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Theoretical, Numerical and Experimental Research of Single Stage, Radial Discharge Centrifugal Pump Operating in Turbine Mode

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Abstract—In this study, the best efficiency point of end suction, radial discharge, centrifugal pump operated in turbine mode was arrived applying numerical and experimental analysis. The pump was simulated both in direct and turbine modes using Star CCM+ CFD software. Characteristic curves were developed for the pump in direct and turbine modes. A monoblock centrifugal pump of specific speed 35.89 (m, m^3/s) was used for this study. The pump was tested experimentally in turbine and pump mode .The theoretical and numerical results were verified by those obtained through experimentation. Some of the correlations proposed by earlier researchers for performance prediction of pump in reverse mode were also tested.

Keywords: Computational Fluid Dynamics, Experimental, Pump as Turbine, Radial discharge centrifugal pump, Renewable energy, Pump as Turbine (PAT), Best Efficiency Point (BEP).

I. INTRODUCTION

Depleting fossil fuel reserves and their deteriorating environmental affect has shifted the focus of researchers towards renewable energy.Around 14% of the world's populations are living without electricity and about 84% out of these live in rural areas. Around 95% of those living without electricity are located in countries in Africa and developing Asia [1]. Many of these countries have utilized only about one fourth of the possible hydropower. The hydropower projects face a lot of challenges such as complicated and time consuming procedures to obtain necessary approvals towards environmental permits, land acquisition, obtaining long-term finance and also face public opposition. Small hydropower projects can play an important role in meeting the power requirements of remote rural, high terrain areas. Run-of-river projects do not require expansive reservoirs and resettlement of people and so there is no difficulty in obtaining public acceptance. Rural population of around 20% i.e about 220 million people live without electricity as extending grid supply to these areas is not practical due technical and economic reasons [2].

Pump as turbine technology for rural villages in high terrain area, where water streams are naturally available proves to be a much feasible and practical solution for off grid supply. Pump manufacturer do not provide turbine mode curves for their pump. Selection of a pump suitable for a particular site for turbine mode operation requires these curves. Correlations based on theoretical approach are proposed by some researchers to estimate the performance of a pump in turbine mode, results of these correlations however are not so accurate and show around 20% deviation between the experimental and predicted turbine mode operation of pumps. Experimental evaluation of the turbine mode characteristics is specific to the pumps tested and do not serve as a generalized tool for pump selection. The advantage of experimental study helps in validating results obtained through theoretical and modeling analyses.

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II. THEORETICAL, NUMERICAL AND EXPERIMENTAL INVESTIGATION

The pump manufacturers provide performance curves for the pump mode operation. These curves provide information about flow rate, head and the efficiency in pump mode. The turbine mode performance predictioncan be obtained from the pump mode information by simple calculations. Theoretical approach uses head and flow correction factors which are derived from pump mode information to predict turbine mode performance. Relations proposed by earlier researchers are mainly based on pump mode best efficiency point or based on pump specific speed.The theoreticalapproach based results although have not been very satisfactory. Many researchers have emphasized on ascertaining theoretical approach results through experimental investigation.The experimental investigation has limitations as all the pumps cannot be tested. Computational fluid dynamic software's have been very useful tool in the performance prediction. Many researchers have attempted performance prediction of pump in turbine mode using these software's. Flow through pump is three dimensional with turbulence and unsteadiness, the results of simulations are not exact, moreover the results also dependent on mesh, boundary conditions applied and selection of appropriate turbulence model. In this study Star CCM+ CFD software and Realizable k-ε turbulence model is used.

The grid independence study was performed .A mesh with total mesh count 1,886,400 gave least error results.The results obtained by simulation were validated by comparing them against those available through manufacturer provided curves. The results of simulation were also compared with experimental results. A centrifugal pump of specific speed 35.89 (m, m^3/s) was used for this study. The schematic lay out of the experimental test rig is seen in the Fig. 2. Various methods are used by researchers in predicting pumps performance in reverse mode operation. Based on pump geometry few methods adopted are theoretical [3-5]. The difficulty using theoretical methods based on geometry is the availability of detailed geometry of the pumps to be studied in turbine mode which manufacturers hesitate to share. Theoretical methods for developing reverse pump characteristics are mainly based on either best efficiency point in pump mode or based on pump specific speed number. [6]. Specific speed is useful in estimating a pump and a hydro turbine performance. The impeller shape of the pump predicted through specific speed becomes a basis for selection of an appropriate pump and hydro turbine suitable for a particular site. The specific speed based prediction is proposed by some researchers. [7, 8, 9, 10, 11] other authors [12, 13, 14, 15] attempted PAT performance prediction based on the pump mode BEP.

