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DETERMINING OF IMPELLER DIMENTIONS FOR ACIEVING LOW FLOW/HIGH HEAD IN A CENTRIFUGAL PUMP

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ABSTRACT

This paper deals with the development of Low Flow/High Head Pump. In this paper, a study of centrifugal pump with single-stage end suction is worked out to design a new low flow/high head pump with certain specification. The highlighting and most important parts which enhances the properties of the pump and on which changes are made to vary the output are the impeller and the body which is the casing. The design of Low Flow/High Head centrifugal pump are chosen because of its need and importance in chemical industry. Low flow/High Head pump demands was born when the demand of more efficient and improved chemical reactions started requiring higher process pressure and temperatures. It is the most useful mechanical rotor-dynamic machine in fluid works which widely used in pump Seawater booster for critical services, Light hydrocarbon boosting, Petrochemical processing, Light-Vacuum and Heavy-Vacuum-Gas-Oil ie LVGO and HVGO respectively, Heavy-duty chemical processing etc. A brief history of these pump types are taken for study and comparing the data such as type of impellers, efficiency, head coefficient, head-capacity curves and relative cost of the current designs and study the current needs. Evaluating the needs and study of advantages and disadvantages from the design point of view to develop an effective plan of execution. With summing up all data collected and the current market requirement data, mathematically design a pump with property of Low flow/High Head.

KEYWORDS: Pumps, LVGO and HVGO

A pump is a machine, which changes over mechanical vitality to hydraulic energy. Pumps has an imperative part in numerous household and industrial purposes. Different assortment of pumps is utilized everywhere throughout the world for various scope of applications. In the first place before 1930's things were entirely basic. Yet, as the requirement of exchanging fluids from one place to other expanded, new innovations of pumping came up. Low flow/High Head pump demands was conceived when the request of more effective and enhanced chemical process began requiring higher process pressure and temperatures. Presently a day’s Low Flow/High Head Pumps are broadly required in oil and gas, hydrocarbon preparing industry and power plants. They are principally used to pump Seawater promoter for basic administrations, Light hydrocarbon boosting, Petrochemical processing, Heavy-obligation synthetic handling and so forth. So as to chip away at this task a concise history of these pump types are taken for study and looking at the information, for example, sort of impellers, efficiency, head coefficient, head-capacity curves and relative cost of the present plans and study the present needs. The design of the radiating pump impeller isn't an all-around institutionalized one. So there is no specific strategy to take after the fundamental simply begin with finding the specific speed and choosing the impeller compose for the parameters given. Each firm count on their designer’s experience, ability and focused instinct to outline a decent impeller. The theory is not a fixed one for pumps so developers have to go for old plan of tied and tested strategies.

MOTIVATION

According to now there are for the most part three pumps which serve this highlights. They are rotating casing pump, high speed centrifugal pump and early regenerative turbine pumps. Be that as it may, all the present design faces challenges on the grounds that a speed increasing gear is ordinarily required to acquire high head and when speed is increased, high internal relative speeds influence pump to subject to erosion if abrasives are available. This high speed increases the chance of wear ring leakage which in turn effect the efficiency of the pump. As due to this high radial load in single stage design, the shaft may have large amount of deflections which in turn affects the pumps badly. Many companies use an add on Inducer to produce High discharge head but it has limitations of achieving the output at near Best Efficiency Point and the also has limitations to the fluids which can be used to pump through. So an improved design which eliminates the use of inducer and which doesn’t have higher operating speed has to be designed.

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SOLUTIONS

So it is necessary to have an improved design which can offer the Low flow/High Head which produces less relative internal velocities and steady output. As per current needs in industries evaluated, Pumps which serve Low flow/High head has certain difficulties when it comes in producing high head. To achieve the same, a brief history of these pump types are taken for study and comparing the data such as type of impellers, efficiency, head coefficient, head-capacity curves, relative cost of the current designs and study the current needs and also study of advantages and disadvantages from the design point are evaluated to prepare an effective plan for execution. With summing up all data collected and the current market requirement data, mathematically design a pump with property of Low flow/High Head pumping.

