Design of Solar Powered Submersible Pump

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Abstract- *Agricultural technology is changing rapidly. Farm machinery, farm building and production facilities are constantly being improved. Nowadays, Solar Power is used for many applications in agricultural field. Solar powered water pumping systems have become the interest of many people in the recent years. Acknowledging that nature has provided a bounty of energy which can be converted into electrical energy has created innovative ways of discovering materials that can be used to make a system that supports turning heat into electricity.*

The idea of this project is to create a solar panel system that would be used with centrifugal submersible pump. A submersible pump is very useful type of pump to lift the underground water , so a submersible pump is selected. Pumps which are available in the market are costly as well as of having high power, so the pumps that are available are not appropriate for small scale farming . So, the purpose was to design a pump which can work on low power but should have a high head. The solar system is also selected for minimum cost .Solar systems includes solar panels and charge controller with effective working and also with minimum cost.Solar water pumping system operate on direct current. The output of solar power system varies throughout the day and with changes in weather conditions. Photovoltaic module, the power source for solar pumping, have no moving parts, requires no maintenance and last for decades. A properly designed solar pumping system will be efficient, simple and reliable.

Keywords- Solar power, Submersible ,Centrifugal, Pump, Impeller Design, Design Calculations, Solar Panel System.

I. INTRODUCTION

This Centrifugal pumpis of a very simple design. The two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving part, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, polycarbonate, cast iron, stainless steel as well as other materials. The diffuser (also called as volute)houses the impeller and captures and directs the water off the impeller. Water enters the center (eye) of the impeller and exits the impeller with the help of centrifugal force. As water leaves the eye of the impeller a lowpressure area is created, causing more water to flow into the eye. Atmospheric pressure and centrifugal force cause this to happen. Velocity is developed as the water flows through the impeller spinning at high speed. The water velocity is collected by the diffuser and converted to pressure by specially designed passage ways that direct the flow to the discharge of the pump, or to the next impeller should the pump have a multi-stage configuration. The pressure (head) that a pump will develop is in direct relationship to the impeller diameter, the number of impellers, the size of impeller eye, and shaft speed. Capacity is determined by the exit width of the impeller. The head and capacity are the main factors, which affect the horsepower size of the motor to be used. The more the quantity of water to be pumped, the more energy is required.

II. ADVANTAGES OF CENTRIFUGAL SUBMERSIBLE PUMP

- These pumps are extremely efficient
- A lot of the energy can be conserved
- Handles solids & liquids
- These pumps become quiet in the water
- These pumps are versatile
- No priming is necessary
- Greater efficiency
- Both solids and liquid are managed easily
- Internal pressure is enough for pumping of water inside and no additional components are required.
- When the water flows throughout the pump, then cavitation doesn‟t occur due to no "spike" within the water force.

III. AIM

- To study the existing submersible pump available in the market.
- To increase the head of the pump of 1 HP submersible pump.
- To design and manufacture solar powered submersible pump in low cost.

IV. LITERATURE REVIEW

In order to know more about the pump we referred various research papers which gave us idea of the previous work done in this field.

We referred research paper „Design , development & testing of an impeller for performance improvement‟ by Paint m. patil , Sangram s. patil which was devoted to focus on water pump design by the modern development of CFD also research devoted to use theoretical analysis in combination with CFD concepts foe determination of optimum value of blade angle for max. Pressure head & discharge for water head.

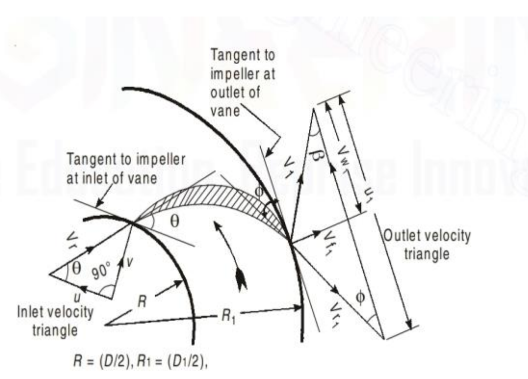
We referred research paper Improving efficiency of4"submersible pump using CFD analysis. This project deals with modification in existing design of radial flow impeller. After modelling the impeller they calculated the equivalent theoretical efficiency so as to prove that the impeller that designed newly is of better design than previous one. The sole objective was to increase efficiency of the pump. We referred research paper „Design , construction & performance evaluation of submersible pump with numerical experimentation‟ by ErtanEngin (middle east tech. institute). In this thesis ,non clog pump is designed & constructed on the basis of suitable approach of known pump design. Close agreement between results of actual test & num. experiment performed by CFD code shows CD analysis is quite useful tool is predicting the hydraulic characteristics of non clog pumps.

