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Design of Impeller Blade by Varying Blades and Type of Blades Using Analytical

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ABSTRACT:

Pumps are used in the process of transferring fluids from one place to other and these pumps have a vital role in the domestic and industrial areas. This project deals with the application and need of Structural analysis in the pump industries. For this purpose, we have made a design and analysis of impeller used in Domestic Open well Radial flow water pumps. The impeller selected is of enclosed type, which is commonly used in domestic water pumps. In this project we have designed an impeller for a domestic need using formulas formulated by Dr. K.M Srinivasan, in the book Rotodynamic Pumps. The impeller is modeled using CAD software and analyzed using Ansys package. The Ansys output is cross checked with desired requirements, so as to state the accuracy and need of Ansys analysis. In this study, the performance of impellers with the same outlet diameter having different blade numbers for centrifugal pumps is thoroughly evaluated The impeller outlet diameter, the blade angle and the blade numbers are the most critical parameters which affect the performance of centrifugal pumps. The model pump has a design rotation speed of 4000rpm and an impeller with 4, ,8 & 12 numbers of blades has been considered. The inner flow fields and characteristics of centrifugal pump with different blade number are simulated and predicted by using Ansys software. The simulation is steady design pressure to take into account the impeller. For each impeller, static pressure distribution, total pressure distribution and the changes in head as well as efficiencies of centrifugal pump are discussed.

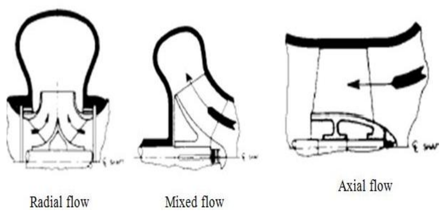
With the increase of blade number and Type of blade like forward, backward and radial blades types also design. By checking material variation and find out best material for best design type will be finalized.

INTRODUCTION

TYPES OF PUMPS:

- Axial flow pumps
- Single stage or multistage
- Open impeller
  - Fixed pitch
  - Variable pitch
- Closed impeller
- Radial flow pumps
- Single suction or double suction
- Self priming or non priming
- Single stage or multistage
  - Open impeller
  - Semi open impeller
  - Closed impeller
- Mixed flow pumps
- Single suction or double suction
- Self priming or non priming
- Single stage or multistage
- Accelerate flow by imparting kinetic energy
- Decelerate (diffuse) in stator
- Results in increase in fluid pressure

Impeller profiles



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THEORY:

In this chapter the relevant physics of a pump will be described, and methods to calculate the performance will be given. A brief introduction to methods for measuring the velocity of a fluid will also be given.

2.1 PUMP THEORY:

2.1.1 VELOCITY TRIANGLES:

In turbo machinery the motion of the fluid needs to be specified according to the rotational motion of the impeller. The absolute velocity c can be regarded as the velocity relative to a stationary part, such as the housing or the diffuser. This can be seen as the sum of two velocities: the peripheral velocity of the impeller u, and the fluid velocity relative to the impellers w.

$$c = u + w \qquad (2.1)$$

When these velocities are plotted, they form a velocity parallelogram or a velocity triangle. The velocities are normally given subscript 1 or 2, where 1 corresponds to impeller inlet, and 2 to impeller outlet. The subscripts 3 and 4 corresponds to the inlet and outlet of the diffuser, while 5 and 6 correspond to the inlet and outlet of the return channels of multistage pumps. The velocity parallelograms can be seen in Figure 2.1. Figure 2.2 shows how these can be rearranged in order to form velocity triangles that show the relation between the relative and absolute velocities. α and β represent the angles of the absolute and relative velocities at the inlet and outlet of the impeller. When dealing with an axial inlet we usually assume zero swirl, meaning $\alpha_1 = 90^\circ$. In multistage pumps this is difficult to obtain because of disturbances in the flow caused by the previous stage.

