EXPERIMENTAL AND NUMERICAL INVESTIGATION OF CENTRIFUGAL PUMP PERFORMANCE IN REVERSE MODE`

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ABSTRACT

Pumps as turbines have been successfully applied in a wide range of small hydro sites in the world. Since the overall efficiency of these machines is lower than the overall efficiency of conventional turbines, their application in larger hydro sites is not economical. Therefore, the efficiency improvement of reverse pumps is essential. In this study, attempt made to justify the simulation results with the experimental results for the two centrifugal pumps with different specific speed. Experiment performed for different shaft speed for each pump. Dimensionless parameters such as power number, head number and efficiency are compared, obtained from the experiments and obtained with the use of CFD. The results obtained from the simulation of the PAT is agrees well with the experimental results. It has seen some deviation in the efficiency but simulation gives good results for the head number and power number.

Keywords: CFD-Computational Fluid Dynamics, PAT- Pump As Turbine

I. INTRODUCTION

The current energy crisis - with rising fuel prices - might be enough to boost the world's appetite for renewable energy. Renewable energy technologies produce profitable energy converting natural phenomenon into useful forms of energy. One of renewable energy technology is hydropower. These systems convert the hydraulic energy of water, potential and kinetic energy of water into mechanical or electrical work. But due to very high cost of conventional turbines etc it is not possible to make as much hydraulic plants as needed. Micro-hydropower is a practical and potentially low-cost option for generating electricity at remote sites, particularly for small villages in hilly areas. Running costs for such schemes are very low, but the initial capital cost can be relatively high and any reduction in equipment costs will make the technology more accessible. One way to reduce the equipment cost is to use a standard pump unit as an alternative to a conventional turbine.

Pumps are mass-produced, and as a result, have the following advantages for micro hydro compared with purpose-made turbines: Integral pump and motor can be used as a turbine and generator set, Available for a wide range of heads and flows, Available in a large number of standard sizes, Short delivery time, Spare parts such as seals and bearings are easily available, Easy installation - uses standard pipe fittings.

II. A BRIEF OF HISTORICAL BACKGROUND EAHE SYSTEMS IN THE WORLD

The use of pumps as turbines has been topic of research for over 70 years. There are many researchers who have already tried to give the best prediction methods to find the turbine performance curves but none of them were able to give an accurate method.

The research on using pumps-as- turbines was started around 1930. Thoma and Kittredge (1931) published the complete characteristics of few pumps; they accidentally found that pumps could operate very efficiently in the turbine mode. Punit Singh **[1]** recommended little changes in boundary conditions for the 3 PAT's. For the 35.3 rpm and the 39.7 rpm, the boundary conditions should comprise of total pressure at the inlet measurement plane and the mass flow rate at the exit of the draft tube The boundary conditions for the 24.5 rpm PAT has been reversed with specification of mass flow rate at the inlet measurement plane and atmospheric pressure at the draft tube exit. The simulations are carried out at identical load point from experimental study. The variables that are transferred from experimental measurements are discharge (or mass flow rate) and operating speed (N). The discharge used here is the net discharge across the PAT control volume without any considerations for leakage flow. The CFD model then essentially evaluates the net head (H) across the inlet and exit measurement planes, and the hydraulic output torque (T). In agricultural facilities (animal buildings) and horticultural facilities (greenhouses) Earth–air heat exchangers have been used over the past several decades and have been used in conjunction with solar chimneys in hot arid areas for thousands of years, probably beginning in the Persian Empire.

Sania raval, J.T.kshirsagar**[5]** based on experimental data carried out at a major university in Karlsruhe, Germany by Dr Punit singh, presented CFD results in qualitative and quantitative form. The following five flow rates were used for CFD analysis: 0.075 m3/sec, 0.100 m3/sec, 0.126 m3/sec, 0.151 m3/sec and 0.176 m3/sec. CFD results give a higher value of power output and efficiency as losses are not considered. The future work will concentrate on improving mass quality and numerical schemes for both experimental and numerical results by testing several pumps at different operating specific speed.

Sania raval, J.T.kshirsagar[5] based on the numerical models, investigated various parameters that cannot be measured experimentally like flow pattern and internal hydraulics variables etc. Numerical approach is useful in identifying loss in individual component like draft tube impeller and casing. The authors analyzed the accuracy of the CFD models using experimental data as standard. The entire geometry consisting of casing, impeller, and draft tube is modeled for numerical analysis. An unstructured tetrahedral intermediate size mesh is used for increasing the grid density at locations where gradients of flow variables are observed to be high .Leakage through wear rings or packing is not included in the model. Frozen rotor concept is used for the interface between stationary and rotating components. Boundary condition are specified in the form of total pressure at the inlet flange of casing and mass flow rate at the exit of the draft tube.