LITREATURE REVIEW

Fan Meng conducted an experimental investigation on pressure fluctuations in centrifugal pump volute casing. The situation was taken under consideration mainly under part load condition. He took two diffuser volutes for testing, one with a radial diffuser and other with tangential diffuser. Both volute casing was designed for same impeller parameters, where 3 impeller diameter was taken for the detailed study. The pump parameters include rotation speed n=2900rpm, design flow Q_{des}=330m3/h and the design head H_{des}=48m and the specific speed Ns= 162.3.A computational grid was generated by ANSYS ICEM-CFD. With the help of numerical simulations Head Vs Flow chart and Efficiency Vs Flow chart is plotted for three different flow conditions. Here it was observed two diffusers were showing almost same results and the volute with radial load had higher efficiency value at part load and other conditions had almost same results in efficiency. This showed the necessity of detailed look at part-load condition so the pressure fluctuations were studied in detailed using statistical analysis method. The result was observed and concluded that in radial volute the pressure fluctuation intensity was more average so reduced load on impeller. As my design was planned for low flow / high head pumps which are mainly required in chemical plants, pressure fluctuation is an important thing to look into. So a more average pressure fluctuation intensity can be selected over the other, so the study and development would be focusing on volute with radial diffuser

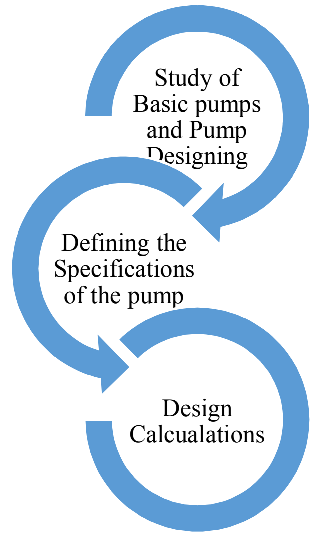
The study done by Shalin P Marthe and Rishi R Saxena on the performance characteristic of centrifugal pump on centrifugal pump. They did their study on all the three types of impeller blade that is the backward, radial and forward bladed impellers and performance characteristic where evaluated. The design of impeller was done with creo parametrics with outlet blade angles of 70^o,80^o,90^o and 100o. In comparison of streamline it was observed that turbulence was least at angle 70^o and high at angle 100^o.If turbulence is more it can lead to cavitation so 70^o is a safe angle comparatively with other angles taken. Another simulation done for water vapor contour it was observed that water vapor formation increases with increase in outlet blade angle. It was also observed that low pressure zone develops with backward to forward bladed pump. It was concluded that lower the outlet blade angle the head achieved is low. As my study is for high head pumps the higher the value of blade outlet angle would benefit me but the chances of high turbulence may lead to low pressure zones which in turn can cause cavitation.

A study done by Conard C Mowrey explains the history and theory behind high head /low flow pumps. The study included comparison of 3 major types of pumps i.e. Rotating casing pump, High speed centrifugal pump and Regenerative turbine pumps. Among the all three the high speed centrifugal pumps are the one to be discussed as this is the one which will help the study of my centrifugal pumps of low flow/high head category. High speed centrifugal pump had the working principle same but the change is that they use a different impeller named Barske which needs high speed to pump the fluid and uses a gear drive to achieve this job. Conard compare the efficiencies and report the drawbacks of all pumps. The common drawback he found was high internal velocity of these pumps are a problem when it comes to pumping a fluid which has abrasives and in turn increases wear ring leakage. They can be only used for low suction applications and as it is high speed shaft deflection is major issue. So we needed conclude the design should be for normal speed where fluids with abrasives can be pumped and eliminating the extra gear drive which may increase maintenance issues.