We referred „Design & optimization techniques in horizontal open well submersible pump‟ This project deals with modification in existing design of radial flow impeller made out of plastic material in many case may be called rotor. By modifying the design of impeller .They could achieved considerable increase in o/p discharge which will be shown in their work.

We referred „Performance influence in submersible with different diffuser inlet width‟ byQingshun Wei ,Xihan Sun . Study is based on diffuser inlet width which is key geometric parameter that affects performance of submersible pump. Diffuser with different inlet widths& same impeller were equipped to construct submersible pump model through autocad software. The performance curves of the submersible pump ,with six diffuser inlet width were obtained using computational fluid dynamics method as well as simulation results were tested with experimental method are presented in article.

1) IMPELLER PROFILE

The expression for the work done (or the energy supplied) by the impeller of a centrifugal pump on the liquid flowing through it may be derived in the same way as for a turbine. The liquid enters the impeller at its centre and leaves at its outer periphery. Figure shows a portion of the impeller of a Centrifugal pump with one vane and the velocity triangles at the inlet and the outlet tips of the vane.For the sake of convenience the same system of notation is employed as that for turbines. Thus, V is absolute velocity of liquid, u is peripheral (or tangential) velocity of the impeller Vr is relative velocity of liquid, Vf is velocity of flow of liquid, and Vw is velocity of whirl of the liquid at the entrance to the impeller. Similarly V1, u1 Vr1, Vf1 and Vw1 represent their counterparts at the exit point of the impeller.Further θ represents the impeller vane angle at the entrance and φ represents the impeller vane angle at the outlet. Similarly α is the angle between the direction of the absolute velocity of entering liquid and the peripheral velocity of the impeller at the entrance, and β is the angle between the absolutevelocity of leaving liquid and the peripheral velocity of the impeller at the exit point.At the entrance to the impeller since there are no guide vanes (as in the case of turbines) thedirection of the absolute velocity of liquid at this point of the impeller is not directly known. However,for best efficiency of the pump it is commonly assumed that the liquid enters the impeller radially thatis the absolute velocity of the liquid at the entrance to the impeller (or at the inlet tip of the impellervane) is radial in direction. Thus, in this case α = 90° and the velocity of whirl Vw at inlet is equal tozero.



Further it is desired that the liquid enters and leaves the vane without shock. This can be ensured if the inlet and outlet tips of the vane are parallel to the direction of the relative velocities at the two tips. As such it is assumed that the relative velocities Vr and Vr1 are parallel to the tangents to the vane at the inlet and outlet tips respectively as shown in Fig 3.1

Velocity Triangle for impeller.

2) DESIGN CALCULATION FOR IMPELLER

- Design centrifugal submersible pump for following specifications
  - Total static head = 20m
  - Discharge = 165 lpm = 0.00275 m^3/s

Step 1) Determination of manometric head

- Total static head = 20m
- Discharge = 165 lpm = 0.00275 m^3/s

* As per standard reference static suction head should be greater than 10% of total head.

Hs= hs +hd..........(1)

- Let hs = 0.1 × Hs = 0.1 × 20 = <u>2 m</u>

Hence hd= 20-2= <u>18m</u>

*Length of suction pipe and delivery pipe are not given then assume,

- Length of suction pipe as

Ls = 1.4 × hs = 1.4 × 2 = <u>2.8 m</u>

Ld = 1.4 × hd = 1.4 × 18 = <u>25.2 m</u>

- Selection of pipe diameters

Suction pipe:-

Q= As × Vs……….(2)

As = Area of suction pipe

Vs = velocity of liquid in suction pipe

*As per standard pump references in order to avoid cavitation Vs< 3.5 m/s

- Let us select Vs = 1.5 m/s

Q = $\pi/4$ Ds^2 × V………..(3)

Ds= 0.04831 m = 48.31 mm ≈ 50 mm

Hence , selecting standard pipe of 50 mm

Recalculating velocity in suction pipe

Vs= 1.4m/sec < 3.5 m/sec

Hence selection is safe.