III.METHODOLOGY

3.1 PAT working principal

Pumps are designed for particular head and flow rate. In pumps fluid enters from suction side at low pressure side and discharge to high pressure side. It takes energy from impeller to convert fluid low pressure to high pressure

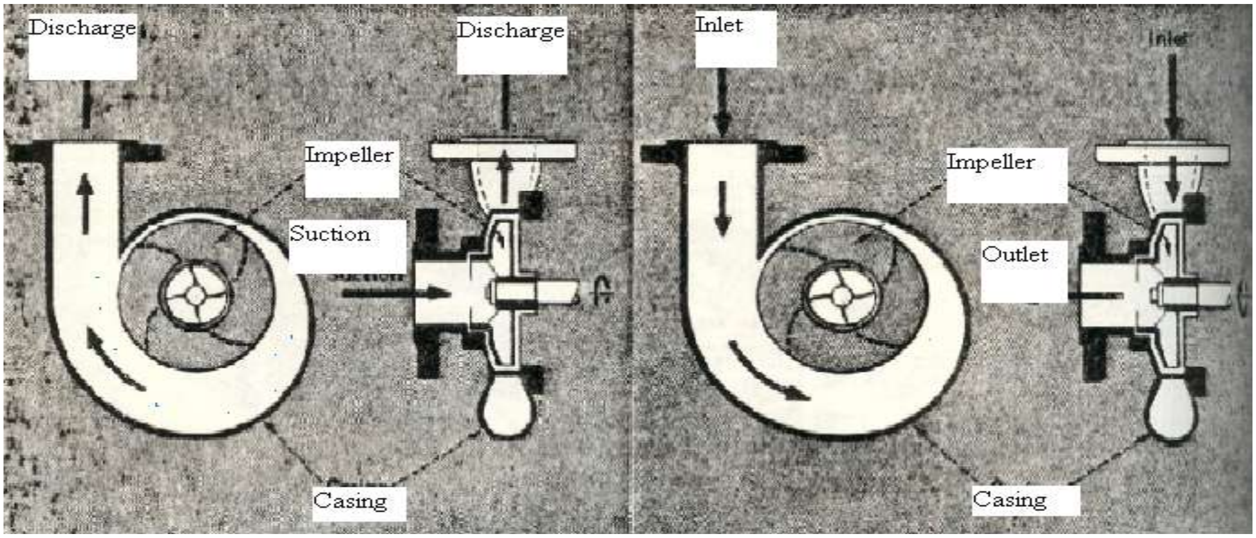

Fig.1: Pump in pumping and turbine mode

In case of pump as turbine (PAT) pump operates in reverse mode. In PAT mode water enters in casing at high pressure and transfer convert pressure and kinetic energy in to mechanical energy and fluid exit from pump the eye at low pressure. And fluid exit from pump the eye at low pressure. In the reverse operation of pump it may be less efficient because the direction of flow is reverse and hydraulic and frictional losses increase sharply. When fluid enters in to impeller because of Sharpe edges of impeller at the end flow separation occurs at inlet of PAT due to that separation losses temperature accordingly. The pipe outlet is given where the space needs to be air conditioned in industrial or livestock buildings etc. By taking benefits of this costless energy we can reduce our energy consumption for air conditioning of space hence a very useful technique it is.

3.2 Experimental Set up

A compact open loop-test rig was used for testing the pump in turbine mode. The test rig consists of one feed pump of radial flow type for pumping water at high pressure to provide the necessary head and flow. The pump has been connected with the motor. The motor connects with VFD (variable frequency drives) drive. The motor speed can be easily varied and controlled with PLC (Programmable logic controller). Thus it improves the efficiency of motor and also it matches the speed according to changing load requirements and gives accurate. Continuous process control over a wide range of speeds. From the sump the water supplied with high pressure in the centrifugal pump being used as a turbine. After imparting motion to the pump impeller, water is discharged through the conical draft tube connected to the pump outlet. Water goes back through the same sump channel, from which the radial flow pump again pumps the same water back to the PAT.

