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Designing an optimally efficient impeller for a submersible turbine pump

Arjun Asudani
arjunasudani@gmail.com
Dhirubhai Ambani International School, Mumbai, Maharashtra

ABSTRACT

This paper aims to tackle the problem of designing an optimally efficient impellers for SUBMERSIBLE TURBINE PUMP (being developed first time in India under the 'Make In India Aatmanirbhar' scheme by Govt. of India), while taking into account several factors like the material of the impeller and its components. The paper suggests a mechanical solution to the problem that is practical and efficient, involving a constructive overview of the design process of each component in the impeller –the number of vanes and the outlet angle, the shut off head, the diameter, the width, the eye diameter, the shaft diameter and more. This paper presents the construction and function of each component of the proposed system and illustrates the same using graphs and three-dimensional computer-generated models. It also takes into account the effectiveness of the material used in the construction of the impeller, whilst articulating safety requirements and ensuring that the design abides by them. Finally, it concludes by summarizing the effectiveness of the proposed design specifications.

Keywords: *Impeller, Outlet Angle, Shutoff Head, Shaft Diameter, and Polyoxymethylene.*

1. INTRODUCTION

In simple words, efficiency is essentially how well a system can convert one form of energy to another. Its importance stems from the fact that efficiency prevents the loss of energy in a system, and therefore allows any mechanical device to execute a better performance or deliver a greater output. For a mechanical system like that of a submersible turbine pump, a greater efficiency value will generate a greater conversion rate of kinetic energy to pressure energy. An l submersible turbine pump is a multi-stage stacked centrifugal pump whose stages are determined by bottom hole pressure and desired flow rate, and whose energy conversion is executed by impellers.

The world's pump market is now achieving the emergence of intelligence pump system and micro disc pump technology. Hence, it becomes necessary for India to develop creative, innovative ideas to improve pump systems and their designs. Recently, in India, submersible pump has gained a large market share in both the industrial and domestic sector. Due to their high versatility and reliability, they need no priming and don't cause cavitation issues. As of now, India is developing and manufacturing submersible pumps only for water use only but the impeller proposed in the following paper is for a submersible Turbine Pump used in dispensing motor fuel (Petroleum). The impeller design becomes the most challenging part to make this a highly efficient pump as it is the component most responsible for the pump's performance. Therefore, all possible care must be taken in designing and maintenance of impellers, which has got direct relation to the efficiency of pump.

2. SUBMERSIBLE TURBINE PUMP (STP) DESCRIPTION

The STP is designed for use only at facilities dispensing motor fuel. The Submersible Turbine Pump (STP) is responsible for driving fuel from the storage tank, through the piping infrastructure and into the vehicle through the use of pressure energy. It optimizes fuel flow and dispensing and its advanced manifold design makes it the industry's easiest and safest STP to install and service. STP has built-in check valve, switching valve, safety valve & vacuum valve. The submersible pump can operate several nozzles simultaneously & can be for petroleum products such as gasoline, kerosene, diesel etc.

3. PROJECT CONSTRAINTS

Majority of project constraints revolved around the design of the impeller. These include
1. Size of impeller to be assembled to the designed motor of 1.5HP at 2850 rpm using single phase power supply and across stages.

2. Impeller had to be Anti-static due to the flameproof nature of the pump.
3. Design of impeller involved achievement of desired efficiency to be able to fit into the pump capsule.
4. Selection of material that are flameproof in nature along with physical properties that provide longevity.

4. PROBLEM STATEMENT AND SPECIFICATIONS

Ideally, mechanical engineers should be able to utilise centrifugal pumps according to their desired customer requirements. This would require each component of the pump, especially the impeller, to have an efficiency that generates results for the requirements in an economically feasible manner. Usually, this requires a pump to have a hydraulic efficiency of about 65% or more, which can become very difficult to execute in most cases.

