To Enhance Efficiency of Submersible Pump

Kishan P. Nagar[1] Jaykrushna U. Chhatbar[2] Avinash V. Parmar[3] Bhautik M. Mathukiya[4] Parag V. Vekariya[5]
[5]Assistant Professor
[1,2,3,4,5]Department of Mechanical Engineering
[1,2,3,4,5]Dr. Subhash Technical Campus, Junagadh (362001), India

Abstract— Submersible pumps are widely used for irrigation, water supply plants, sewage and oil wells because of their suitability in practically any service. Therefore it is necessary to find out the design parameters and working conditions that yield optimal output and maximum efficiency with lowest power consumption. Study indicates that Computational fluid dynamics (CFD) analysis is being increasingly applied in the design of Submersible pumps. Our objective is to increase the efficiency of Submersible pump by changing the blade angle of impeller by theoretical calculation and then comparing it with software base calculation through CFD analysis. With the aid of the CFD approach, the complex internal flows in water pump impellers, can be well predicted, to speed up the pump design procedure.

Key words: Submersible Pump, CFD, Impeller

I. INTRODUCTION

A. Submersible Pump

A submersible pump is a type of centrifugal pump that operates while completely submerged in the liquid which is to be pumped. A centrifugal pump combines the hydraulic design of an end-suction pump with a submersible motor.

Submersible pumps are always of the close-coupled type – meaning that the impeller mounts directly on the end of the motor shaft, and the pump casing attaches directly to the motor frame. Submersible pump motors are designed with a water-cooled jacket. In most cases, the liquid being pumped is circulated around the motor to provide cooling for the motor. The internal parts of the motor are protected by a water-tight enclosure, which prevents the entry of any liquid.

In addition, larger submersible pumps include a sensor placed between the pump and motor which senses when the liquid has gotten past the motor seals and prevents the unit from further operation until repairs have been made. It can be economically designed for both oil and water wells. It can pump fluid from depths up to 15,000 feet. They can be used in crooked or deviated wells. It is relatively simple to operate. It generally provide low lifting costs for high fluid volumes. The advantages of a submersible pump are numerous.

First, it has the advantage of being self-primed because the substance it is pumping, usually water is right there at the pump itself. Further, the submersible pump may actually have to do less work than a standard pump simply because it is closer to the liquid being pumped. The pumps are typically electrically powered and referred to as Electrical Submersible Pumps (ESP).

B. CFD

Computational Fluid Dynamics usually abbreviated as CFD, is a branch of fluid mechanics uses numerical methods and algorithms to solve and analyze problems that involve fluid flow. Computers are used to perform the calculations required to simulate the interaction of liquids and gases with surfaces defined by boundary conditions. Computational techniques replaces the governing partial differential equations with algebraic equations that are much easier to solve using computer.

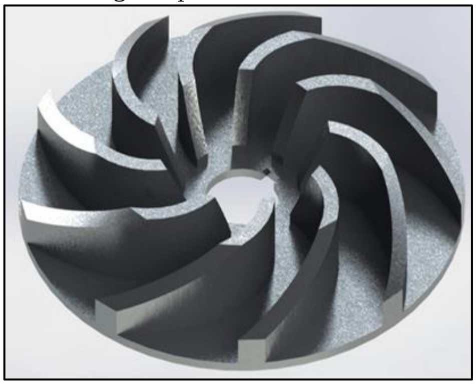


Fig. 1: Semi-Closed Impeller used in Submersible Pump

II. DESIGN OF IMPELLER & CFD ANALYSIS

We had collected input data from industry in which we have been working for project.Input data for Impeller design are Head (H) = 60 meter, Discharge (Q) = 0.000383 m3/ second, Speed (N) = 2800 RPM and Stages (m) = 6