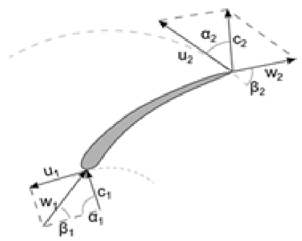

Figure 2.1: Velocity diagram in an impeller stage

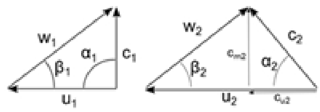

Figure 2.2: Velocity triangles

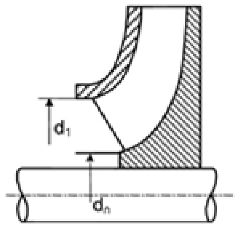

Figure 2.3: Inlet dimensions

According to theory , the angle at the outlet, β_2 , aligns with the camber angle of the impeller. In reality this angle deviates due to slip and blade blockage, as we shall see later. The peripheral velocity u may easily be calculated by knowing the rotational speed n of the impeller, by the following relation:

$$u = \pi d \frac{n}{60} \qquad (2.2)$$

in which d is the diameter where the velocity is evaluated. The absolute velocity 'c' can be decomposed into meridional and peripheral components with subscripts m and u. With zero swirl at the inlet c_{u1} is negligible, and $c_{m1} = c_1$. By taking into account conservation of mass, the relation between c_{m1} and c_{m2} can be found.

$$C_{m2} = \frac{Q_{La}}{A_2} = C_{m1} = \frac{A_1}{A_2} \qquad (2.3)$$

In Equation (2.3), Q_{La} is the volume flow passed through the impeller, A_1 is the area at the inlet and A_2 is the outlet area of the impeller. The areas are calculated from equations (2.4) and (2.5), where d_n is the hub diameter as seen in Figure 2.3, d_1 is the impeller eye diameter, d_2 is the diameter at the outlet and b_2 is the height of the outlet of the impeller.

Inlet area: $A_1 = \frac{\pi}{4} d_1^2 - d_n^2 \qquad (2.4)$

Outlet area: $A_2 = \pi d_2 b_2 \qquad (2.5)$

The calculation of c_{u2} is a bit more difficult due to slip, and will be discussed further in subsection 2.1.4.

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This basic knowledge of velocity triangles will be used throughout the following sections, and they are important parameters when designing the diffusing elements.

2.1.2 SPECIFIC SPEED:

In order to classify pumps into different categories, the specific speed was first introduced by Camerer in 1914 and further developed by Stepanoff in 1948

$$n_q = n\frac{\sqrt{Q_{La,opt}}}{H_{opt}^{0.75}} \qquad (2.6)$$

When calculating the specific speed with Equation (2.6) H_{opt} is the head. The sub-script opt indicates that they are evaluated at the best efficiency point of the pump, also called BEP. By calculating the specific speed it is possible to classify which kind of pump would be suitable for different applications, and it is also possible to compare pumps in different operating conditions. n_q is not dimensionless, but is a number used for classifications in the same way as the Reynolds' number.

2.1.3 EULER'S EQUATION, THEORETICAL HEAD

By combining Newton's 2.law and the law of momentum, Euler's equation can be obtained in order to calculate the theoretical head of the pump.

$$H_{th\infty} = \frac{U_2 C_{U2} - U_1 C_{U1}}{g} \qquad (2.7)$$

In a pump with an axial inlet c_{u1} is negligible and Euler's equation reduces to:

$$H_{th\infty} = \frac{U_2 C_{U2}}{g} \qquad (2.8)$$

To obtain this theoretical head we assume an infinitely amount of infinitely thin blades. Reducing the number of blades reduces the friction area in the pump, but also increases the pressure differences between the suction side and pressure side of the blades. When this difference grows we experience a flow pattern on the trailing edge of the blade called slip.

This will be further discussed in the following section.

2.1.4 SLIP:

To fully understand what happens at the trailing edge of the blades, it is necessary to know a bit about what goes on within the impeller. The impeller is a curved channel in constant movement in which the blades act upon the fluid to create an increased velocity and pressure. This leads to the fact that the pressure will be higher at the pressure side than at the suction side of the blades. Since the pressure distribution correlates with the velocities, there must be a difference in the velocity of the fluid at these two surfaces. The flow is therefore not able to follow the blade exactly, and deviates from shape of the blade. When the flow passes the trailing edge, the pressure difference immediately vanishes and the streamlines curve around the trailing edge to satisfy the outlet conditions. Slip is an important design parameter when designing pumps, and it has a significant influence when computing head and flow properties. Although slip is a well known phenomena, exact calculations of the process can only be done by testing. In Figure 2.5 the effect of the slip on the outlet angle β2 can be seen. The Figure uses a slightly different notation, but the difference in outlet angle δ between the flow and the blade angle β_{2B} can clearly be seen, as well as the changed peripheral component of the absolute velocity c_{u2} . Getting a complete knowledge of the effects of slip is a difficult operation, with several uncertainties. Nevertheless it is important to take the slip into account when designing centrifugal pumps, and the following approaches give us good approximations in design operations.