Designing a **three-stage** impeller of this calibre will require companies to take into account several customer specifications. Let us assume the following specifications –

- Head (i.e. height to which the liquid has to be pumped) = 18 metres.
- Capacity of the pump required by the customer = 250 litres per minute (LPM) = $\frac{250}{16.67 \text{ X } 3600}$=0.0041658335≈ 0.004$m^3$/sec
- Rotations per minute (rpm) = 2850
- Liquid to be pumped = M.S (Motor Spirit)
- **Specific Gravity = 775 kg/m3**
- Single phase motor

The following paper will deal with how, using the specifications above, the design of the impeller can provide an optimal hydraulic efficiency of 69%. The paper aims to delineate the construction of such a system; one that is built entirely using mechanical components and designed to be completely self-sufficient. This will allow for economic efficiency for any company as well, given the fact that the useful output is a large value.

5. DESIGN CONSIDERATION

There are various factors or design considerations which influence the design of the element or, perhaps, the entire system. These factors have to be considered in any design problem. In a given design problem, It's need to identify the various design considerations and incorporate them in the design process in their order of importance.

Some of the important factors which influence the design, are as follows :

- ➢ It should have more Mechanical strength.
- ➢ It should be Anti-friction.
- ➢ It should be Anti-erosion and Anti-corrosion.
- ➢ It should be Anti-static.
- ➢ Density should be low.
- ➢ It should have high thermal stability.

6. DESIGN PROCESS

6.1. Definition Problem

Impeller have to give output 250LPM at 18m of head with 69% of efficiency. To design Impeller to achieve high efficiency the biggest challenge is size and weight constraints. It has to fit in 4" (INCH) size pump with limited space of length.

6.2. Determination of type of Impeller

To design high efficient pump the selection of type of Impeller becomes important factor. There are three types of impeller, Closed impeller, Semi-open impeller and Open impeller. Closed impeller can reach a high efficiency with a high head. So selecting closed type of impeller is perfect to achieve high efficiency.

6.3 Analysis of Impeller components

To illustrate the functionality of each component within the impeller in terms of its mobility, a 3D render of the design was created.

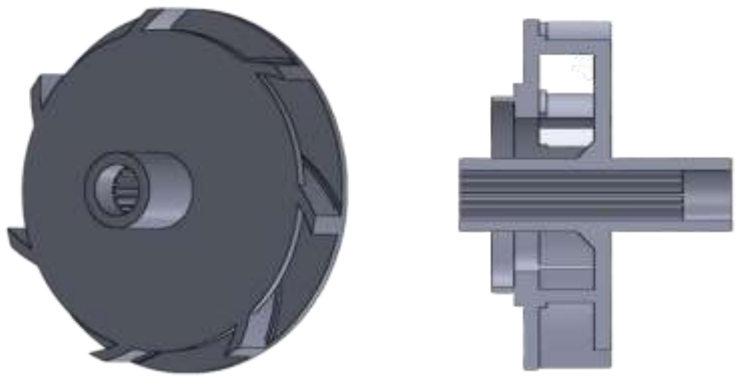


Figure 1: 3D render of impeller

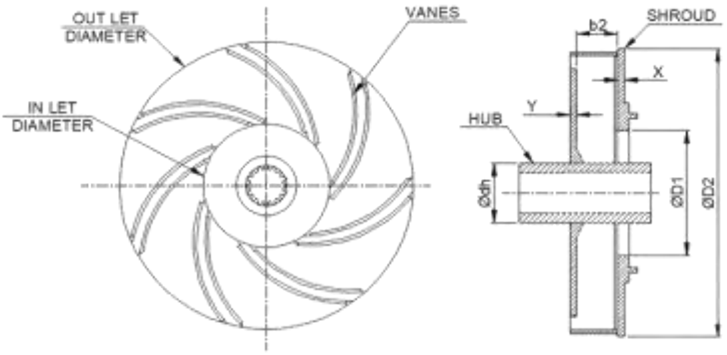


Figure 2: Cross sectional of impeller

Here,
ϕD1 = Impeller eye (inlet) diameter (in mm)
ϕD2 = Impeller outlet diameter (in mm)
ϕdh = Impeller hub diameter (in mm)
X, Y = Impeller shroud thickness (in mm)
b2 = Impeller width (in mm)