Let, D_1=diameter of impeller at inlet
D_2=diameter of impeller at outlet
N=speed of impeller, rpm
u_1=tangential velocity of impeller at inlet =$\frac{\pi N D_1}{60}$
u_2=tangential velocity of impeller at outlet =$\frac{\pi N D_2}{60}$
V_1=absolute velocity of liquid at inlet
V_{r1}=relative velocity of liquid at inlet
α_1=angle made by V_1 at inlet with direction of motion of vane
β_1=angle made by V_{r1} at inlet with direction of motion of vane And V_2,V_{r2},α_2 and β_2 are the corresponding angles at outlet

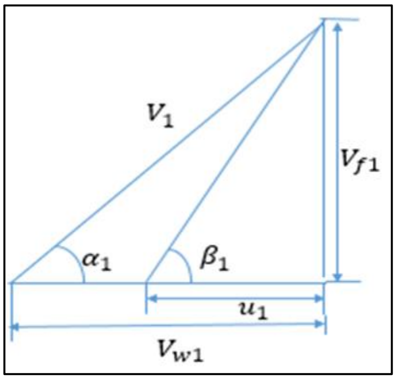


Fig. 2: Inlet Angle

Here, $\sin\alpha_1 = \frac{7.36}{9.74} = 0.75564$

α_1=$\sin^{-1}(0.75564)$
Inlet angle,α_1=49.08=49°
Diameter at inlet,D_1= 1.766
Absolute velocity, $u_1=\frac{\pi D_1 N}{60}=\frac{\pi\times1.766\times2800}{60}=0.2589\ {}^{m}/_{s}$
Absolute velocity, $u_2=\frac{\pi N D_2}{60}=\frac{\pi\times7.5\times.001\times2800}{60}=1.099\ {}^{m}/_{s}$
Here,$\tan\beta_1=\frac{V_{f1}}{u_1}$
$\therefore \tan(28°)=\frac{V_{f1}}{0.2589}$
$\therefore V_{f1}$ =0.2589× tan(28°)
Flow velocity,V_{f1}=0.1376 ${}^{m}/_{s}$
Here, $\tan\alpha_1=\frac{V_{f1}}{V_{w1}}$
$\therefore V_{w1}=\frac{V_{f1}}{\tan\alpha_1}=\frac{0.1376}{\tan(49°)}$
Whirl velocity at inlet, V_{w1}=0.1196 ${}^{m}/_{s}$
Here,$\sin\alpha_1=\frac{V_{f1}}{V_1}$
$\therefore V_1=\frac{V_{f1}}{\sin\alpha_1}$
$\therefore V_1$=0.1823 ${}^{m}/_{s}$

