TURBOMACHINERY 2024 Solved Question Paper

SPPU Mechanical Telegram Channel - Created by @vaibhavpandit_tele
(Reference - Technical Publication & Chat GPT)

Q1) a) Give classification of turbomachines with suitable example. **[6]**

1.2 Classification of Turbo Machines

SPPU : M

The turbo machines may be classified as follows :

1) According to the direction of energy flow in the machines

i) Energy flows from fluid to rotor (hydraulic turbines and steam turbines)

ii) Energy flows from rotor to fluid (pumps, compressors, fans, blowers, etc)

2) According to the type of fluid used

i) Hydraulic turbo machines (pumps, water turbines etc.)

ii) Thermal turbo machines (fans, comprossors, blowers etc.)

3) According to the dynamic action of fluid

i) Impulse based machines (Pelton turbine)

ii) Reaction based machine (Francis or Kaplan turbine)

4) According to the direction of fluid flow

i) Tangential flow machines (Pelton turbine)

ii) Radial flow machines (Centrifugal pump or compressor)

iii) Axial flow machines (Kaplan turbine or axial compressor)

iv) Mixed flow machines (Francis turbine)

b) A Pelton wheel is having a mean bucket diameter of 1 m and is running at 1000 rpm. The net head on pelton wheel is 700m. If the side clearance angle is 15º and discharge through the nozzle is 0.1 m^3/s. Determine **[8]**

i) Power available at the nozzle.

ii) Hydraulic efficiency of the turbine.

Turbo - 2024

Q1] b)

bucket diameter = 1 m, RPM = 1000

Net head = 700 m   side clearance = 15°.

Discharge Rate = $0.1\ m^3/s$

Determine → Power available at nozzle
Hydraulic effcy of the turbine.

Assume ⇒ $C_v = 0.98$ $g = 9.81$

Power available at nozzle is water power.

$$WP = \rho \times g \times Q \times H = 1000 \times 9.81 \times 0.1 \times 700.$$

$$= 686.7 \times 10^3\ W = 686.7\ kW$$

Hydraulic Efficiency (η) of the turbine -

$$u = \frac{\pi D N}{60} = \frac{\pi \times 1 \times 1000}{60} = 52.35 \text{ m/sec}$$

The velocity of jet is,

$$V_1 = C\sqrt{2gH} = 0.98 \times \sqrt{2 \times 9.81 \times 700}$$

$$V_1 = 114.84 \text{ m/sec}$$

(Hydraulic Efficiency)
$$\eta_h = \frac{2\left[(V_1 - u)(1+\cos\phi)\right]u}{V_1^2}$$

$$= \frac{2\left[(114.84 - 52.35)(1+\cos 15^\circ)\right]52.35}{(114.84)^2}$$

---- use calci

$$\eta_h = 0.9752$$

$$\Rightarrow 97.52\%$$

Q2) a) Explain construction and working of Kaplan Turbine with neat sketch. **[6]**

SPPU : Oct. 2019

3.11.2 Kaplan Turbine

- If the vanes on the hub are fixed but adjustable then the turbine is called as **Kaplan turbine**. It is a modified propeller turbine.
- Kaplan turbine is an axial flow turbine which is suitable for low heads and hence it requires large quantity of water to develop large amount of power.
- It was developed by V. Kaplan an Austrian Engineer.
- It operates in an entirely closed conduit from the head race to tail race.
- Fig. 3.30 shows the arrangement of Kaplan turbine. (See Fig. 3.30 on next page)
- The main elements of Kaplan turbine are similar to Francis turbine.
- In these turbines, water turns between the guide vanes and runner at right-angle into the axial direction and then passes through the runner.
- The vanes attached to the hub or boss are so shaped that the water flows axially through the runner.
- The main advantage of Kaplan turbine is that, its runner blades can be turned about their own axis for adjusting the angle of inclination.

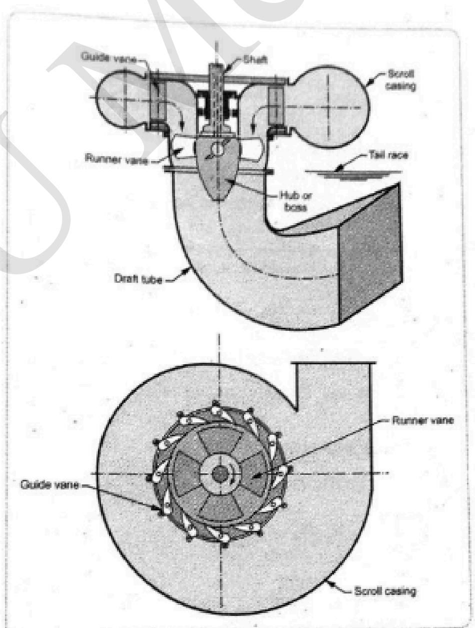


Fig. 3.30 : Kaplan turbine

- The working principle of Kaplan turbine is similar to Francis turbine.
- The method of drawing the velocity diagram and the expressions for workdone, power and efficiency of a Kaplan turbine are similar to Francis turbine.

b) The external and internal diameters of inward flow reaction turbines are 1.20 m and 0.6 m respectively. The head on the turbine is 22 m and velocity of flow through the runner is constant and equal to 2.5 m/s. The guide blade is given as 10° and runner vanes are radial at inlet. If the discharge at outlet is radial, determine **[8]**

i) The speed of the turbine

ii) The vane angle at the outlet of the runner

iii) Hydraulic efficiency of the turbine.

Q2] b] Inward flow - reaction turbine.

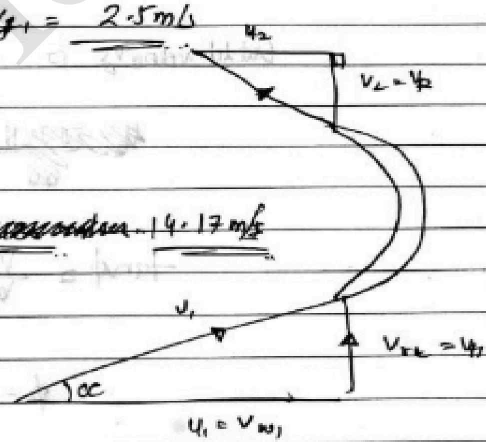
External dia = 1.2m

Internal dia = 0.6m

Head of turbine = 22 m   velocity of flow = 2.5 m/s

(α) Guide blade angle = 10° $\theta = 90°$

$V_{w1} = u_1 \quad V_{f1} = V_{f1} = 2.5\ m/s$



$\tan\alpha = \dfrac{V_{f1}}{u_1}$

$u_1 = \dfrac{2.5}{\tan 10} = 14.17\ m/s$

$V_{w1} = u_1 = 14.17\ m/s$

Also $u_1 = \dfrac{\pi D_1 N}{60}$

$14.17 = \dfrac{\pi \times 1.2 \times N}{60}$ $N = \dfrac{14.17 \times 60}{\pi \times 1.2}$

$N = 225.52$ rpm

Blade velocity at outlet :

