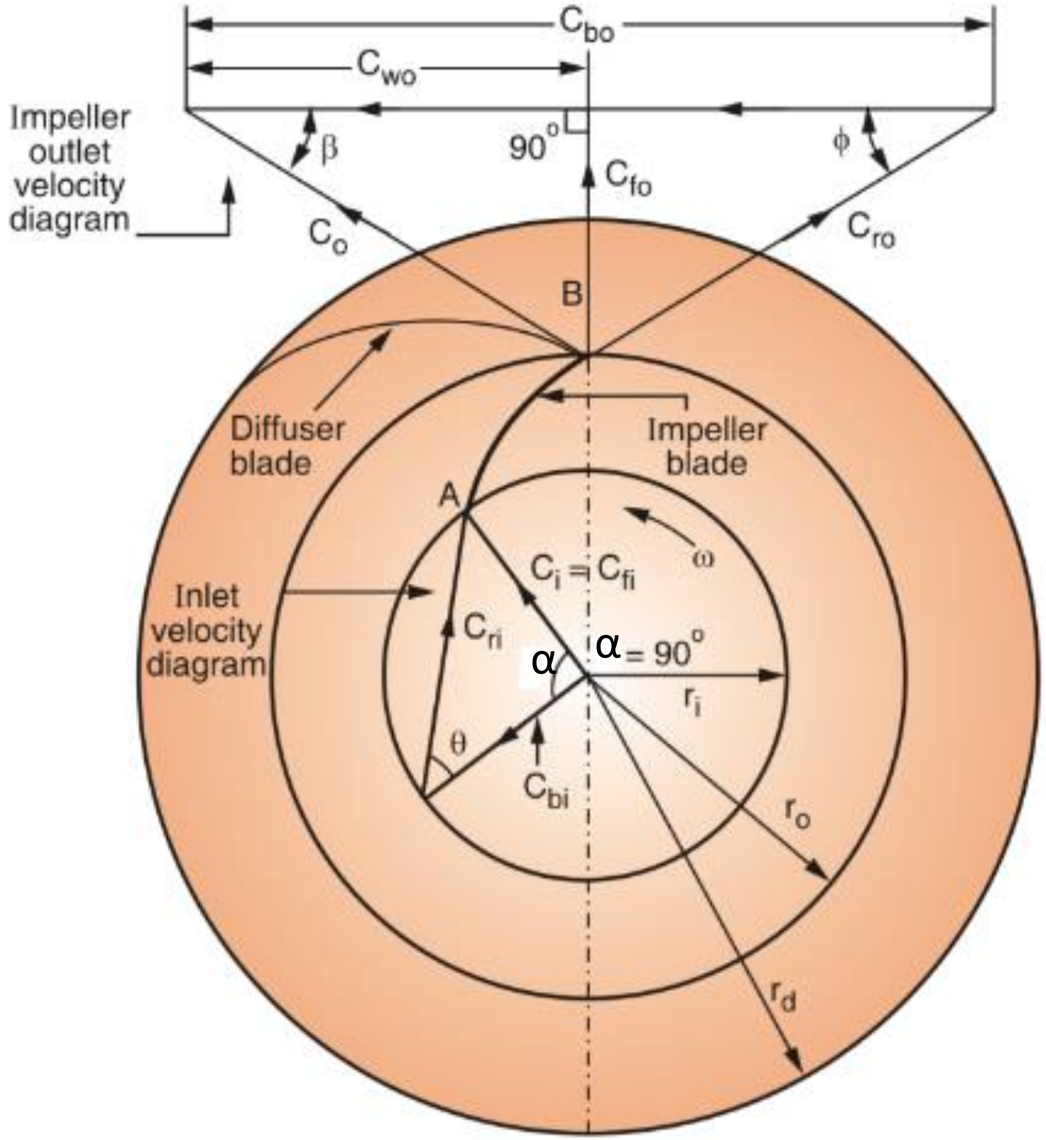
$$T_{02}' = (284.76)\left(\frac{4.38}{1.0094}\right)^{\frac{1.4-1}{1.4}}$$

$$\therefore \boxed{T_{02}' = 433\ K}$$

$$\therefore \eta_{isentropic\,(stagnation)} = \frac{433 - 284.76}{479 - 284.76}$$

$$\eta_{isentropic\,(stagnation)} = \frac{148.24}{194.24} = \underline{\underline{0.763)}}$$

Velocity diagram



Assumptions :

1. The flow of gas is steady.
2. There is no separation of flow.
3. There is no formation of shock wave anywhere during the gas flow.
4. The flow through the impeller is frictionless.

Newton's 2nd law of motion, Torque exerted is equal to rate of change of angular momentum.

$$\text{Torque } (T) = \dot{m}\left[C_{wo} \cdot r_o - C_{wi}\, r_i\right]$$

Assuming radial entry $C_{wi} = 0$

$$T = \dot{m} \cdot C_{wo} \cdot r_o$$

$$\dot{W} = T \times \omega$$

$$= \left[\dot{m} \times C_{wo} \times r_o\right] \times \omega$$

$$= \dot{m} \times C_{wo} \times \left[r_o \times \omega\right]$$

$$\dot{W} = \dot{m} \times C_{wo} \times C_{bo}$$

$$\therefore \boxed{W/kg = C_{wo}\, C_{bo}}$$

$\phi = 90^\circ$, (exit from impeller radial)

$$C_{wo} = C_{bo}$$

$\therefore$ Ideal Work $= C_{bo}^2$ ← Euler work

Ideal Power $= \dot{m} C_{bo}^2$

From velocity triangles,

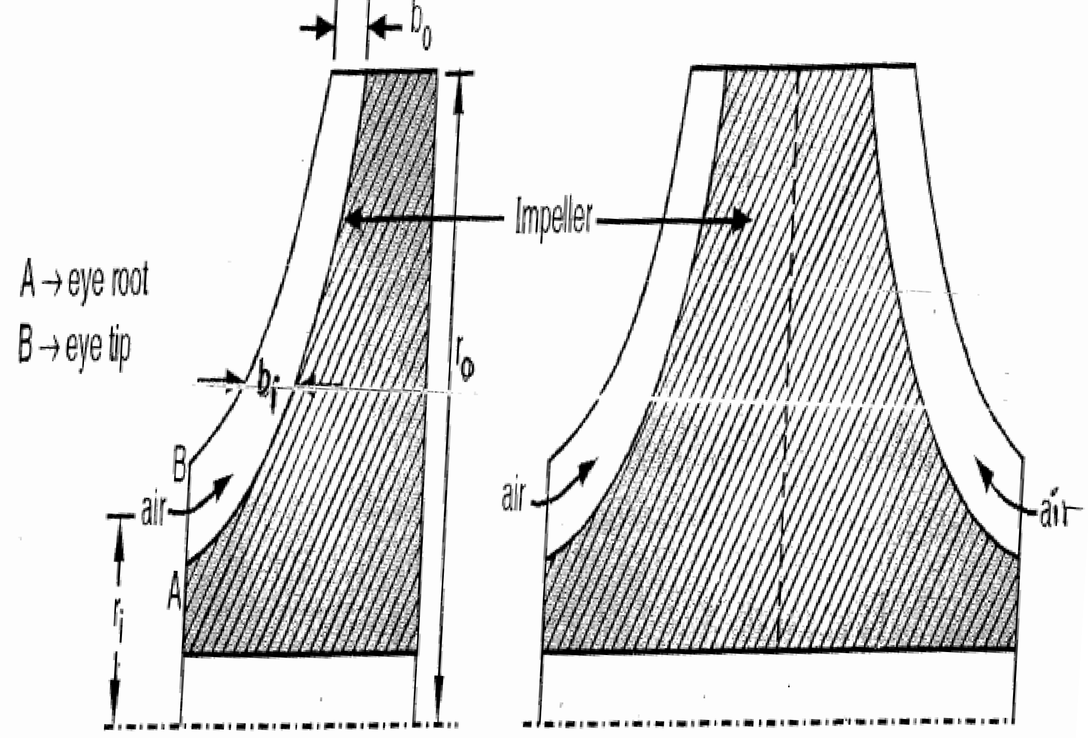
$$C_{bo}.C_{wo} = \underbrace{\frac{C_{ri}^2 - C_{ro}^2}{2}}_{1} + \underbrace{\frac{C_o^2 - C_i^2}{2}}_{2} + \underbrace{\frac{C_{bo}^2 - C_{bi}^2}{2}}_{3}$$

$$\therefore \text{ Euler work} = \frac{C_{ri}^2 - C_{ro}^2}{2} + \frac{C_o^2 - C_i^2}{2} + \frac{C_{bo}^2 - C_{bi}^2}{2}$$

$$\text{Euler work} = C_p(T_{o2} - T_{o1})$$

1 & 3 $\Rightarrow$ static head $\Rightarrow$ [Decrease in rel. vel. + Increase in centrifugal head]

2 $\Rightarrow$ Dynamic head.