6.4 Working and Analysis of Three-Stage Impeller

Figure 3: 3D render of STP Motor Assembly

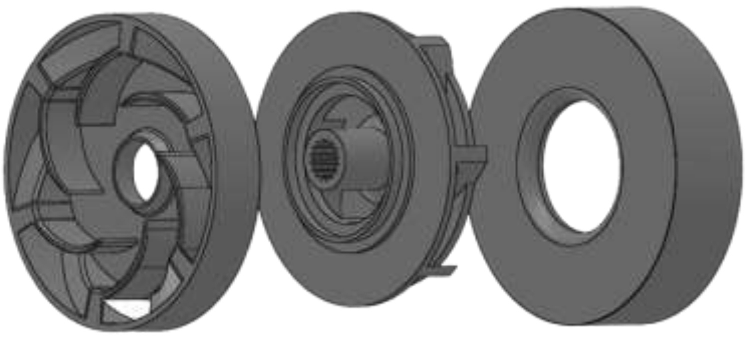


Figure 4: 3D Image of Diffuser, Impeller and Cover Plate

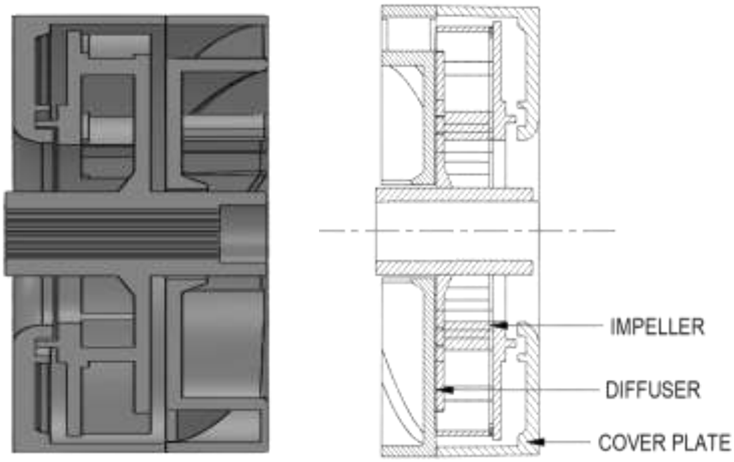


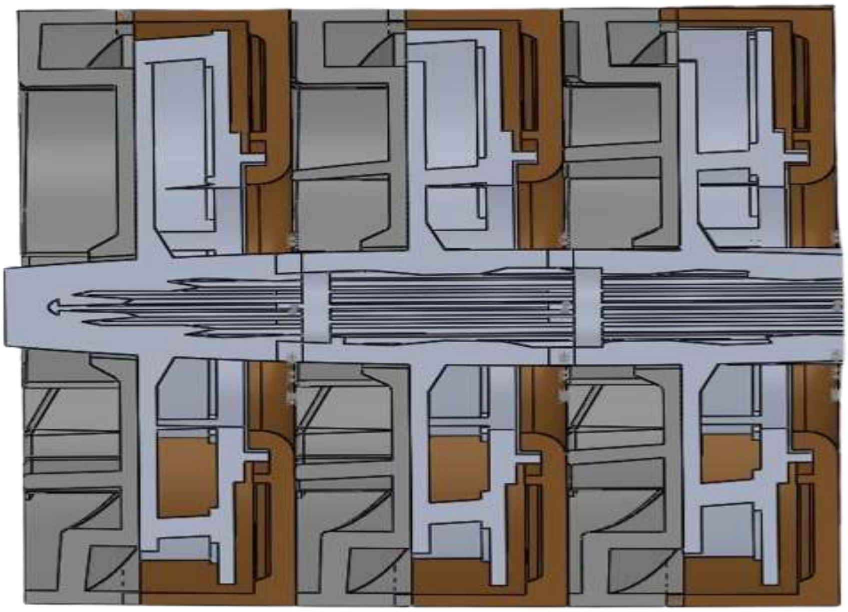
Figure 5: Assembly of Diffuser, Impeller and Cover plate