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.6 \times 225.52}{60}$$

$$u_2 = 7.084 \ m/s$$

Outlet velocity :

$$\cancel{u_2 = \frac{\pi D_2 N}{60} = \pi \times \ldots \times 0.6 \times}$$

$$\tan\phi = \frac{V_{f2}}{u_2} = \frac{2.5}{7.084} = 0.352$$

$$\phi = \tan^{-1}(0.352)$$

$$\phi = 19.43^\circ$$

Hydraulic efficiency of turbine

$$\eta_h = \frac{V_{w1} \ u_1}{gH} = \frac{14.17 \times 14.17}{9.81 \times 22} = 0.93035$$

$$\eta_h = 90.35\%$$

Q3) a) What is compounding? Explain the need of compounding. Explain any one method of compounding in steam turbines. **[6]**

4.7 Compounding of Steam Turbine

SPPU : May 2011, Dec. 2013, May 2014, May 2015, April 2016, April 2017, May 2019, Dec. 2019

- When the steam is expanded from boiler pressure to condenser pressure in a single stage, the absolute velocity of steam leaving the turbine is very high.
- Also this results in the tremendously high blade velocity and consequently the high rotor speed upto 25000 to 30000 rpm.
- Hence the various methods should be adopted to reduce this rotor speed to a lower value.
- These methods includes mounting the number of rotors on a common shaft. Due to this the jet velocity is absorbed in stages as the steam flows over blades. This is known as **compounding of steam turbine.**
- The following are the methods used for compounding of steam turbine :

(i) Velocity compounding (ii) Pressure compounding (iii) Pressure-velocity compounding

4.7.1 Velocity Compounding

SPPU : April 2017

- Fig. 4.7 shows the velocity compounding arrangement in which velocity is decreased gradually over the set of moving and fixed blades.
- The steam is expanded from boiler pressure to condenser inlet pressure through a stationary nozzle. During this the pressure drop occurs and velocity and kinetic energy of steam increases.

- Now the portion of the increased kinetic energy is absorbed by a row of moving blades. This results in the decrease in the velocity as the work is done on the moving blades.
- The steam is then flows through the row of fixed blades. These fixed blades are used to direct the steam over next row of moving blade without altering the velocity.
- During the flow of steam over next rows of moving blades where again the work is done and steam leaves the turbine with lower value of velocity.
- As the velocity is gradually decreased over the compounded stages, the method is known as the velocity compounding.

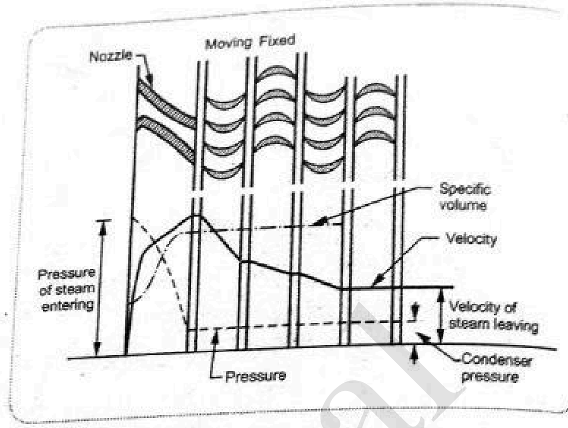


Fig. 4.7 : Velocity compounded impulse turbine

Advantages of velocity compounding

- Less number of stages are employed to reduce the velocity.
- Low initial cost.

Disadvantages of velocity compounding

- High steam consumption.
- Low efficiency.

4.7.2 Pressure Compounding

SPPU : Dec. 2018

- Fig. 4.8 shows the pressure compounding arrangement in which pressure of steam decreases gradually over the set of nozzles and blades.
- In first stage, the steam expands partially from boiler pressure in the first set of nozzles. The kinetic energy increases during the flow of steam through nozzle and again get absorbed at the moving blades where pressure remains constant.
- During second stage again the steam expands in the nozzle and pressure falls to a certain value and velocity of steam again increases.
- The kinetic energy obtained is then again absorbed by the row of moving blades where pressure remains constant.
- This process repeats in the next stages untill the condenser pressure and lower value of velocity is achieved.
- As the pressure is gradually decreased over the compounded stages of nozzles and blades, the method is known as pressure compounding.

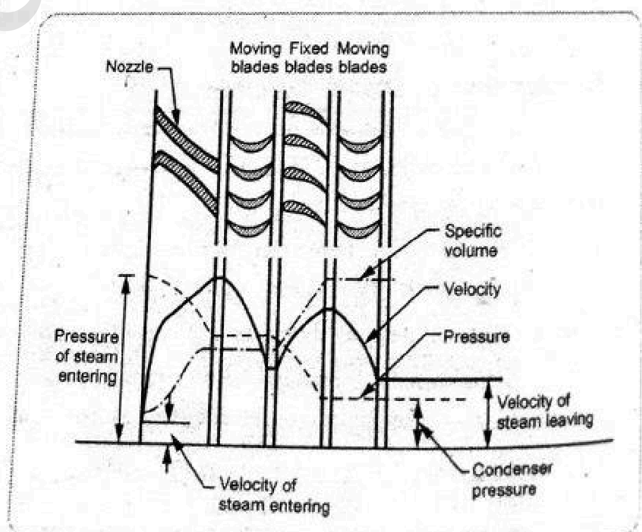


Fig. 4.8 : Pressure compunded steam turbine

Advantage :

- The speed ratio remains constant throughout the turbine.

Disadvantages :

- Larg number of stages are used in pressure compounding.
- Expensive method of compounding.

4.7.3 Pressure Velocity Compounding

- Fig. 4.9 shows the pressure - velocity compounding arrangement which is the combination of both the methods discussed earlier.
- It consist of number of stages of nozzles, fixed and moving blades. The pressure drop from boiler pressure to condenser pressure is divided into the stages and lower value of velocity also.
- The nozzles are fitted at the begining of each stage and pressure drop occurs only during the flow through nozzle. The pressure remains constant during remaining stage.

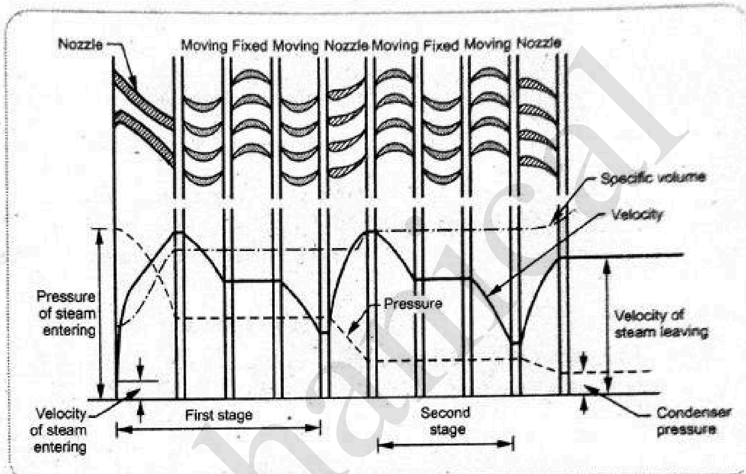


Fig. 4.9 : Pressure and velocity compounded steam turbine

- Also during the flow of steam through nozzle the velocity of steam as well as kinetic energy increases. This increased kinetic energy is absorbed during the flow of steam over moving blades.
- As both the pressure as well as velocity is gradually decreased over the compounded stages it is known as pressure velocity compounding.