III. EXPRESSIONS

$$Pmech.Output = \tau \times \omega \text{ (Watt)} \quad …….. \quad (1)$$

Where

τ is the torque

ω is the angular velocity

The net head across PAT is denoted as H.

Hydraulic power input to the PAT is expressed as

$$P_{hyd.input} = \rho g Q H \text{ (Watt)}…. \quad (2)$$

Where,

ρ is the water density

g is the gravitational acceleration

Q is the flow rate

H is the net head across PAT

The efficiency of the PAT is then expressed as

$$\eta_{PAT} = \frac{Pmech.Output}{P_{hyd.input}} \times 100 \quad ………. \quad (3)$$

Loss in power is mainly accounted for mechanical and hydraulic loss. Mechanical friction power loss is because of friction between the fixed and rotating parts in bearings and stuffing box shaft seals.

Hydraulic loss includes shocked entry and incidence losses, disc friction, recirculation, separation, and leakage loss.

IV. PUMP MODE CFD MODEL VALIDATION

The CFD model was validated by comparing manufacturer provided head and flow values with those obtained through CFD simulation in pump mode. The results are displayed in Table-I. A less than 6% error validates the reliability of results. Fig.1 shows the head vs discharge curve for pump mode based on manufacturer data and CFD simulation results.

Table- I: Pump mode CFD and Manufacturer data

Flow Rate (lps)	CFD Head (m)	Manufacturer Data (m)	% Error
1	15.24	15.75	3.25
2	15.71	16.13	2.55
3	15.56	16.00	2.74
4	15.55	15.63	0.51
5	15.37	14.88	-3.31
6	14.26	13.75	-3.74
7	12.64	12.00	-5.36

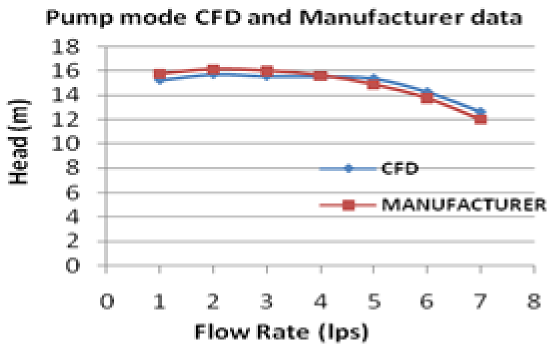

Fig. 1. CFD & Manufacturer data based head vs flow curve in pump mode

V. EXPERIMENTAL TEST RIG

The hydraulic test rig included feed pump that was 5 H.P, 2800 rpm, mono block, radial discharge centrifugal pump with 32 m head and 7 l/s flow at B.E.P. The pump tested as PAT was a 2 H.P, 2800 rpm, mono block, radial discharge centrifugal pump of specific speed 35.89 (m, m^3/s) with 13.8 m head and 7.2 l/s flow at B.E.P. Piping system was made from uPVC (polyvinyl chloride). Water tank capacity was 1500 liters. Control valves included uPVC ball valves and a gun metal gate valve. Pressure gauges used were digital display PG20 pressure gauges. A digital display electromagnetic flow meter was used for recording flow. The braking system used a belt brake dynamometer with digital display spring balance.

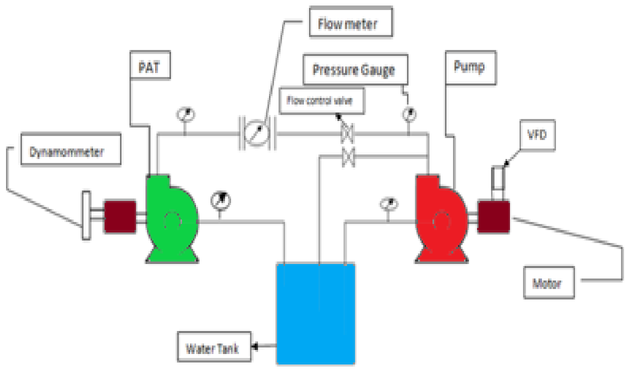

Fig. 2. Schematic layout of the experimental test rig.