Mr Ragoth Sing and M. Natraj performed a detailed analysis on a self-designed pump impeller which they designed using Solid works. SolidWorks Flow Simulation is a method to approach using computerized method to get detailed results of flow in centrifugal pump. The performance of the centrifugal pump relays on the pump impeller parameters ad CFD analysis helps the designer to get optimum parameters by simulations. The methodology they followed contained 6 stages where initially the specification such as flow and head to obtain in the final pump were selected, next step was to calculate the design calculations which included calculation of specific speed, output power and torque and efficiencies hence with the help of this base calculations find the shaft diameter, velocities and blade angles. Next step was to develop vane profile development where the vane was designed using Circular arc method and point to point approach with the calculated values of velocities and angles. Then modelling of impeller using solid works was done. Hence it was found that backward curved vanes had better flow distribution than forward curved. From the results it was notices backward curved impeller had better efficiency but the head pressure was more for forward curved impeller. A comparison of both circular methods designed impeller and point to point approach was done and it was seen the circular arc method designed impeller showed more efficiency because the variation was minimum. So it can be concluded to follow the circular arc method to design so as to achieve more efficient results.

E.C. Bacharoudis and A.E. Filios0 performed a study on centrifugal pump impellers. The Study was done by varying outlet blade angles. Impeller diameter, blade angle and number of blades are most important parameters which alters the pump properties. Here, Bacharoudis do the study of 3 impellers where all 3 have same diameter but different outlet blade angle. The pump impellers he studied was having outlet blade diameter of 20^o ,30^o and 50^o and using Fluent an analysis was done for turbulent flow and for the study of different impellers configuration when the parameters are varied. The analysis was done on a pump designed on one dimensional flow theory and had characteristics $Q=43m^3/h$ and Head 10m and Speed is 925rpm. According to CFD predictions at nominal flow rate the value of head was found H=9m.It was concluded that the 10 percentage of the head loss may be because of not considering 3 dimensional flow structure and less concentration on hydraulic losses. It was also observed that there is great increase in flow rate this may also have affected the reduction in head. Considering the 3 impeller outlet blade angle it was observed that when the outlet blade angle was increased from 20^o- 50^o there was a rise of 6% in head value but overall hydraulic efficiency went down by 5%. But in when it was working in off-design it was observed that a good improvement was observed in hydraulic efficiency with the increase of blade angle when it was working at higher flow rates.

METHODOLOGY



PUMPS

A device or machinery used for raising, compressing or transferring fluid is termed as pump. Fluid can either be defined in gaseous state or liquid state. Pumps being the most commonly sold and used mechanical device is a common sight in every industry. This is the reason why pumps are available in a wide range. extremes in the centrifugal pump spectrum known as mixed flow impellers. Low specific speeds are the Characteristic for radial pumps. As shown in the diagram below there are many options in pump design.

Main parts of pumps are Impeller and Casing. Type of impellers and casing are the vital factors to be considered, where the type is decided based on the output required.

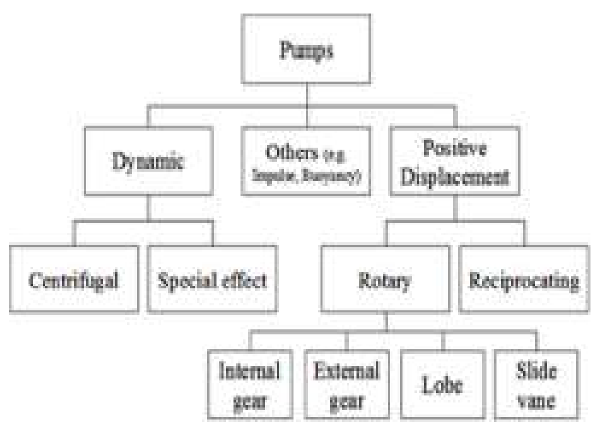


Figure 4.1 The pump classification according to the motion of work

Impeller

Impeller is a rotating device, used to transfer energy to fluid. This occurs while energy is being transferred from the motor which drives the pump to the fluid, which is pumped by accelerating the fluid outwards from the centre of rotation.

Impellers can be split into 3 as Closed impeller, Semi-Open Impeller and Open impeller

As the design is planned for a process pump it should have capability of handling solids too. So the design is done as semi-open impeller.