Slip calculation

Figure 2.5 introduces the slip coefficient γ. The slip coefficient is defined by Gülich and Tuzson as:

$$C_{u2\infty} - C_{u2} = (1-\gamma)\, u_2 \qquad (2.9)$$

In Equation (2.9) $c_{u2\infty}$ represents the peripheral component of the absolute velocity with infinite number of blades. c_{u2} is the peripheral component of the real velocity, taking slip into account.

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The most accurate values which exist for the slip coefficient were calculated by Busemann in 1928, and later reviewed and adjusted by Wiesner in 1967. Wiesner derived the following expression for calculating slip coefficient with a standard deviation of about ±4%:

$$\gamma = f_1 \left(1 - \frac{\sqrt{\sin \beta_{2B}}}{Z_{La}^{0.70}}\right) \qquad (2.10)$$

In Equation (2.10) β_{2B} is the blade angle at the outlet and z_{La} is the number of impeller blades. The factor f_1 is for radial impellers set to 0.98. With $\gamma = 1$ there is no slip. This equation is valid for a limited range of mean diameter ratios, given by the following expression:

$$\varepsilon_{lim} = \exp\left(-\frac{8.16 \sin \beta_{2B}}{Z_{La}}\right) \qquad (2.11)$$

The limit is defined as $\frac{d_{1m}}{d_{2m}} = \varepsilon_{lim}$, where the subscript m represents a mean streamline. The mean streamline corresponds to the streamline ending on the geometric mean diameter at the outlet $\frac{d_{1m}}{d_{2m}} > \varepsilon_{lim}$, the right side of the equation (2.10) can be multiplied by the factor kw, calculated by the following equation

$$k_w = 1 - \left(\frac{\frac{d_{1m}}{d_{2m}} - \varepsilon_{lim}}{1 - \varepsilon_{lim}}\right)^3 \qquad (2.12)$$

Shape No. Range	Shape of the Impeller
$10<n_s<50$	Radial Type
$50<n_s<150$	Mixed Type
$150<N_s<400$	Axial Type

PFLEIDERER'S CORRECTION

Another approach, presented by Stepanoff and by Lazarkiewicz and Troskolanski, is to use Pfleiderer's correction factor Cp to calculate the theoretical head with a finite number of blades directly. The relation is given in Impeller Pumps as:

$$H_{th} = \frac{1}{1+C_p} H_{th\infty} \qquad (2.13)$$

The correction factor is by Pfleiderer defined by the semi-empirical formula [2, p. 94]

$$C_p = 2 \frac{\psi}{Z_{La}} \frac{1}{1-\left(\frac{r_{1m}}{r_{2m}}\right)^2} \qquad (2.14)$$

in which z_{La} is the number of impeller blades, r_{1m} and r_{2m} are the inner and outer mean radius, while ψ can be calculated from the following formula: s, while ψ can be calculated from the following formula:

$$\psi = f(1 + \sin \beta 2B)\left(\frac{r_{1m}}{r_{2m}}\right) \qquad (2.15)$$

where f is chosen between 1.0 and 1.2. The Pfleiderer correction thus gives a simple way to calculate the reduced head due to slip, but the two methods give slightly different results, as we shall see later.

2.1.5 IMPELLER OUTLET VELOCITIES:

To accurately calculate the outlet velocities from the impeller, it is important to obtain as thorough information about the flow as possible. This includes slip, blade profile, trailing edge profile, and of course the main parameters such as flow, head, rotational speed and so on. To calculate the velocity triangle at the outlet, the preceding knowledge is used combined with geometry. The meridional component of the absolute velocity c_{m2} is calculated with Equation (2.3), while the peripheral component 'c_{u2}' can be found from geometry in Figure 2.5:

$$C_{u2} = \gamma u_2 - \frac{C_{m2}}{\tan \beta_{2B}} \qquad (2.16)$$

These values represent the velocities at the mean streamlines. They can also be calculated at the inner and outer streamlines, which is recommended in detailed design.

