# Design Aspects
# Turbo-machines Laboratory
## Turbo-machines

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# Centrifugal Impeller

There are three main categories of impeller type of impeller's vane, which are used in the centrifugal pumps as:

Radial Vane

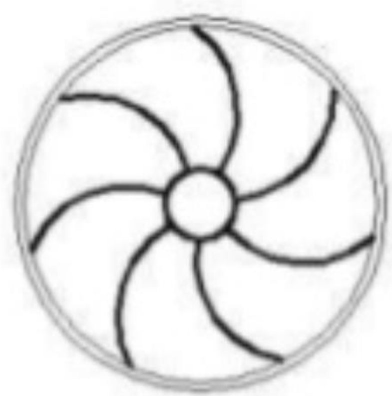
Backward Vanes

Forward Vanes

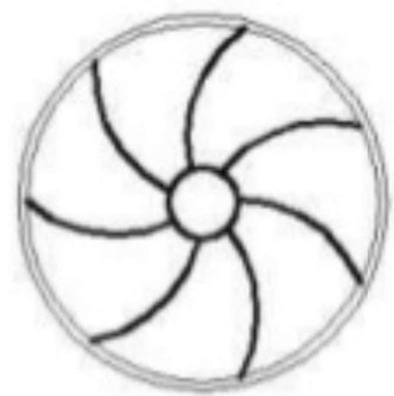
# Centrifugal Impeller

There are three main categories of impeller type of impeller's vane, which are used in the centrifugal pumps as:

Radial Vane



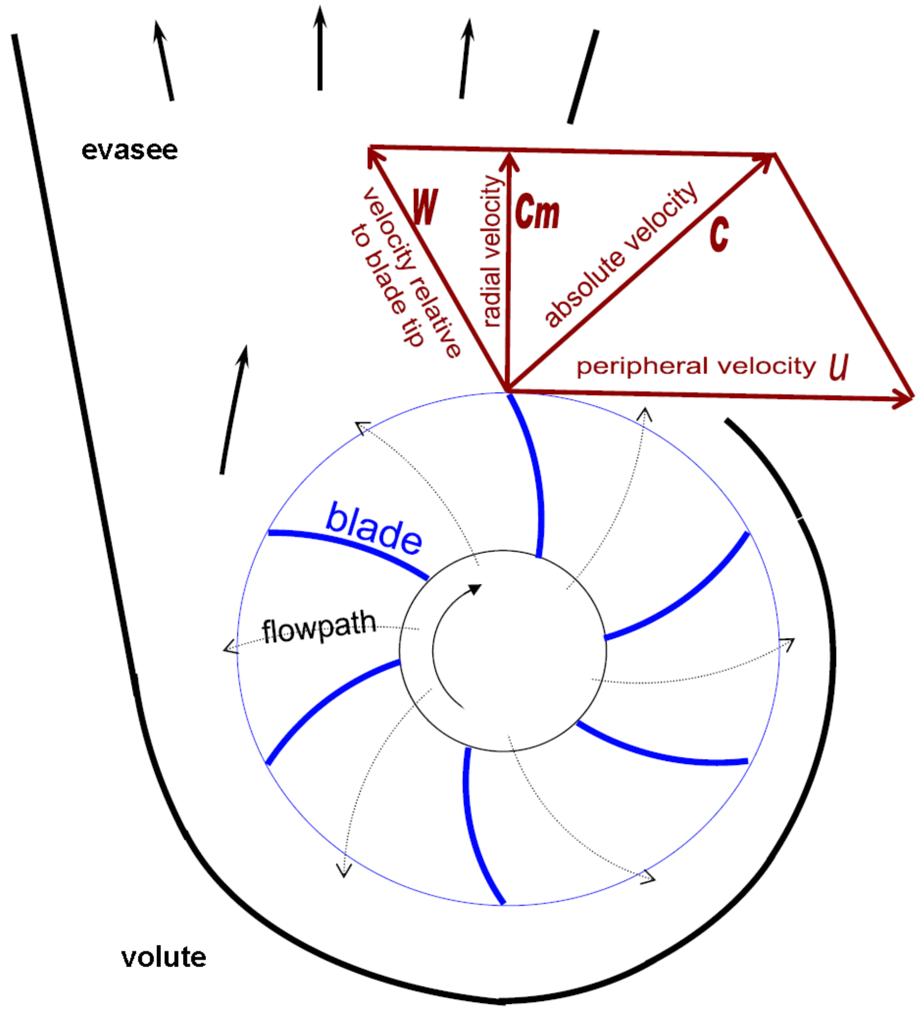
Backward Vanes



Forward Vanes

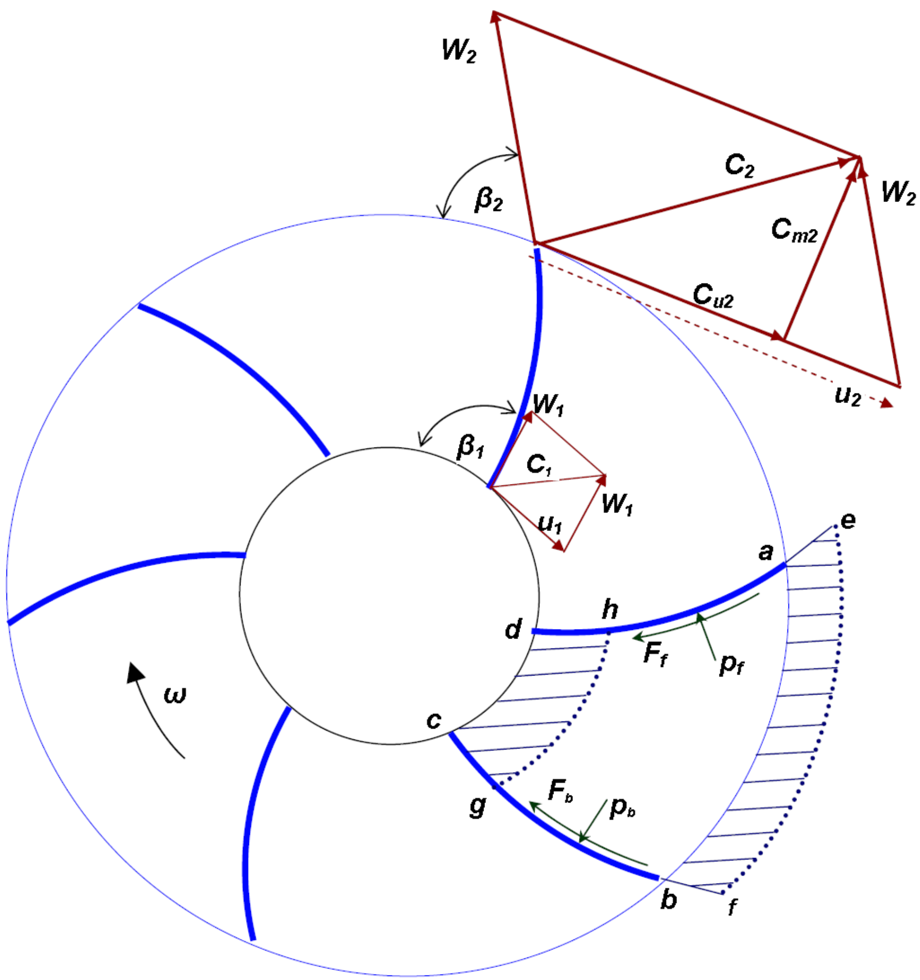
Design of Impeller

# Backward Bladed Centrifugal Impeller




Velocity Triangle of backward bladed centrifugal impeller

# Backward Bladed Centrifugal Impeller




Subscript 1: Inlet

- $\omega$ : Angular velocity (radians/s)
- $u$ : Peripheral speed of blade tip (m/s)
- $C_m$ : Radial component of fluid velocity (m/s)
- $\beta$ : Vane angle
- $p_f$ : Pressure on front of vane (Pa)
- $F_f$ : Shear resistance on front of vane (N/m$^2$)

Subscript 2: Outlet

- $C$ : Absolute fluid velocity (m/s)
- $W$ : Fluid velocity relative to vane )m/s)
- $C_u$ : Peripheral component of fluid velocity (m/s)
- $p_b$ : Pressure on back of vane (Pa)
- $F_b$ : Shear resistance on back of vane (N/m$^2$)

Velocity Triangle at Inlet and Outlet

# Backward Bladed Centrifugal Impeller

If a mass, *m*, rotates about an axis at a radius, *r*, and at a tangential velocity, *v*, then it has an angular momentum of *mrv*. Furthermore, if the mass is a fluid that is continuously being replaced then it becomes a mass flow, d*m*/d*t*, and a torque, *T*, must be maintained that is equal to the corresponding continuous rate of change of momentum

$$T = \frac{dm}{dt}(rv) \qquad \text{Nm or J} \qquad (10.4)$$

In the case of the centrifugal impeller depicted in Figure 10.3, the peripheral component of fluid velocity is $C_u$. Hence the torque becomes