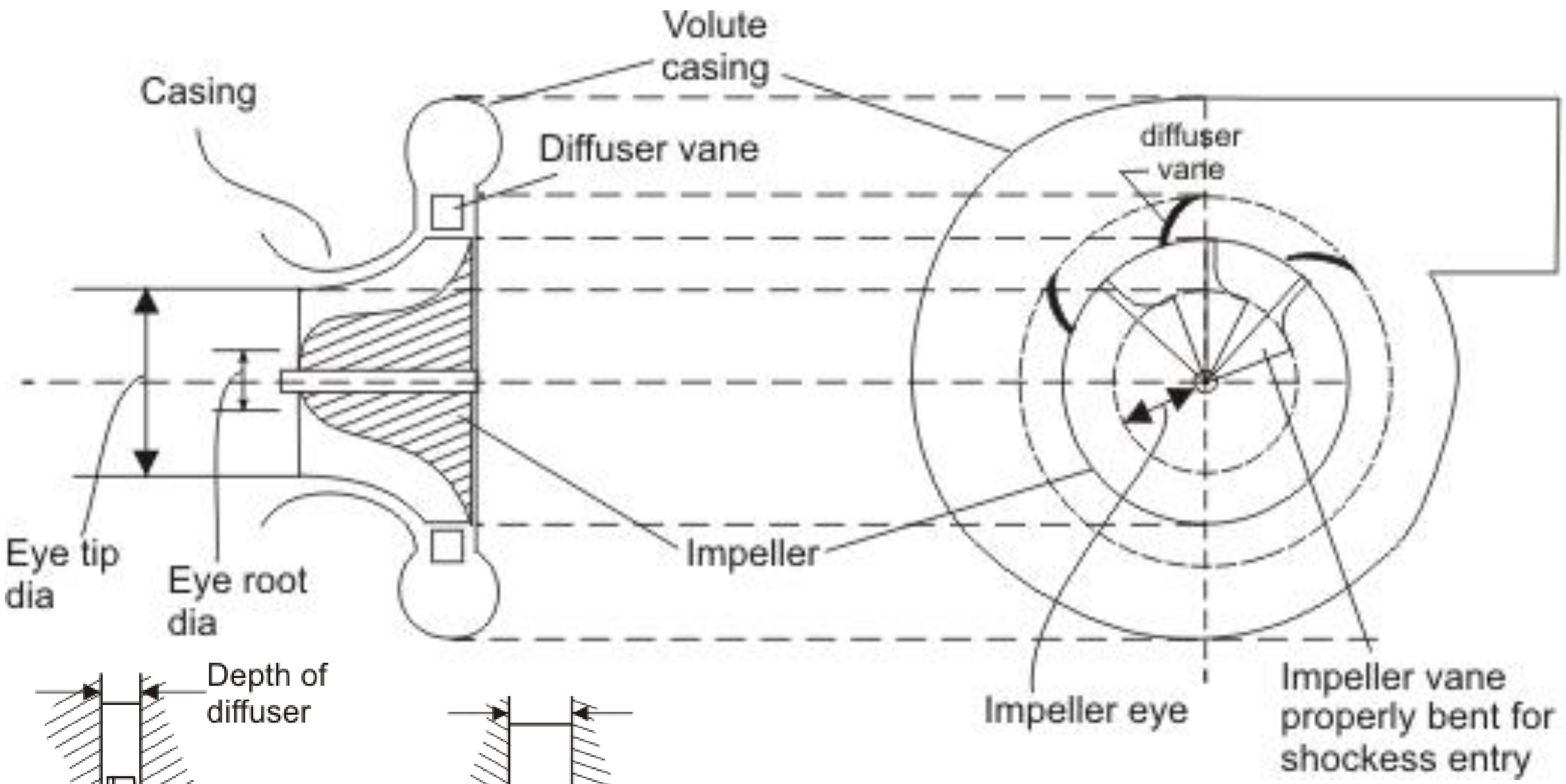
(a) Single sided

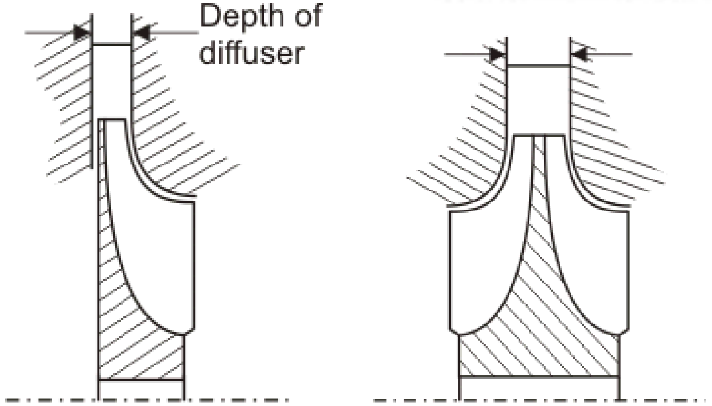
(b) Double sided

Single and double sided impellers

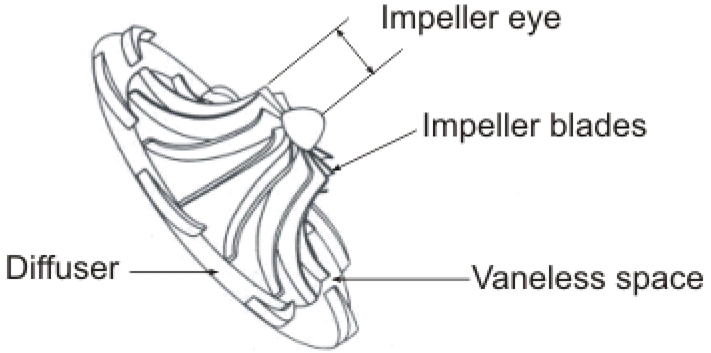
mass flow rate $= \dot{m} = \dfrac{\text{Area of flow} \times \text{velocity of flow}}{\text{spc volume}}$

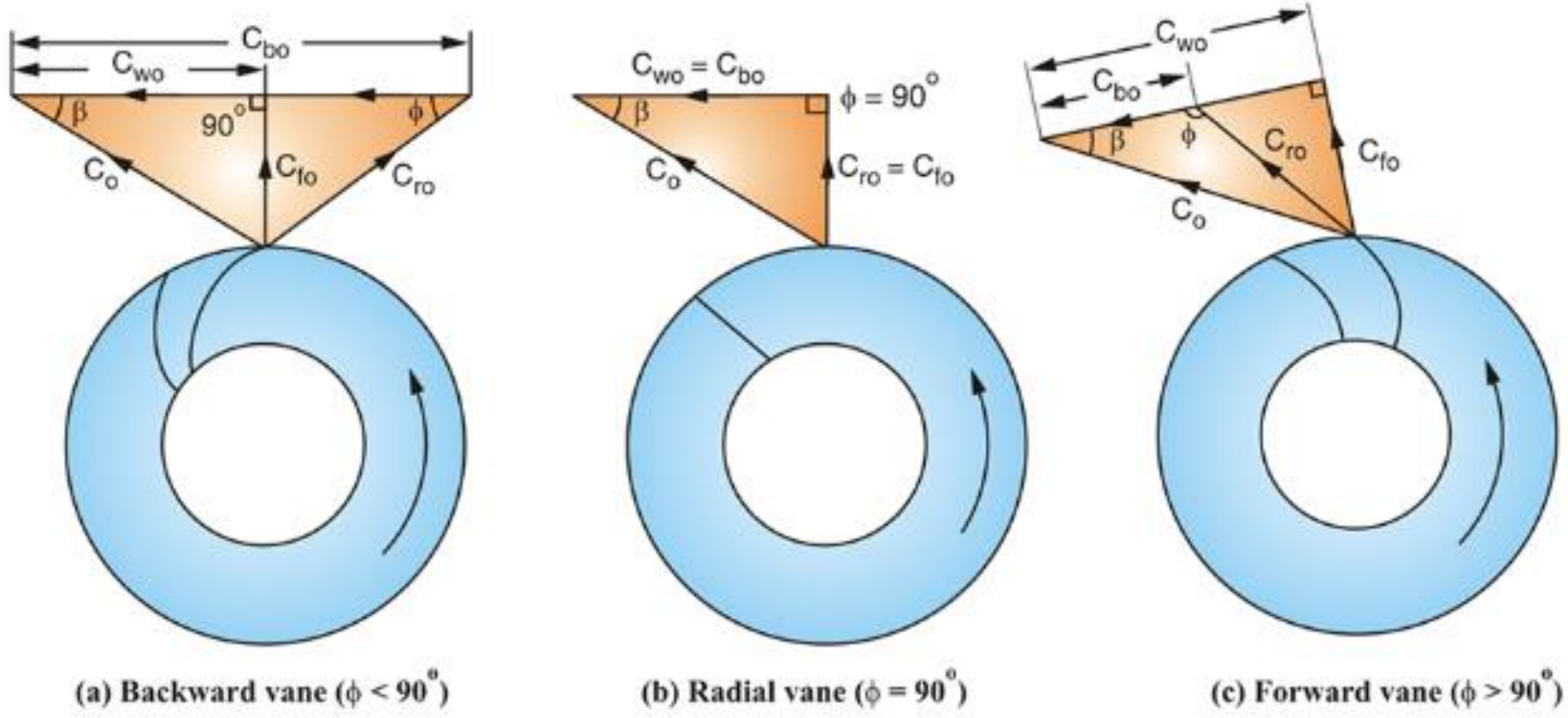
$$\dot{m} = \frac{(2\pi r_i b_i) C_{fi}}{v_i} = \frac{(2\pi r_o b_o) C_{fo}}{v_o}$$



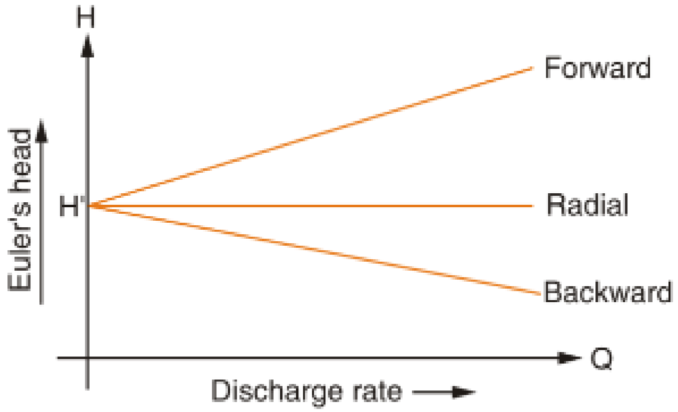


Centrifugal Compressor





Shape of vanes of a centrifugal compressor with outlet velocity diagram



Discharge - Head characteristics of forward, radial and backward vanes

A centrifugal compressor compresses air at the rate of 2kg/sec from a static pressure of 1 bar and static temp. of 20°c to a stagnation pressure at outlet equal to 5 bar. The velocity of air at inlet is 150 m/s and compresser runs at 20,000 rpm. Assume $C_p = 1.005$ KJ/kgK, slip factor $= 0.9$, isentropic efficiency $= 80\%$.

Calculate: i) The change in stagnation temp. from inlet to outlet.