The Submersible Turbine Pump works on the same principle as a centrifugal pump. Three impellers are connected to a single shaft and this shaft is driven by an induction motor. The liquid enters through the eyes of the impeller and it is then thrown out radially due to centrifugal action. This way the liquid particles gain kinetic energy. Liquid needs to pass through the 1st stage impeller to the 2nd stage impeller efficiently. A stationary device called a diffuser is used for this purpose and the stationary cover plate, which covers the impeller, guides the liquid through the diffuser efficiently. The liquid flows through impeller eyes and is thrown by its outlets towards the diffuser, which converts kinetic energy to flow energy. The liquid flows from the impeller and enters the diffuser; the diffuser then deflects the inlet liquid and makes it ready for the next impeller stage. The next impeller is connected to outlet of the diffuser. The series of connected impeller is multiplies the pressure gain at each stage. This is the reason multistage submersible pump produce huge amount of pressure head.

Generally, a centrifugal pump driven by an induction motor. In this submersible pump, an induction motor is used to run the impellers. The power supplied to an induction motor is given to the stator which is in single phase power. The motor produces immense heat during operation and due to this reason, the motor is either cooled by product (pumped liquid) or oil.

The entire impeller, diffuser, cover plate, and motor assembly are immersed in working fluid. This means the priming is not needed in a submersible pump and due to this, no cavitation occurs. This way the liquid is pumped by the submersible motor and is thrown towards (1 ½”) discharge end.

3 important factors to consider:
- To achieve high head with high or low discharge the impellers are always connected in series.
- Head depends upon the diameter of the impeller.
- Discharge capacity depends upon the width of impeller (width between both shroud).



Stage 1 Stage 2 Stage 3

Figure 6: 3D render Assembly of three stage impeller

7. SELECTION OF MATERIAL

The STP is used for petroleum products and hence must flame proof. It becomes necessary to select a material with low water absorption and high dielectric strength – so the pump can deal with temperature and friction. Polyoxymethylene (POM) is an engineering thermoplastic used in precision parts that require high stiffness, low friction and excellent dimensional stability – making it the ideal material for the impeller.

Properties of Polyoxymethylene that make it the most feasible material:
- Excellent short-term mechanical properties such as tensile strength, toughness and rigidity.
- Low tendency to creep and fatigue.
- Low linear coefficient of thermal expansion.
- Tendency to maintain good mechanical and electrical properties at 140° C (for short time) 90° C (for long time).
- Good mechanical properties over a large range of temperatures (up to -40° C).
- Excellent resistance to most of chemicals, organics solvents, and fuels at room temperature.
- Good dimensional stability and environment stress-cracking resistance.
- Low permeability and water absorption.
- High hardness, a glossy and smooth moulded surface.

8. SELECTION OF TYPE OF VANES

There are three types of impeller vanes: Forward-inclined, radial and backward-inclined vanes. In a centrifugal pump, the backward-inclined vanes are most common. These yield the highest efficiency of the three because fluid flows into and out of the vanes with the least amount of turning.

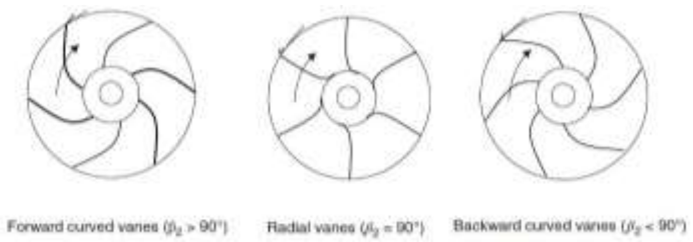


Figure 6: Exit velocity diagrams

9. DESIGN CALCULATION OF IMPELLER

9.1 Calculation of power absorbed by pump

- Technical parameters:
- Discharge (Q) = 250 LPM
- Head (H) = 18 Meters
- Speed = 2850 RPM
- Efficiency (ή%) = 69%
- Liquid = M.S
- Specific gravity = 775 kg/m3 = 0.775 kh/dm3
- Power input = Single phase
- No. of stages = 3

$$P = (Q \times H \times S.G)/(1.34 \times 45 \times ή\%) \text{ kw}$$

$$\therefore P = (250 \times 18 \times 0.775)/(1.34 \times 45 \times 69) \text{ kw}$$

$$\therefore P = 0.83 \text{ kw} \text{-------------------- (1)}$$

Therefore, the power absorbed by the pump is 0.83 kw.