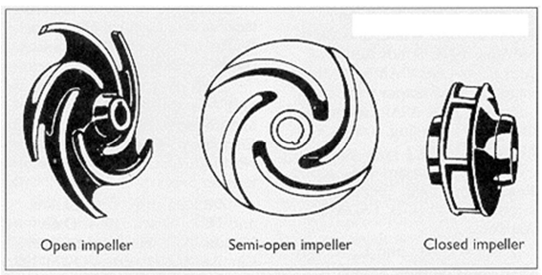


Figure 4.2 Open, Semi-Open and Closed Impeller designs

In semi open impeller it can be used for single phase and two phase. It has much less efficient than Closed impeller design but it can handle multiple phase makes a plus for it.

The impeller blade can also be designed into 3 division

Which is

Forward Swept vanes, Where outer blade angle greater than 90^o

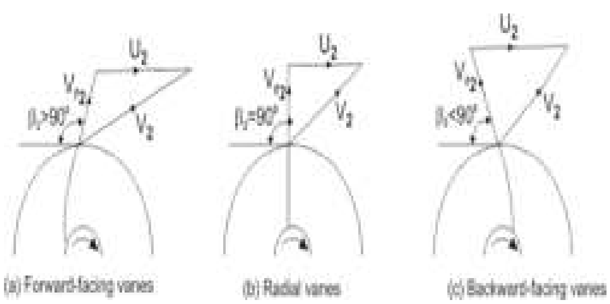


Figure 4.3 Forward, Radial and Backward facing impeller vanes

Radial Exit vanes, Where the outlet blade angle is 90^o

Backward swept vanes, Where the outlet blade angle is less than 90^o

As from studies we can discuss these three blading in view to achieve high head.

The performance of the centrifugal pump depends upon the size and configuration of the vanes used. As we can see in the head – discharge curve plotted for the same impeller blades previously simulated, it was seen that for higher outlet blade angle a high head was observed compared to other blades. So, we conclude while designing a pump for high head forward and radial exit vanes are preferred over the backward curved.

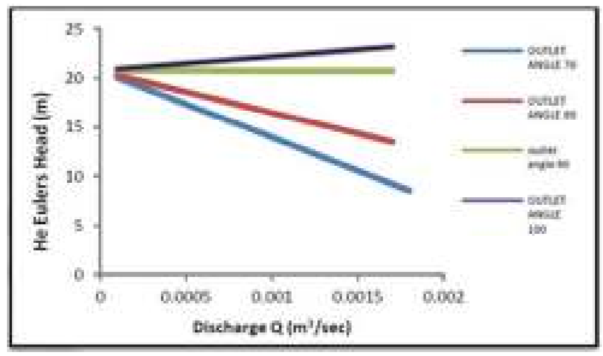


Figure 4.4 Head-Discharge characteristics of different blades

THEORETICAL DESIGN - LOW FLOW/HIGH HEAD PUMP

Impellers are designed with the help of discharge and head. The buyer will generally mention the preferred head, capacity and need. Remaining pump specifications are derived and calculated with relating formulas.

Duty Conditions

Final pump specifications are needed to be assumed or the pump is designed for certain duty condition.

The main specification required are

H - Head, Piezo metric height in meter of water

Q - Discharge (Flow rate), in m3/hr

And Number of poles of motor to be used

Calculations

Pump Rotational Speed – N (RPM)

Rotational speed, n is number of revolutions the shaft makes within a certain amount of time. Pump speed is generally given in min^{-1} (rpm).

The pivoting recurrence of the pump shaft subsequently portrays a pump's rotational speed(n). It ought not be mistaken with specific speed (N_s) and is constantly characterized as a positive figure. The pump heading of revolution is determined as clockwise or opposite.

The determination of pump rotational speed is firmly identified with the characteristics of the pump hydraulic system (circumferential speed, impeller, specific speed), as the over-all strength and efficiency of the pump and drive framework should be considered.