RADIAL PUMP IMPELLER DESIGN:

- Design a rotor(impeller) of a radial water pump for the following given values
- Q = 95 lpm =1.583 lps
- H = 24 m
  n = 2880 rpm
  Power: 1hp = 746W
  Pipe size: 25 X 25 mm
- The **Specific speed** for the pump is calculated from the following formula for the given values

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$$n_s = 3.65\frac{n\sqrt{Q}}{H^{3/4}} = \frac{3.65 * 2880\sqrt{0.00158}}{24^{3/4}} = 38.5$$

- After finding the specific speed the shape of the impeller can be decided using the following table.

➢ The specific speed is in the radial type pump range

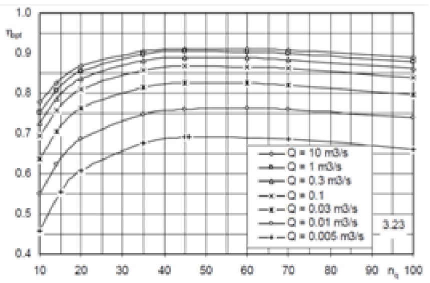
- The shaft Power is given as:

$$P = \frac{\rho QgH}{\eta_o}$$

- The overall efficiency η_o for a single stage, single entry, radial pump can be read from the figure below to be $\eta_o = 0.85$.

- The power pump becomes,

$$P = \frac{10^3 \times 0.00158 \times 9.81 \times 24}{0.85} = 1.84\text{hp}$$



Efficiencies of single-stage, single-entry, radial pumps

(Source: Centrifugal Pumps, Johann Friedrich Gülich, 2nd ed)

- Shaft diameter can be found using the following formula

$$d_s = \sqrt[3]{\frac{16}{\pi}\frac{T}{\tau_t}}$$

- The torque can be calculated as it follows:

$$T = \frac{P}{\omega} = \frac{P}{2\pi n} = \frac{55000}{2\pi \times 2880/60} = 362\text{Nm}$$

- The allowable shear stress for most of the shafts is in the range between 40 -60 N/mm2. Thus the shaft diameter becomes:

$$d_s = 32.13 - 35.64\text{mm}, \text{say } d_s = 40\text{mm}$$

ψ	Pump Type
0.7-1.3	Radial Impeller
0.25-0.7	Mixed-flow impeller
0.1-0.4	Axial Impeller

- The hub diameter, d_h = 1.1 to 1.3 d_s

$$d_h = 1.2d_s = 50\text{mm}$$

- The eye diameter (D1) can be calculated by assuming inlet number . The radial velocity at the inlet is given by:

$$C_{om} = \varepsilon\sqrt{2Y}$$

- The volume flow rate at the suction end is given by:

$$\varrho' = C_{om}\frac{\pi}{4}(D_1^2 - d_h^2), \text{where } \varrho' = \frac{\varrho}{\eta_V}$$

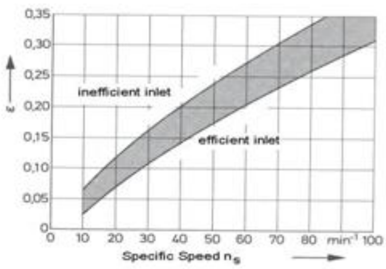
- Calculating for the eye diameter,

$$D_1 = \sqrt{\frac{4\varrho}{\pi \times C_{om} \times \eta_V} + d_h^2}$$

- The volumetric efficiency is given by:

$$\frac{1}{\eta_V} = 1 + \frac{0.287}{n_s^{2/3}}, = 1 + \frac{0.287}{22.3^{2/3}}, = 1.0362 \text{ or } n_V = 0.965$$

- The inlet number can be found from the following table.