Table 2. Instrumentation used with experimental test rig

Instrument	Type	Accuracy
Flow meter	Electromagnetic flow meter	+/- 0.5% of flow
Digital pressure gauge	PG/DPG-20	Accuracy:+/- 2.5% F.S
Rope brake Dynamometer	0-50 kg, digital display	+/- 0.5 %
Speed Digital Tachometer	Non Contact Photo Electric	+/- 0.05%
Variable Frequency Drive	Fuji VFD 10HP	0.01% of max. frequency

VI. CFD SIMULATION

In this study Star CCM+ CFD software and Realizable k-ε turbulence model is used.The grid independence study was performed .A mesh with total mesh count 1,886,400 gave least error results. Fig.3. Shows the solid geometry of the pump impeller and volute casing used in this study.

Single vane passage model of the pump impeller is insufficient to investigate the variation of the flow caused by volute circumferentially. To study the pump behavior in the turbine model through CFD simulation hence a 3D model of the complete impeller and casing is generated.

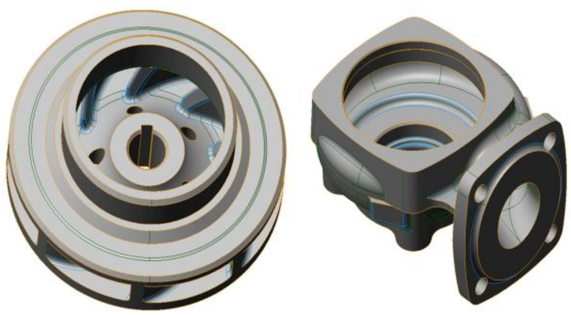

Fig.3. Solid geometry of pump impeller and casing

VII. MESH DESCRIPTION

Type of Mesh: Polyhedral + Prisms
First Prism Layer Height: 0.005 mm
Total Height: 1 mm
Number of Layers: 10

Fig.4, 5 & 6 show the meshing details of the simulation model.

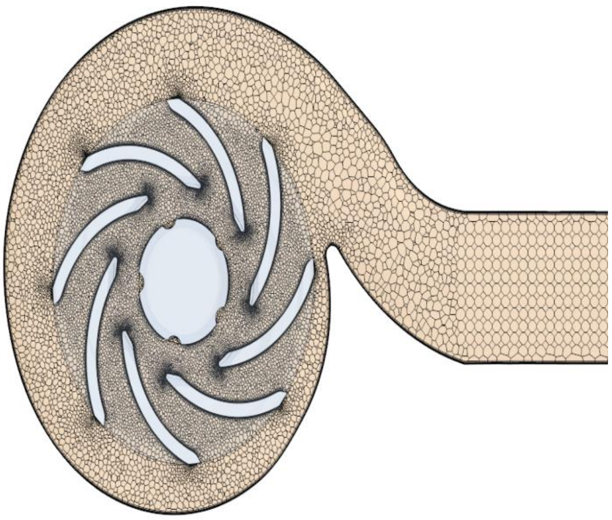

Fig.4.Cross-sectional view of Turbine model

Results of and numerical simulation are only approximations and depend on mesh and boundary conditions defined. Accurate mesh and boundary conditions result in better and accurate results.