Most pumps operate at rotational speeds between 950 and 2900 rpm but frequently reach in excess of 6,000 rpm with special gearing and turbine drives.

$$n = (f * 120)/p$$

f - Frequency (Hz) = 50Hz

p - Number of Poles = 2

$$n = (50 * 120)/2 = 3000 rpm$$

$$n = (f * 120)/p$$

f - Frequency (Hz) = 50Hz

p - Number of Poles = 4

$$n = (50 * 120)/4 = 1500 rpm$$

So we can see for 4 pole it is 1500rpm and 2 pole it is 3000rpm, but we have to consider slip also. So a for motor with 4 poles synchronous speed is 1500rpm but full load speed is considered as 1450rpm as slip comes to act. Like same for 2 poles the full load speed is 2900 rpm.

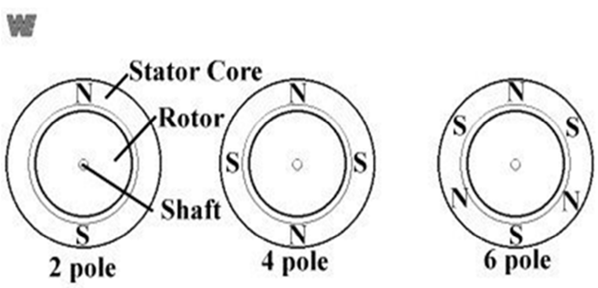


Figure 5.1 Representation of magnetic poles in a pump motor

Specific Speed-Optimum Geometry Versus Specific Speed

Crucial to any arrangement of ordering pumps is the rotor geometry that is ideal for each sort, as delineated in Figure regarding the specific speed Ns.

Here Q is the volumetric flow rate or capacity

n is the rotational speed

Ω is the angular speed

ΔH or just H is the pump head—all at the best efficiency point (BEP).

Here head coefficient ψ = $g\Delta H/(\Omega^2 r^2)$ is related lone by the flow coefficient $Qs = Q/(\Omega r^3)$.

$$N_s = \frac{n * (Q)^{1/2}}{(H)^{3/4}} \qquad \text{Eq 5.1}$$

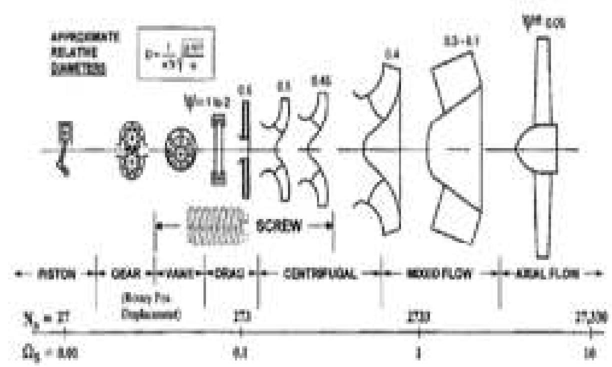


Figure 5.2 Optimum geometry as a function of BEP specific speed (for single stage rotors).

The radial vane impeller is described by specific speed 273 to 2733. Whereas from 2733 -27330 it shows the Mixed flow and Axial Flow.

So from figure find out the Specific Speed Ns

And ψ value , which can be obtained from the figure

Approximate Relative Diameter of the impeller (d_2)

$$d_2 = \frac{1}{\pi N}\sqrt{g\Delta H / \psi}$$

n is the rotating speed and = 1450rpm

ΔH is the pump head

The head coefficient (ψ)

Or by deriving Impeller diameter, d_2

We Know,

$$dF = dmr\omega^2$$

$$dP = dF/A$$

$$dm = \rho d\forall = \rho b r d\phi dr$$

Substitute for dF and dm . Write dP

$$\int_1^2 dP = \int_1^2 \frac{\rho b r d\phi dr \cdot r\omega^2}{b r d\phi} = \rho\omega^2 \int_1^2 r dr = \frac{\rho\omega^2}{2}(r_2^2 - r_1^2)$$

$$P_2 - P_1 = \frac{\rho\omega^2}{2}(r_2^2 - r_1^2)$$

divide by ρg

$$\frac{P_2 - P_1}{\rho g} = \frac{\left(\frac{\rho\omega^2}{2}(r_2^2 - r_1^2)\right)}{\rho g}$$

$$\frac{P_2}{\rho g} - \frac{P_1}{\rho g} = \frac{\left(\omega^2 r_2^2 - \omega^2 r_1^2\right)}{2g}$$