Guidelines to choose ε

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- From the table,

  $\varepsilon = 0.08 - 0.13, \text{for } n_s = 22.3$
- Thus,

  $C_{om} = 2.5 - 7.25 m/s \approx 4 m/s$
- Another option to estimate the value of inlet number is to use the formula by Pfleiderer,

  $\varepsilon = (1.5 - 3).10^{-2} n_s^{\frac{2}{3}}$
- Thus, D1 becomes,

  $D_1 = \sqrt{\frac{4\times0.0833}{\pi\times4\times0.965} + 0.05^2} = 0.173m \approx 0.175m$
- The outer diameter D2 can be calculated using:

1) The head coefficient (ψ)

$$\psi = \frac{gH}{U_2^2/2}$$

$$U_2 = \frac{\pi D_2 n}{60} = \sqrt{\frac{2gH}{\psi}}$$

$$D_2 = \frac{60}{\pi n}\sqrt{\frac{2gH}{\psi}}$$

- The head coefficient for different type of pumps is given below.
- Choosing $\psi = 1$,

$U_2 = \frac{31.32m}{s}$, and

$D_2 = 0.412m$

2) The specific diameter (δ)

$$\delta = \frac{\psi^{1/4}}{\phi^{1/2}}$$

$$\psi = \frac{2gH}{U_2^2/2} = \frac{2Y}{\pi^2 n^2 D^2}$$

$$\phi = \frac{C_m}{U} = \frac{Q}{AU} = \frac{Q}{\frac{\pi D^2}{4}\pi D_n} = \frac{4Q}{\pi^2 D^3 n}$$

- The specific diameter becomes,

$$\phi = \frac{\psi^{1/4}}{\phi^{1/2}} = \left[\frac{2Y}{\pi^2 n^2 D^2}\right]^{1/4} \Big/ \left[\frac{4Q}{\pi^2 D^3 n}\right]^{1/2} = \frac{1.05 D_2 Y^{1/4}}{Q^{1/2}}$$

- D2 becomes,

  $D_2 = \frac{\delta}{1.05(gH/Q^2)^{1/4}}$

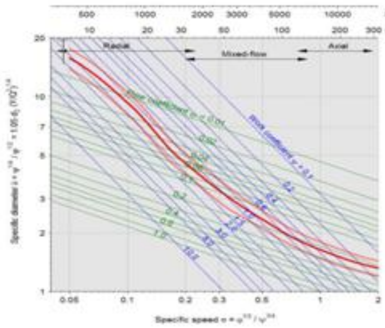
- The specific diameter is plotted in the following diagram for various specific speeds,

- From the table, $\delta = 6.5$.

$$D_2 = \frac{6.5}{1.05(9.81\times50/0.0833^2)^{1/4}} = 0.397mm$$

- Thus the outer diameter can be taken to be

$D_2 = 0.4m$



Cordier Diagram

- Blade width b1

$$b_1 = \frac{Q'}{\pi D_1 C_{om}} = \frac{Q}{\eta_V \pi D_1 C_{om}}$$

$$= \frac{0.0833}{0.965\times\pi\times0.175\times4} = 0.039m \approx$$

40m

- The Blade width at the outlet b2

$$b_2 = \frac{Q'}{\pi D_2 C_{2m}} = \frac{Q}{\eta_V \pi D_2 C_{2m}}$$

- C_{2m} can be found from figure below.

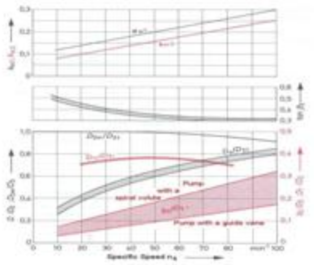
$$C_{2m} = k_{m2}\sqrt{2gH},$$

k_{m2}=0.11, from the figure

$$C_{2m} = 3.45 m/s$$

$$b_2 = \frac{0.0833}{0.965 \times \pi \times 0.4 \times 3.45} =$$

$0.0199 m \approx 20 m$



- Blade angle β_0

$$\beta_0 = \tan^{-1}\left[\frac{C_{om}}{U_1}\right] = \tan^{-1}\left[\frac{C_{om}}{\pi D_1 n}\right]$$

$$= \tan^{-1}\left[\frac{4}{\pi \times 0.175 \times 1450/60}\right] = 16.75°$$

- Blade angle β_2 can be estimated from the previous figure.