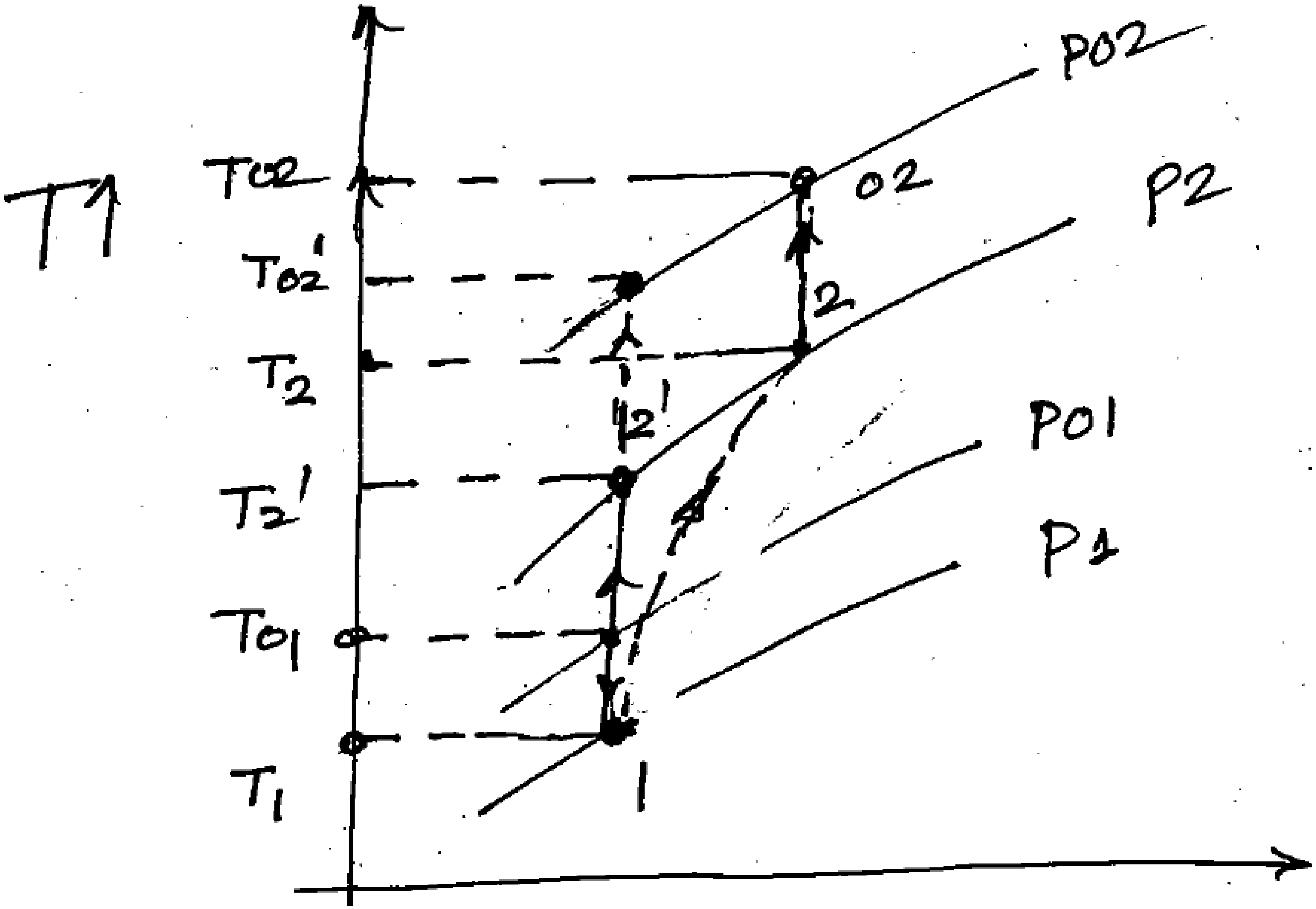
ii) Impeller diameter at outlet

iii) Power required to drive the compressor.

$\dot{m} = 2$ kg/sec, $P_1 = 1$ bar, $T_1 = 20°c = 293$ K

$P_{02} = 5$ bar, $C_1 = 150$ m/sec, $C_p = 1.005$, $\psi_s = 0.9$

$\eta_i = 80\%$.



T
T02
T02'
T2
T2'
T01
T1
P02
02
P2
2
2'
P01
P1
1

$$T_{01} = T_1 + \frac{C_1^2}{2C_p}$$

$$= 293 + \frac{150^2}{2 \times (1.005 \times 1000)}$$

$$\underline{T_{01} = 304.19\ K}$$

$$\frac{T_{01}}{T_1} = \left(\frac{P_{01}}{P_1}\right)^{\gamma-1/\gamma}$$

$$\frac{P_{01}}{P_1} = \left(\frac{T_{01}}{T_1}\right)^{\gamma/\gamma-1}$$

$$P_{01} = (1)\left(\frac{304.19}{293}\right)^{\frac{1.4}{1.4-1}}$$

$$= 1.14\ \text{bar.}$$

$$\frac{T_{02}'}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\gamma-1/\gamma}$$

$$T_{02}' = (304.19)\left(\frac{5}{1.14}\right)^{\frac{1.4-1}{1.4}}$$

= 464.07

$$\eta_{isentropic} = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}}$$

$$0.8 = \frac{464.07 - 304.19}{T_{02} - 304.19}$$

$$\therefore \underline{T_{02} = 504.04\,K}$$

change in stagnation temp. $(T_{02} - T_{01})$

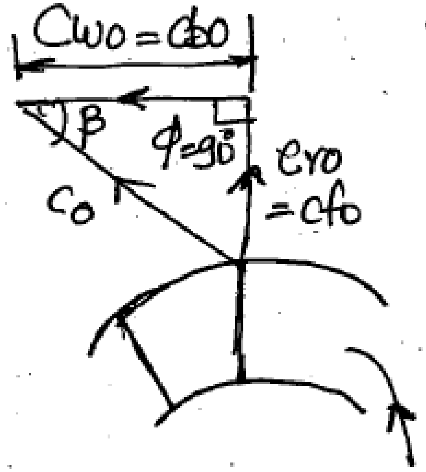
$$= [504.04 - 304.19]$$

$$= 199.85\,K$$

ii) Impeller diameter at outlet:

$$W_a = C_p (T_{02} - T_{01})$$

$$W_a = 1.005 (199.85) = 200.85 \text{ kJ/kg}$$



$$W_a = \psi_s . [C_{bo} . C_{wo}]$$

Assuming radial discharge at outlet

$C_{bo} = C_{wo}$

$(\phi = 90°)$

$$200.85 \times 10^3 = 0.9 (C_{bo})^2$$

$$\therefore \quad C_{bo} = 472.4 \text{ m/sec}$$

$$C_{bo} = \frac{\pi D_o N}{60} \quad , \quad \underline{D_o = 0.451 \text{ m}}$$

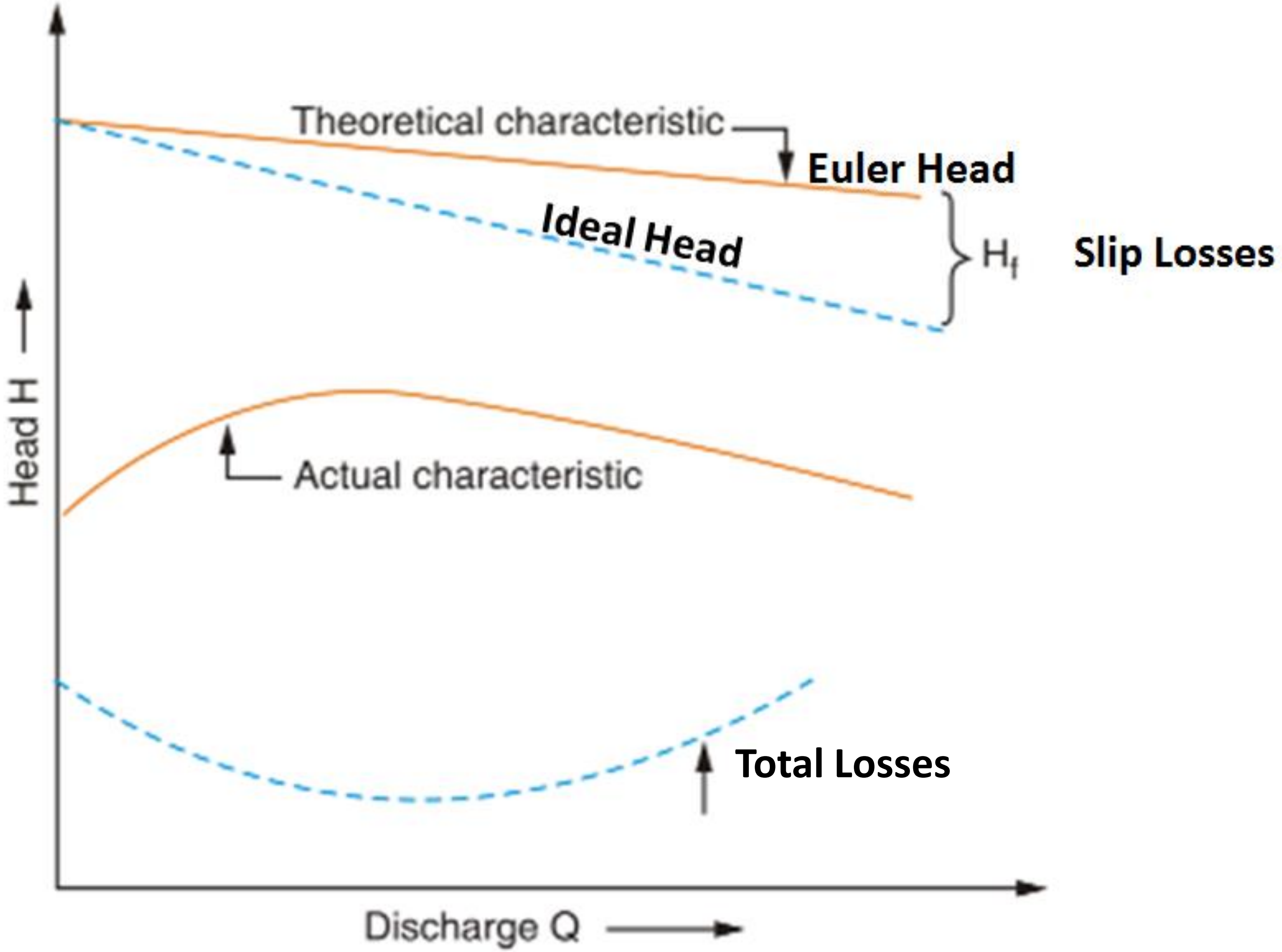
$$472.4 = \frac{\pi \times D_o \times 20000}{60}$$