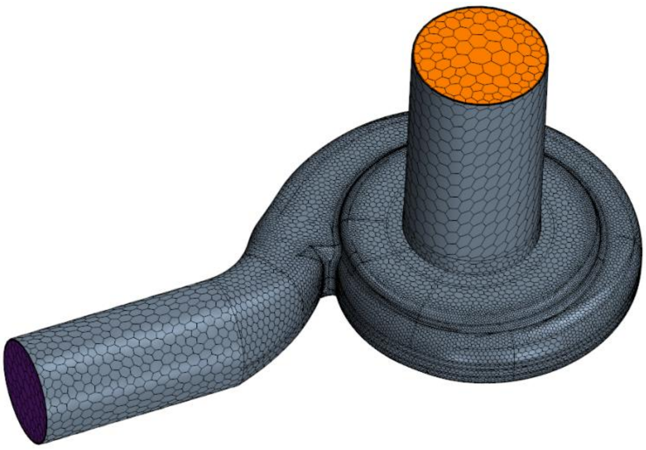

Fig.5. Volume mesh for the Turbine model

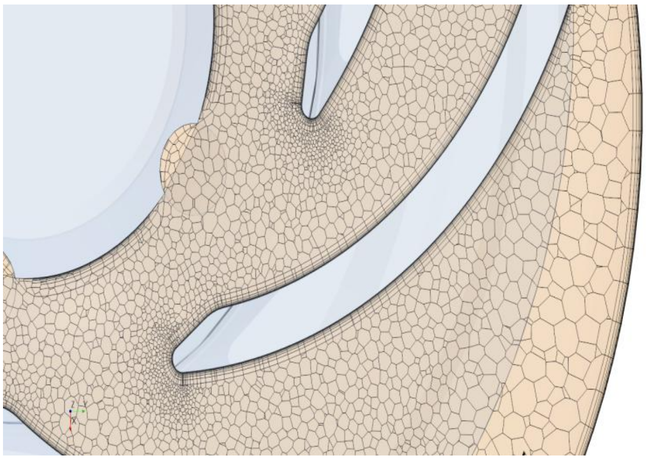

Fig.6. Closer view of mesh distribution inside the turbine model

To achieve reduced simulation run time smallest mesh which gives mesh independent results is used. The accuracy of CFD results is dependent on the boundary conditions and physics used and should be independent of the mesh resolution. The simulation results are validated by comparing them with experimentally obtained results. Frozen rotor concept adopted in CFD software is used to model the interface between the stationary and rotating components. Fig. 7, 8 & 9 show the velocity colour plots for the turbine mode simulation for PAT rpm 2600, 2800, and 3000 respectively.

It is evident from the velocity colour plots that at higher heads and flow there is an higher pressure zone at the vane tips seen in red colour. The B.E.P is reached in PAT mode for the three rpm 2600, 2800, 3000 below 400kPa pressure and flow rates between 13.5 lps to 16 lps. The B.E.P in turbine mode is achieved at higher head and flow as compared to the turbine mode operation.

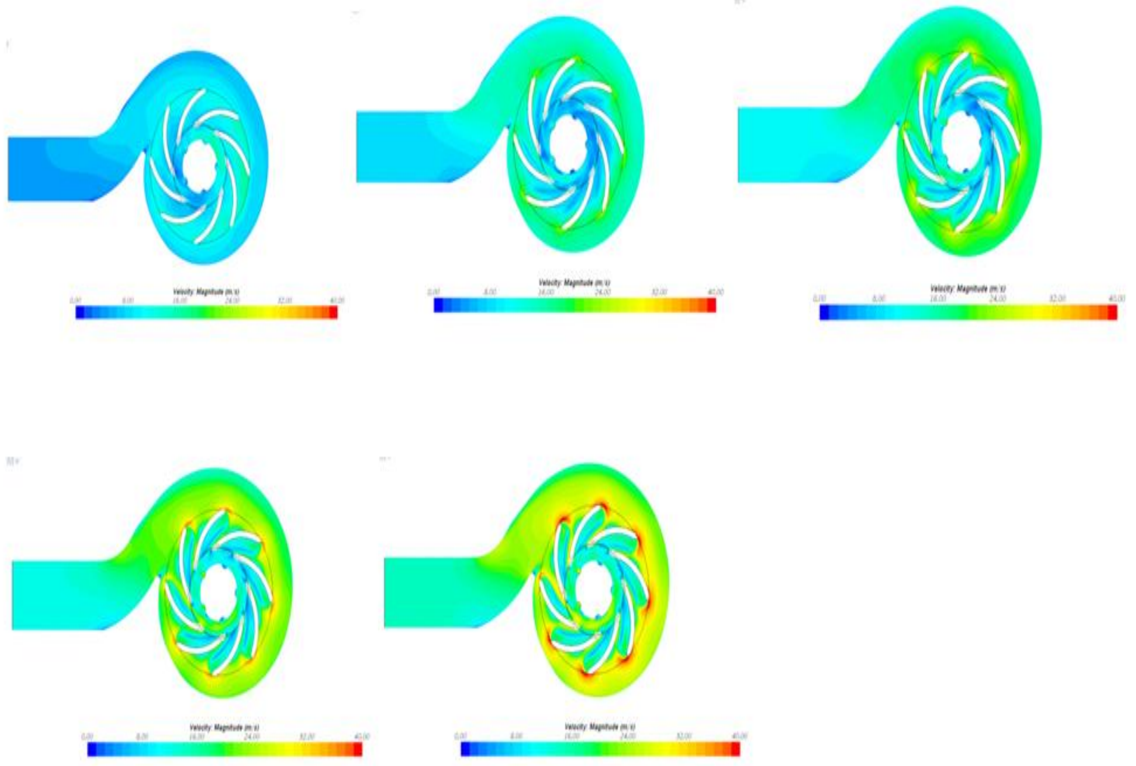

Fig.7. Velocity plots at 196 kPa , 400 kPa, 600 kPa, 900 kPa, 1200 kPa for 2600 RPM