$$\tan\beta_2 \approx 0.4 - 0.43$$

$$\beta_2 = 21.8° \text{ to } 23.3°$$

- Number of blades

$$Z \approx \frac{\beta_2}{3} \approx 7, \text{Stepanoff approach}$$

$$Z = k\frac{D_1+D_2}{D_2-D_1}\sin\left(\frac{\beta_1+\beta_2}{2}\right), \text{taking the value of } k = 7$$

$$Z = 6$$

IMPELLER DESIGN:

Design is the application of scientific and mathematical principles to practical ends to form efficient and economical structures, machines, processes, and systems.Design of centrifugal impeller is done. Impeller design parameters are calculated using his procedure by giving head, volume flow rate and pump speed as input.

Specification of pump
Head: 24m
Discharge: 95lpm = 1.583lps
Power: 1hp = 746W
Speed: 2880rpm
Pipe size: 25 X 25 mm
Calculated parameters
The following are the parameters calculated using the above said method and these are the parameters which help in generating the impeller vane profile.
Specific Speed = 38.5
Power input to the pump = 1.84hp
Shaft Diameter = 25 mm
Outer Diameter of impeller = 144 mm
Velocity of fluid at the impeller inlet = 5.43 m/s
Inner Diameter of impeller = 36 mm
Inlet blade angle = 19.25°
Impeller width at inlet = 10 mm
Blade angle at outlet = 23.76°
Number of blades = 4
Vane profile development:
The vane profile can be developed by using Point by Point method, Single arc method, Multi arc method and Error Triangle method. In this work, Point by Point method is selected, and the vane profile parameters are calculated and the profile is traced. The more number of points we employ, the tracing of the profile is made easy. In this case we have selected 7 trace points.

S.NO	r	Cm	B	W	Cm/w	δ	t	δ/t	sin ß	ß
1	18	1.44	32	5.46	0.265	5	28.27	0.17	0.44	26.14
2	27	1.39	28.3	5.06	0.27	5	42.41	0.11	0.39	23.13
3	36	1.34	24.7	4.67	0.29	5	56.54	0.08	0.37	22.11
4	45	1.29	21	4.27	0.3	5	70.68	0.07	0.37	21.98
5	54	1.25	17.3	3.87	0.32	5	84.82	0.05	0.38	22.4
6	63	1.2	13.73	0.48	0.35	5	98.96	0.05	0.39	23.3
7	72	1.15	10	3.08	0.37	5	113.09	0.04	0.41	24.72

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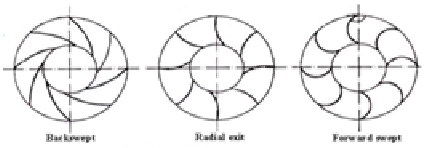
S.NO	B Δ	r	x	$\Delta\theta$	Radians	Degrees
1	113.18	0.009	99.94	0.89	0 0	0
2	86.69	0.009	77.52	0.69	0.895	51.5
3	68.35	0.009	61.7	0.55	1.597	91.5
4	55.05	0.009	49.98	0.44	2.156	123.3
5	44.91	0.009	40.87	0.36	2.602	149.1
6	36.83	0.009	33.5	0.3	2.97	170.2
7	30.16	-		-	3.272	187.5

CALCULATION PART:

Pump Impeller Types

Centrifugal pump impeller types are:

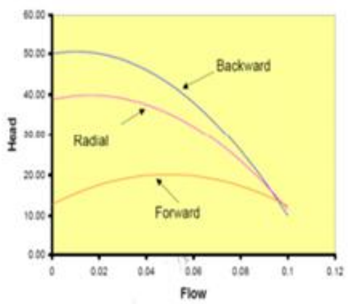
1. Forward swept impeller
2. Radial exit impeller
3. Backswept impeller



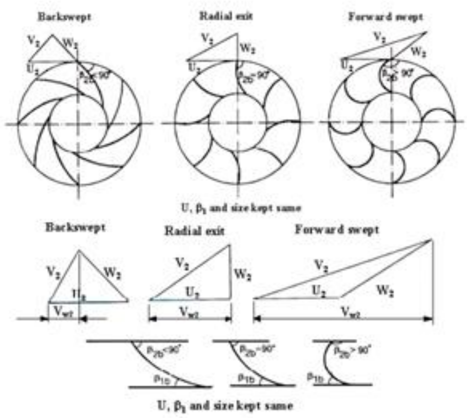
1.Forward swept impeller is used for low flow rate and head.