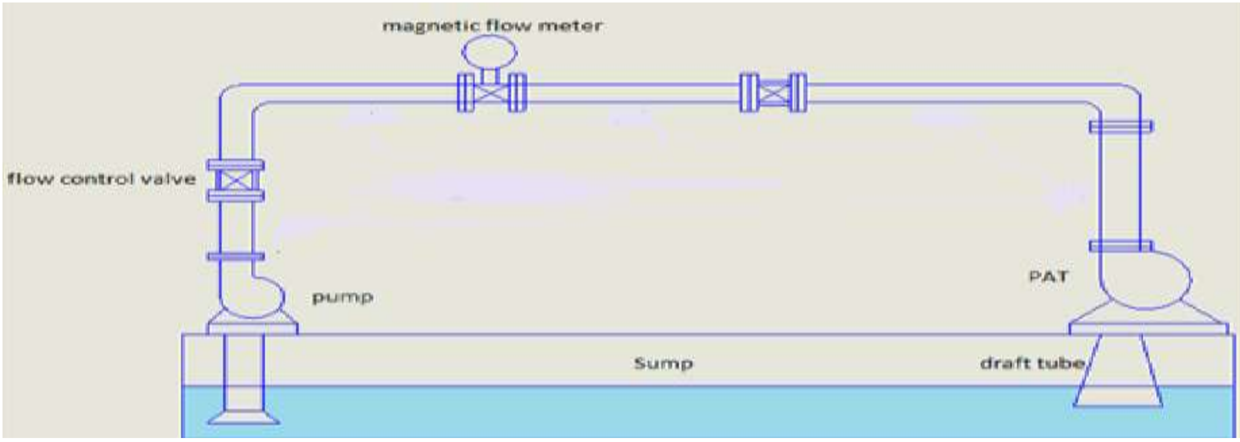

Figure : 2 Layout of test rig

IV. DIMENSIONLESS ANALYSIS

The Buckingham's Pi-Theorem of dimensionless analysis deals with the physical phenomenon of turbine operation using all these variables alone. During study of a particular type of turbine or pump, the design parameters can be neglected except the diameter of runner. The other important parameters that need to be considered is the speed N, the net head H, the discharge Q, and output power P. Density and acceleration due to gravity are necessary to complete dimensionless property of these groups. All these variables can be simplified using Buckingham's theorem for a turbine in separate ways like based on constant speed and based on constant head. As we performed all the tests at constant speed so the result of constant speed will be presented. Three groups are formed which define the entire turbine performance. All these groups carry the variable speed N hence it is to be based on constant speed. The dimensionless
groups are,

Flow Coefficient (ϕ): The combination of speed and discharge making it dimensionless.
It also includes a geometrical parameter diameter. It is given by,

$$\text{Flow Coefficient } (\phi) = \frac{Q}{ND^3}$$

Head Coefficient (ψ): This dimensionless group is a combination of net head and speed.
It also includes parameter of diameter and acceleration due to gravity (g).

$$\text{Head Number } (\psi) = \frac{gH}{N^2D^2}$$

Power Coefficient (π): This is a dimensionless group combining speed and output power
with density and diameter.

$$\text{Power Coefficient}\,(\pi) = \frac{P}{\rho N^3 D^5}$$

V. EXPERIMENTAL RESULTS

Figure3 indicates that the 38.47 rpm PAT have about 66 % efficiency at 1100 rpm shaft speed. The flow number at best efficiency point is about 0.191 .at best efficiency point the power number available from the PAT is about 1.3. At full load condition the power number is about 3 . And at no load condition the power number is about 0.250.