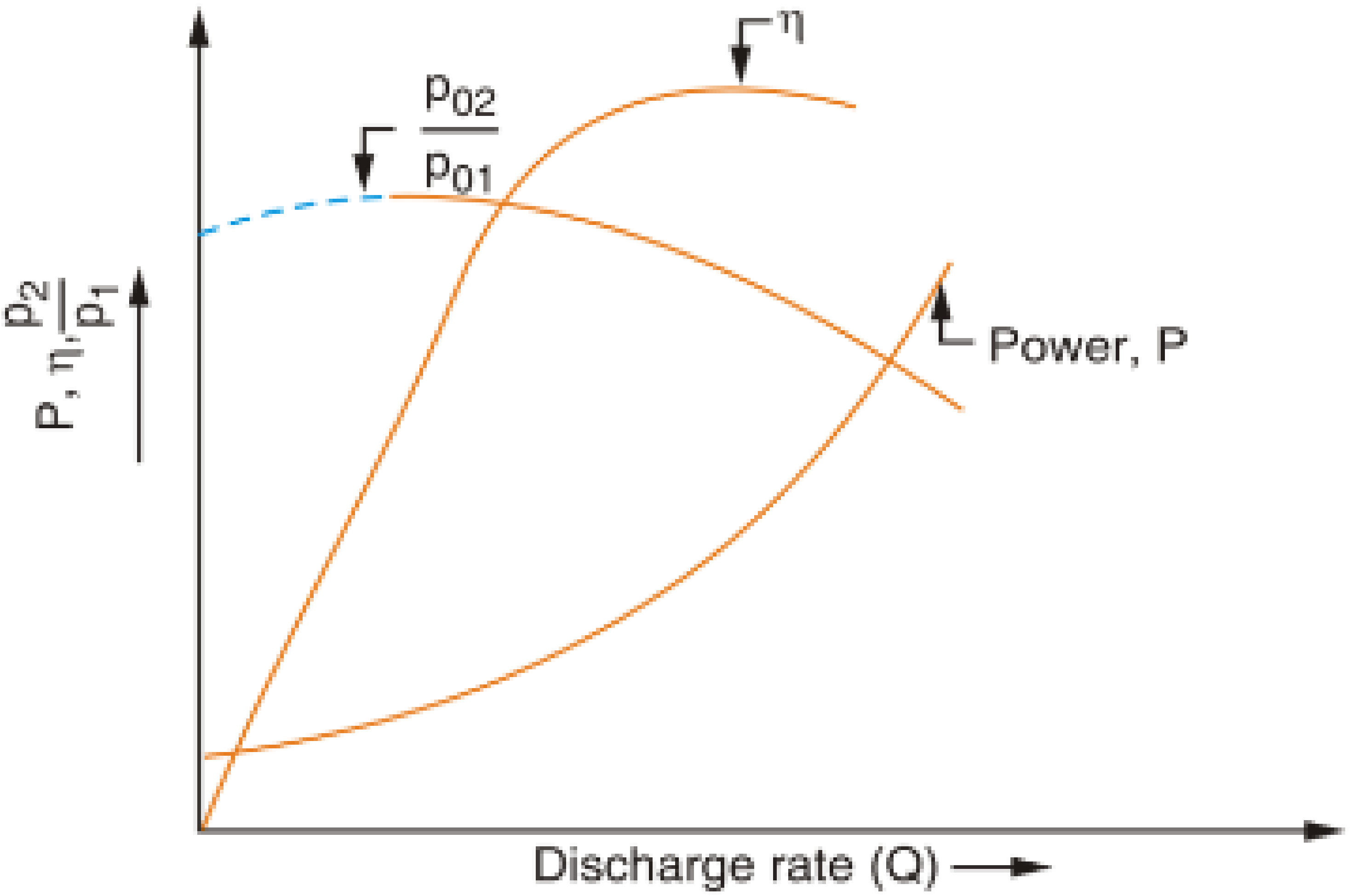
iii) Power required to drive the compressor

$$P = \dot{m} \times W = 2 \times 200.85$$

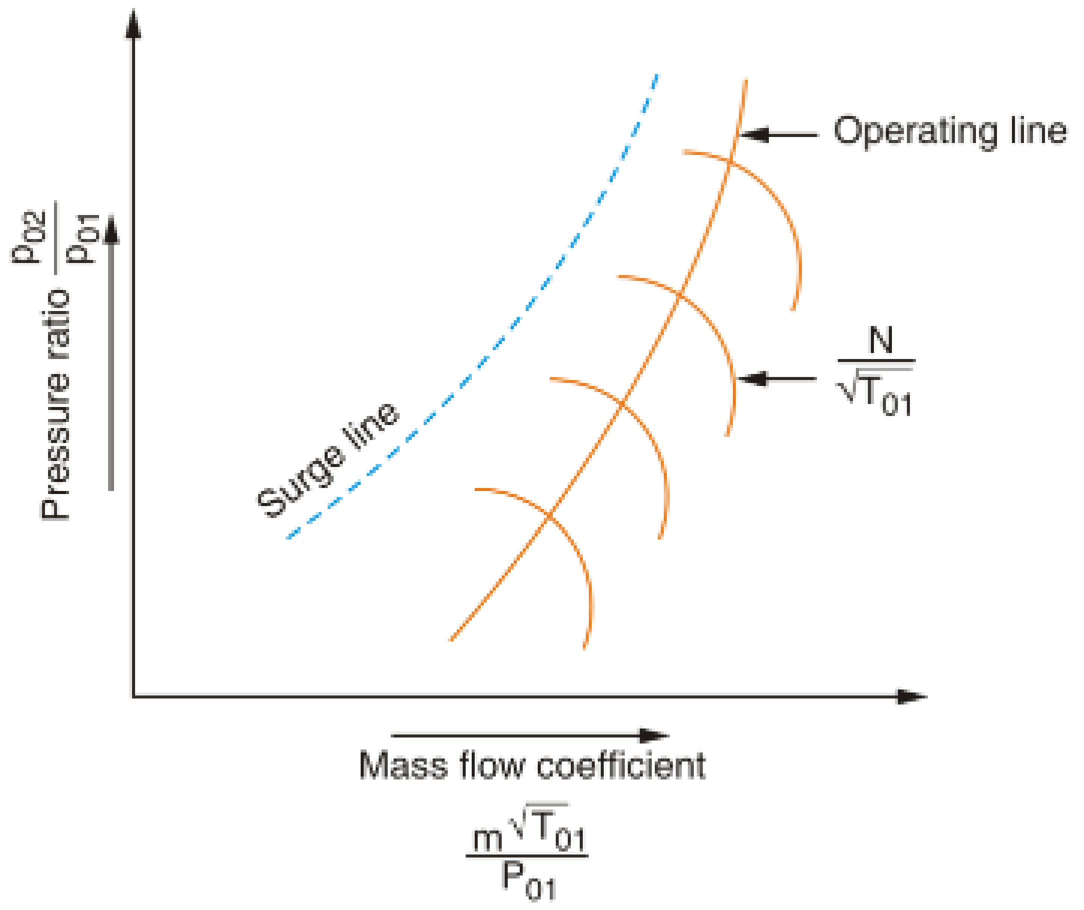
$$\boxed{P = 401.70 \text{ kW}}$$

Ideal and Actual (H-Q) Characteristics curve (backward curved vanes)