2. Backswept impeller is used for high flow rate and head.

3. Radial exit impeller is used for medium head and flow rate



Impeller Types and Velocity Triangle



Terminology of Centrifugal Pumps

Suction Head (hs):

It is the vertical height of the center line of the pump above the water surface in the sump. This height is the suction lift and is denoted as hs

Delivery Head (hd):

It is the vertical height between the center line of the pump and water surface in the tank to which water is delivered. It is denoted as hd

Static head (Hs):

Static head is the vertical distance between the liquid level in the sump and the delivery tank. It is denoted by Hs. Therefore static head,

Hs = hs+hd

Manometric Head (Hm) or effective head:

It is the total head or lift that must be produced by the pump to satisfy external requirement. It includes all the losses.

$$h = H_d - H_s$$

$$\left[\frac{P_d}{\gamma} + \frac{V_d^2}{2g} + Z_d\right] - \left[\frac{P_s}{\gamma} + \frac{V_s^2}{2g} + Z_s\right]$$

Head Developed by a Pump

Total dynamic head

□For a horizontal pump the total dynamic head is defined as

$$H = H_d - H_s + \frac{V_d^2}{2g} - \frac{V_s^2}{2g}$$

For a vertical pump with pumping element submerged, the total dynamic head is defined as

$$H = H_d - H_s + \frac{V_d^2}{2g}$$

Design Parameters

Impeller Speed

• Pipe Connections and Velocities

• Impeller Inlet Dimensions and Vane Angles

• Impeller Outlet Dimensions and Vane Angles

• Impeller Vane Shape

Pump Design Procedure

In most cases, rotational speed, N, design flow rate, Q, and head, H, will be prescribed

 Check whether a reasonable impeller size can achieve the desired head,

$H = U_2^2 / g$

 A target efficiency η is assumed, eliminating the effect of losses. This is checked against the efficiency from the chart.

 Now $H_{th} = H / \eta$

 To facilitate choice of subsequent variables, a dimensionless plot is introduced, which shows theoretical head, represented by head coefficient, Ψ, against the flow rate, represented by flow coefficient, Φ

Now Head coefficient, $\psi = \frac{gH}{\eta U_2^2} = \frac{C_{t2}}{U_2}$

flow coefficient, $\phi = \frac{Q}{\pi D_2 B_2 U_2} = \frac{C_{m2}}{U_2}$

Also $\Psi = \sigma - \Phi \tan \beta 2$

 Ct2 is proportional to the theoretical head and Cm2 is proportional to the flow rate

 These velocities fully determine the pump performance

 Head coefficient would generally decline linearly from about Ψ = 0.75 at Ns = 1000 to Ψ = 0.45 at Ns = 4000

 Typical values for Ψ and Φ are in the range of 0.4-0.7 and 0.05-0.2 respectively

 Choice of head and flow coefficients from the plot also determines the Impeller exit blade angle β2. Usually it is taken as 68° from radial

Exact impeller diameter D2 can then be chosen to give the necessary tip speed, U2

$$H_{th} = \frac{H}{\eta} = \frac{U_2 C_{t2}}{g} = \frac{\psi U_2^2}{g}$$

Effective Impeller exit width b2 is determined from the expression for flow rate,

$$Q = \pi D_2 B_2 C_{r2} = \pi D_2 B_2 U_2 \varphi$$

Number of Impeller blades can be chosen from the given chart using exit blade angle β2, and slip coefficient σ.

More blades will guide the flow better, increase the slip coefficient and increase the head

Impeller Speed and Pipe Conditions

Impeller Speed

• Speed of the drive unit may be specified by the customer.

• It generally follows standard motor speeds, e.g. 1440, 2880 rpm, etc.

• In such cases, the specific speed is fixed unless the head is high enough to use a multistage machine.

• Then the specific speed may be varied by using different number of stages.

• If the operating speed is not given, it may be chosen to give a specific speed corresponding to high pump efficiency and the prevailing suction conditions.