9.2 Design of vane

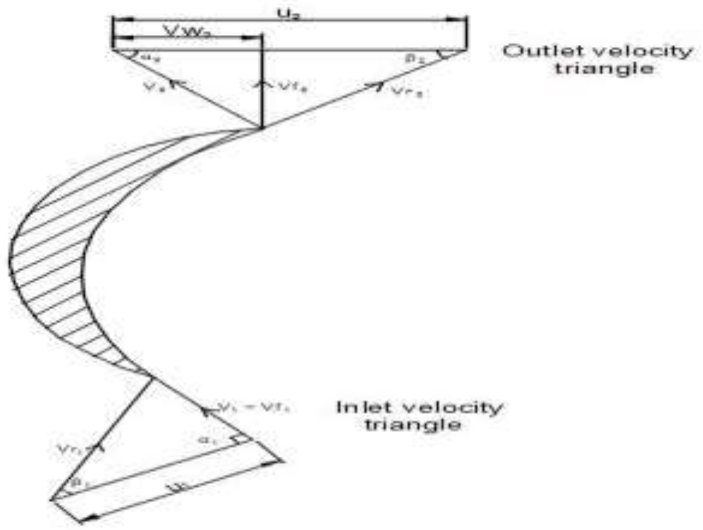


Figure 7: Vane design diagram

These velocity diagrams are drawn in order to shown the motion and velocity of a fluid particle while flowing through an impeller. It is drawn for the STP to determine the various velocities of the liquid during the discharge of the pump.

Here (1 = For inlet and 2 = For outlet):

- V_1 = Actual Inlet velocity
- V_{f1} = Flow velocity
- V_{r1} = Relative velocity
- u_1, u_2 = Tangential velocity of impeller (Blades / Vanes velocity) inlet and outlet.
- V_1, V_2 = Absolute velocity of liquid at inlet and outlet
- V_{r1}, V_{r2} = Relative velocity of liquid with respect to vanes velocity at inlet and outlet
- V_{f1}, V_{f2} = Flow or radial component of absolute velocity at inlet and outlet
- V_{w1}, V_{w2} = Whirl component of absolute velocity at inlet and outlet
- α1, α2 = Angle with which liquid is leaving the impeller
- β1**,** β2 = Vane angle at inlet and outlet

9.3 Calculation of D1 (Inner diameter) and D2 (Outer diameter)

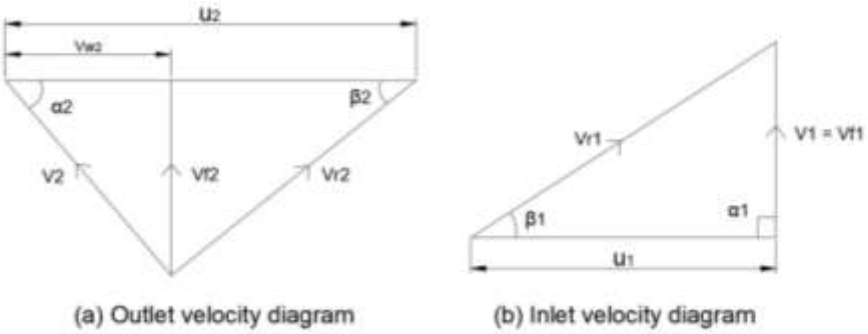


Figure 8 (a) and (b): Outlet and Inlet Velocity Diagrams

We know that impeller velocity at outer periphery is:

$$U2 = ku2\sqrt{2.g.Hm}$$

Here, ku2 = Slip factor = 0.95 to 1.25
0.95 = For low speed
1.25 = For high speed
g = Acceleration due to gravity = 9.81 m/s^2

Total Head to develop by pump is 18m. Since this is the three stage pump the manometric head is divided by 3 Hm

$$Hm = \text{Manometric head (in meters)} = \frac{18}{3} = 6 \text{ meters}$$

$$U2 = 1.25\sqrt{2 \times 9.81 \times 6}$$

$$u2 = 13.56 \text{ m/s} \quad \text{-------------------- (2)}$$

Also,

$$u2 = \frac{\pi \times D2 \times N}{60}$$

$$D2 = \frac{60 \times u2}{N \times \pi}$$

$$D2 = \frac{60 \times 13.56}{2850 \times 3.14}$$

$$D2 = 0.09091 \text{ m}$$

$$D2 = 91 \text{ mm} \quad \text{-------------------- (3)}$$

Outer diameter of the impeller is 91 mm.