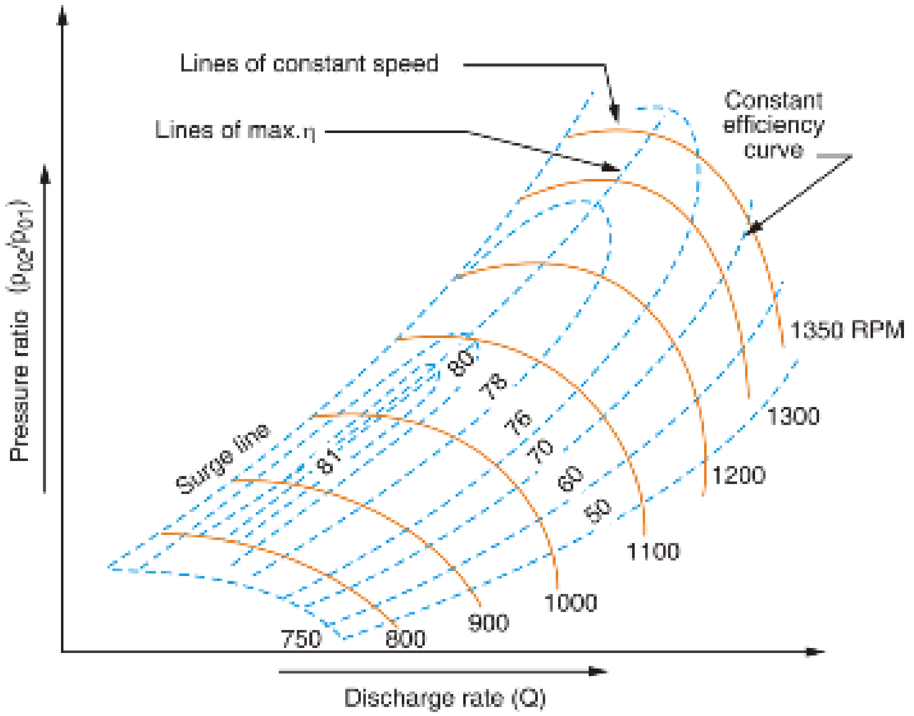




Characteristics curve for rotary compressor

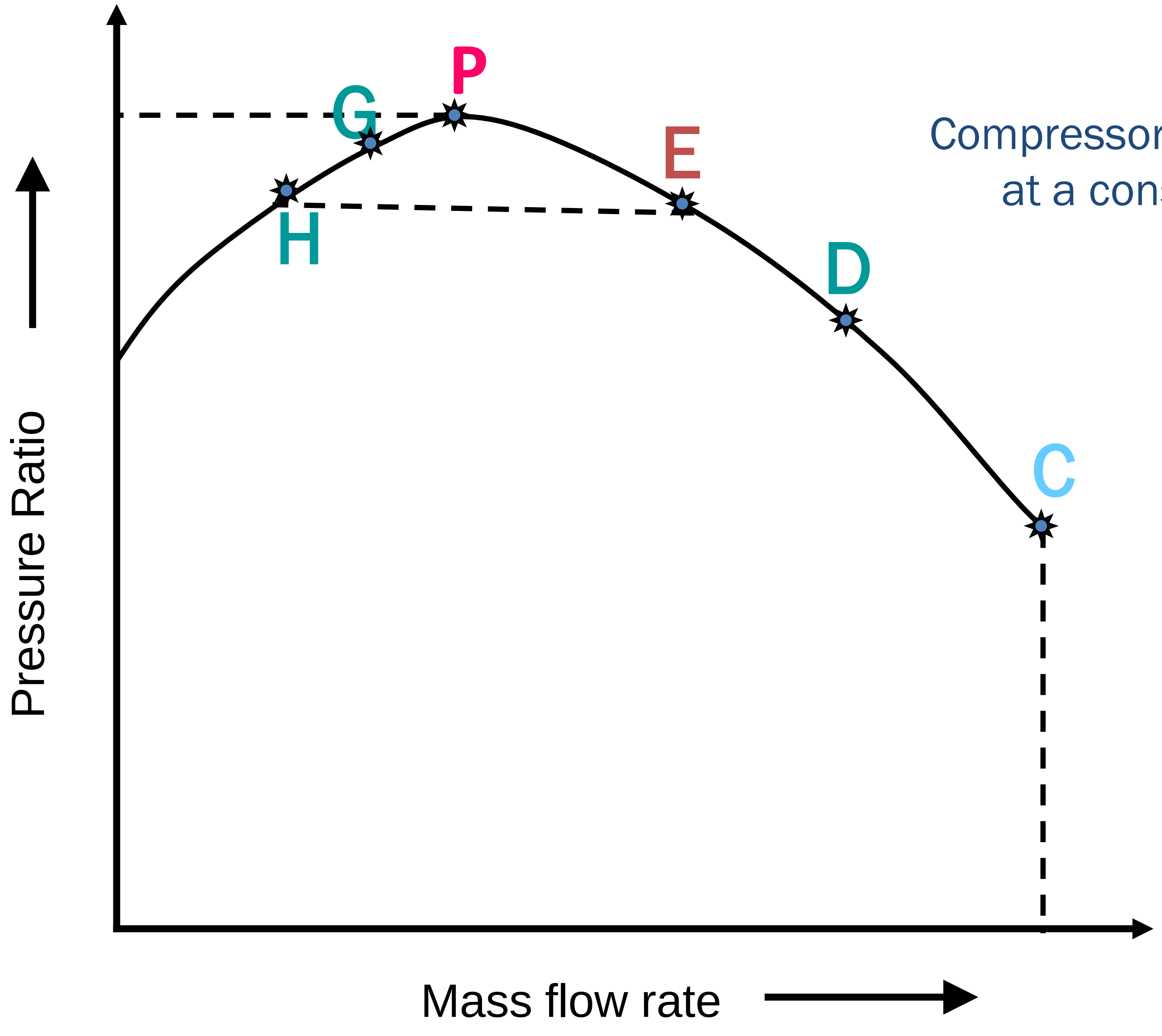


Plot of pressure ratio against mass flow coefficient at various speed parameters



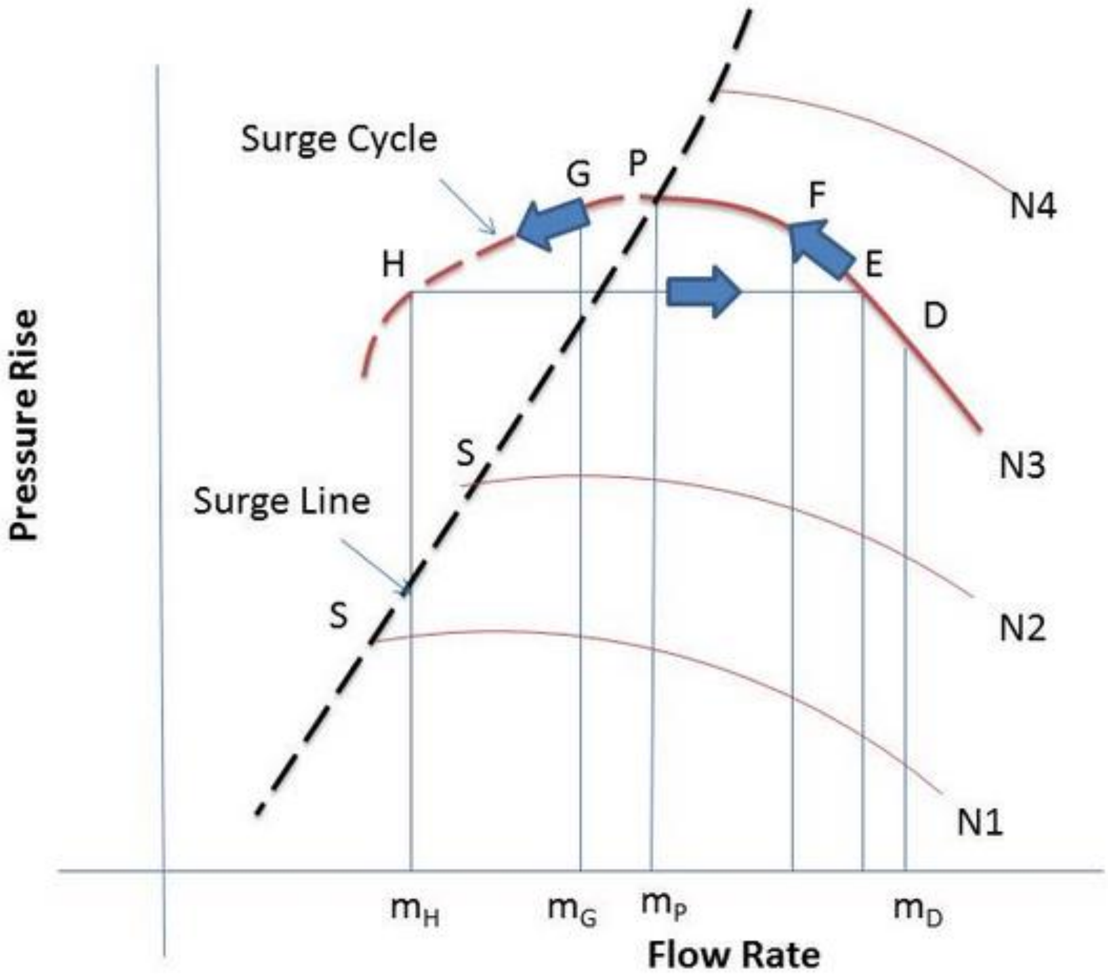
Performance characteristic curves of a centrifugal compressor

Surging and Choking



Compressor characteristics
at a constant speed

Surging



Surge Cycle
G
P
N4
F
H
E
D
N3
S
Surge Line
N2
S
N1
m_H
m_G
m_P
m_D
Flow Rate
Pressure Rise

Surging

Surging is the complete breakdown of steady flow in the compressor which occurs at low flow rate.

Surging takes place when compressor is operated off the design point and it affects the whole machine and this is aerodynamically and mechanically undesirable.

It can damage the rotor bearings, rotor seals, compressor driver and affect the whole cycle operation.

It results in high temperature, high vibration and leads to flow reversal.

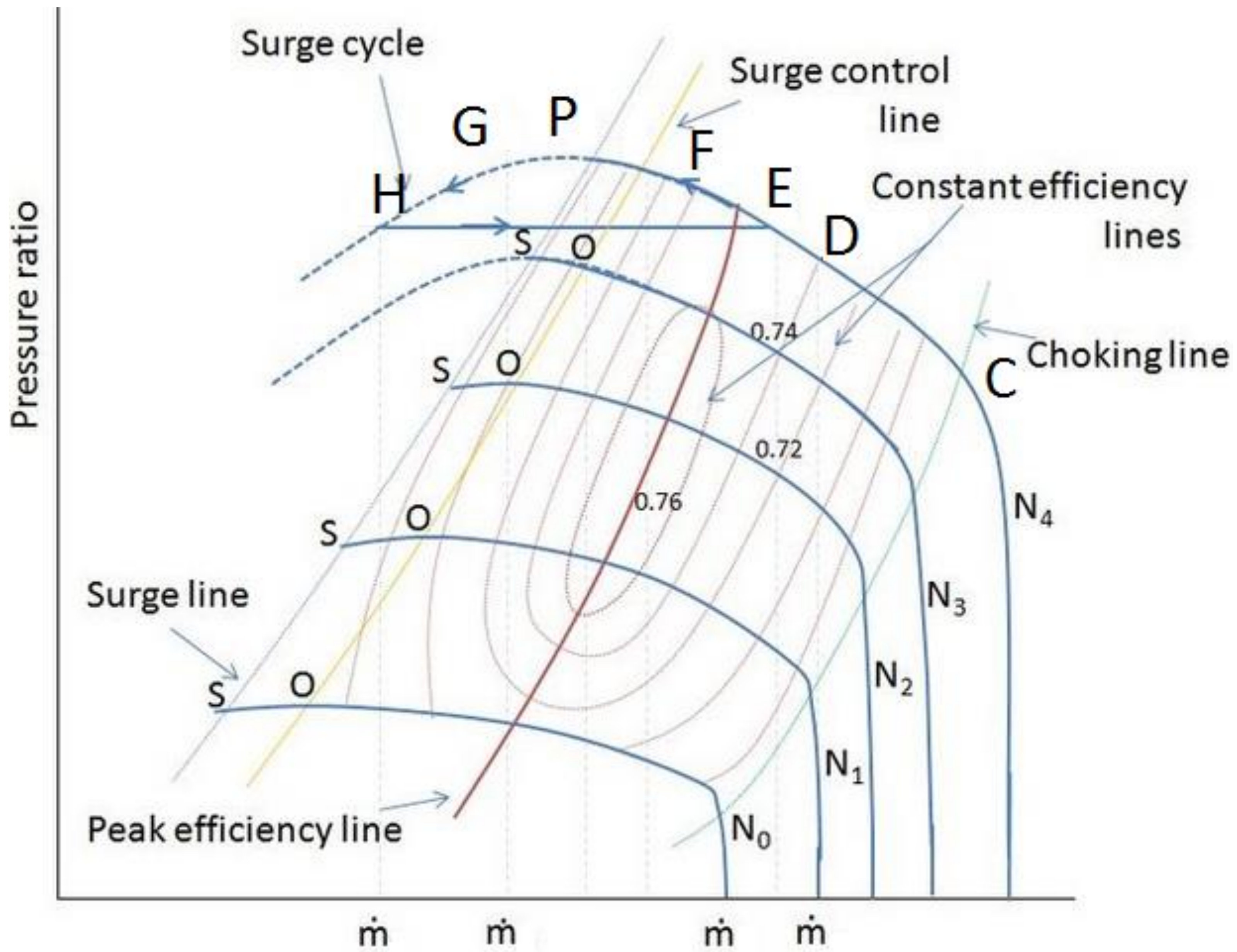
Surge process

- Assuming the compressor operates at point D on the characteristic curve (let at constant speed N_3) . Now if the flow rate is reduced to by closing a control valve on the delivery pipe, the static pressure upstream of the valve is increased. This increased pressure is then matched by the increased delivery pressure (at E) which is developed by the compressor. Now further reducing the flow the increased pressures in the delivery pipe are again matched by the compressor delivery pressures at F and P on the characteristic curve.

- On the characteristic curve at the flow rates below $\dot{m}P$ provides lower pressure as seen in the fig. at G and H. But now the pipe pressures due to further reduction of flow by valve (let at point G) will be higher than the pressure at G and H. This unbalance between the pipe pressure and the compressor delivery pressure only exist for a very short time. This is because there is higher pressure in the pipe than the air pressure produced by the compressor and due to this reversing of the flow takes place and it leads to a complete break-down of the normal steady flow from the compressor to the pipe.

- **Surge cycle**
- Due to flow reversal, pressure in the pipe falls and the compressor regains its normal stable operation (let at point E) delivering the air at higher flow rate (m_E). But the control valve is still corresponds to the flow rate m_G. Due to this compressor's operating conditions will again return to G through points F and P. And due to lower compressor pressure, the pressure falls further to P_H and the entire phenomenon from point H to G repeats again and again and this cycle **HEFPGH** known as the **surge cycle**.

- **Surge point**
- **Surge points** are the peak points on the characteristic curves left of which the pressure generated by the compressor is less than the pipe pressure and these points initiates the surge cycle. These points on the curves are shown in the fig. by point S.
- **Surge line**
- **Surge line** is the line which connects the surge points (S) on each characteristic curve corresponding to different constant speeds. The stable range of operation for the compressor is on the right hand side of the surge line.



Surge cycle
Surge control line
G
P
F
H
E
D
Constant efficiency lines
S
O
0.74
Choking line
C
0.72
0.76
N_4
N_3
Surge line
N_2
N_1
Peak efficiency line
N_0
Pressure ratio
ṁ
ṁ
ṁ
ṁ

Choking

Choking is the condition which occurs in the compressor in which it operates at very high mass flow rate and flow through the compressor can't be further increased as mach number at some part of the compressor reach to unity i.e. to sonic velocity and the flow is said to be choked.