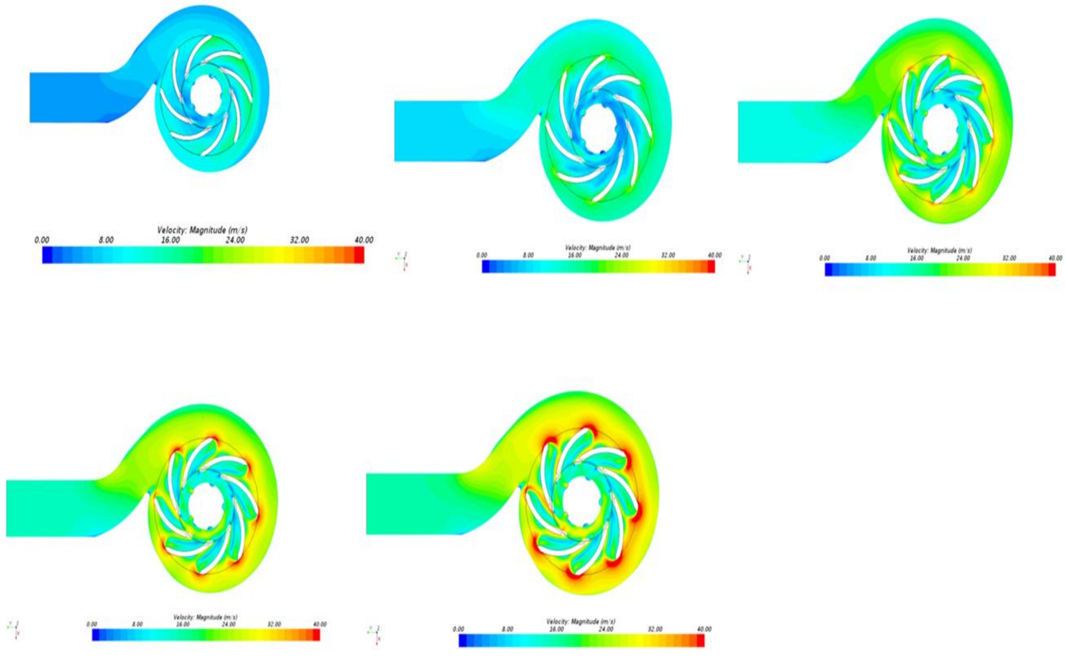

Fig.8. Velocity plots at 196 kPa , 400 kPa, 900 kPa, 1200 kPa, 1500 kPa for 2800 RPM

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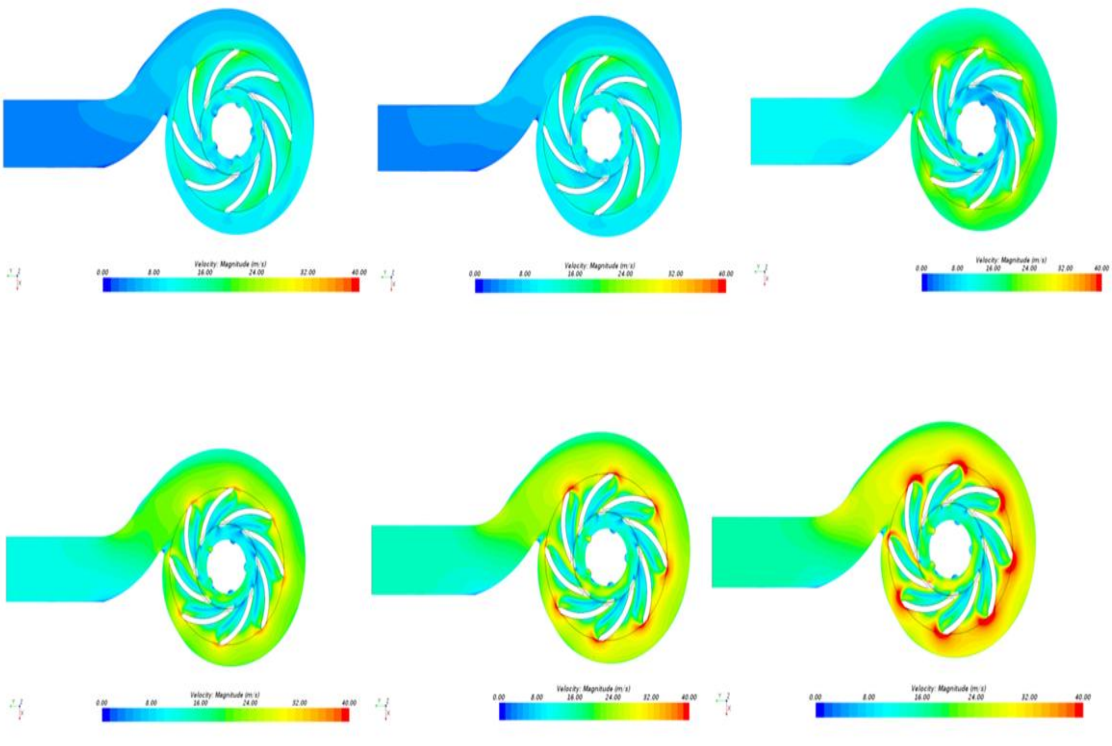