Now final,

$$u2 = \frac{\pi \times D2 \times N}{60}$$

$$u2 = \frac{\pi \times 0.091 \times 2850}{60}$$

$$u2 = 13.57 \text{ m/s} \quad \text{--------------------(4)}$$

Normally ratio of outlet diameter to inlet diameter is 2 to 2.5.

$$\frac{D2}{D1} = 2 \text{ to } 2.25$$

For high speed ratio is 2.25.

$$D1 = \frac{0.091}{2.25}$$

$$D1 = 0.040 \text{ Meter}$$

$$D1=40\ mm \qquad \text{--------------------- (5)}$$

Therefore, the inlet diameter of impeller is 40mm.

9.4. Blade / vane design

- **Inlet condition : No pre-whirl**

$$\alpha 1 = 90°$$

$$Vw1 = 0$$

$$V1 = V_{f1} = V_s$$

Where V_s = Suction velocity
Ds = Inlet diameter of impeller / suction pipe

Calculation of suction velocity:
Discharge (m3/s) = Area of suction pipe x velocity

$$0.004 = \frac{\pi}{4} X\ ds^2\ X\ Vs$$

$$0.004 = \frac{\pi}{4} X\ (0.04)^2\ X\ Vs$$

$$Vs = 3.18\ m/s \qquad \text{------------ (6)}$$

Therefore,suction velocity is 3.18 m/s

- Exit condition : $\beta 2 < 90°$
- From inlet velocity triangle:

$$u1 = \frac{\pi\ X\ D1\ X\ N}{60}$$

$$u1 = \frac{3.14\ X\ 0.040\ X\ 2850}{60}$$

$$u1=5.96\ m/s \qquad \text{-------------------- (7)}$$

Using trigonometric function we get,

$$\tan\beta 1 = \frac{V1=Vf1=Vs}{u1}$$

$$\tan\beta 1 = \frac{3.18}{5.96}$$

$$\tan\beta 1 = 0.536$$

$$\beta 1 = \tan^{-1}(0.536)$$

$$\beta 1 = 18.5°$$

$$\beta 1 = 20° \qquad \text{-------------------- (8)}$$

This value is increased slightly to account for the contraction of the stream as it passes the inlet edges and the pre-rotation of liquid.

- **From outlet velocity triangle**

Using trigonometric function we get,

$$\tan\beta 2 = \frac{Vf2}{u2-Vw2}$$

$$Vf2 = kf2\sqrt{2.g.Hm}$$

Where kf2 is the flow coefficient at outlet = 0.16 to 0.25

$$Vf2=0.25\sqrt{2\ X\ 9.81\ X\ 6}$$

$$Vf2=2.71\ m/s \qquad \text{-------------------- (9)}$$

Manometric (manufacturer standard) efficiency is 69%.

$$ή_{mano} = \frac{Hm}{(\frac{Vw2 \text{ X } u2}{g})}$$

$$0.69 = \frac{6}{(\frac{Vw2 \text{ X } 13.57}{9.81})}$$

$$Vw2 = 6.28 \text{ m/s} \quad \text{--------------------} \ (10)$$

Therefore,

$$\tan\beta 2 = \frac{2.71}{13.57-6.28}$$

$$\tan\beta 2 = 0.371$$

$$\beta 2 = 20.4°$$

$$\beta 2 = 25° \quad \text{--------------------} \ (11)$$

β2 is always larger than β1 (25° > 20°).
Therefore,

$$\tan\alpha 2 = \frac{Vf2}{Vw2}$$

$$\tan\alpha 2 = \frac{2.71}{6.28}$$

$$\tan\alpha 2 = 0.431$$

$$\alpha 2 = \tan\ (0.431)$$

$$\alpha 2 = 23.3°$$

9.5 Calculation of no. of blades / vanes
This can be calculated by using the formula below

$$\text{No. of blade (Z)} = 6.5 \text{ X } [\frac{D2+D1}{D2-D1}] \text{ X } \sin(\frac{\beta 1+\beta 2}{2})$$