The point on constant speed line at which choking occurs is known as **choke point** .

Choke line is the line joining the choke points on different constant speed lines . The operation on right side of choke line is not possible.

Centrifugal Compressors

- Stall

- Defined as the (aerodynamic stall) or the break-away of the flow from the suction side of the blades.
- A multi-staged compressor may operate safely with one or more stages stalled and the rest of the stages unstalled . but performance is not optimum. Due to higher losses when the stall is formed.

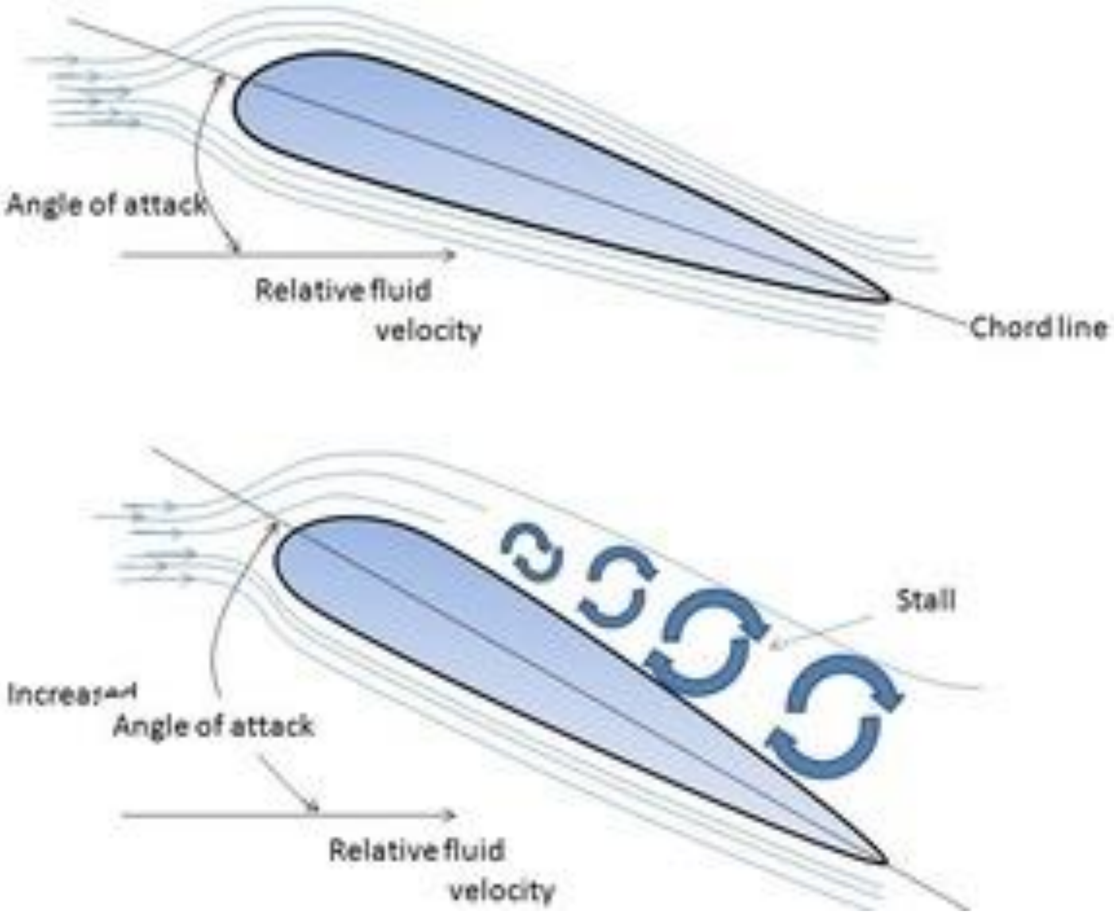
- Surge

- Is a special fluctuation of mass flow rate in and out of the engine. No running under this condition.
- Surge is associated with a sudden drop in delivery pressure and with violent aerodynamic pulsation which is transmitted throughout the whole machine.

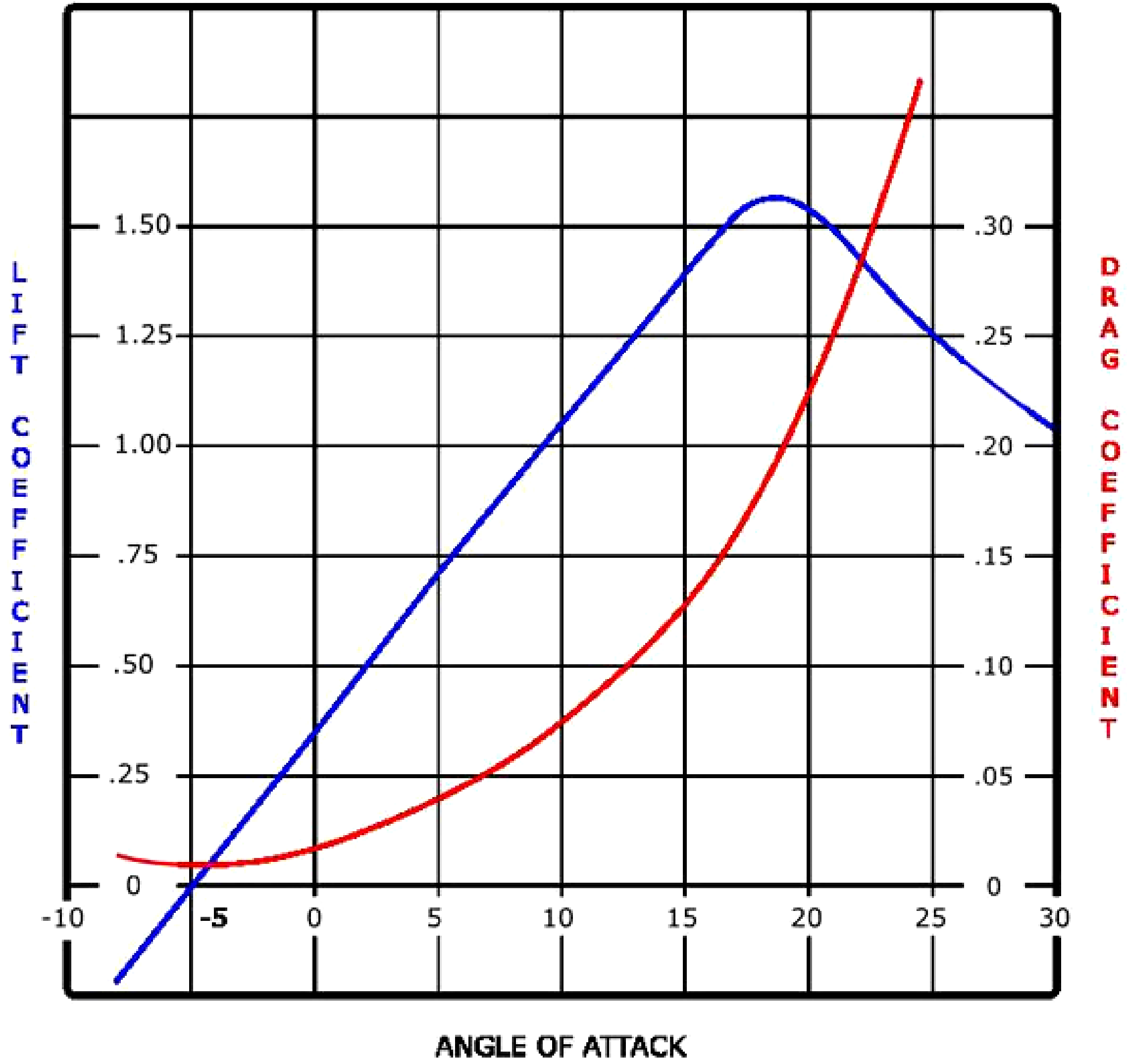
Surge Description

- Flow reverses in 20 to 50 milliseconds
- Surge cycles at a rate of 1/3 to 3 hertz
- Compressor vibrates
- Temperature rises
- "Whooshing" noise
- Trips may occur
- Conventional instruments and human operators may fail to recognize surge

STALLING


Angle of attack
Relative fluid
velocity
Chord line
Stall
Increased
Angle of attack
Relative fluid
velocity