4.8 Velocity Diagram for Moving Blades of Impulse Turbine

- To find the force on the blades and power developed by the turbine, it is necessary to determine the rate of change of momentum of steam across the moving blades.
- For this purpose, the velocity diagram at inlet and outlet is drawn for the moving blades.
- Let,

u = Circumferential or tangential velocity of the blade = $\frac{\pi d_m N}{60}$

d_m = Mean diameter of blade drum in m

N = Speed of turbine in rpm

V_1 = Absolute velocity of steam at inlet to moving blades (exit velocity of steam from nozzle or fixed blades)

V_2 = Absolute velocity of steam at exit to moving blade (Inlet velocity of steam to second ring of nozzles or fixed blades)

V_{w1} = Tangential component of V_1 (velocity of whirl at inlet to moving blades)

V_{w2} = Velocity of whirl at outlet of moving blades

V_{r1} and V_{r2} = Relative velocity of steam at inlet and outlet of moving blades respectively

V_{f1} and V_{f2} = Axial component of V_1 and V_2 respectively (flow velocity at inlet and outlet)

α = Exit angle of nozzle or the angle with which steam enters the moving blades

θ = Inlet angle of moving blades

ϕ = Outlet angle of moving blades

β = Angle with which steam at V_2 leaves the moving blades

(Inlet angle of fixed blades)

$\dot{m}$ = Mass flow rate of steam in kg/sec.

K = Blade velocity coefficient or friction factor = $\frac{V_{r2}}{V_{r1}}$

H = Height of blade in m

- Fig. 4.10 shows the velocity diagram of the moving blade at inlet and outlet for an impulse turbine.
- The steam jet from nozzle is impinged on the moving blade at an angle 'α' and the blade starts rotating with velocity 'u'.
- The tangential component of V_1 produces work and the axial component of V_1 is responsible for the flow of steam in axial direction.
- If there is no friction between the steam and the moving blade suface then $V_{r2} = V_{r1}$ $(K = 1)$.
- In this chapter, for convenience the velocity diagram at inlet and outlet of moving blade is combined.
- While drawing the combined velocity diagram, the blade velocity 'u' is taken as common for both the triangles. Refer Fig. 4.11.
- In case of impulse turbine, the relative velocity of steam remains constant or it is reduced slightly due to friction.
- But in case of reaction turbine, the steam expands as it flows over the moving blades. This results in increase in relative velocity of steam at outlet. It means,

For impulse turbine $V_{r2} \le V_{r1}$

For reaction turbine $V_{r2} > V_{r1}$

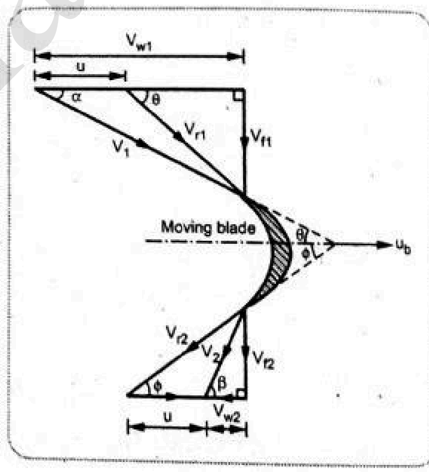


Fig. 4.10 : Inlet and outlet velocity diagram for impluse turbine

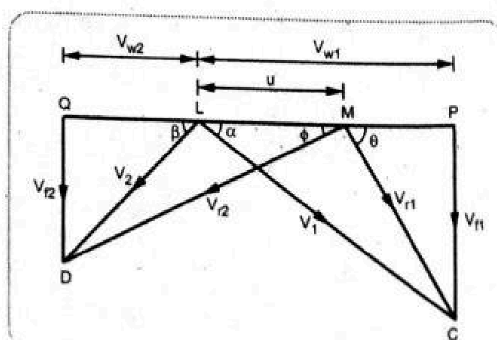


Fig. 4.11 : Combined velocity diagram for impulse turbine

b) In De Laval turbine, steam is issued from the nozzle with a velocity of 1500 m/s whereas the mean blade velocity is 500m/s. The nozzle angle is 20° and the inlet and outlet angles of blades are equal. The mass of steam flowing through the turbine is at the rate of 1200kg/hr. Assuming blade velocity coefficient k=0.8, Draw velocity diagram and determine **[6]**

i) The Blade angles

ii) The power developed by turbine

iii) The Blade Efficiency

Solution : Given :

$V_1 = 1500$ m/s, $u = 500$ m/s, $\alpha = 20°$, $\theta = \phi$,

$m = 1200$ kg/hr $= 0.33$ kg/s, $\dfrac{V_{r2}}{V_{r1}} = 0.8$

To find : i) θ and ϕ ii) P iii) η_b

Step 1 : Calculate blade angles

From velocity triangle,

$$V_{w1} = V_1 \cos\alpha = 1500 \times \cos 20 = 1409.5 \text{ m/s}$$

$$V_{f1} = V_1 \sin\alpha = 1500 \times \sin 20 = 512 \text{ m/s}$$

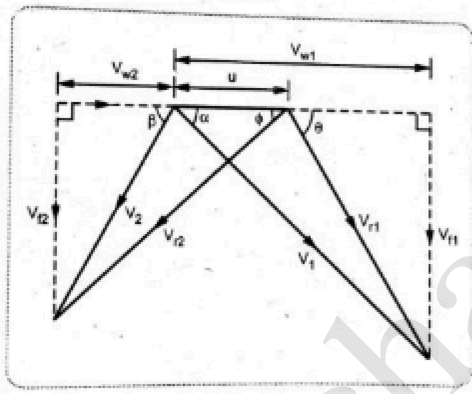


Fig. 4.28

$$\tan\theta = \frac{V_{f1}}{V_{w1} - u_1} = \frac{512}{1409.5 - 500}$$

$$\therefore \quad \theta = \mathbf{29.37°} \text{ and } \theta = \phi = \mathbf{29.37°}$$

Step 2 : Calculate power developed by turbine

From velocity triangle,

$$V_{r1} = \sqrt{V_{f1}^2 + (V_{w1} - u_1)^2} = \sqrt{512^2 + (1409.5 - 500)^2}$$

$$\therefore \quad V_{r1} = 1043.711 \text{ m/s}$$

$$V_{r2} = 0.8\,V_{r1} = 834.96 \text{ m/s}$$

$$\cos\phi = \frac{V_{w2} + u}{V_{r2}} \qquad \therefore \cos 29.4 = \frac{V_{w2} + 500}{834.96}$$

$$\therefore \quad V_{w2} = 227.9 \text{ m/s}$$

Power is given by,

$$P = \frac{\dot{m}\,(V_{w1} + V_{w2}) \times u}{1000} = \frac{0.33\,(1409.5 + 227.9) \times 500}{1000}$$

$$\therefore \quad \mathbf{P = 270.17\ kW}$$

Step 3 : Calculate blade efficiency

We know that,

$$\eta_b = \frac{2(V_{w1} + V_{w2})}{V_1^2} \times u = \frac{2(1409.5 + 227.9) \times 500}{1500^2}$$

$$\therefore \quad \eta_b = \mathbf{0.728 = 72.8\ \%}$$

Q4) a) Explain governing of steam turbine with any one method. **[6]**

- Bleeding of steam for heating the feed water in between the stages.