$$T = \frac{dm}{dt}(r\,C_u) \qquad \text{Nm or J} \qquad (10.5)$$

# Backward Bladed Centrifugal Impeller

Consider the mass of fluid filling the space between two vanes and represented as *abcd* on Figure 10.3. At a moment, d*t*, later it has moved to position efgh. The element abfe leaving the impeller has mass d*m* and is equal to the mass of the element cdhg entering the impeller during the same time. The volume represented by abgh has effectively remained in the same position and has not, therefore, changed its angular momentum. The increase in angular momentum is that due to the elements abfe and cdhg. Then, from equation (10.5) applied across the inlet and outlet locations,

$$T = \frac{dm}{dt}\left[r_2 C_{u2} - r_1 C_{u1}\right] \qquad \text{J} \qquad (10.6)$$

$$\frac{dm}{dt} = Q\rho \qquad \frac{\text{kg}}{\text{s}}$$

where $Q$ = volume flow ($m^3/s$)
and $\rho$ = fluid density ($kg/m^3$)

# Backward Bladed Centrifugal Impeller

Design of Impeller

Giving

$$T = Q\rho[r_2 C_{u2} - r_1 C_{u1}] \qquad \text{J} \qquad (10.7)$$

Now the power consumed by the impeller, $P_{ow}$ is equal to the rate of doing mechanical work,

$$P_{ow} = T\omega \qquad \text{W} \qquad (10.8)$$

where $\omega$ = speed of rotation (radians/s)

giving $P_{ow} = Q\rho\omega\,[\,r_2\,C_{u2} - r_1\,C_{u1}\,]$ W (10.9)

But $\omega\,r_2 = u_2$ = tangential velocity at outlet
and $\omega\,r_1 = u_1$ = tangential velocity at inlet.

Hence $P_{ow} = Q\rho[u_2\,C_{u2} - u_1\,C_{u1}]$ W (10.10)

# Backward Bladed Centrifugal Impeller

The power imparted by a fan impeller to the air was given by equation (5.56) as $p_{ft}\, Q$

where $p_{ft}$ = rise in total pressure across the fan.

In the absence of frictional or shock losses, $p_{ft}\, Q$ must equal the power consumed by the impeller, $P_{ow}$. Hence

$$p_{ft} = \rho(u_2 C_{u2} - u_1 C_{u1}) \quad \text{Pa} \qquad (10.11)$$

This relationship gives the theoretical fan total pressure and is known as Euler's equation.

The inlet flow is often assumed to be radial for an ideal centrifugal impeller, i.e. $C_{u1} = 0$, giving

$$p_{ft} = \rho\, u_2 C_{u2} \quad \text{Pa} \qquad (10.12)$$

# Backward Bladed Centrifugal Impeller

Euler's equation can be re-expressed in a manner that is more amenable to physical interpretation. From the outlet vector diagram

$$W_2^{\,2} = C_{m2}^{\,2} + (u_2 - C_{u2})^2$$

$$= C_{m2}^{\,2} + u_2^{\,2} - 2u_2\, C_{u2} + C_{u2}^{\,2}$$

or

$$2u_2 C_{u2} = u_2^{\,2} - W_2^{\,2} + (C_{m2}^{\,2} + C_{u2}^{\,2})$$

$$= u_2^{\,2} - W_2^{\,2} + C_2^{\,2} \qquad \text{(Pythagorus)}$$

Similarly for the inlet,

$$2u_1 C_{u1} = u_1^{\,2} - W_1^{\,2} + C_1^{\,2}$$

Design of Impeller

# Backward Bladed Centrifugal Impeller

Euler's equation (10.11) then becomes

$$p_{ft} = \rho\left\{\frac{u_2^{\,2} - u_1^{\,2}}{2} - \frac{W_2^{\,2} - W_1^{\,2}}{2} + \frac{C_2^{\,2} - C_1^{\,2}}{2}\right\} \text{ Pa} \qquad (10.13)$$

centrifugal effect | effect of relative velocity | change in kinetic energy

Gain in static pressure + Gain in velocity pressure

10.3.1.2. Theoretical characteristic curves for a centrifugal impeller

Euler's equation may be employed to develop pressure-volume relationships for a centrifugal impeller. Again, we must first eliminate the $C_u$ term. From the outlet vector diagram on Figure 10.3,

$$\tan\beta_2 = \frac{C_{m2}}{u_2 - C_{u2}} \qquad \text{giving} \qquad C_{u2} = u_2 - \frac{C_{m2}}{\tan\beta_2}$$