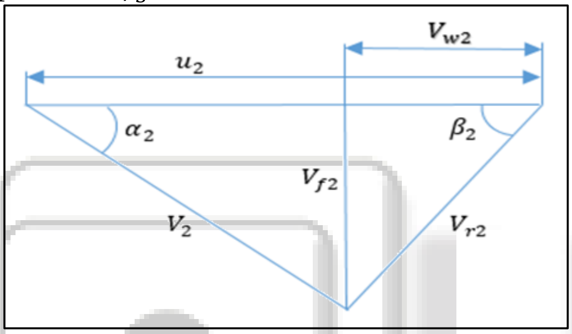


Fig. 3: Outlet Triangle

Now for outlet triangle, β_2=11.6°
Here, $\frac{V_{f1}}{V_{f2}}$=0.8
$\therefore V_{f2}=\frac{0.1376}{0.8}=0.172\ {}^{m}/_{s}$
P= m $(V_{w2}u_2 - V_{w1}u_1)$
1600=0.38(V_{w2} ×1.099−0.1198×0.2589)
$\frac{1600}{0.38}$ = (V_{w2} ×1.099−0.0309644)
0.0309644+4210.5263=V_{w2} ×1.099
$\therefore V_{w2}=\frac{4210.55728}{1.099}$
V_{w2}=3831.26 ${}^{m}/_{s}$
$\tan\alpha_2=\frac{V_{f2}}{V_{w2}}$
$\therefore \alpha_2=\tan^{-1}(\frac{0.172}{1.081})$
$\therefore \alpha_2$=9.048°
Specific speed,$N_{sh}=\frac{1000\times{}^{N}/_{60}\times\sqrt{Q}}{(gH)^{\frac{3}{4}}}$
$N_{sh}=\frac{1000\times{}^{2800}/_{60}\times\sqrt{.000383}}{(9.81\times60)^{\frac{3}{4}}}$
$N_{sh}=\frac{1000\times46.667\times0.0194}{119.49}$
N_{sh}= 7.6132
Hydraulic efficiency, $\eta_{hy}=1-\frac{0.42}{(\log d_e-0.172)^2}$
$\eta_{hy}=1-\frac{0.42}{(\log 14-0.172)^2}$
η_{hy}=55% =0.55

A. *Design Optimization & CFD*

The design optimization has been made on the outlet blade angle as 44° and 49° to compare the hydraulic efficiency of the existing impeller having the outlet blade angle of 54°.

Impeller design	Inlet blade angle(β_i)	Outlet blade angle(β_o)
Existing model	28°	12°
Impeller 1 (optimum)	69°	44°
Impeller 2 (optimum)	69°	54°

Table 1:

B. *Design of an Impeller Models*

The outlet blade angle of the existing impeller is modified by using some design software SOLID WORKS.

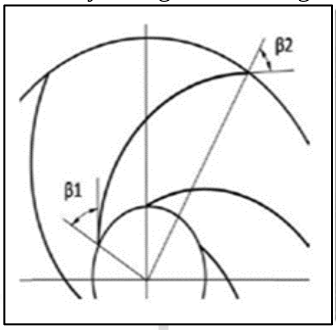


Fig. 4:

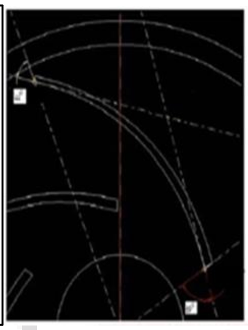


Fig. 5:

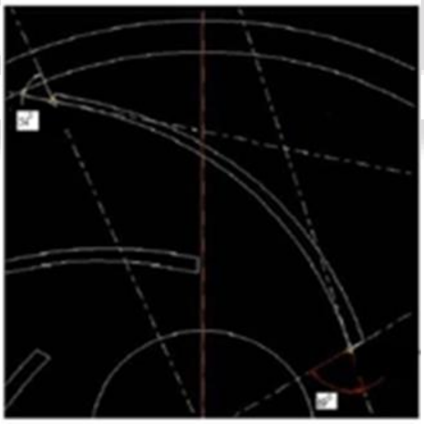


Fig. 6:

The figure-1 shows that discription of the inlet blade angle(β_1) & outlet blade angle(β_2) of the impeller.In an existing impeller, the outlet blade angle has been modified to 44° and 54° kept the inlet angle of the impeller is same as that of the existing one which is 69°. The modified outlet blade angle for the impeller models have been given figure-2.

C. *Software Analysis Model*

Analysis for Existing Impeller - Static pressure (Pascals), relative velocity magnitude (m/s), Velocity Vector

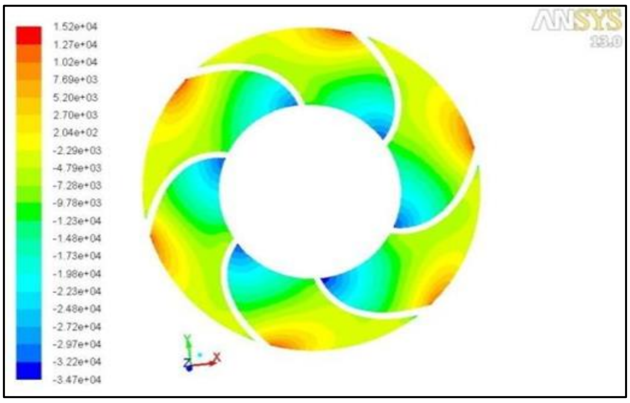


Fig. 7: Static Pressure (Pascals)