4.16 Governing of Steam Turbines

SPPU : May 2011, Dec. 2011, Dec. 2013, May 2014, Dec. 2019

- We know that, the purpose of governing is to maintain the speed of turbine fairly constant irrespective of the load.
- The output power of turbine is controlled by varying the steam flow with the help of valves interposed between the boiler and the turbine.
- Depending upon the methods of varying steam flow rate, there are various governing methods used.
- The following are the various governing methods used for the steam turbine :

(i) Throttle governing	(ii) Nozzle governing
(iii) By-pass governing	(iv) Combination of above

4.16.1 Throttle Governing

SPPU : May 2011, Dec. 2013, May. 2014, Dec. 2015, May 2019, Dec. 2019

- Fig. 4.49, shows the simple throttle arrangement. The purpose of throttle governing is to throttle the steam whenever there is reduction in the load as compared to design load before it is supplied to the turbine.
- This helps in maintaining the speed of the turbine.
- To start the turbine for full load running the steam inlet is opened i.e. the throttle valve is opened.
- If the load on the turbine is reduced, the energy supplied to the turbine will be in excess and hence speed of the turbine increases.
- Due to this increased speed, the governor sleeve will lift as well as the pilot piston valve spindle will also get lifted.
- The upper port is then opened to oil pressure and lower port to the oil return.
- The relay piston will thus close the throttle valve partially and amount of steam supplied to the turbine will reduce and hence the speed on turbine comes to normal.
- The lowering of throttle valve will also lower the pilot piston spindle and close the ports to stabilize the relay piston.
- Due to restriction of passage in the valve, the steam is throttled from p_1 to p_2 and specific ideal output of turbine thus reduces from $h_1 - h_2$ to $h_3 - h_4$.

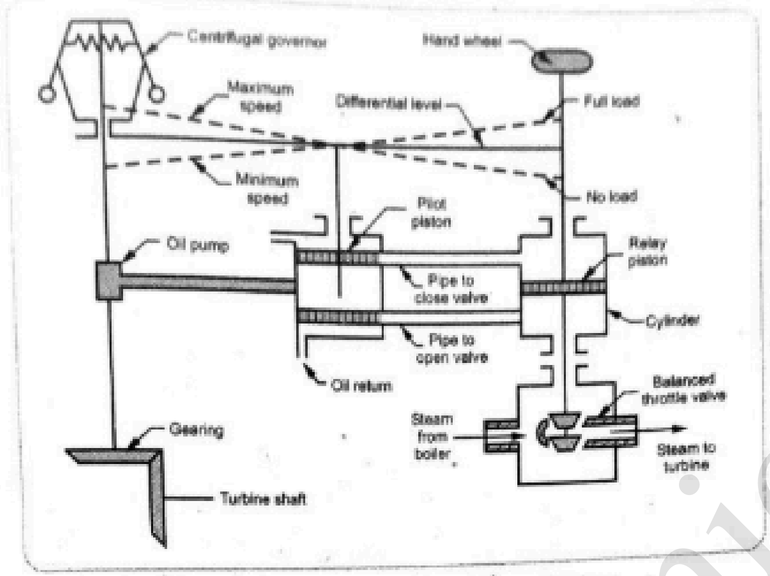


Fig. 4.49 : Throttle governing of steam turbine

- Though the effort of the governor may not be sufficient to move the throttle valve against the piston, the oil operated relay (servo-mechanism) is used in circuit to amplify the small force produced by the governor.

Advantages :

- Simple in operation.
- Less initial cost.
- Lower admission losses.
- Used on small turbines (impulse as well as reaction turbines).

Disadvantages :

- Severe throttling losses.
- Thermodynamically not efficient as available heat drop is low.
- Reduced efficiency of turbine if throttling is carried out at low loads.

SPPU : May 2014, April 2016, Dec. 2017

4.16.2 Nozzle Governing

- Nozzle governing is the more efficient type of governing in a steam turbine.
- In this type of governing, the nozzles are grouped together and each group is controlled by a seperate valve. Refer Fig. 4.50.
- The groups may contain number of nozzles (3 to 5 or more) as shown in Fig. 4.50 by N_1, N_2 and N_3. These groups are controlled by individual valve V_1, V_2 and V_3 respectively.
- Under the full load condition the valves remains fully open.

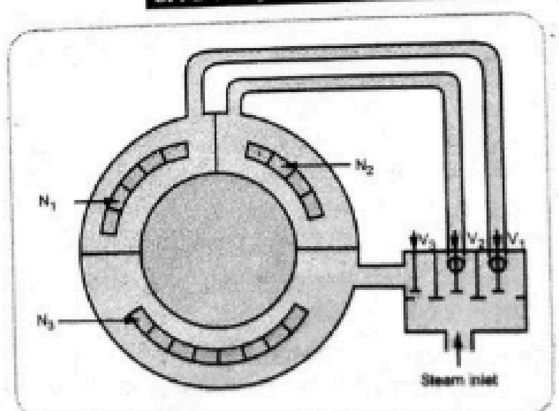


Fig. 4.50

- With the variation in load, the supply from the steam nozzle may be varied accordingly by shutting off or opening the nozzle.
- The nozzle control can only be applied to the first stage of a turbine.
- Also it is suitable for impulse turbine and larger turbines having an impulse stage followed by the impulse-reaction turbine.

Advantages :

- No throttling losses.
- More efficient than throttle governing as available heat drop is high.
- Used for medium and large turbines having initial impulse stage.

Disadvantages :

- High admission losses.
- Only applied to the first stage of a turbine.
- Pressure drop at entry to second stage when some of the nozzles cut off.

4.16.3 Comparison between Throttle Governing and Nozzle Governing

SPPU : Dec. 2017

Sr. No.	Throttle Governing	Nozzle Governing
1.	In this governing admission losses are less.	In this governing admission losses are more.
2.	As available heat drop is low, this governing is not efficient.	As available heat drop is high, this governing is more efficient.
3.	Throttling losses are more.	In this case, no throttling losses.
4.	It is used on small size turbines (impulse and reaction turbines)	It is used on medium and large size turbines having initial impulse stage.

4.16.4 By-pass Governing

- By-pass governing is used when the turbine is throttle governed.
- The steam turbine working under designed or economic load have full admission of steam in the high pressure stages.
- At the maximum load, the turbine would require the additional steam. As this additional steam could not pass through the first stage since the additional nozzles are absent.
- Thus, the by-pass governing is used to admit the additional steam through by-pass valve to the later stages. Refer Fig. 4.51.
- This by-pass valve opens when throttle valve has opened and steam is by-passed through the valve to lower stages in the turbine.

