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Department: Mechanical Engineering

Subject: ME- 603 (Turbomachinery)

Unit: IV

Topic: Theory of Compressor

**Pressure ratio**: As compared to reciprocating compressors the velocities encountered in centrifugal compressors are very large and therefor total head quantities should be considered while analysing centrifugal compressors. The total head quantities take into account the kinetic energy of the air passing through the compressor.

Let us consider, a horizontal passage of varying area through which air is flowing. Applying steady flow energy equation to the system for 1 kg of air flow (assuming no external heat transfer and work transfer to the system), we get

$$U_1 + p_1 v_1 + (C_1)^2 / 2 = U_2 + p_2 v_2 + (C_2)^2 / 2$$

$$H_{1+}(C_1)^2/2 = H_2 + (C_2)^2/2$$

$$C_p T_1 + (C_1)^2/2 = C_p T_2 + (C_2)^2/2$$

$$C_pT + (C_2)^2/2 = constant \\$$

Temperature 'T' is called the static temperature or the temperature of the air measured by the thermometer when the thermometer is moving at the air velocity. If the moving air is brought to rest under reversible conditions, the total kinetic energy of the air is converted into heat energy, increasing the temperature and pressure of the air. The temperature and pressure of the air is known as 'stagnation' or 'total head' temperature and pressure. The total head temperature and pressure are denoted by a suffix notation '0'.

$$C_pT + (C)^2/2 = C_pT_0$$

Where T<sub>0</sub> is known as total head or stagnation temperature



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 $T_0 - T = C^2/2c_p$ 

 $H_0 - H = C^2/2$ 

For finding the total head pressure, use the equation

 $P_0/P = (T_0/T)^{\gamma/(\gamma-1)}$ 

P= static pressure,

T= static temperature,

 $P_0$  = stagnation pressure, and

 $T_0$  = stagnation temperature

### Velocity Diagrams and Theory of Operation of Centrifugal Compressors

Let  $U_1$  = Mean blade velocity at entrance,

 $U_2$  = Mean blade velocity at exit,

 $V_1$  = Absolute velocity at inlet to the rotor,

 $V_2$  = Absolute velocity at outlet to the rotor,

 $V_{r1}$ = Relative velocity of air at entry of rotor,

V<sub>r2</sub>= Relative velocity of air at exit of rotor,

V<sub>w1</sub>= Velocity of whirl at inlet,

V<sub>w2</sub>= Velocity of whirl at outlet,

 $V_{f1}$  = Velocity of flow at inlet,

 $V_{f2}$  = velocity of flow at outlet,

 $\alpha_1$  =Exit angle from the guide vane or inlet angle of the guide vane,

 $\alpha_2$  = Inlet angle of the diffuser,

 $\beta_1$ = Inlet angle to the rotor or impeller,

 $\beta_2$ = Outlet angle to the rotor or impeller,



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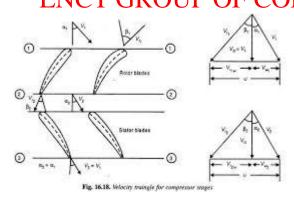


Fig. shows the velocity diagrams for the inlet and outlet of the impeller. It is assumed that the entry of the air is axial therefore the whirl component at the inlet  $(V_{w1})$  is zero and therefore  $V_1 = V_{f1}$ . The enlarged views of inlet and outlet velocity diagrams are shown in fig. To avoid shock at entry and exit the blade must be parallel to the relative velocity of air at outlet or inlet and therefore  $\beta_1$  and  $\beta_2$  are the impeller blade angles at the inlet and outlet. The diffuser blade angle must be parallel to the absolute velocity of air from the impeller  $(V_2)$  therefore  $\alpha_2$  is the diffuser blade angle at the inlet and  $\alpha_3$  is the diffuser blade angle at the outlet. If the discharge from the diffuser is circumferential, then the blade angle at outlet  $\alpha_3$  should be as small as possible.

### Work done by impeller:

The work done per unit time by the impeller can be calculated by finding out firstly the tangential force on the impeller and then multiplying it with the impeller blade velocity at the outlet, tangential force  $(F_t)$  x blade velocity  $(U_2)$ . According to newton's second law of motion, force applied is proportional to the rate of change of momentum. So the tangential force  $F_t$  = mass of air entering and leaving the impeller in a unit time x rate of change of tangential velocity in a unit time.

$$F_t = m (V_{w2} - V_{w1})$$

 $V_{w1} = 0$  as air enters radially

$$F_t = m (V_{w2})$$

Where m is the mass of air flow through the compressor in kg/s.