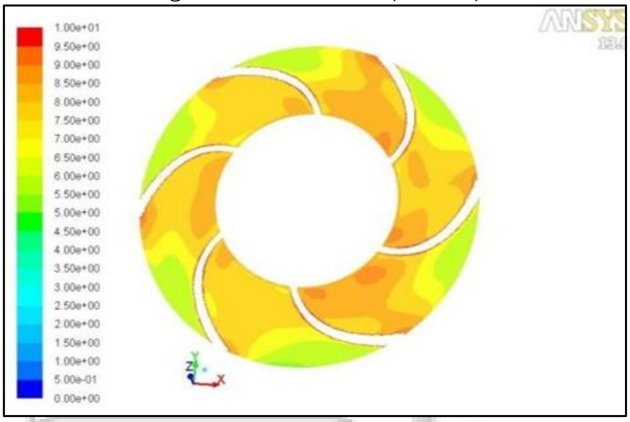


Fig. 8: Relative Velocity Magnitude (m/s)

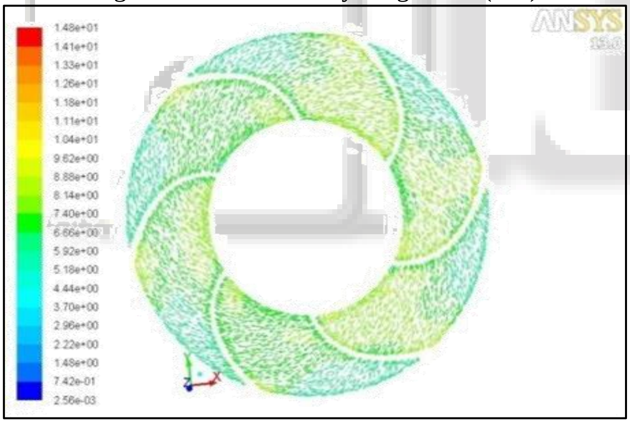
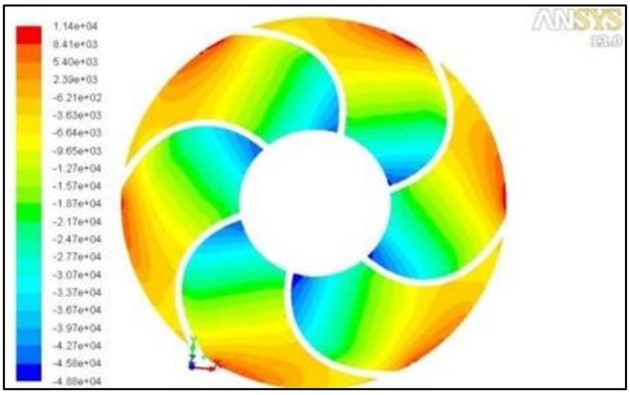


Fig. 9: Velocity Vector

Analysis for Impeller (Model-1 & 2) - Static pressure (Pascals), relative velocity magnitude (m/s), Velocity Vector