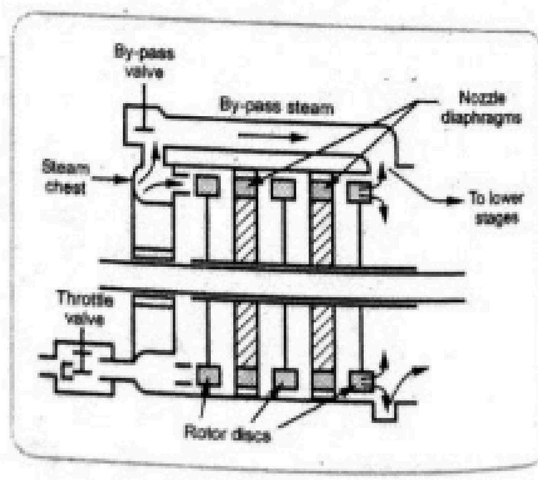


Fig. 4.51 : By-pass governing

- The supply of steam in lower stages increase the work output in these stages but the overall efficiency is reduced.

b) In Parson's reaction turbine running at 500rpm with 50% reaction develops 75 kW per kg per second of steam. The exit angle of blades is 20º and steam velocity is 1.5 times the blade velocity. Determine **[6]**

i) Blade Velocity

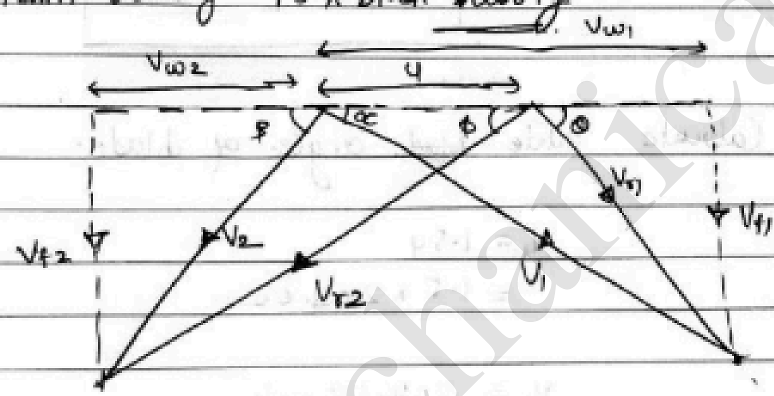
ii) Inlet angle of moving blade.

Q4] b) In Parson reaction Turbine..

N = 500 rpm 50% reaction develops

P = 75 kW blade angle $\alpha = \phi = 20°$

Steam velocity = 1.5 x blade velocity



$$V_{w1} = V_1 \cos\alpha = (1.54)(\cos 20) = 1.4094$$

$$V_{r2} = V_1 = 1.54$$

$$V_{w2} = V_{r2} \cos\phi - u = 1.54 \cos 20 - u$$
$$= 0.40954$$

$$\text{Power } P = \frac{\dot{m} \times (V_{w1} + V_{w2})\,u}{1000}$$

$$75 = \frac{1 \times (1.4094 + 0.40954) \times u}{1000}$$

$$\frac{75\times1000}{1.8189} = u^2$$

$$u = \sqrt{\frac{75\times1000}{1.8189}}$$

$$\boxed{u = 203.06\ m/s}$$

Calculate inlete ~~blade~~ angle of blade.

$V_1 = 1.5u$

$= 1.5\times203.06$

$V_1 = 304.59\ m/s$

From velocity triangle

$V_{f1} = V_1 \sin\alpha = 304.59\times\sin20^\circ = 104.17\ m/s$

$V_{w1} = V_1\cos\alpha = 304.59\times\cos20 = 286.22\ m/s$

$$\tan\theta = \frac{V_{f1}}{V_{w1}-u} = 1.25$$

$$\theta = \tan^{-1}(1.25)$$

$$\boxed{\theta = 51.39^\circ}$$

Q5) a) State & explain: **[6]**

i) Unit Speed

ii) Unit Discharge

iii) Unit Power

i) Unit Speed

Definition

Unit speed is the **speed of a turbine when operating under a unit (1-meter) head.**

Expression

$$N_u = \frac{N}{\sqrt{H}}$$

Explanation

- The speed of a turbine is directly related to the square root of the head.
- To compare or test turbines under different heads, their speeds are converted to an equivalent speed at **1 m head**.
- This helps in performance comparison and model testing.

ii) Unit Discharge

Definition

Unit discharge is the **discharge through a turbine when operating under a head of 1 meter**.

Expression

$$Q_u = \frac{Q}{\sqrt{H}}$$

Explanation

- Discharge also varies with the square root of the head.
- Converting actual discharge to discharge under unit head helps compare hydraulic designs and performance curves for turbines running at different heads.

iii) Unit Power

Definition

Unit power is the **power developed by a turbine under a unit (1-meter) head**.

Expression

$$P_u = \frac{P}{H^{3/2}}$$

Explanation

- Power varies with the **3/2 power of the head** ($H^{3/2}$).
- Converting actual power to unit power allows easy comparison of turbines or turbine models for similar operating conditions.
- Used in characteristic curves and scaling laws.

b) The outer diameter of an impeller of a centrifugal pump is 400mm and outlet width is 50mm. The pump is running at 800 rpm and is working against a total head of 15 m. The vane angle at outlet is 40° and manometric efficiency is 75% Determine **[6]**

i) Velocity of flow at outlet

iii) Velocity of water leaving the vane

iii) Angle made by absolute velocity at outlet

Q4) b)

N = 800 rpm   outer diameter = 400 mm

outlet width = 50mm

Total head = 15m

Vane angle = 40°   manometric efficiency = 75% = 0.75

① Velocity flow at outlet

Tangential velocity of an impeller at outlet

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.4 \times 800}{60}$$

$$u_2 = 16.75 \text{ m/sec}$$

Manometric effi'y of pump

$$\eta_{mo} = \frac{g \cdot H_m}{V_{w2} \; u_2}$$

$$0.75 = \frac{9.81 \times 15}{V_{w2} \times 16.75}$$

$$V_{w2} = 11.71 \text{ m/sec}$$

$$\tan\phi = \frac{V_{f2}}{u_2 - V_{w2}}$$

$$\tan(90) = \frac{V_{f2}}{16.75 - 11.71}$$

$$V_{f2} = \underline{\underline{4.22 \text{ m/sec}}}$$

Velocity of water leaving the vane

$$V_2 = \sqrt{V_{w2}^2 + V_{f2}^2} = \sqrt{(11.71)^2 + (4.22)^2}$$

$$V_2 = \underline{\underline{12.44 \text{ m/sec}}}$$

Angle made by absolute velocity at outlet (β)

$$\tan\beta = \frac{V_{f2}}{V_{w2}} = \frac{4.22}{11.71} = 0.36.$$

$$\beta = \tan^{-1}(0.36)$$

$$\beta = \underline{\underline{19.81^\circ}}$$

Q6) a) Explain various heads in centrifugal pump with a neat sketch. **[6]**

SPPU : May 2012, May 2013

5.5.1 Heads of Centrifugal Pump

- The heads of a centrifugal pump are as follows :

(a) Suction head	(b) Delivery head
(c) Static head	(d) Manometric head

a) Suction Head (h_s) : It is the vertical distance between level of sump and eye of an impeller. Refer Fig. 5.2. It is also called as **suction lift.**

b) Delivery Head (h_d) : It is the vertical distance between the eye of an impeller and the level at which water is delivered.