So we Can write

$$\frac{P_2}{\rho g} = \frac{\left(\omega^2 r_{2\,1}^{2\,2}\right)}{2g}$$

We know

$$H = \frac{P}{\rho g}$$

$$U = r\omega$$

So,

$$H_2 = \frac{U_2^2}{2g}$$

Substituting d_2 /2(ω) for U_2 and solving for D_2

$$d_2 = \frac{2\sqrt{2gH_2}}{\omega} = \frac{2\times 60\sqrt{2\times 9.81\times H_2}}{2\pi N} = \frac{84.6\sqrt{H_2}}{N}$$

So impeller diameter d_2

Multiply the other side by an experimentally determined coefficient □

$$d_2 = \frac{84.6\times\Phi\times\sqrt{H_2}}{N} \qquad \text{Eq 5.2}$$

Most of the plotted points fall within a range of 0.9 to 1.1. for □.

So the value of d_2 is to be found with the equation 5.2.

Pump Power Output or Hydraulic Power

Hydraulic power is the power transferred to the fluid by the pump or the power which is obtained by the fluid. $P = Q * \rho * H * 9.81$

P = Hydraulic power in Watt.

Q = Flow in m3/hr

ρ = Density of the liquid in kg/m^3

H = Piezo metric height in meter of water

Average Intensity of gravity, g =9.81m/s^2

Liquid is lift against gravitational force. so

W = Force x distance

= weight of fluid x head = mgH

Power is the rate of doing work so.

P= mgH/t = (m/t) gH

= (mass flow rate) gH

= (density of fluid x volumetric flow rate) gH

$$P = Q * \rho * H * 9.81 \qquad \text{Eq 5.3}$$

Hydraulic power is calculated so as to get torque out of it. So from Eq 5.3 Power value is calculated within the limits.

Torque

The torque or turning moment, for a pump can be calculated by

$$\tau = 9552 * \frac{P(kW)}{N} \qquad \text{Eq 5.4}$$

P = Power transmitted to the fluid by the pump in Watt.

N = Rotational speed (RPM)

Brake Horse Power (BHP)

Before brake horse power we should look into efficiency of a pump

Efficiency is equal to water horse power divided by brake horse power

$$\eta = \frac{WHP}{BHP}$$

Hence,

$$BHP = \frac{WHP}{\eta} \qquad \text{Eq 5.5}$$

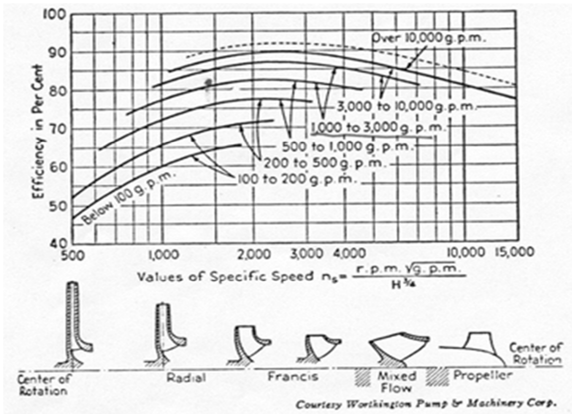


Figure 5.3 Approximate relative impeller shapes and efficiencies as related to specific speed

So we know from previous figure we take

$$\eta = 70\%$$

And from equation 5.5, we get the Brake horse power, that is the input power to the pump .

This input power is to be calculated so as to get shaft diameter. As this power is needed to be transferred through this shaft, the torque which has effect on the shaft is taken, so as to calculate the diameter of the shaft. To calculate shaft torque we use Brake horse power, that is the power output from the actuator motor.

Shaft Torque

$$T = 9552 * \frac{P(kW)}{Rpm} \qquad \text{Eq 5.6}$$

P = Power transmitted to the pump by the actuator motor with the help of a shaft in Watt.