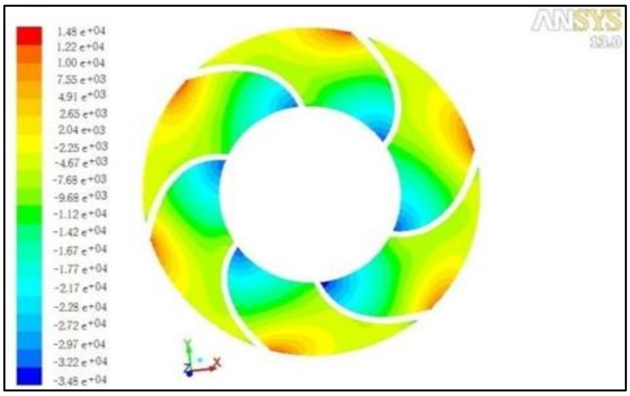
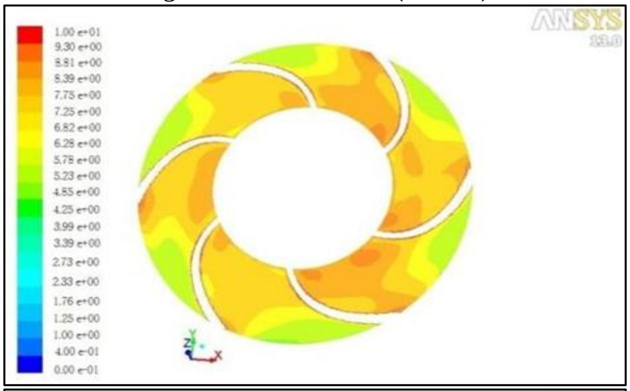


Fig. 10: Static Pressure (Pascals)



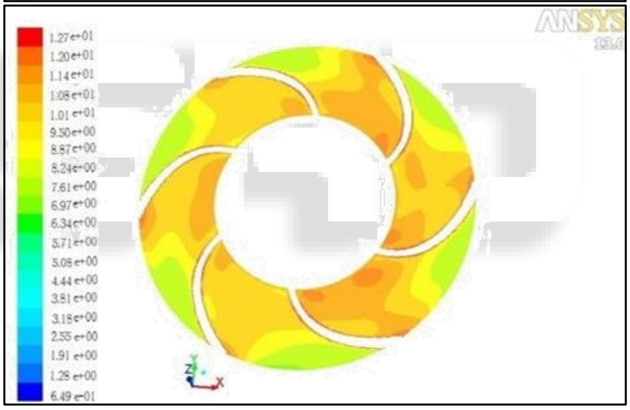
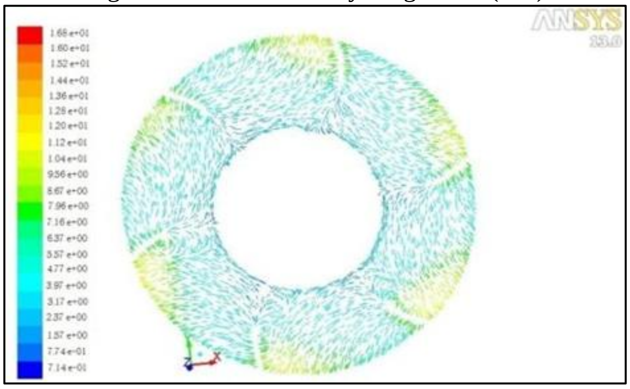


Fig. 11: Relative Velocity Magnitude (m/s)



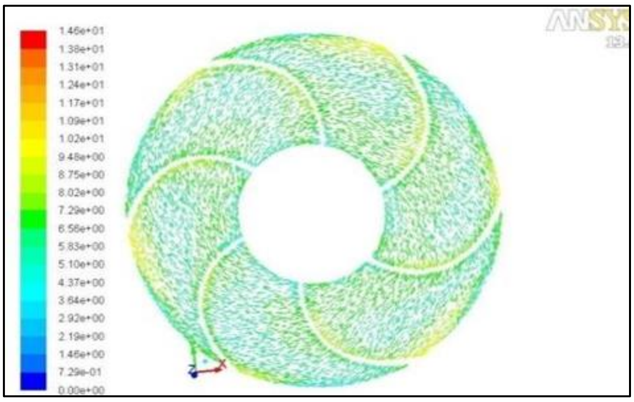


Fig. 12: Velocity Vector

III. RESULT & DISCUSSION

In this graph we can find that at 54° outlet blade angle the discharge and head will be maximum when compared to the outlet blade angle 44° and 49°. Hence it is conclude that at 54° angle the head and discharge will be higher than the other angles 44° and 49°. Various results are shown in the following figures.