c) Static Head (H_s) : It is the sum of suction head and delivery head. It is given by,

$$H_s = h_s + h_d \quad \text{... (5.5}$$

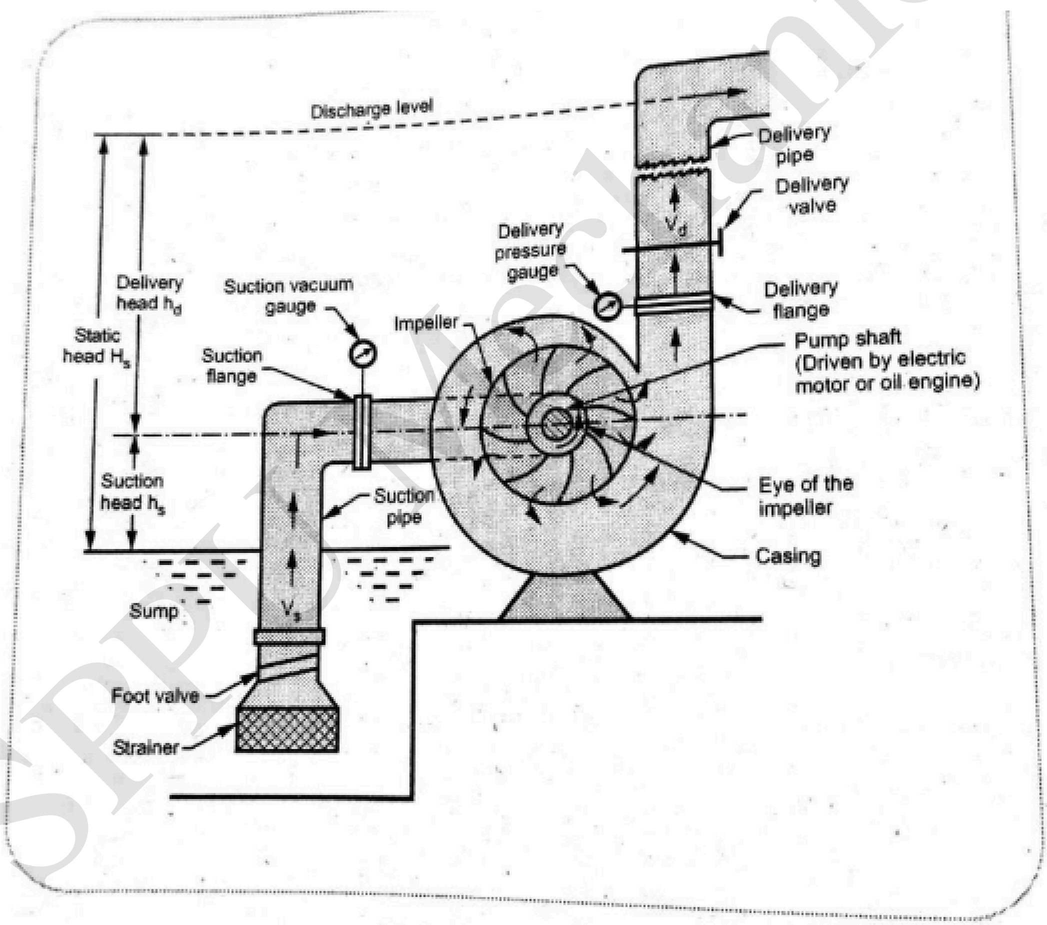


Fig. 5.2 : Components of a centrifugal pump

d) **Manometric Head (H_m)** : The head against which the centrifugal pump has to work is called as manometric head.

- It is given by the following equations :

(i)
$$H_m = \begin{pmatrix}\text{Head imparted by the}\\ \text{impeller to the water}\end{pmatrix} - \begin{pmatrix}\text{Loss of head in the pump}\\ \text{impeller and casing}\end{pmatrix}$$

$$\therefore \quad H_m = \frac{V_{w2}u_2}{g} - (h_{Li} + h_{Lc}) \quad \text{... (5.6)}$$

where, h_{Li} and L_{Lc} = Losses of head in the impeller and casing

$$H_m = \frac{V_{w2}u_2}{g} \quad \text{... if losses are negligible ... (5.7)}$$

(ii)
$$H_m = \text{Static head + Losses in pipes + Kinetic head at delivery}$$

$$\therefore \quad H_m = H_s + (h_{fs} + h_{fd}) + \frac{V_d^2}{2g}$$

$$= (h_s + h_d) + (h_{fs} + h_{fd}) + \frac{V_d^2}{2g} \quad \text{... (5.8)}$$

where,

h_s and h_d = Suction and delivery head

h_{fs} and h_{fd} = Loss of head due to friction in suction and delivery.

V_d = Velocity of water in delivery pipe.

(iii)
$$H_m = \begin{pmatrix}\text{Total head at outlet}\\ \text{of pump}\end{pmatrix} - \begin{pmatrix}\text{Total head at inlet}\\ \text{of pump}\end{pmatrix} \quad \text{... (i)}$$

$$\text{Total head at outlet} = \frac{p_d}{\gamma} + \frac{V_d^2}{2g} + Z_d = h_d + \frac{V_d^2}{2g} + Z_d$$

$$\text{Total head at inlet} = \frac{p_s}{\gamma} + \frac{V_s^2}{2g} + Z_s = h_s + \frac{V_s^2}{2g} + Z_s$$

where,

$\frac{p_d}{\gamma}$ and $\frac{p_s}{\gamma}$ = Pressure head at outlet and inlet of pump

$\frac{V_d^2}{2g}$ and $\frac{V_s^2}{2g}$ = Velocity head at outlet and inlet of pump

Z_d and Z_s = Vertical height of outlet and inlet of pump from datum line

From equation (i),

$$H_m = h_d + \frac{V_d^2}{2g} + Z_d - h_s + \frac{V_s^2}{2g} + Z_s \quad \text{... (5.9)}$$

b) A centrifugal pump delivers 1565 LPS against a manometric head of 6.1m. When the impeller rotates at 200 rpm. The impeller diameter is 1.22 m and area at outer periphery is 6450 cm^2. If the vanes are set back at angle of 26° at the outlet. Determine **[6]**

i) Manometric Efficiency

ii) Power required to drive the pump

iii) Minimum starting speed if ratio of external to internal is 2.

Solution : Given data :

$Q = 1565$ lps $= 1.565\ m^3/sec$, $H_m = 6.1$ m, $N = 200$ rpm, $D_2 = 1.22$ m,

$A_{f2} = 6450\,cm^2 = 0.6450\ m^2$, $\phi = 26°$, $\frac{D_2}{D_1} = 2$, $\therefore\ D_1 = \frac{D_2}{2} = \frac{1.22}{2} = 0.61$ m

To find : i) η_{ma} ii) P iii) N_{min}

Step 1 : Calculate the manometric efficiency

The tangential or peripheral velocity at the outlet is,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 1.22 \times 200}{60} = 12.7758 \text{ m/sec.}$$