Pipe Connections and Velocities

• To avoid cavitations, the diameter of the suction pipe is usually larger than the pump suction flange, and both are larger than the discharge flange and pipe.

• Suction flange diameter is selected to achieve design point fluid velocity, which may range from 1.3 to 6 m/s.

• Discharge flange diameter is also based upon the design point fluid velocity in the range of 4 to 14 m/s.

Sizing of Impeller Inlet

First step is to determine the shaft size.

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• Shaft size should take care of the torque and bending moment, should avoid excessive lateral deflection, and should keep the critical speed away from operating speed

Shaft diameter, $D_s = \sqrt[3]{\frac{16T}{\pi f_s}}$ or $\sqrt[3]{\frac{32M}{\pi f_t}}$

Based on torque alone Based on bending strength alone

A more accurate check on the stresses and the deflection may be made after
the impeller is designed and the loads are known
Impeller Vane Inlet Angle
Fluid is assumed to enter the Vane radially. So that $\alpha_1 = 90°$

The Vane inlet angle β_1 is given by: $\tan\beta_1 = \frac{V_{r1}}{\mu_1}$

This Value of is slightly increased to take care of the stream contraction at the inlet edges and of the fluid prerotation. It is increased more for high suction lifts and smaller D2/D1 (i.e., higher specific speed impellers). The inlet angle usually falls in the range of 10° -25°.

Inlet guide vanes could be placed before the impeller to give a negative Vu1 and thus raise the head and overall efficiency at the design condition, but the head and efficiency drop off more rapidly under off design operating conditions.

The virtual head for $\alpha_1 = 90°$ is given by: $H_{vir,\infty} = \frac{1}{g}\left[u_2^2 - \frac{u_2 V_{r2}}{\tan\beta_2}\right]$

Total head H is given by: $H = K\frac{u_2 V_{u2}}{g} = KH_{vir,\infty}$

Combining the above equations, we get : $\frac{H_g}{K} = u_2^2 - \frac{u_2 V_{r2}}{\tan\beta_2}$

Solving for u_2: $u_2 = \frac{1}{2}\left[\frac{V_{r2}}{\tan\beta_2} + \sqrt{\left(\frac{V_{r2}}{\tan\beta_2}\right)^2 + \frac{4gH}{K}}\right]$

For radial impeller, varies between 0.65 and 0.75 and K varies between 0.6 and 0.7, the larger Values applying to lower specific speed impellers.
The outer diameter can be more easily obtained considering the overall head coefficient at the head H at best efficiency point

$$u_2 = \phi\sqrt{2gH}$$

The value of varies between 0.9 and 1.2 with average value close to unity. The main influencing factors are head, flow capacity and specific speed. The value can be read from available charts.
• In general, backward curved vanes are used for pump impellers, because they have lower exit velocity compared to radial or forward curved vanes.
Results:
Input for Blade design are given below:
Specific Speed = 38.5
Power input to the pump = 1.84hp
Shaft Diameter = 25 mm
Outer Diameter of impeller = 144 mm
Velocity of fluid at the impeller inlet = 5.43 m/s
Inner Diameter of impeller = 36 mm
Inlet blade angle = 19.25^{oP}
Impeller width at inlet = 10 mm
Blade angle at outlet = 23.76^{oP}
Number of blades = 4

CONCLUSION:

An impeller was designed for the input details using standard formulas and the vane profile was traced accordingly. Since the simulation is done only for the impeller's blade and cover part, the front and rear shrouds and hub portions in the modeling. The output of the simulation is close enough to the theoretical calculations. Hence it can be stated that, the usage of ANSYS structural analysis is worthy when compared to Forward, Backward and Radial are analyzed with changing no. of blades as 4,8 and 12 steps of blades. From the Results and plot table it is concluded that Backward blade reduction in deformation, Stress and Strain when comparing other types by increases in

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blade nos. thus by results we reducing the time for various prototypes, and reducing the financial investments for each trials. Percentage of reduction of stress comparing Backward blade with Forward & Radial is varies from 6 to 14 % for 4no. blade, 8 to 32 % for 8 no. blade,31 to 69 % for 12 no. blade. By results we come to conclusion by increasing no. of blade in backward flow is best design for high head and low stress and Conclude with best results.

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