N = Rotational speed (RPM)

Shaft torque is represented by the same equation 5.4, but here the Power, P we replace with Brake horse power, so as to get the torque value which the shaft experience.

Shaft Diameter

Calculation of shaft diameter in the base of calculated torque value. The pressure distribution on the shaft influences the bending moment and if there is any unbalanced radial thrust acting it will also have an effect on bending moment

Diameter of shaft is equal to

$$d_{sh} = \sqrt[3]{\frac{16T}{\pi s_s}}$$

Where T is the torque to be transmitted through the shaft and S_s is Shear stress and is taken as 4000psi(Assuming)

So , Shaft Diameter d_{sh}

$$d_{sh} = \sqrt[3]{\frac{16T}{\pi s_s}} \qquad \text{Eq 5.7}$$

The Inlet Meridional Velocity

Before that we need to know basic of velocity calculations. There are mainly 3 velocities used for velocity triangle plotting and calculations. They are the absolute velocity c , peripheral velocity of the impeller u and the relative fluid velocity to impeller w. . □ and β are angle which represent angle of the absolute and relative velocities at inlet and outlet of impeller.

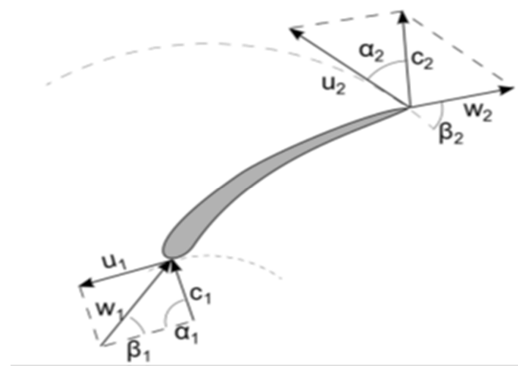


Figure 5.4 Velocity diagram in an impeller stage

c= u+w

The absolute velocity c , can be split into meridional velocity c_m and peripheral velocity c_u . With swirl = zero at the inlet c_{u1} is negligible so $c_{m1} = c_1$.

$$c_{m_1} = K_{cm1}\sqrt{2gH} \qquad \text{Eq 5.8}$$

where K_{cm1} is the velocity coefficient by Stepanoff (1957) modified form in figure

C_{m1} is Inlet meridional velocity which can be obtained from the equation 5.8

Table 1: Stepanoff (1957) modified form to find Kcm1 and Kcm2 (Equations for data: K_{cm1} = 0.001923(Ns) + 0.0615, K_{cm2} = 0.001923(Ns) + 0.0615)

N rpm	N_s	K_{cm1}	K_{cm2}	c_{m1} m/s	c_{m2} m/s
1000	40	0.1384	0.1670	3.065	3.699
1100	44	0.1461	0.1743	3.235	3.859
1200	48	0.1538	0.1815	3.406	4.018
1300	52	0.1615	0.1887	3.576	4.178
1400	56	0.1692	0.1959	3.746	4.338
1500	60	0.1769	0.2031	3.917	4.498
1600	64	0.1846	0.2103	4.087	4.657
1700	68	0.1923	0.2176	4.257	4.8.17
1800	72	0.2000	0.2248	4.428	4.977
1900	76	0.2076	0.2320	4.598	5.137
2000	80	0.2153	0.2392	4.768	5.297

Where

H = Head, Piezo metric height in meter of water

And g = Acceleration due to gravity

To find, The inlet cross section area (A_0)

$$A_0 = Q^0 / C_{m1} \qquad \text{Eq 5.9}$$

A common volumetric efficiency for centrifugal pumps is 96%. Therefore, the design Q becomes

$$Q^0 = Q / 0.96$$

As we see equation 5.9 it can be simply described in terms of a circle, Q^0 is the flow through the circle m^3/h and C_{m1} is the velocity in m/s

So we can divide both in terms of units

$$\frac{m^3/h}{m/s} = \frac{(m^3/h)}{(m/h)*3600} = \frac{m^2}{3600}$$ which is Area of circle