Velocity of flow Q, at outlet is

$$V_{f2} = \frac{Q}{A_{f2}} = \frac{1.565}{0.6450} = 2.4263 \text{ m/sec.}$$

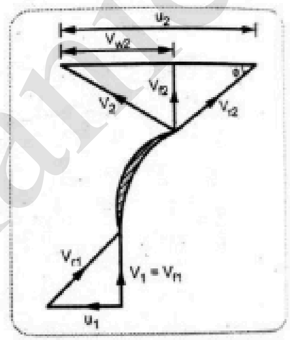


Fig. 5.22

From outlet velocity triangle,

$$\tan\phi = \frac{V_{f2}}{u_2 - V_{w2}} \qquad \therefore \tan 26 = \frac{2.4263}{12.7758 - V_{w2}}$$

$$V_{w2} = 7.8010 \text{ m/sec.}$$

Now the manometric efficiency is given by,

$$\eta_{ma} = \frac{g H_m}{V_{w2} u_2} = \frac{9.81 \times 6.1}{7.8010 \times 12.7758} = \mathbf{0.6004 = 60.04\ \%} \quad \text{... Ans}$$

Step 2 : Calculate the power required to drive the pump

Neglecting mechanical losses ($\eta_m = 100\%$), the mechanical efficiency is given by,

$$\eta_m = \frac{\text{Impeller power}}{\text{Shaft power}} = 1$$

$$\therefore \quad \text{Shaft power} = \text{Impeller power} = \rho Q\left(\frac{V_{w2} u_2}{g}\right) \times g$$

$$= 1000 \times 1.565 \times \left(\frac{7.8010 \times 12.7758}{9.81}\right) \times 9.81$$

$$\mathbf{P = 155.9741 \times 10^3 W = 155.9741\ kW}$$

Step 3 : Calculate the minimum starting speed required for pump

The minimum starting speed for pump is given by,

$$N_{min} = \frac{120 \times \eta_{ma} \times V_{w2} \times D_2}{\pi (D_2^2 - D_1^2)} = \frac{120 \times 0.6004 \times 7.8010 \times 1.22}{\pi (1.22^2 - 0.61^2)}$$

$$N_{min} = \mathbf{195.5245\ rpm}$$

Q7) a) Explain Construction and working of axial flow compressor with a neat sketch. **[4]**

Construction

- The axial flow compressor is often described as a *reversed reaction turbine*. The energy transfer in the compressor is the reverse of that in the turbine.
- *In this compressor the air flows parallel to the axis of compressor*. It consists of adjacent rows of moving blades (rotor) and fixed blades (stator).
- The moving blades are mounted on the rotating drum and the fixed blades are attached to the casing.
- A row of moving blades followed by a row of fixed blades is called as **one stage**.
- The main function of the fixed blades is to receive high velocity air from the preceding rotor blades and to direct the flow to the succeeding rotor blades.
- Fig. 7.1 shows the axial flow compressor with drum type rotor. The compressor blades have aerofoil section whereas the turbine blades have profile section.

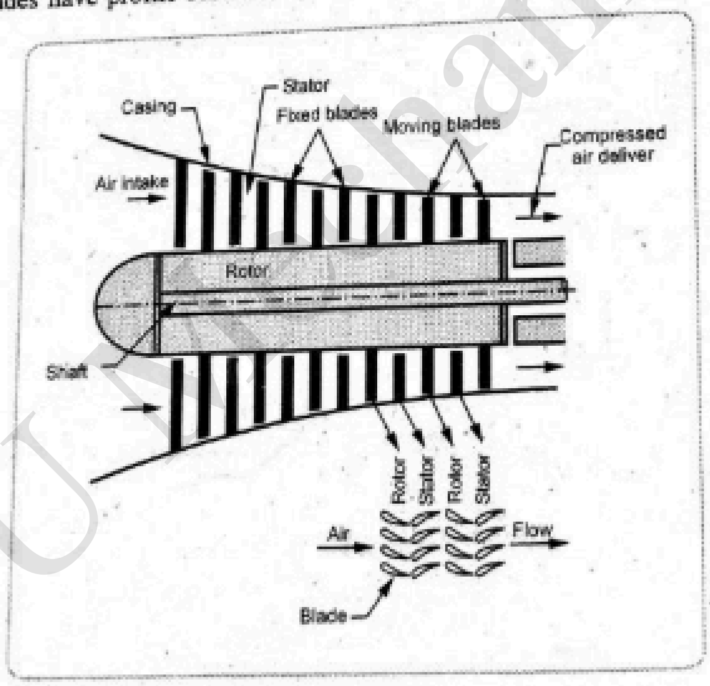


Fig. 7.1 : Drum type axial flow compressor

- At the inlet of compressor an extra row of fixed vanes is provided which is called as **Inlet Guide Vane (IGV)**.

- The acceleration process carried in the converging blade passage of reaction turbine is more efficient and stable as compared to diffusing (decelerating) process carried in the diverging passage of blades of axial compressor.
- The annular area is reduced from inlet to outlet of the compressor which results in *constant flow velocity throughout the compressor length.*
- Due to diffusion there is rise in temperature in the diverging passage of the moving blades. Also, the absolute velocity is increased due to work input.

Working

- The basic working principle of axial flow compressor is similar to the centrifugal compressor.
- The input energy of the rotor shaft is transferred to the air by moving blades and thus accelerating the air.
- The blades are arranged in such a way that the spacing between the blades form a diffuser. Hence, the velocity of air relative to the blades is decreased ($V_{r2} < V_{r1}$) and there is a rise in pressure.
- After moving blades, the air is further diffused in stator blades which also form diffuser passage.
- While passing through the stator blades, the air is turned through an angle so that it can easily pass to a second row of moving blades.
- The pressure rise per stage in axial compressors is 1.1 to 1.3 hence a number of stages (generally 5 to 14 stages) are used for pressure ratio upto 12.
- The multistage axial compressor can supply air upto 30,000 m^3 / min.

b) A rotary air compressor working between 1 bar and 2.5 bar has internal and external diameter of impeller as 300 mm and 600 mm respectively. The vane angle at inlet and outlet are 30° and 45° respectively. If air enters impeller at 15 m/s, Find **[8]**

i) Speed of impeller in rpm

ii) Work done by compressor per kg of air.

Example 6.9 *A rotary air compressor working between 1 bar and 2.5 bar has internal and external diameters of impeller as 300 mm and 600 mm respectively. The vane angle at inlet and outlet are 30° and 45° respectively. If air enters impeller at 15 m/s, find i) Speed of impeller in r.p.m. ii) Work done by compressor per kg of air.* **SPPU : Dec. 2009, 10 Marks**

Solution : Given data :

$p_1 = 1$ bar, $p_2 = 2.5$ bar, $D_1 = 300$ mm $= 0.3$ m, $D_2 = 600$ mm $= 0.6$ m, $\theta = 30°$, $\phi = 45°$, $V_{f1} = V_{f2} = 15$ m/sec.