# Backward Bladed Centrifugal Impeller

For radial inlet conditions, Euler's equation (10.12) then gives

$$p_{ft} = \rho u_2 C_{u2} = \rho u_2 \left\{ u_2 - \frac{C_{m2}}{\tan \beta_2} \right\} \quad \text{Pa} \qquad (10.14)$$

But $$C_{m2} = \frac{Q}{a_2} = \frac{\text{volume flowrate}}{\text{flow area at impeller outlet}}$$

Equation (10.14) becomes

$$p_{ft} = \rho {u_2}^2 - \frac{\rho u_2}{\tan \beta_2} \frac{Q}{a_2} \quad \text{Pa} \qquad (10.15)$$

# Backward Bladed Centrifugal Impeller

For a given impeller rotating at a fixed speed and passing a fluid of known density, $\rho$, $u$, $a$ and $\beta$ are all constant, giving

$$p_{ft} = A - BQ \qquad \text{Pa} \qquad (10.16)$$

where constants $A = \rho u_2^{\,2}$

and $B = \dfrac{\rho u_2}{a_2 \tan \beta_2}$

The flowrate, $Q$, and, hence, the pressure developed vary with the resistance against which the fan acts. Equation (10.16) shows that if frictional and shock losses are ignored, then fan pressure varies linearly with respect to the airflow.

We may apply this relationship to the three types of centrifugal impeller:

# Backward Bladed Centrifugal Impeller

*Radial bladed*

$\beta_2 = 90°$ and $\tan\beta_2 =$ infinity

giving B = 0

Then

$$p_{ft} = \text{constant A} = \rho\, u_2^{\,2}$$

i.e. theoretically, the pressure remains constant at all flows [Figure 10.4 (a)]

*Backward bladed*

$\beta_2 < 90°$ , $\tan\beta_2 > 0$

$$p_{ft} = A - BQ$$

i.e. theoretically, the pressure falls with increasing flow.

*Forward bladed*

$\beta_2 > 90°$, $\tan\beta_2 < 0$

$$p_{ft} = A + BQ$$

i.e. theoretically, pressure rises linearly with increasing flow.

The latter and rather surprising result occurs because the absolute velocity, $C_2$ is greater than the impeller peripheral velocity, $u_2$, in a forward bladed impeller. This gives an impulse to the fluid which increases with greater flowrates. (In an actual impeller, friction and shock losses more than counteract the effect.).

Design of Impeller

# Backward Bladed Centrifugal Impeller

The theoretical pressure-volume characteristic curves are shown on Figure 10.4 (a).

The theoretical relationship between impeller power and airflow may also be gained from equation (10.16).

$$P_{ow} = p_{ft}\, Q \quad = \quad AQ - BQ^2 \qquad \text{W} \qquad (10.17)$$

The three power-volume relationships then become:

*Radial bladed*
$B = 0$ and $P_{ow} = AQ$ (linear)

*Backward bladed*
$B > 0$ and $P_{ow} = AQ - BQ^2$ (falling parabola)

*Forward bladed*
$B < 0$ and $P_{ow} = AQ + BQ^2$ (rising parabola).

The theoretical power-volume curves are shown on Figure 10.4 (b). Forward bladed fans are capable of delivering high flowrates at fairly low running speeds. However, their high power demand leads to reduced efficiencies. Conversely, the relatively low power requirement and high efficiencies of backward bladed impellers make these the preferred type for large centrifugal fans.