Delivery pipe :-

Q = A_d×V_d

Ad=Area of delivery pipe

Vd = Velocity of liquid in delivery pipe

*As per standard pump references in order to avoid cavitation vd< 4m/sec

Vd= 3.5 m/sec

Q = π/4 Dd^2 × Vd

Dd=0.03162 m = 31.63 mm ≈ 32 mm

Recalculating Vd

Vd= 3.42 m/sec < 4 m/sec

Hence is safe.

- Manometric head

Hm = Hms + Hmd………….(4)

- Hms = Static suction head + Velocity head loss + loss due to friction in suction pipe + head loss due to strainer + head loss due to bends.

Static suction head = <u>2m</u>

Velocity head loss = V_s^2 / 2g = <u>0.0998 m</u>

Loss in suction pip, hfs = $(4fL_sV_s^2)/2gD_s$=(4× 0.008 × 2 × 1.4^2)/2× 9.81 × 0.050=<u>0.1278 m</u>

Head loss due to strainer = <u>0.2 m</u> (assume)

Head loss due to bends = n × loss due to each bend = 2 × 0.1 = <u>0.2 m</u>

Hms = 2+0.0998 + 0.1278 + 0.2 + 0.2 = <u>2.6276 m</u>

- Hmd = static suction head + velocity head loss + loss due to friction in delivery pipe + head loss due to strainer + head loss due to bends

statics delivery head = <u>18 m</u>

velocity head loss = V_d^2 / 2g = <u>0.5926 m</u>

loss in delivery pipe h_{fd} = $(4fL_dV_d^2)/2gD_d$ = (4× 0.008 × 25.2 × 3.41^2)/2× 9.81 × 0.032 = <u>14.93 m</u>

Head loss in delivery valve = <u>0.2 m (</u>Assume)

Head loss due to bends = n × loss due to each bend = 4 × 0.1 <u>= 0.4 m</u>

H_{md} =18+0.5926+14.93+0.2+0.4 =<u>34.1226 m</u>

Hence ,H_m= H_{md} + H_{ms} = 2.6276 + 34.1226 = **<u>36.7502 m</u>**

Hm≈ 37 m

*Now entire design to design against head 37 m

Step 2) Motor Selection

P = ($\rho g Q Hm$)/η_0............(5)

ρ = 1000 Kg/m^3

g = 9.81 m/sec^2

Q = 0.00275 m^3/sec

H_m= 37 m

$\eta_o = \eta_{manometric} \times \eta_{volumetric} \times \eta_{Mech}$ = 0.35 × 0.95 ×0.9 = **0.7265**

P = (1000 × 9.81 × 0.00275 × 37)/0.7265

P = 1373.94 W

P = 1.37 kW

- Selection of RPM of motor

$$N_s = N\sqrt{Q}/H^{3/4} \ldots\ldots\ldots(6)$$

*N_s can be taken 15 to 40 for centrifugal pump we will found specific speeds corresponding to standard RPM values.

N= 960 RPM -N_s= 5.23

N=1440RPM - N_s = 7.98

N= 2880RPM- N_s = 15.96 (This is in the prescribed range)

*Hence selecting standard 3Ø motor of 1.5 kW at eated speed of 2880 RPM

Step – 3) Impeller Design

- Material used as cast iron

Type – Since clear water to pump selecting closed impeller.

*Peripheral velocity of impeller at exit given by

$$u_2 = k_u\sqrt{2gHm} \ldots\ldots\ldots(7)$$

Let, K_u = 1.05

H_m= 37 m

u_2= 28.29 m/sec

u_2= 28.29 m/sec = $\frac{\pi D2N}{60}$

Hence, **D_2 = 0.1876 m**

D_1 = (1/3 to 1/2) × D_2

Let **D_1 = 0.5 × D_2 = 0.0938 m**

u_1 = $\frac{\pi D1N}{60}$ = 14.14 m/sec

*Check head developed by the pump

$(u_2^2-u_1^2)/2g$ = 30.60 m <H_m (37 m)