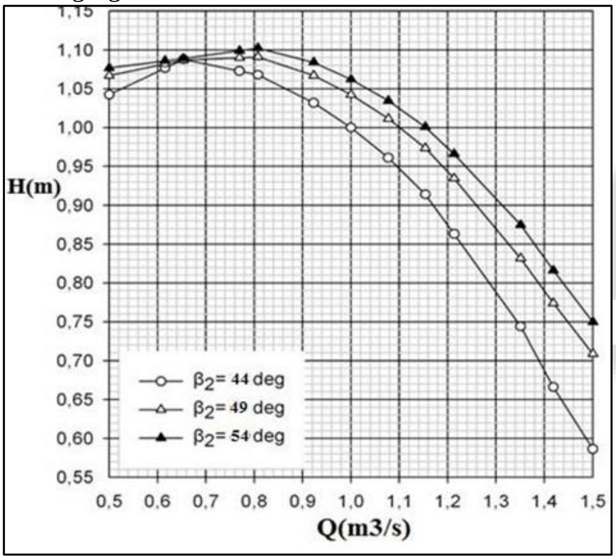


Fig. 13: Head & Discharge Curve

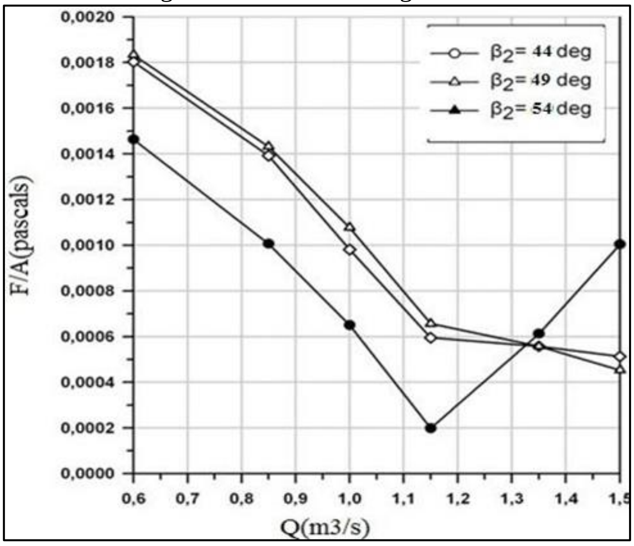


Fig. 14: Pressure & Discharge Curve

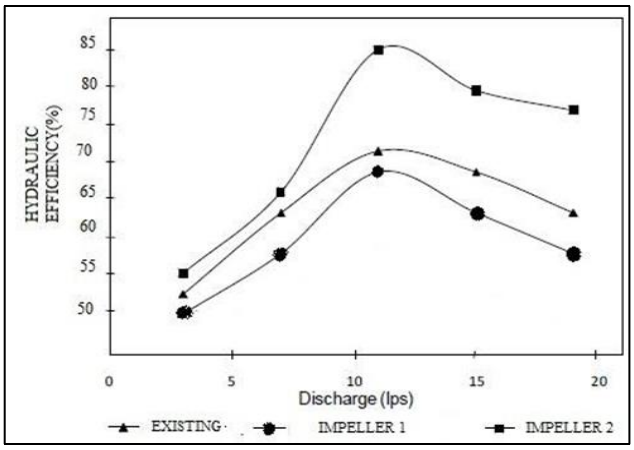


Fig. 15: Hydraulic Efficiency & Discharge

S.No	Impeller Models	Inlet Blade Angle(B1)	Outlet Blade Angle(B2)	Hydraulic Efficiency(Hh)
1	Existing model	28^0	12^0	55%
2	Model 1	69^0	44^0	67.12%
3	Model 2	69^0	54^0	83.35%

Table 2: Hydraulic Efficiencies of the Impeller Models

IV. CONCLUSION

In mathematical analysis the existing impeller model have a hydraulic efficiency of 65.3% and furthur the same model has been analysed made by using Computational Fluid Dynamics (CFD) software then it has been 73.5%. From the above results it has been concluded that analytical results of the existing impeller is higher than the mathematical results, further the same existing impeller has been optimum at the outlet blade angle region which is increased to 54°. The existing impeller has an outlet blade angle of 49° and the modified angle is 44° for model 1 and 54° for model 2.