$$Z = 6.5 \text{ X } [\frac{91+40}{91-40}] \text{ X } \sin(\frac{20+25}{2})$$

$$Z = 6.27 \text{ ; however } Z > 6.27$$

$$\therefore Z = 7$$

Therefore, total no of vanes in impeller will be 7.

9.6 Calculation of width of impeller
Thickness of vanes is as per manufacturer standard

$$t1 = 5 \text{ to } 8 \text{ mm}$$

$$t2 = 4 \text{ to } 6 \text{ mm}$$

The shroud thickness can be assumed as 2 mm as each shroud, with close consideration given to safety and longevity.

9.7 Calculation of radius of curvature

$$\text{Radius of curvature} = \frac{R2^2 - R1^2}{2[R2\cos(\beta 2) - R1\cos(\beta 1)]}$$

$$= \frac{(45.5)^2 - (20)^2}{2[45.5\cos(25°) - 20\cos(20°)]}$$

$$= 37 \text{ m}$$

9.8 Calculation of shaft diameter under impeller eye
Material of impeller = Polyoxymethylene (POM)

Tensile strength / Yield stress = 62 N/mm2

$$\therefore \quad \sigma y = 62 \text{ N/mm2}$$

$$\sigma t = \frac{\sigma y}{FOS}$$

Where, σy = Yield stress

σt = True stress

FOS = Factor of safety = 4

τ = Shear stress

$$\sigma t = \frac{62}{4}$$

$\therefore \quad \sigma t = 15.5$ N/mm2

Now to find shear stress, formula we have,

$$\tau = \frac{\sigma t}{2}$$

$$\tau = \frac{15.5}{2}$$

$$\therefore \quad \tau = 7.75 \text{ N/mm2}$$

Methodology: Evaluate shaft diameter based on twisting moment initially and then check for actual stress using "exact estimation"

This is the calculation of the torque is transmitted to the shaft. It is denoted by mt.

$$P = \frac{2\pi Nmt}{60}$$

Here,

P = Power absorbed by pump (in watts) = 0.83 Kw

N = Speed of pump (in RPM) = 2850 RPM

mt = Torque (in Nmm)

ds = Shaft diameter (in mm)

$$\therefore P = \frac{2\pi Nmt}{60}$$

$$\therefore 0.83 \text{ X } 10^3 = \frac{2 \text{ X } 3.14 \text{ X } 2850 \text{ X } mt}{60}$$

$$\therefore mt = \frac{0.83 \text{ X } 10^3 \text{ X } 60}{2 \text{ X } 3.14 \text{ X } 2850}$$

$$\therefore mt = 2.78 \text{ X } 10^3 \text{ Nmm}$$

Considering torsional failure,

$$mt = \frac{\pi}{16} \text{ X } \tau \text{ X } ds^3$$

$$\therefore ds^3 = \frac{2.78 \text{ X } 10^3 \text{X } 16}{3.14 \text{ X } 7.75}$$

$$\therefore ds = 12.22 \text{ mm} \sim 13 \text{ mm}$$

This diameter is calculated to the basis of shear with the help of low allowable stresses to compensate for the uncertainties in the total value of the load. Assuming a safety factor of 10% on the shaft diameter, a shaft diameter of 13 mm is chosen.

The hub diameter dh (as in the figure shown earlier) should be 10-20% more than the shaft diameter. This is to prevent failure of the impeller which can occur when the hub shears due to the torque transmitted by the shaft. Therefore, in order to avoid any such eventuality which will cause a complete breakdown of the pumping operation, the size is increased. Another important factor affecting the strength of the hub diameter is the size of the keyway of the impeller. Thus, it is increased by 10 – 20% of the shaft diameter.

Therefore,

Hub diameter = 1.5 X 13

= 19.5 mm.