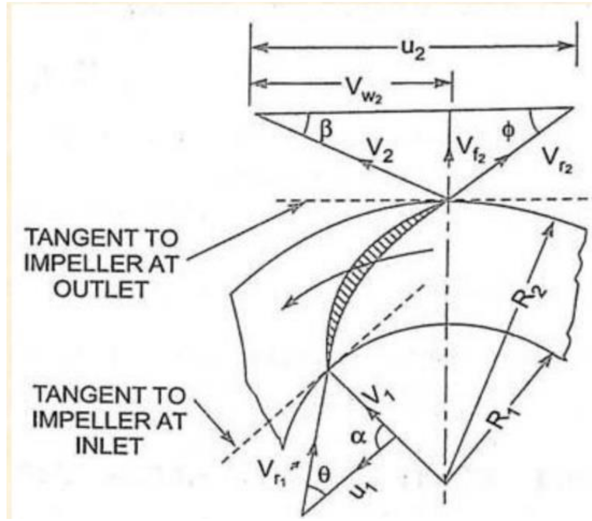
*Hence we need to increase the diameter of the impeller

Let **D_2=0.2 m ∴ u_2=31.66 m/sec**

D_1=0.105 m ∴ u_1= 15.38 m/sec

$(u_2^2-u_1^2)/2g$ = 38.31 m <H_m (37 m)

Diameter D2(m)	θ1	θ2	Width B1(mm)	Width B2(mm)	Head (m)
0.2100	5.20°	12.52°	7	2	20
0.1270	8.32°	13.03°	11	3	14
0.1016	10.35°	13.11°s	14	5	9



Step 4) Determination of velocity components

1. Inlet Vane angle

$\tan\theta_1 = v_1/u_1$

*Assuming $V_1 = V_s$ = 1.4 m/sec

$$\textbf{Tan}\ \theta_1 = V_1 / u_1 \ldots\ldots\ldots(8)$$

u_1= 15.38

θ_1= 5.20°

Whirl velocity at exit of impeller

$$\eta_{mano} = gH_m/V_{w2} x u_2 \ldots\ldots\ldots(9)$$

η_{mano}= 0.85 (assumed)

u_2= 31.66 m/sec

V_{w2}= 13.48 m/sec

2. Now velocity at exit of impeller

$$V_{f2} = \varphi\sqrt{2gHm} \ldots\ldots\ldots(10)$$

φ= 0.15

H_m= 37 m

V_{f2}=4.04 m/sec

3. Vane angle at exit

$$\tan\theta_2 = V_{f2}/u_2 - V_{w2} \ldots\ldots\ldots(11)$$

θ_2=12.52°

4. Blade angle at exit

$$\tan\alpha 2 = v_{F2}/V_{w2} \ldots\ldots\ldots(12)$$

α_2=16.68^0

5. Determination of number backward curved vanes

$$Z=6.5\left(\frac{D2+D1}{D2-D1}\right)\sin\left(\frac{\theta1+\theta2}{2}\right) \text{..........(13)}$$

Z=3

We select , Z≈6

*Calculating other parameters assuming different diameter for impeller

Let D_1 = 0.1016 m therefore ,**u_2=15.32 m/sec**

D_2 = 0.0508 m therefore , **u_1=7.66 m/sec**

Hence ,

Head will be **$H = u_2^2 - u_1^2 = 8.97$ m**

Power required will be

P = <u>0.74 kW</u>

*Calculating velocity Components

$\tan\theta_1 = v_1/u_1$

Hence , using $V_1 = V_s$ = 1.4 m/sec

$\theta1 = 10.35°$

Whirl velocity at exit of impeller

I. $\eta_{mano} = gH_m / v_{w2} x u_2$.............(9)

η_{mano}= 0.85 (assumed)

u_2= 15.32 m/sec

V_{w2}= 6.78 m/sec

1. Now velocity at exit of impeller

$$V_{f2}=\varphi\sqrt{2gH} \text{..........(10)}$$

φ= 0.15

H= 8.97 m

V_{f2}=1.99 m/sec

2. Vane angle at exit

$\tan\theta_2 = V_{f2}/u_2 - V_{w2}$...............(11)

θ_2=13.11°

3. Blade angle at exit

$\tan\alpha 2 = v_{F2}/V_{w2}$..............(12)

$\alpha_2 = 16.35^0$

$$Z=6.5\left(\frac{D2+D1}{D2-D1}\right)\sin\left(\frac{\theta1+\theta2}{2}\right)$$