The Inlet Diameter (D_1)

Area A can be written in terms of diameter

$A = \frac{\pi d^2}{4}$ so writing in terms of diameter $d = \sqrt{\frac{4A}{\pi}}$

So we will find the area by equation,5.9, substituting the same , find d_1

$$d_1 = \sqrt{\frac{4A_0}{\pi}} \qquad \text{Eq 5.10}$$

Blade Inlet Angle (β_1)

For axial inlet of pump, swirl is assumed to be zero, meaning $\square_1 = 90^o$

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 so $c_{m1} = c_1$

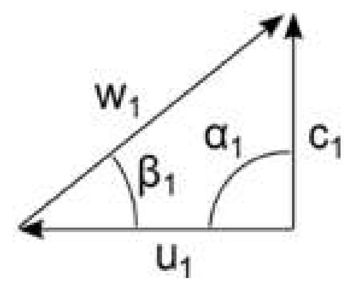


Figure 5.5 Inlet Velocity Triangle

$$\tan \beta_1 = \frac{C_{m1}}{u_1} \qquad \text{Eq 5.11}$$

The velocity u, ie the peripheral velocity can be determined. It requires speed n in rpm and impeller inlet diameter. Both values are found by previous equations, substitute the same and find velocity, u

$$u_1 = \frac{\pi d_1 n}{60} \qquad \text{Eq 5.12}$$

So from the equation 5.12 we find u_1 and from equation 5.8 we found C_{m1}

Now by rearranging Equation 5.11 we can see

$$\beta_1 = \tan^{-1}\left(\frac{C_{m1}}{u_1}\right) \qquad \text{Eq 5.13}$$

So just substitute the Cm1 and u1 and find Blade inlet angle

Fluid Velocity Relative to the Impellers

So from figure 5.6 we can see

Cos β_1=u_1/w_1

We need to find w_1 ie fluid velocity relative to the impellers

So by rearranging we get

w_1=u_1/ Cos β_1 Eq 5.14

RESULTS

- n is pump rotational speed (rpm) = 1450rpm, 2900rpm
- d_2 Approximate Relative Diameter $d_2 = \frac{1}{\pi N}\sqrt{g\Delta H / \psi}$
- Brake Horse Power(BHP)

$$BHP = \frac{WHP}{\eta}$$

- Shaft Torque at N rpm

$$T = 9552 * \frac{P(kW)}{Rpm}$$

- Diameter of shaft d_{sh}

$$d_{sh} = \sqrt[3]{\frac{16T}{\pi s_s}}$$

- Inlet meridional velocity c_m

$$c_{m_1} = K_{cm1}\sqrt{2gH} \text{ /s}$$

- The inlet cross section area A_0

$$A_0 = Q^0 / u_1$$

- The inlet diameter d_1

$$d_1 = \sqrt{\frac{4A_0}{\pi}}$$

- Blade inlet angle β_1

$$\beta_1 = \tan^{-1}\left(\frac{C_{m1}}{u_1}\right)$$

- The inlet peripheral velocity of the impeller u_1

$$u_1 = \frac{\pi d_1 n}{60}$$

- The inlet fluid velocity relative to the impeller w_1

$$w_1 = u_1 / Cos(\beta_1)$$

CONCLUSION

In this project till now a brief study of history of Low flow/ High head pumps are done. A broad study on the design of pumps was done in which the design of casing and impeller was separately taken. The design includes points from two design method berman method and stepanoff way of mathematical designing. To design a centrifugal pump impeller a procedure is proposed. The design procedure leads to good results in a lesser time. From the calculations we got the impeller diameter and inlet conditions. From studies we saw that a radial volute casing will be appropriate to be designed to have a more average pressure fluctuation intensity. As we have to go for High head, we studied that the outlet blade angle has a large influence on head. The effect of the forward swept vane, radial and backward swept vane were studied and found a radial and a forward swept blading can achieve a higher head. The methodology includes five steps, where in this paper we deal with only study of design, Specification of pump and design calculations.

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