For these two model an analysis is done by using same CFD software and it shows the results of hydraulic efficiency as 67.12% for model 1 and 83.35% for model 2. It has been seen that by increasing the blade angle, the hydraulic efficiency of the impeller gets increased but the pressure head range of the impeller is decreased when compared to the existing model and optimum model 1. Thus by increasing the outlet blade angle the hydraulic efficiency has been increased by 9.8% when compared to the existing model.

Validation:

Q =0.00383 $^{m^3}/_{s}$ H = 89 m = 434.90 feet

$$\eta = \frac{WHP}{HP\ of\ Pump}$$

$$\therefore WHP = \frac{HQ}{3960}$$

$$= \frac{\frac{434.90\ (\text{feet}) \times 6075 (\text{gpm})}{3960}}{1.01}$$

$$\therefore \eta = 66.05\ \%$$

V. LITERATURE SURVEY

Michael R. Berry and Yasser Khan Bangash et. al. 1 [2003] has investigated that the present invention provides an electric submersible pumping assembly that includes an encapsulated pumping device containing a pump, an electric submersible motor, a sealing device at the top, and an opening device at the bottom. The lubricant-filled, initially sealed, encapsulated pumping device allows the pump and motor to be run in the wellbore without contamination and be left intact until operated.

K.J. Saveth (Consultant) et. al. [2] [1993] has investigated that With operating costs playing a larger and larger role in the economics of producing a well, optimizing equipment selection by means of an efficient artificial lift system is critical in reducing operating costs for a given field. Six wells in Carter County, Oklahoma, utilizing three different methods of artificial lift were looked at to compare overall system efficiencies between each method. Progressing cavity, reciprocating, and electric submersible pumps were compared on per barrel of fluid produced basis and the electrical power usage converted to kw-hr/hr.

Feltus Henry James et. al. [3] [1963] has investigated that In relation to deep well pumps as contemplated within the scope of the present invention, reference is had to operations in conjunction with wells extending in depth from in the neighbourhood of 1500 feet to and beyond 10,000 feet. At the present time, the more conventional type of deep well pump i.e. Sucker rod type and others are more increasingly being replaced by centrifugal pumps of the single or multistage type having, in association therewith, electric motors which may be either positioned above or below the pump but in close proximity there to and which are kept dry by sealed housings filled with oil or wherein the sealed windings of the motors are insulated by being potted in plastic resin material. Pump motor systems of this type are normally built in one long casing and are adapted for a particular capacity suitable for pumping purposes within a discrete range of conditions.

James F. Lea and J.L. Bearden et. al. [4] [1982] has investigated that the presence of free gas has been recognized as having, in general, a detrimental effect on the performance of a submersible centrifugal pump. Gas locking occurs when the pump ingests too much gas and actually stops pumping because its head (or pressure) production is drastically decreased. This causes the motor to unload and to shut down because of excessive under load (control protection).

Jerzy A. Lorett and Joseph E. Vandevier et. al. [5] [1987] has investigated that a low volume submersible pump assembly has features to stabilize the operation. The pump assembly includes a centrifugal pump driven by a submerged electrical motor. The centrifugal pump has an impeller configuration that is designed for high specific head, but potentially unstable operation. At the design speed, the pump potentially could deliver an indeterminate flow rate for a given head and also at the design speed, the design flow rate will yield a design head that is very close to the maximum head at zero flow. To stabilize the potentially unstable pump, a variable speed drive varies the speed of the motor to maintain constant torque. Maintaining a constant torque allows a considerable variance in head with only a small variance in flow rate resulting.