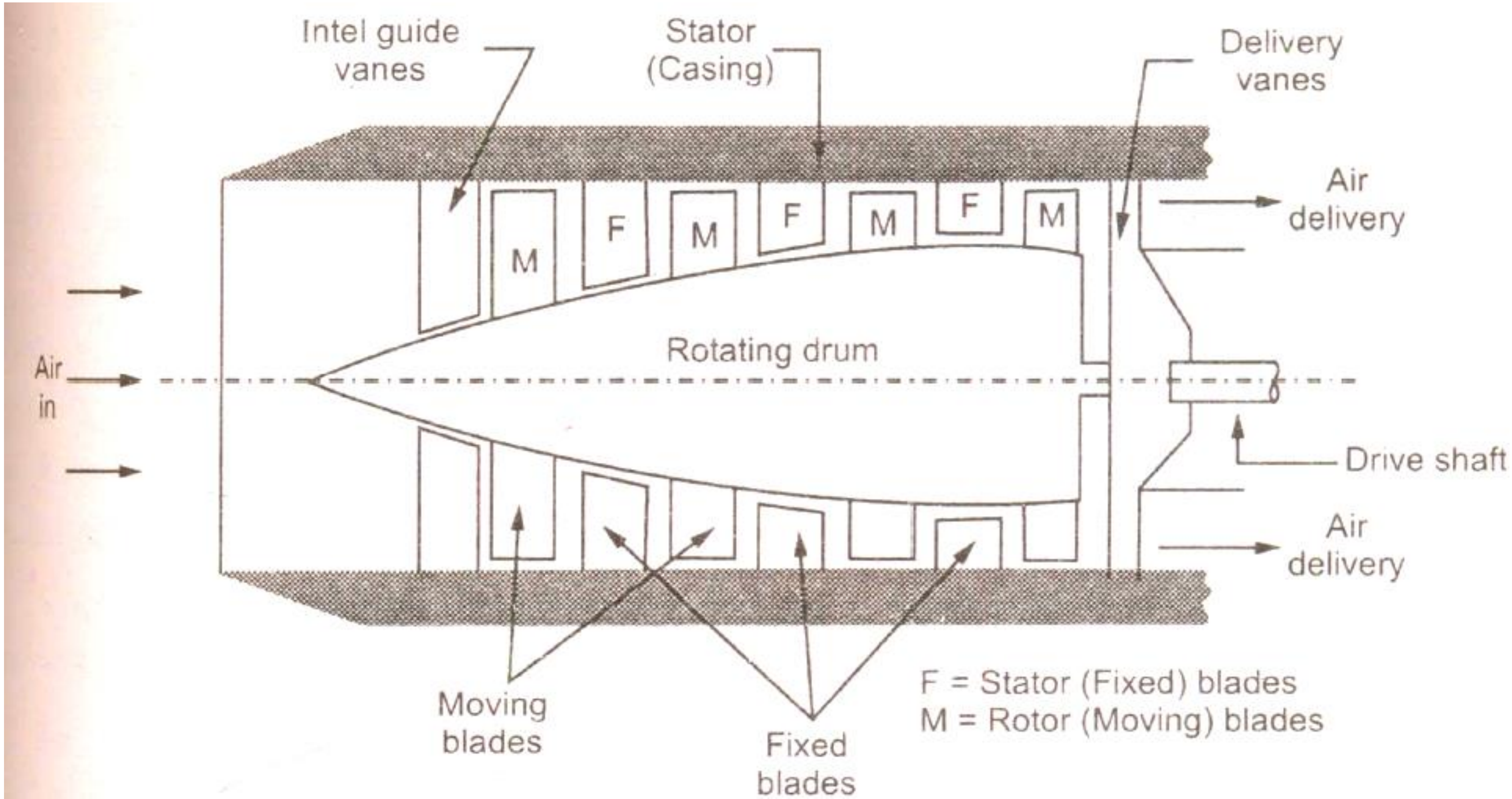
airfoil

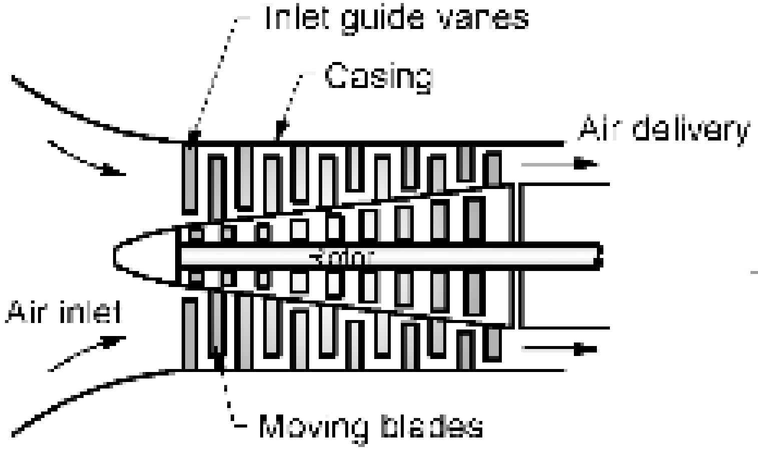

LIFT COEFFICIENT
DRAG COEFFICIENT
1.50
1.25
1.00
.75
.50
.25
0
.30
.25
.20
.15
.10
.05
0
-10
-5
0
5
10
15
20
25
30
ANGLE OF ATTACK

- **Stalling** is the separation of flow from the compressor blade surface .
- At low flow rates the incidence angle or angle of attack increases and due to this there occurs the flow separation on the suction side of the blades which is known as positive stalling.
- If the flow separation occurs on the pressure side of the blade then it’s known as negative stalling and this occurs due to negative incidence angle.
- But generally positive stalling is taken into consideration.

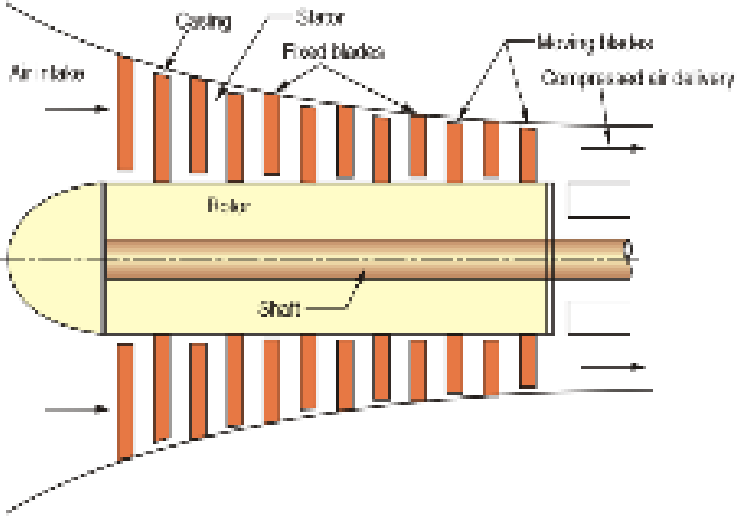
- In the compressor at high pressure stages if there occurs a deviation from design point (at which compressor is designed to operate) the angle of attack exceeds its stalling value and stall cells (which are the regions where fluid starts to whirl at a particular location and don’t moves forward) to form at hub and tip of the blade. The size of these cells increases with decreasing flow rate. If the flow rate is further reduced these cells grow larger and it affects the whole blade height and this causes significant drop in the delivery pressure and at very low flow rate, flow reversal takes place which is known as surge. It also results in drop in stage efficiency of the compressor and its delivery pressure