# Backward Bladed Centrifugal Impeller

10.3.1.3. Actual characteristic curves for a centrifugal impeller

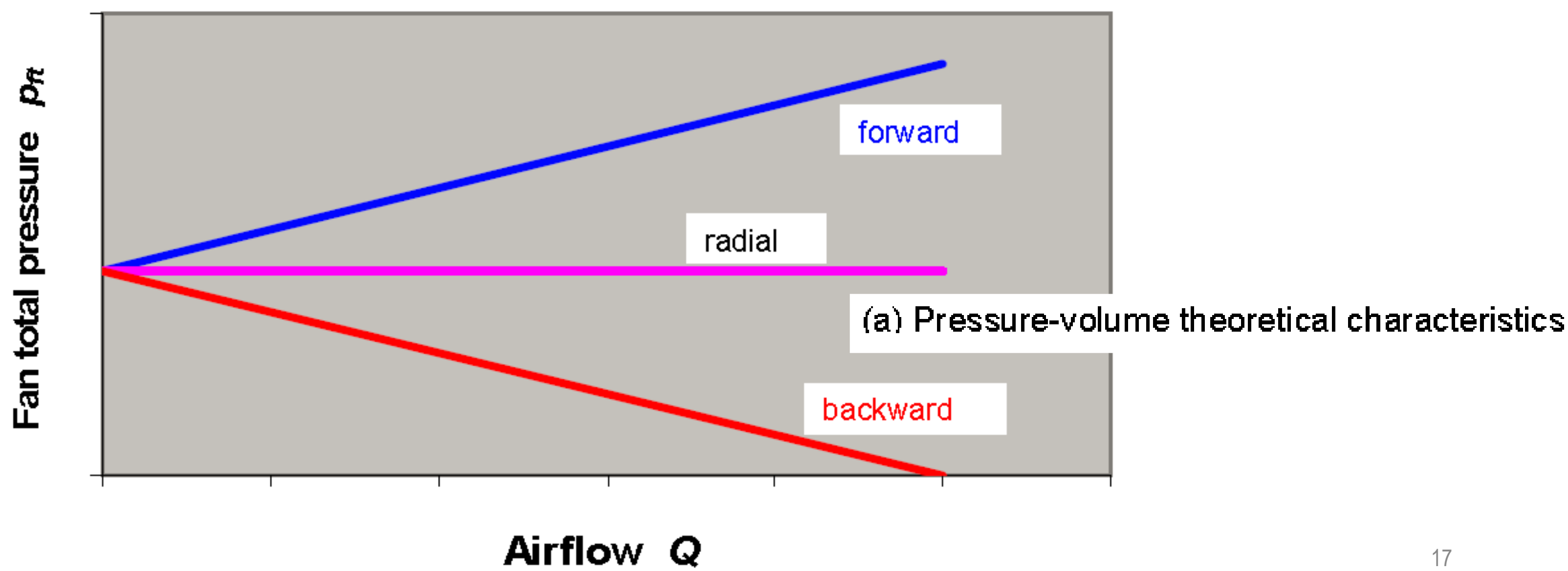
The theoretical treatment of the preceding subsections lead to linear pressure-volume relationships for radial, backward and forward bladed centrifugal impellers. In an actual fan, there are, inevitably, losses which result in the real pressure-volume curves lying below their theoretical counterparts. In all cases, friction and shock losses produce pressure-volume curves that tend toward zero pressure when the fan runs on open circuit, that is, with no external resistance.

Figure 10.5 shows a typical pressure-volume characteristic curve for a backward bladed centrifugal fan. Frictional losses occur due to the viscous drag of the fluid on the faces of the vanes. These are denoted as $F_f$ and $F_b$ on Figure 10.3. A diffuser effect occurs in the diverging area available for flow as the fluid moves through the impeller. This results in a further loss of available energy. In order to transmit mechanical work, the pressure on the front face of a vane, $p_f$ is necessarily greater than that on the back, $p_b$. A result of this is that the fluid velocity close to the trailing face is higher than that near the front face. These effects result in an asymmetric distribution of fluid velocity between two successive vanes at any given radius and produce an eddy loss. It may also be noted that at the outlet tip, the two pressures $p_f$ and $p_b$ must become equal. Hence, although the tip is most important in its influence on the outlet vector diagram, it does not actually contribute to the transfer of mechanical energy. The transmission of power is not uniform along the length of the blade.

# Backward Bladed Centrifugal Impeller

The shock (or separation) losses occur particularly at inlet and reflect the sudden turn of near 90° as the fluid enters the eye of the impeller. In practice, wall effects impart a vortex to the fluid as it approaches the inlet. By a suitable choice of inlet blade angle, $\beta_1$, (Figure 10.3) the shock losses may be small at the optimum design flow. An inlet cone at the eye of the impeller or fixed inlet and outlet guide vanes can be fitted to reduce shock losses.

In the development of the theoretical pressure and power characteristics, we assumed radial inlet conditions. When the fluid has some degree of pre-rotation, the flow is no longer radial at the inlet to the impeller. The second term in Euler's equation (10.11) takes a finite value and, again, results in a reduced fan pressure at any given speed of rotation.