Therefore , Z = 3.6 , we select Z≈6

- Width of the blade b

$Q = \pi D_1 b_1 k V_{f1}$

$b_1 = Q/\pi D_1 k V_{f1}$

<u>**b_1=13.67≈14 mm**</u>

$Q = \pi D_2 b_2 k V_{f2}$

$b_2 = Q/\pi D_2 k V_{f2}$

<u>**b_2=4.8 mm≈5 mm**</u>

*Using Different Diameters we have compared our Results below

- **Shaft design**

$P=2\pi N[M_t]/60$

P=1.5 kW

N=2880 rpm

$[M_t]=Px60/2\pi N$

- Selecting material for shaft c-40 with ....(From Psg)

δy=330 N/mm

Considering FOS=3

[δt]=110 N/mm^2

[τ] = 40 N/mm^2=40 x 10^6 N/mm^2

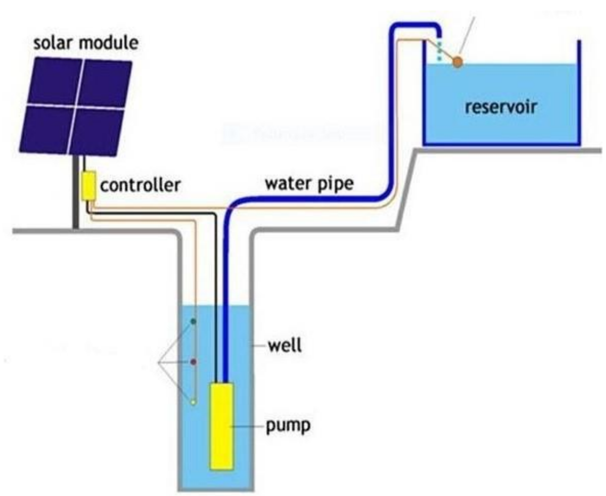
$[M_t]=\frac{\pi}{16}d_s^3[\tau]$

d_s=0.0085m = 8.5 mm

$d_{s\ exact}$ = d_2=1.3 x 8.5=11.05 mm ≈15.62 mm

Impeller hub diameterd_h=1.5 x d_2=1.5 x 15.62 = **23.43≈24 mm**

3) **SCHEMATIC DIAGRAM OF SETUP**



4) SELECTION OF SOLAR PANEL

Solar Panels (in W)	No. of panels used	Power Gereratad (in W)	Cost (in Rs)
Luminous 80 W	12	960	46680
Luminous 105 W	9	945	38682
Luminous 165 W	6	990	36138
Luminous 325 W	3	975	25550
Blue Bird 320 W	3	960	36138
Loom Solar 390W	3	1170	51000
Microtek 325 W	3	975	22425

With reference to the above table we selected three solar panels from Microtek , i.e. Microtek 325W with cost of Rs 7475 each.

5) SELCTION OF CHARGE CONTROLLER

Model Name	Specifications	Price
Luminous N*G+1600	1100VA/24V	9299/-
Microtek PCU 2035/24V	1735VA/24V	15490/-
Microtek MSUN 2035	1735VA/24V	8100/-
Microtek HB 1650	1650VA/24V	7390/-

Therefore with reference to the above table we selected Microtek HB 1650 which costs **Rs 7390**

6) BILL OF MATERIALS

Name of Company	Power (in hp)	Cost (in Rs)
Kirloskar pumps	2 to 5	540000
Padmavati Sales Corp.	2 to 5	350000
Soyo Systems	5 to 27	170000
Akhar Enterprise	10	500000
Pavlo	1	85000
Our Proposal	1	48165

V. CONCLUSION

- The impeller was designed for the input details using standard formulas; hence the result concluded that multistage design is best suitable for our purpose as it have much higher pressure capabilities than a single impeller is able to alone.
- This design can operate with tighter clearances between impellers and the casing. Hence multistage pump is able to achieve higher performance with smaller motor size and using less energy.
- The proposed solar powered submersible centrifugal pump setup is most affordable pump among the available pumps.
- This setup is designed and arranged according to the need of the farmers .

VI. ACKNOWLEDGEMENT

We take this opportunity with great pleasure to express our deep sense of gratitude towards our guide Prof. S. S. Malusare, Finolex Academy of Management and Technology, Ratnagiri for sharing their academic experience with us. His effort, availability, patience and knowhow were very valuable. His valuable suggestion helped us throughout the project and complete it on time.

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Thanking you all.

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