To find : i) N ii) WD_{act}

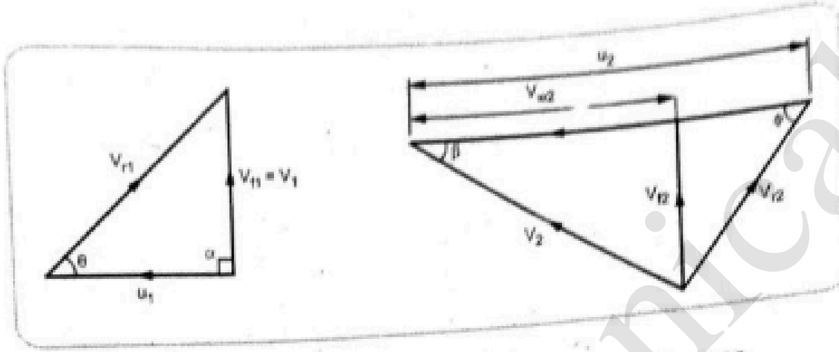


Fig. 6.24

Fig. 6.25

Step - 1 : Calculate the speed of impeller

From Fig. 6.24 , by inlet velocity triangle,

$$\tan\theta = \frac{V_{f1}}{u_1} \quad \therefore \tan 30 = \frac{15}{u_1} \quad \therefore u_1 = 25.9807 \text{ m/sec}$$

Blade velocity at inlet,

$$u_1 = \frac{\pi D_1 N}{60} \quad \therefore 25.9807 = \frac{\pi \times 0.3 \times N}{60}$$

$\therefore$

$$N = \mathbf{1653.9827\ rpm}$$

Step - 2 : Calculate the work done by compressor per kg of air

Outlet blade velocity,

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.6 \times 1653.9827}{60} = 51.9613 \text{ m/sec}$$

From Fig. 6.25, by outlet velocity triangle,

$$V_{w2} = u_2 - \frac{V_{f2}}{\tan\phi} = 51.9613 - \frac{15}{\tan 45} = 36.9613 \text{ m/sec}$$

$\therefore$ Work done by compressor per kg of air,

$$WD_{act} = V_{w2} u_2 = 36.9613 \times 51.9613$$

$$WD_{act} = \mathbf{1920.5571\ J/kg}$$

Q8) a) Differentiate between centrifugal compressor and axial flow compressor. **[4]**

Sr. No.	Centrifugal compressor	Axial flow compressor
1.	The flow of fluid proceeds radially.	The flow of fluid proceeds axially parallel to the axis of compressor.
2.	High pressure ratio per stage (about 4)	Low pressure ratio per stage (about 1.5)
3.	These compressors are easy to manufacture.	These compressors are difficult to manufacture.
4.	Isentropic efficiency of centrifugal compressor is less than axial flow compressor (about 70 %).	Isentropic efficiency of axial flow compressor is about 85 to 88 %.
5.	Power required per kg of air flow is more.	Power required per kg of air flow is less.
6.	Flexible in operation due to adjustable pre-whirl and diffuser vanes.	Not flexible in operation as there is no such arrangement.
7.	It needs large frontal area for given mass flow rates.	It needs less frontal area for given mass flow rates.
8.	Deposite formation on the surface of an impeller does not affect the performance of compressor.	Deposite formation on the surface of an impeller affect the performance of compressor.
9.	It has low starting torque.	It has high starting torque.
10.	It is not suitable for multistaging due to large losses	It is suitable for multistaging .
11.	Delivery pressure is upto 400 bar.	Delivery pressure is upto 20 bar.
12.	The efficiency V/s speed curve is more flat.	The efficiency V/s speed curve is less flat.
13.	**Applications :** Super charging of IC engines, refrigerants and industrial gases, Blowing engine in steel mills, used in air conditoning, fertiliser industry, gas pumping in long distance pipe lines, etc.	**Applications :** In jet engines, gas turbine power plant, steel mills, etc.

b) The impeller of the centrifugal compressor has the inlet and outlet diameter of 0.3 and 0.6 m respectively. The intake is from the atmosphere at 100 kPa and 300 K, without any whirl component. The outlet blade angle is 75°. The speed is 10000 rpm and velocity of flow is constant at 120 m/s. If the blade width at inlet is 6 cm, determine the following **[8]**

i) Specific work

ii) Exit pressure

iii) Mass flow rate

iv) Power required to compressor if the overall efficiency is assumed to be 0.7.

Solution : $D_1 = 0.3$ m, $D_2 = 0.6$ m, $p_1 = 100$ kPa, $T_1 = 300$ K, $V_{w1} = 0$, $\phi = 75°$, N = 10,000 rpm, $V_{f1} = V_{f2} = 120$ m/s,

$B_1 = 0.06$ m, $\eta_0 = 0.7$

To find : i) Specific work ii) Exit pressure iii) $\dot{m}$ iv) power

Step - 1 : Calculate specific work

Refer Fig. 6.34.

We know that,

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.3 \times 10{,}000}{60} = 157.79 \text{ m/s}$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.6 \times 10{,}000}{60} = 314.16 \text{ m/s}$$

From velocity triangle, (Refer Fig. 6.34)

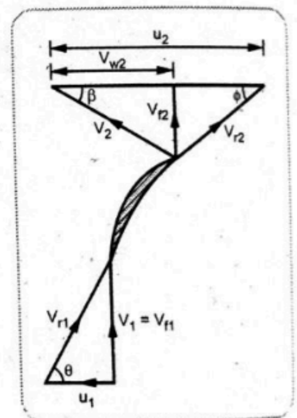


Fig. 6.34

$$V_{w2} = u_2 - \frac{V_{f2}}{\tan\phi} = 314.16 - \frac{120}{\tan 75}$$

$$V_{w2} = 282 \text{ m/s}$$

Specific work is given by,

$$\text{Specific work} = V_{w2}u_2 = 282 \times 314.16 = 88.593 \times 10^3 \text{ J/kg} \quad \text{... Ans.}$$

Step - 2 : Calculate exit pressure

We know that,

$$W = C_p(\Delta T)_{th} = C_p(T_2' - T_1)$$

$$\therefore \quad 88.593 \times 10^3 = 1.005 \times 10^3 (T_2' - 300)$$

$$\therefore \quad T_2' = 388.153 \text{ K}$$

But,

$$\frac{p_2'}{p_1} = \left(\frac{T_2'}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{388.153}{300}\right)^{\frac{1.4}{1.4-1}}$$

$$\therefore \quad \frac{p_2'}{p_1} = 2.463 \quad \therefore p_2 = 2.463\, p_1$$

$$\therefore \quad p_2 = 2.463 \times 100 = \mathbf{2.463 \text{ kPa}} \quad \text{... Ans.}$$

Step - 3 : Calculate mass flow rate

Volume flow rate is given by,

$$Q = \pi D_1 B_1 V_{f1} = \pi \times 0.3 \times 0.06 \times 120 = 6.786 \text{ m}^3/\text{s}$$

We know that,

$$\dot{m} = \rho Q = 1.16 \times 6.786 \quad \text{... (Assume } \rho = 1.16 \text{ kg/m}^3\text{)}$$

$$\therefore \quad \dot{m} = \mathbf{7.871 \text{ kg/s}} \quad \text{...Ans.}$$

Step - 4 : Calculate power required to drive the compressor

Theoretical power is given by,

$$P_{th} = \text{Specific work} \times \dot{m} = 88.593 \times 10^3 \times 7.871$$

$$= 697.315 \text{ kW}$$

Overall efficiency is given by

$$\eta_0 = \frac{\text{Theoretical Power}}{\text{Actual Power}} \quad \therefore \quad 0.7 = \frac{697.315}{\text{Actual Power}}$$

$$\therefore \quad \textbf{Actual Power} = \mathbf{996.164 \text{ kW}} \quad \text{... Ans.}$$