# Backward Bladed Centrifugal Impeller

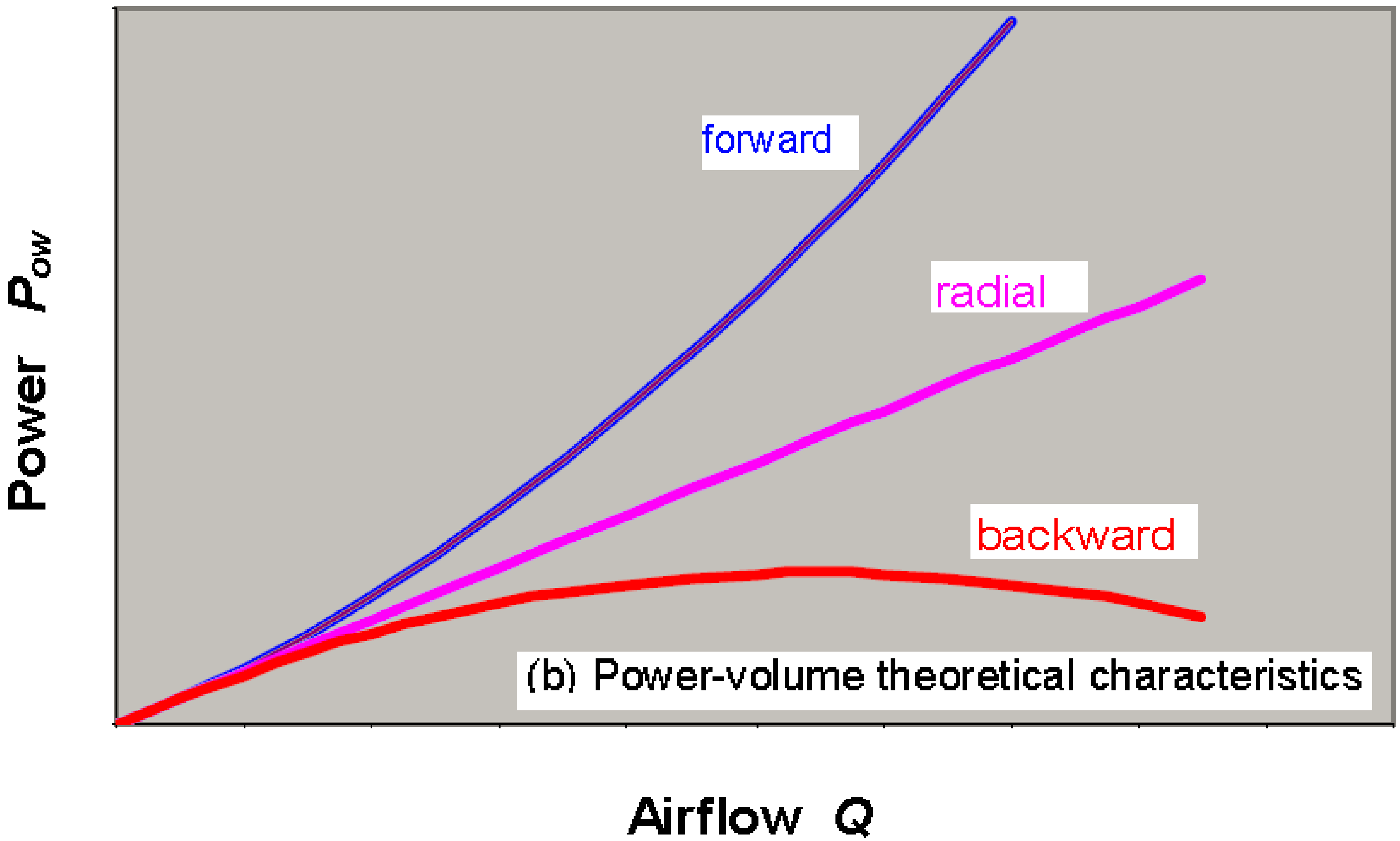



Figure 10.4 Theoretical characteristic curves for a centrifugal fan