V. Ram Kumar and M.Prabhu et. al. [6] [2015] has investigated that this project deals with the modification in the existing design of radial flow impeller made out of plastic material in many cases may be called rotor. By modifying the design of impeller either by changing the blade angle or by changing the width we could achieve a considerable increase in the output discharge which will be shown in the further work. Ansys software is used to analyse the performance of pump and the equivalent output of the new impeller design and use Pro-e for the design of impeller. After modelling the impeller we will calculate the equivalent theoretical efficiency so as to prove that the impeller that is designed newly is of better design than the previous one.

REFERENCES

[1] Michael R. Berry and Yasser Khan Bangash et. al. “Electric Submersible Pump Assembly”, Wood Group Esp, Inc., US 6,595,295 BI ,Issue July 22, 2003.

[2] K.J. Saveth (Consultant) et. al. “Field Study of Efficiencies Between Progressing Cavity Reciprocating and Electric Submersible Pumps”, Society of Petroleum Engineers, SPE-25448-MS, Issue 1993.

[3] Feltus Henry James et. al. “Submersible pump”, Goulds Pumps, US3115840 A, Issue Dec 31, 1963.

[4] James F. Lea and J.L. Bearden et. al. “Effect of Gaseous Fluids on Submersible Pump Performance”, Society of Petroleum Engineers, SPE-9218-PA, Issue December 1982.

[5] Jerzy A. Lorett and Joseph E. Vandevier et. al. “Low volume variable rpm submersible well pump” , Hughes Tool Company , US4678404 A , Issue 7 Jul 1987.

[6] RAMKUMAR and M.PRABHU et. al. “Study Of The Existing Design Of Impeller Of 4” Submersible Pump And Improving It’s Efficiency Using CFDA Through Theoretical Analysis”, ISSN 0976 – 6340 (Print), ISSN 0976 – 6359 (Online) Volume 6, Issue 5, May (2015).

[7] Shukla SN, Kshirsagar JT. Numerical experiments on a cen- trifugal pump. American Society of Mechanical Engineer- ing. 2003. p. 21–30.

[8] Scarbrough TG. Pumps and in service testing. Proceedings of the Ninth NRC/ASME Symposium on Valves; L’Enfant Plaza Hotel Washington, DC. 2006 Jul. p. 1–14.

[9] Oh HW, Chung MK. Optimum values of design variables versus specific speed for centrifugal pumps. Proceedings of the Institution of Mechanical Engineers. 1999; 213(3):219.

[10] Thin KC, Khaing MM, Aye KM. Design and performance analysis of centrifugal pump. World Academy of Science, Engineering and Technology; 2008. p. 422–9.

[11] Singh RR, Nataraj M. Design and analysis of pump impeller using SWFS. World Journal of Modelling and Simulation. 2014; 10(2):152–60.

[12] Church AH, Lal J. Centrifugal pumps and blowers. US: Krieger Publishing Company; 1972.

[13] Nigussie T, Dribssa E. Design and CFD analysis of centrif- ugal pump. International Journal of Engineering

Research and General Science. 2015 May-Jun; 3(3):668–77.
[14] Addison H. Centrifugal and other rotadynamic pumps. 3rd ed. London: Chapman & Hall Ltd: 1966. p. 565.
[15] Liu H, Wang Y, Yuan S, Tan M, Wang K. Effects of blade number on characteristics of centrifugal pumps. Chinese Journal of Mechanical Engineering. 2010. DOI: 10.3901/ CJME.
[16] Weidong Zhou, Zhimei Zhao, T. S. Lee, and S. H. Winoto. A NUMERICAL SIMULATION OF THE IMPELLER-VO- LUTE INTERACTION IN
[17] Manivannan, "Computational fluid dynamics analysis of a mixed flow pump impeller" International journel of science and technology Vol. 2, No. 6, 2010, pp. 200-206.