Fig.9. Velocity plots at 196 kPa , 400 kPa, 600 kPa, 900 kPa, 1160 kPa, 1500 kPa for 3000 RPM

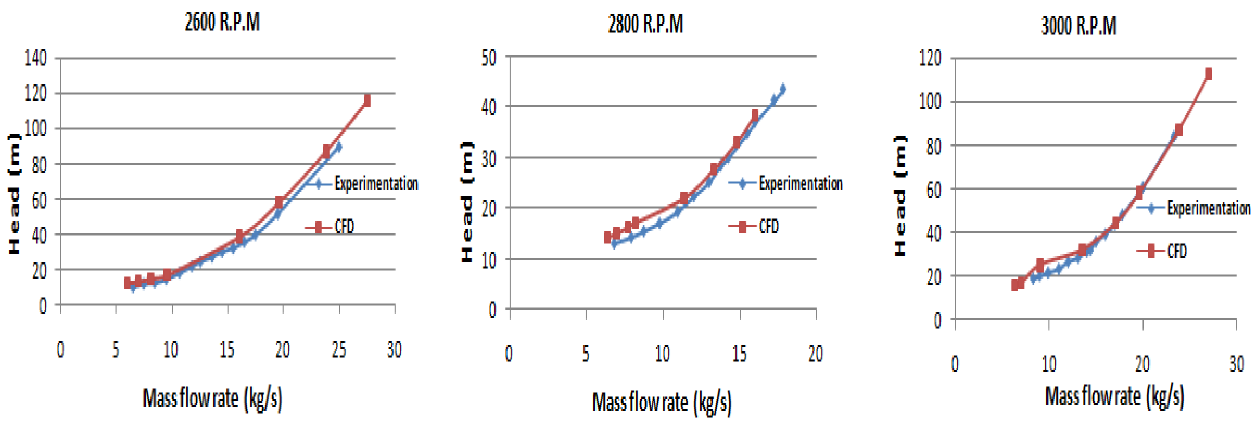

Fig.10. Head vs Flow curve for PAT at 2600, 2800, 3000 R.P.M

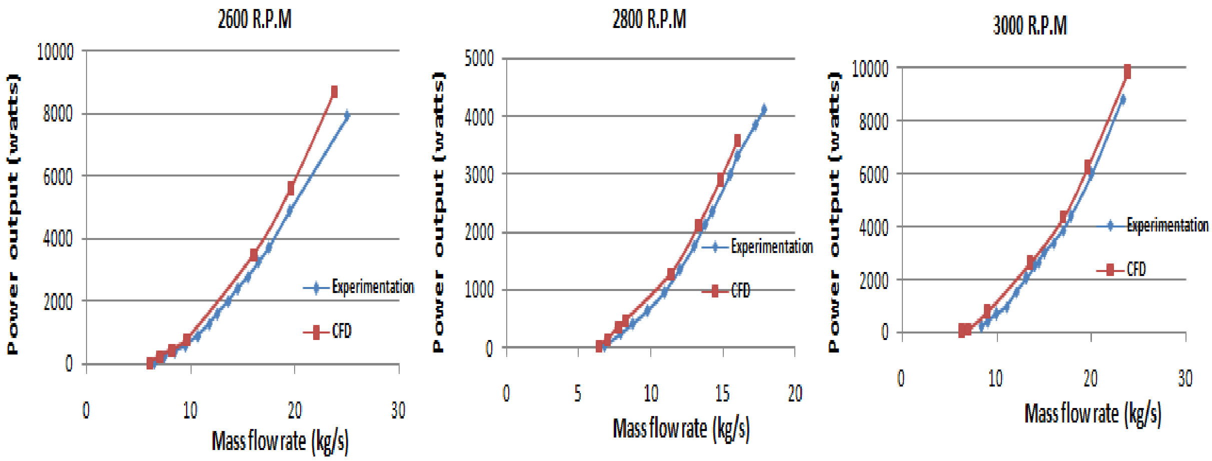

Fig.11. Power vs Flow curve for PAT at 2600, 2800, 3000 R.P.M

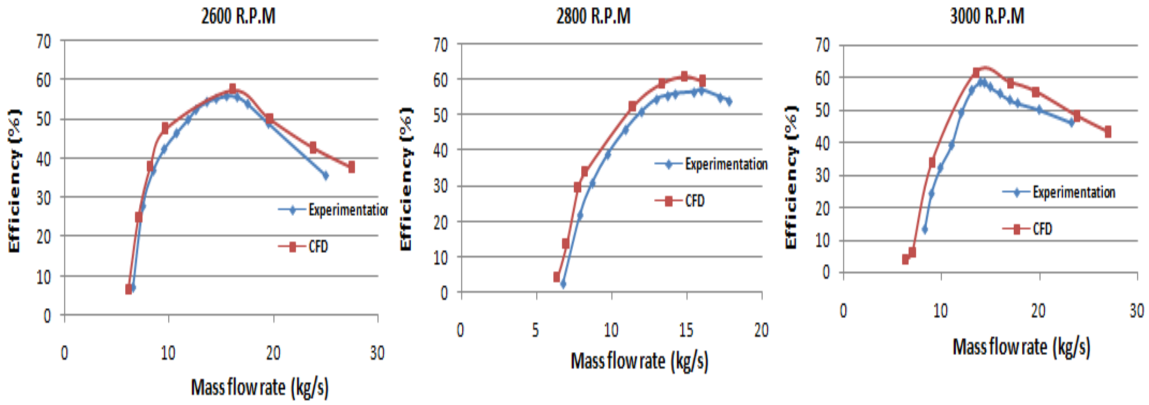

Fig.12. Efficiency vs Flow curve for PAT at 2600, 2800, 3000 R.P.M

VIII. RESULT AND DISCUSSION

This paper presents methodology to predict the reverse mode characteristic of a centrifugal pump. The CFD simulation model was validated with pump mode data available from the pump manufacturers. The results of experimentation are used to validate CFD simulation results for the pump in reverse mode. The results of the proposed methodology are then compared to the prediction methods suggested by various researchers. Steady state readings were obtained after allowing PAT to run for 10 minutes. The rotational speed was maintained fixed by varying load on PAT and the corresponding readings of pressure, flow, torque were noted. The PAT experimentation was performed for three rpm 2600, 2800 and 3000 rpm. The power output is higher in reverse mode than direct mode. The efficiency curve in PAT mode is similar to that of a Francis turbine increasing at lower flow rates and tends to a maximum value at high flow rates.

The highest efficiency in the pump mode was 54 % at 7.4 lps and 13.8 m head. The highest efficiency noted in turbine mode 60.8% is slightly higher than the pump mode. This efficiency is noted at a higher flow and head of 16 lps and 36.99 m respectively at 2800 rpm.

IX. CONCLUSION

A end suction radial discharge mono block centrifugal pump of specific speed of 35.89 (m, m3/s) was simulated and experimentally tested to understand the pump operating in reverse mode. It is revealed that a Centrifugal pump can be operated as hydro turbine satisfactorily without any critical mechanical problems. Higher heads and discharge values at BEP are noted in turbine mode as compared to pump mode. The results of simulation in the present study when compared to the experimental results show errors of less than 10%. The results when applied to correlations suggested by various researchers earlier suggest that no correlation is very close in predicting the performance of a centrifugal pump in reverse mode. The correlations of head and discharge correction factors by Sharma (1985), Stepanoff (1957), Childs (1962), were close in approximation.

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