# Backward Bladed Centrifugal Impeller

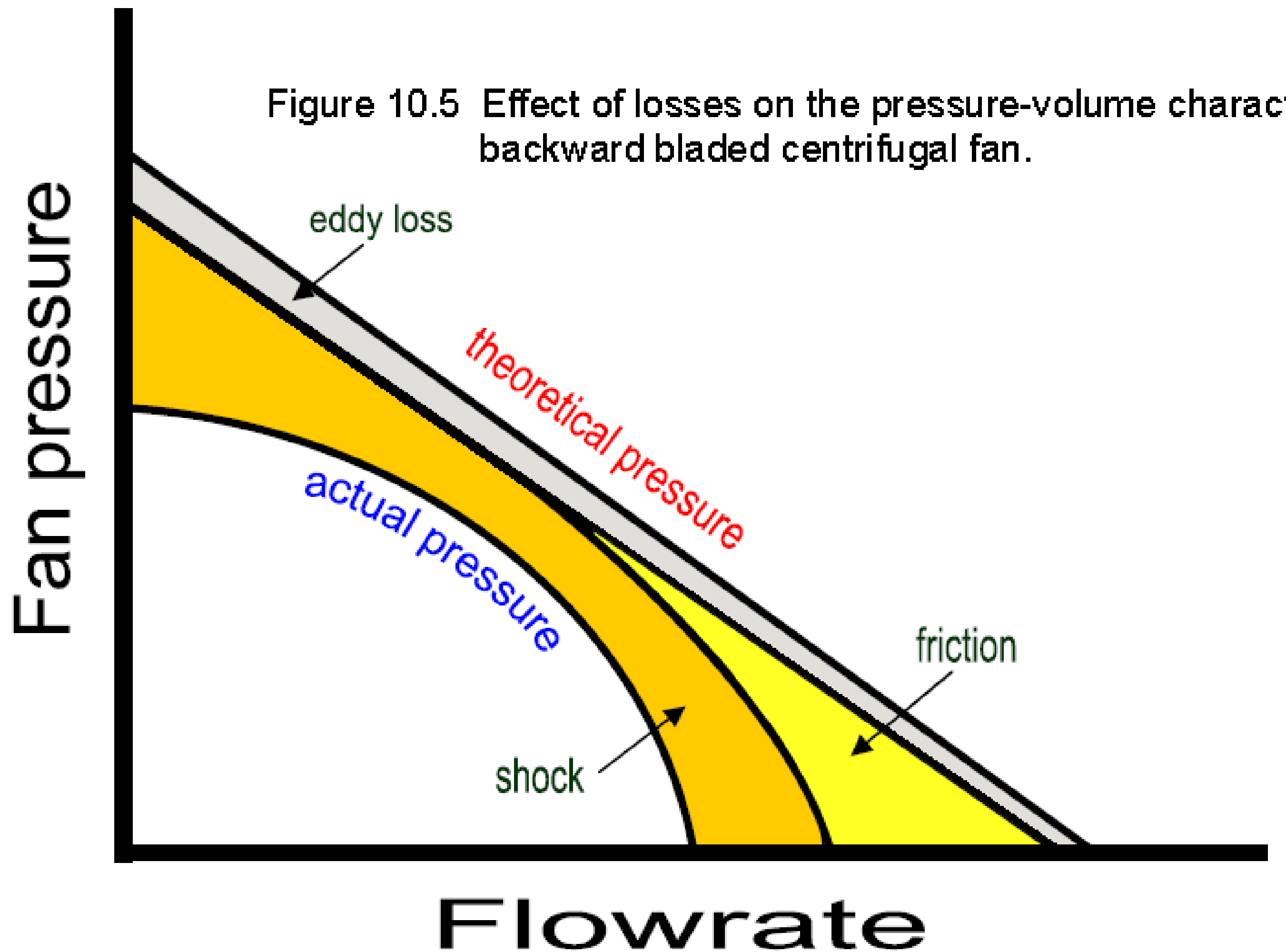
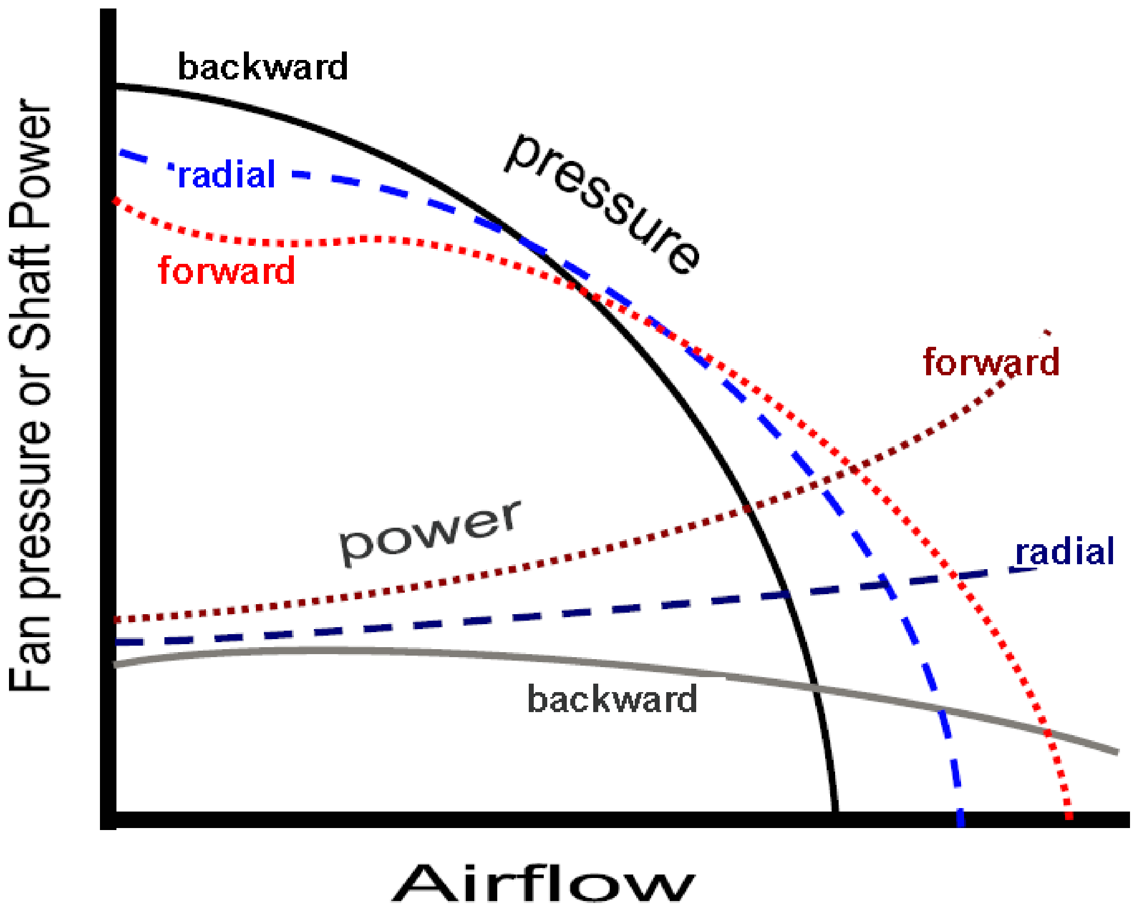