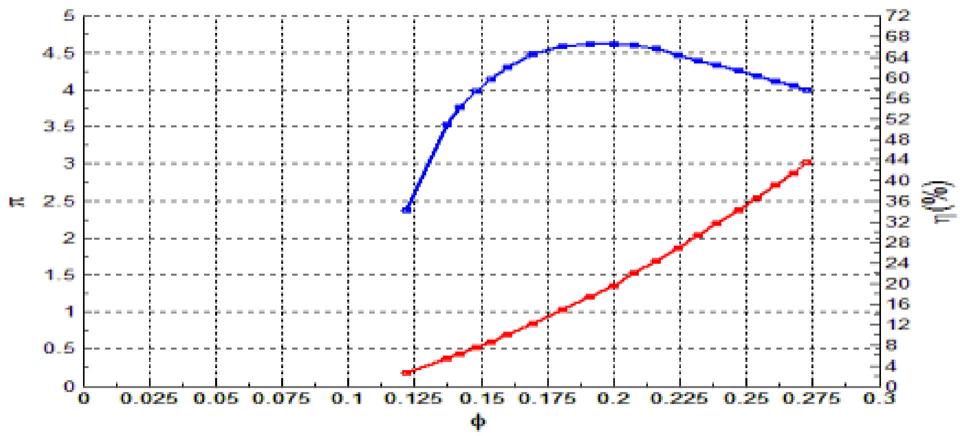

Figure: 3 Power number and efficiency for 38.47 rpm PAT at 1100 rpm speed

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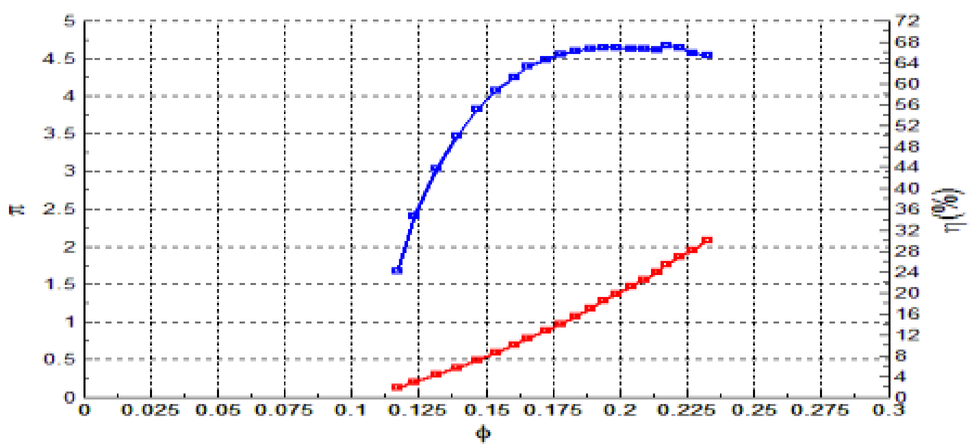


Figure 4: Power number and efficiency for 38.47 rpm PAT at 1300 rpm speed

Figure 4 indicates that the PAT has about 67% best efficiency at 1300 rpm shaft speed. The power number available at BEP from the PAT at 1300 shaft speed is about 1.4. The flow number at BEP is about 0.2. We can see that at full load the efficiency of PAT is 65% and flow number is about0.235. At no load condition the efficiency of PAT is about26% and the flow number is about 0.11. The power number at no load is all most zero.