AXIAL FLOW COMPRESSOR





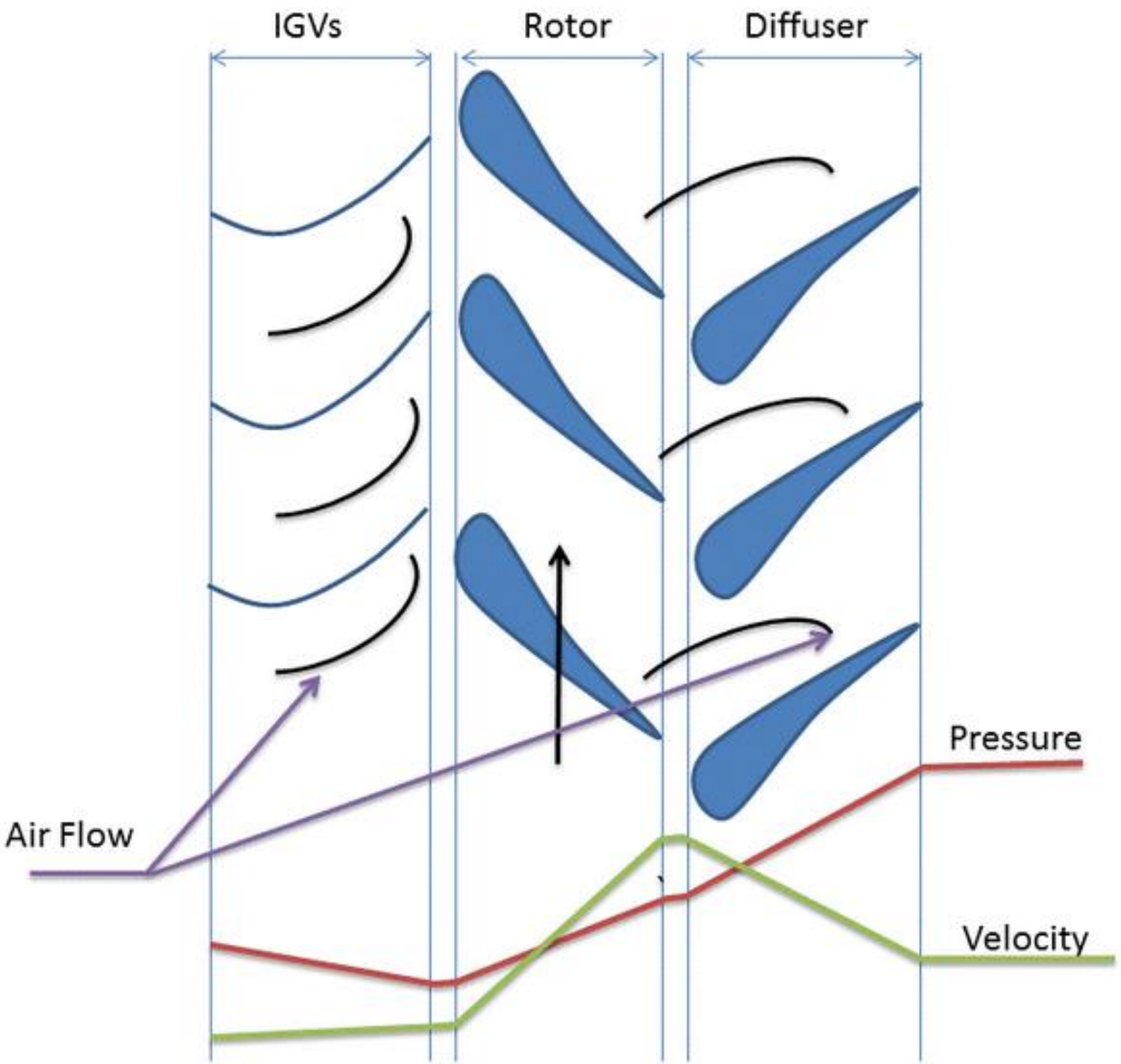
(b) Disc type rotor

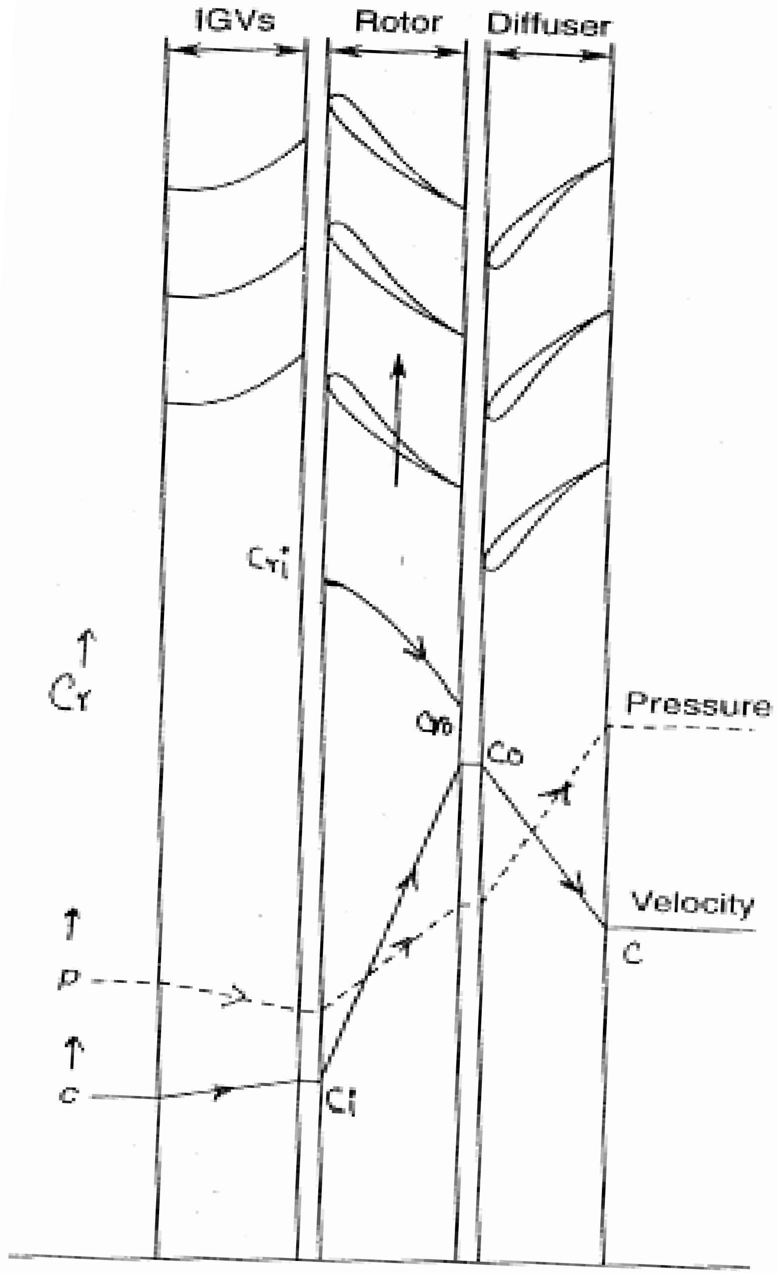


(a) Drum type rotor

Axial flow compressor

Axial Compressor- Stage



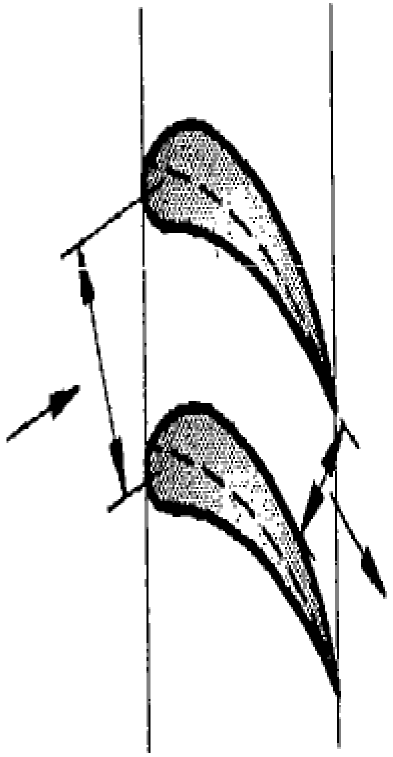


$$\Delta P_{(rotor)} = \frac{Cri^2 - Cro^2}{2}$$

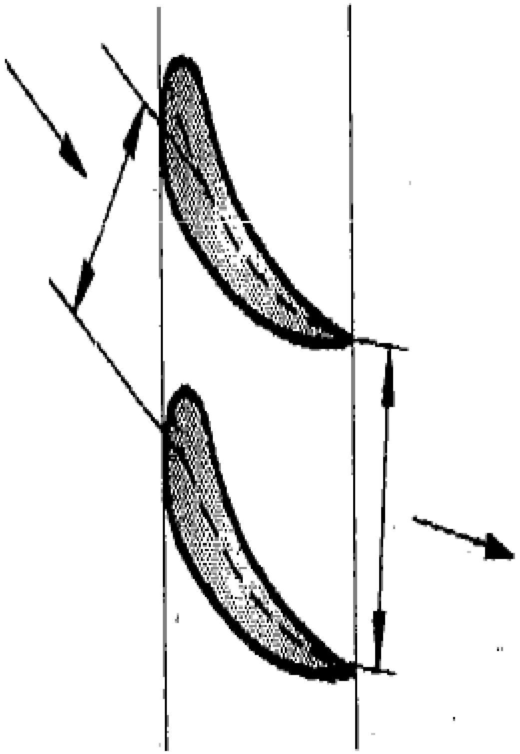
$$\Delta P_{(stator)} = \frac{Co^2 - C^2}{2}$$

$$\text{ideally} = \frac{Co^2 - Ci^2}{2}$$

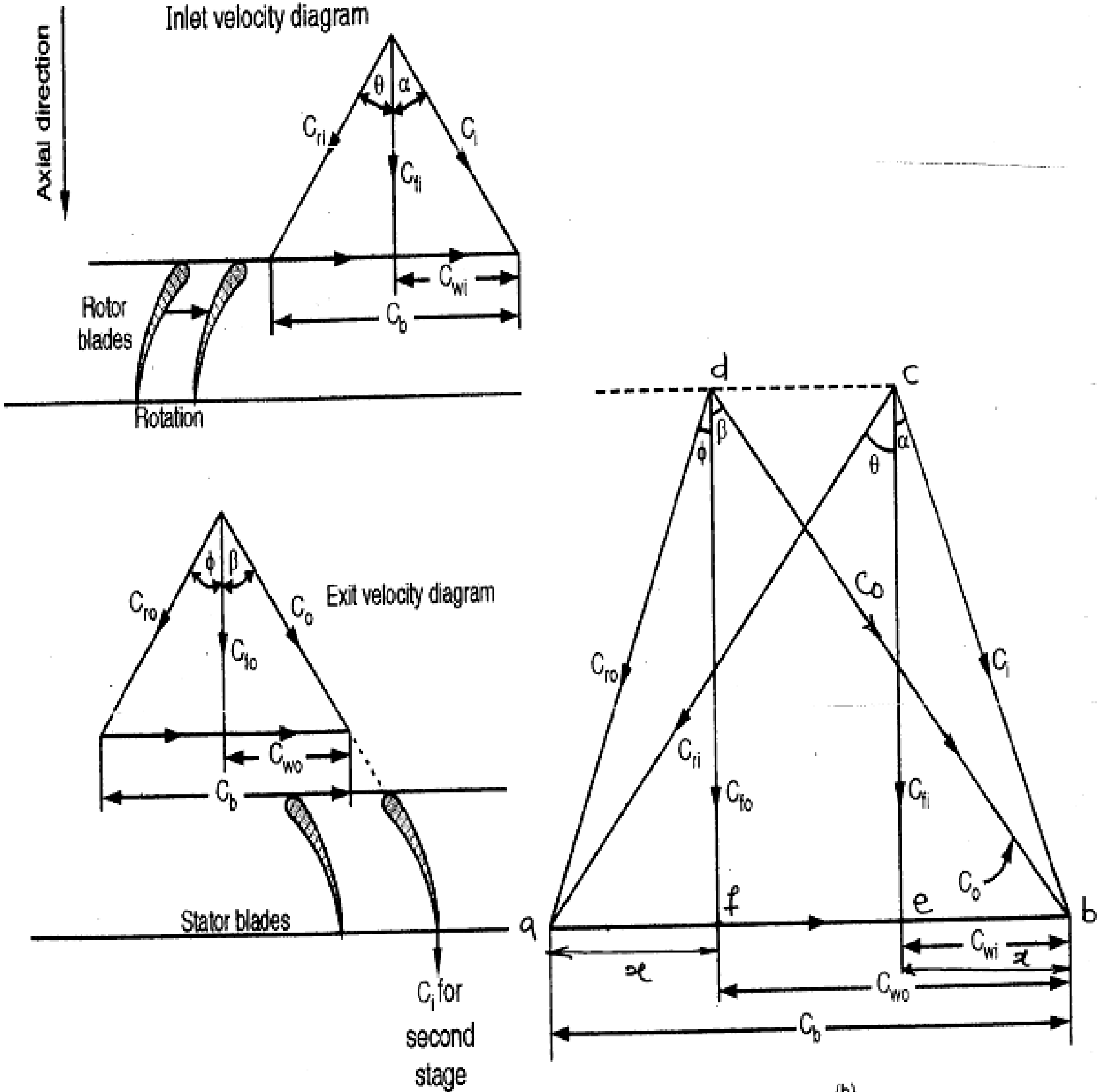
Pressure and velocity variation through a compressor stage (Axial flow)



(a) Turbine blades



(b) Compressor blades



Inlet velocity diagram
Axial direction
θ
α
C_{ri}
C_i
C_{fi}
C_{wi}
C_b
Rotor blades
Rotation
ϕ
β
C_{ro}
C_o
Exit velocity diagram
C_{fo}
C_{wo}
C_b
Stator blades
C_i for second stage
d
c
ϕ
β
θ
α
C_o
C_{ro}
C_i
C_{ri}
C_{fo}
C_{fi}
C_o
a
f
e
b
C_{wi}
x
x
C_{wo}
C_b

(a)

(b)

Work input / kg of air (Newton's 2nd law of motion)

$$W = (C_{wo} - C_{wi})\, C_b$$

Applying S.F.E.E for impeller

$$W = C_p\,(T_{02} - T_{01})$$

$$\therefore \quad C_p\,(T_{02} - T_{01}) = (C_{wo} - C_{wi})\, C_b$$

Degree of reaction: for axial flow compressors as the ratio pr. rise in the rotor blades to pr. rise in the stage.

$$R = \frac{\text{Pr. rise in the rotor blades}}{\text{Pr. rise in (rotor + stator) blades}}$$

$$R = \frac{C_{ri}^2 - C_{ro}^2 / 2}{\frac{C_{ri}^2 - C_{ro}^2}{2} + \frac{C_o^2 - C_i^2}{2}}$$

$$R = \frac{C_{ri}^2 - C_{ro}^2 / 2}{(C_{wo} - C_{wi})\, C_b}$$

For 50% D.O.R, compressor has symmetrical blades. By using velocity triangles & substituting

$$R = \frac{C_f(\tan\theta + \tan\phi)}{2C_b}$$

$$0.5 = \frac{C_f(\tan\theta + \tan\phi)}{2C_b}$$

$$\therefore \quad \frac{C_b}{C_f} = (\tan\theta + \tan\phi). \quad \text{—— (i)}$$

From geometry of velocity Δ's,

$$\frac{C_b}{C_f} = (\tan\theta + \tan\alpha) = (\tan\beta + \tan\phi) \quad \text{— (ii)}$$

Equating (i) & (ii)

$$\boxed{\alpha = \phi \text{ and } \theta = \beta}$$

Sr. No.	Centrifugal compressor	Axial flow compressor
1.	The flow is radial.	The flow is axial.
2.	It develops high pressure ratio per stage of about 4.5:1.	It develops low pressure ratio per stage of about 1.2:1. Therefore to get high pressure ratios, a large number of stages are required.
3.	It has high efficiency over wide range of speeds.	It has high peak efficiency of 86 to 89% in narrow range of speeds when multistage compression is used with aerofoil blades.
4.	Easy to manufacture.	Difficult to manufacture.
5.	It has low starting power requirements.	It requires high starting powers.
6.	It has low weight and cost.	It has high weight and cost.
7.	It needs large frontal area for given mass flow rates.	It needs less frontal area for given mass flow rates.
8.	It is not suitable for multistaging due to large losses in between the stages.	It is suitable for multistaging.
9.	The part load performance is good.	The part load performance is poor.
10.	These are suitable for supercharging of I.C. engines, for compression of refrigerants and other industrial applications.	These are suitable for jet engines, gas turbine power plants and steel mills.