9.9 Calculation of shaft impeller eye area

Eye area = Area at impeller eye – shaft area

$$= 3.141 \text{ X } (R_1)^2 - 3.141 \text{ X } (\tfrac{ds}{2})^2$$

$$= 3.141 \text{ X } (45.5)^2 - 3.141 \text{ X } (6.5)^2$$

$$= 6365 \text{ mm}^2$$

9.10 Calculations of Impeller Width

$$b_2 = \frac{\text{GPM X } 0.321}{C_{m2} \text{X } CD_2H\text{-}ZS_u}$$

where b_2 = impeller width

GPM = LPM/3.8

Z = no. of valves chosen

S_u = thickness of impeller blade

= 1/2" (Usually as per earlier designs)

Now, C_{m2}= outlet velocity of liquid

$$C_{m2} = K_{m2} \text{ X } \sqrt{2gh}$$

K_{m2} = capacity constant

K_{m2} is the constant considering frictional losses in the valves.

Therefore, from the graph shown below

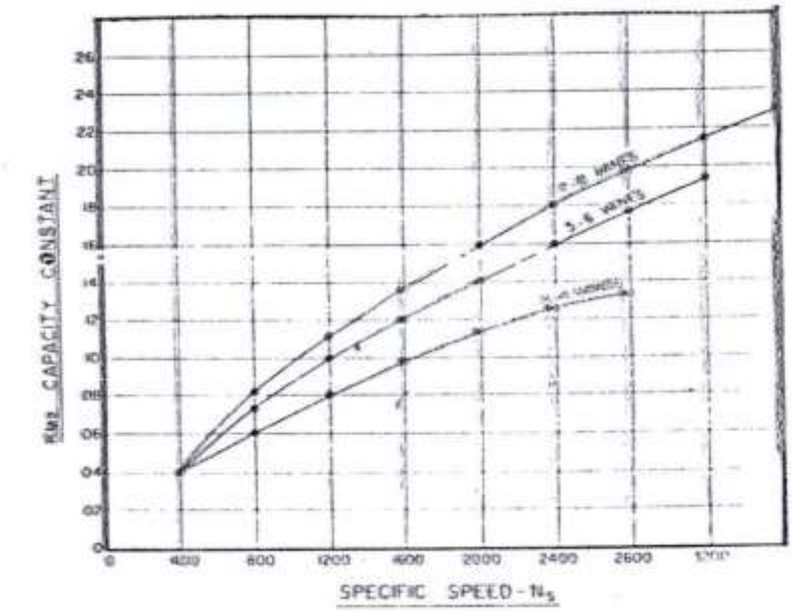


Figure 5: Specific speed vs capacity constant curve

Value of K_{m2} = 0.0665

$$C_{m2} = K_{m2} \text{ X } \sqrt{2gh}$$

g = acceleration in feet/second2

g = 9.81 mts / second2

$$= 9.81 \text{ X } 3.28 ft/s^2 = 32.1768 \text{ ft/s}^2$$

H = head in feet = 7 X 3.28 = 22.96

$$C_{m2} = 0.0665 \text{ X } (2 \text{ X } 32.2 \text{ X } 22.96)^{0.5}$$

= 2.557116537 ft/sec

Therefore, outlet velocity,

$$C_{m2} = 2.557116537 \text{ ft/sec}$$

$$\text{Now impeller width } b_2 = \frac{(GPM \times 321)}{C_{m2} \times (3.141 \times D_2 - Zs_u)}$$

$$GPM = LPM / 3.8$$

$$= 250 / 3.8$$

$$= 65.78947386$$

$$\textbf{Therefore, } b_2 = \frac{65.78947386 \times 321}{2.557116537 \times (3.141 \times 18.335 - 4 \times 0.5)}$$

$$= 0.97” = 24.85mm$$

10. CONCLUSION

From the design of the impeller proposed, we can conclude that the optimal impeller for a submersible turbine pump must be made of polyoxymethylene, must have a outlet diameter of 91mm, an inlet diameter of 40mm, 7 vanes, a shaft diameter of 13mm, and an impeller eye area of approximately 6365 mm^2.

11. REFERENCES

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