Now, work done = 
$$F_t \times U_2$$

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Work done =  $m (V_{w2}) U_2$ 

Work done per kg of air entering and leaving the impeller in a unit time =  $(V_{w2})$   $U_2$ 

This work is known as Euler's work

Power required per impeller for m per kg of air flow in one second,

$$P = (mV_{w2}U_2)/1000$$
 KW

- If the blade is radial then the velocity diagram at the outlet of the impeller is as shown in fig. As  $V_{w2}=U_2$  the work done per kg of air flow per second is given by  $\mathbf{W} = (\mathbf{V}_2)^2$
- Since the air cannot leave the impeller at velocity greater than the impeller tip velocity, the maximum work supplied per kg of air flow per second.
- Now consider the steady flow at the inlet and outlet of impeller, assuming the heat transfer during the flow of air through the impeller is zero.

$$H_{1+}(V_1)^2/2 + W = H_2 + (V_2)^2/2$$

$$W = (H_2 + (V_2)^2/2) - (H_{1+}(V_1)^2/2)$$

$$W = C_p \; (T_2 + (V_2)^2 / 2 C_p) \; \text{ - } C_p \; (T_1 + \! (V_1)^2 / 2 C_p)$$

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

Losses and Isentropic Efficiency of the Compressor:

The following losses occur when air flows through the impeller

- Friction between the air layers moving with relative velocities and friction (i) between the air and flow passages
- Shock at entry (ii)
- (iii) Turbulence caused in air

The losses mentioned above cause an increase in enthalpy of the air without increase of pressure therefore the actual temperature of air coming out from the compressor is more than the temperature of air if it is compressed isentropically. The actual work required for the same increase in pressure ratio is more due to irreversibilities. The actual and isentropic compression for the same pressure ratio is shown in fig.



## "WORKING TOWARDS BEING THE BEST"

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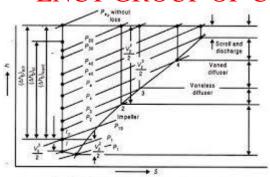


Fig. 16.10. Centrifugal compressor process on h-S chart

The isentropic efficiency is given by the relation,

$$\eta_{isen} = (Isentropic \ work \ / \ Actual \ work) = ((h_{02})^{^{\backprime}} - h_{01}) / \ (h_{02} - h_{01})$$

$$= ((T_{02})' - T_{01})/(T_{02} - T_{01})$$

Provided specific heat at constant pressure (C<sub>p</sub>) remains constant.

If 
$$V_1 = V_2$$
 then  $\eta_{isen} = ((T_2)^{'} - T_1)/(T_2 - T_1)$ 

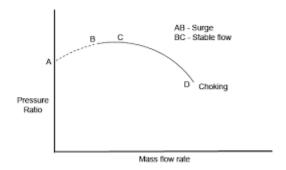
**Slip Factor** ( $\phi_s$ ). It is defined as the ratio of actual whirl component ( $V_{w2}$ ) and the ideal whirl component ( $U_2$ )

$$\phi_s = V_{w2} / U_2$$



### **Surging and Choking**

In the compressor where the flow is against the pressure gradient the incidence loss due to incorrect fluid angles relative to the blades becomes more important and pressure ratio falls sharply at conditions away from design point. This loss, added to the friction loss which will increase with increase of mass flow rate give a pressure ratio- mass flow rate relation as shown in fig.



- 1. At point N, the compressor is choked and is passing the maximum mass flow rate.
- 2. On the section MN of the curve the flow is stable. A fall in mass flow rate will result in a rise in pressure ratio which will tend to restore the fall.
- 3. On the section LM of curve the flow is not stable. A fall in mass flow rate will be accompanied by a fall in pressure ratio. In this situation any small disturbance causing a check in mass flow will cause a fall in pressure ratio and the flow may reverse at some point. When the temporary disturbance is removed, the flow will pick up and it is found that small disturbances cause the flow to oscillate rapidly. The oscillations is noisy and can, if allowed cause structural damage in the compressor. It is called surging.