Figure 10.5 Effect of losses on the pressure-volume characteristic of a backward bladed centrifugal fan.

Design of Impeller

# Backward Bladed Centrifugal Impeller

Figure 10.6 Actual pressure and shaft power characteristics for centrifugal impellers.



# Assignment

## First calculate based on the design parameters

Thickness of channel: 10cm
Inlet axial flow velocity = 2 m/s
Thickness of blade = 3 mm

## Roll Number 1 – 16: Radial Impeller/8 blades

Radius of Eye = (30 + 0.1*x) cm
Radius of Outlet = (50 + 0.2*x) cm

## Roll Number 17 – 32: Forward Curved Impeller/7 blades

Radius of Eye = (30 + 0.15*x) cm
Radius of Outlet = (50 + 0.15*x) cm

## Roll Number 33 – 51: Backward Curved Impeller/7blades

Radius of Eye = (30 + 0.1*x) cm
Radius of Outlet = (50 + 0.1*x) cm

Note:
You can choose appropriate fillet radius in your CAD model.
You can choose blade angles as per your desires condition

# Assignment

## First calculate based on the design parameters

- Use excel or MATLAB to draw performance chart (i.e. del P vs Flow Rate and Power vs Flow Rate) | Deadline: Poush 15
- Orthographic view of impeller in AutoCAD: Deadline: Poush 30
- Create 3D model of the Impeller in 3D software like CATIA or Solidworks: Magh 15
- Use Ansys CFX or Ansys Fluent to simulate the impeller and verify the result with analytical result (i.e. performance chart) – Falgun 15
- Plot following plots
  - Velocity contour field
  - Pressure contour field
  - Streamlines

Files Required:

- Analytical results in Excel or Matlab (.xls or .xlsx or .m format)
- AutoCAD file (.dxf and .dwg) format of orthographic
- 3D CAD Model (.igs and .stp) format
- Report (latex files compressed in .rar file format) – deadline: Falgun 25 (FINAL REPORT)
- Ansys simulation project file (compressed in .rar file format

Design of Impeller

# Assignment

## Guidelines and Code of Conduct

- If found plagiarism
  - Geometry and CAD files 20% of marks reduction
  - Report contents 30% addition reduction of marks
  - Simulation 20% reduction marks
  - Analytical result 10% reduction
  - Late Submission 5% for each day
- Evaluation Scheme
  - Report 50% (Latex 10% + Contents 30% + Report Format 10%)
  - CAD File 10%
  - Ansys CFD File 20%
  - Excel or Matlab File 10%
- Viva will be taken after submission of the report. Date will be notified after submission of report.

To know more about plagiarism please refer to the following website:
https://www.plagiarism.org/
Example for plagiarism checker:
https://www.grammarly.com/plagiarism-checker

If you have any questions regarding design,

**don’t hesitate to ask me**

**!!!**