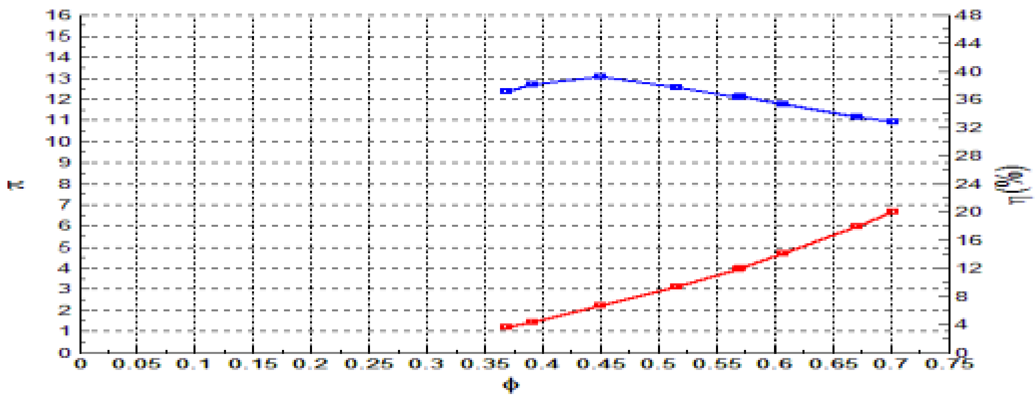
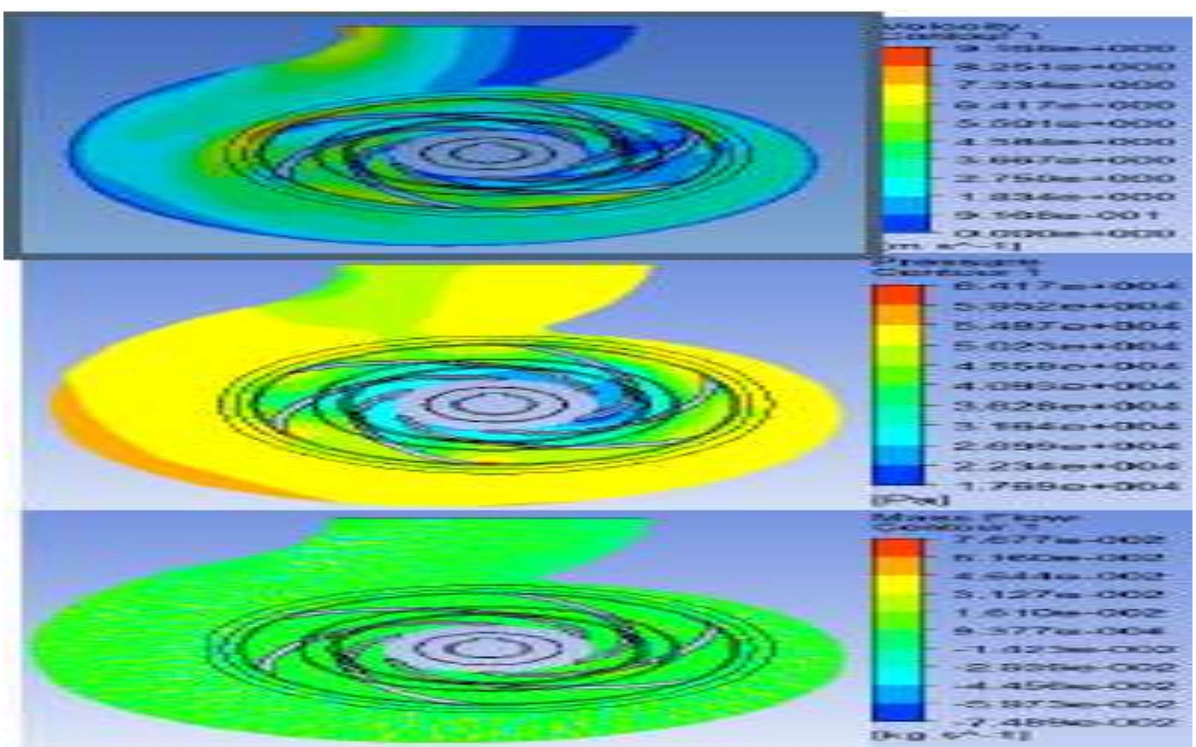
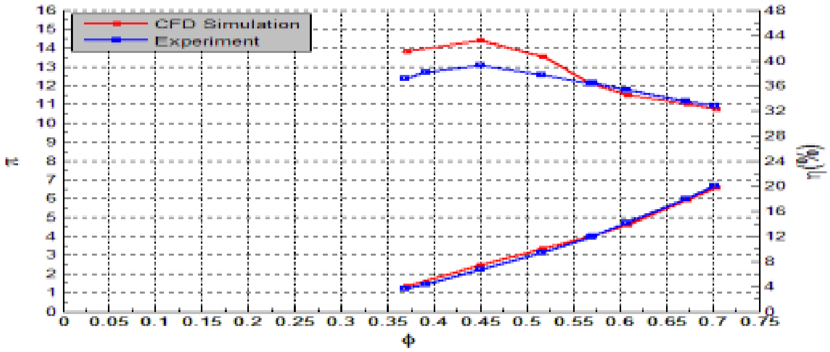


Figure: 5 Head number and efficiency for 53. 83 rpm PAT at 700 rpm speed

Figure 5 indicates that the head number required at BEP is about 12 and flow number is about 0.450 of 53.83 rpm PAT at 700 rpm shaft speed. The head number at full load condition is about 28. At no load condition the head number is about 8. The flow number at full load condition is about 0.7 and at the no load condition the flow number is about 0.35. Same work was simulated using CFD.





Comparison between Power number and Efficiency of experimental results and CFD simulation results for 53. 83 rpm PAT for 700 rpm speed.

VI. RESULT AND CONCLUSION

In this study, two PATs having different specific speed one is 38.47rpm and 53.83rpm are considered. Study shows that the power number and head number obtained from experimental work shows good agreement with CFD simulation results for all range of the flow numbers in both of PATs. CFD simulation efficiency obtained for 38.47rpm PAT has 6% variation compared to Experimental efficiency whereas in 53.83rpm PAT this variation is about 5%. So further study can be carried out using CFD simulation for different operating speed without performing experimental work

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