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Experimental Investigation of Centrifugal Pump Working as Turbine for Small Hydropower Systems

Abstract: An experimental investigation of centrifugal pump has been carried out to study its characteristics in pump and turbine mode operation. By using the experimental results of tested pump and pumps of some previous researchers, new correlations have been developed by using its best efficiency and specific speed in pump mode. Values obtained from the derived correlations show good match with experimental results. These correlations would be very helpful for the performance prediction of pump working as turbine.

Key words: PAT; Centrifugal pump; Specific speed; Best efficiency point

NOMENCLATURE

H	head, m	Ns	specific speed, (m, m^3/s)
Q	discharge, m^3/s	PAT	pump as turbine
D	impeller diameter, m	BEP	best efficiency point
P	power, kW	CFD	computational fluid dynamics
n	rotational speed, rps	EPBT	energy pay back time
g	acceleration due to gravity, m/s^2	GHG	green house gas
h	head ratio at best efficiency, ($Head_{turbine}$/$head_{pump}$)		
q	discharge ratio at best efficiency, (discharge turbine/discharge pump)		

Greek Symbols

φ	discharge number	η	efficiency
π	power number	ψ	head number
ρ	density, kg/m^3		

Subscript

p	pump
t	turbine

1. INTRODUCTION

Centrifugal pump working as turbine is a good alternative for power generation through small and micro hydropower schemes. High equipment cost in small and micro hydropower projects is a major complication in adequate utilization of all small hydropower (especially micro) potential for electricity generation. Due to

rapid increase in energy consumption, the requirement of such alternatives for electricity generation has been increased. Also, high prices, fast depletion rate and environmental implications of fossil fuels create problems in electricity generation through conventional energy sources. Electricity generation through renewable energy sources is appropriate solution for these problems. In addition to this, economic development through renewable energy industry and sustainable energy sector create more employment, which leads to social development of the nation [1]. Among all renewable resources, small hydropower (SHP) is one of the most promising sources of energy generation. In developing countries, small and micro hydropower plants are very effective source for electricity generation. The energy pay-back time (EPBT) and greenhouse gas (GHG) emissions for SHP generation system are less than other conventional electricity generation system [2]. So, encouragement of small hydropower schemes can solve the problem of energy crises of the country.

Reverse running centrifugal pumps are one of the efficient alternatives for generating and recovering power through small and micro hydropower schemes. Centrifugal pump is a common hydraulic machine and it is easily available at all places. Also, the maintenance and installation costs of centrifugal pumps are less. Centrifugal pumps are mainly used in transportation of liquid, industrial process, heating and cooling systems etc. But apart from handling water, pumps can be used to generate electricity when operate in reverse direction. The concept of electricity generation through reverse running centrifugal pump is not new. Around 80 years ago, the research on this field had been started [3]. A large number of theoretical and experimental studies have been done for prediction of performance of reverse running centrifugal pumps. But still there is a need to explore this area more deeply to harness the advantages of this technology for sustainable development.

Selection of a proper pump as turbine (PAT) for a site is a big problem in installation of pump in small hydro-site.   Several researchers viz. Stepanoff, Childs, Sharma, Wong, Williams, Alatorre-Frenk etc. presented some relations for predicting the performance of pump as turbine (PAT). These relations were based upon either pump efficiency or specific speed. But deviation between experimental and predicted reverse operation of standard pumps have been found to be more than 20% [4]. The objective of these relations is to calculate the best efficiency point (BEP) of pump for turbine mode by using the pump operation data provided by the manufacturer. In 1962, Childs [5] presented the PAT prediction method based on the efficiency of pump. Similar type of approach was then presented by McClaskey and Lundquist [6] and Lueneburg and Nelson [7] in 1976 and 1985 respectively.   Hancock [8] stated that for most pumps the turbine BEP is lies within ±2% of the pump mode BEP. Also Grover [9] and Hergt [10] proposed PAT prediction method based on specific speed for turbine mode (obtained similar to pump specific speed). The Grover's method is applicable for the turbine mode specific speed range 10-50 [11]. Comparison between experimental results and methods proposed by the above researchers show relatively large deviations therefore, the use of these formulae had been confined for approximate selection of PAT. Fig 1 shows the classification of methods proposed by various researchers for PAT prediction in chronological order.

Computational fluid dynamics (CFD) have also been applied to the reverse operation of centrifugal pumps. Till now CFD techniques are not successful to present the correct performance prediction of PAT. Many researchers have tried to explore the operation of PAT using CFD but large deviation in results have been found out between experimental and CFD results. Derakhshan and Nourbakhsh [12] compared theoretical, experimental and CFD results of a PAT but large deviations were observed between CFD and experimental results. Modeling of each part of pump is very difficult in CFD due to which the accurate estimation of losses is difficult.

In this study, an experiment was done for a centrifugal pump of specific speed 18 (m, m^3/s) to study the performance characteristics in both the pump and turbine mode. Then, the results of the studied pump and some studied pump by other researchers were used to predict the best efficiency point (BEP) of a centrifugal pump

working in turbine mode. These predicted correlations are developed using the efficiency and specific speed in pump mode. The objective is to develop these relations are to improve the selection criteria for pump which is to be used in turbine mode.
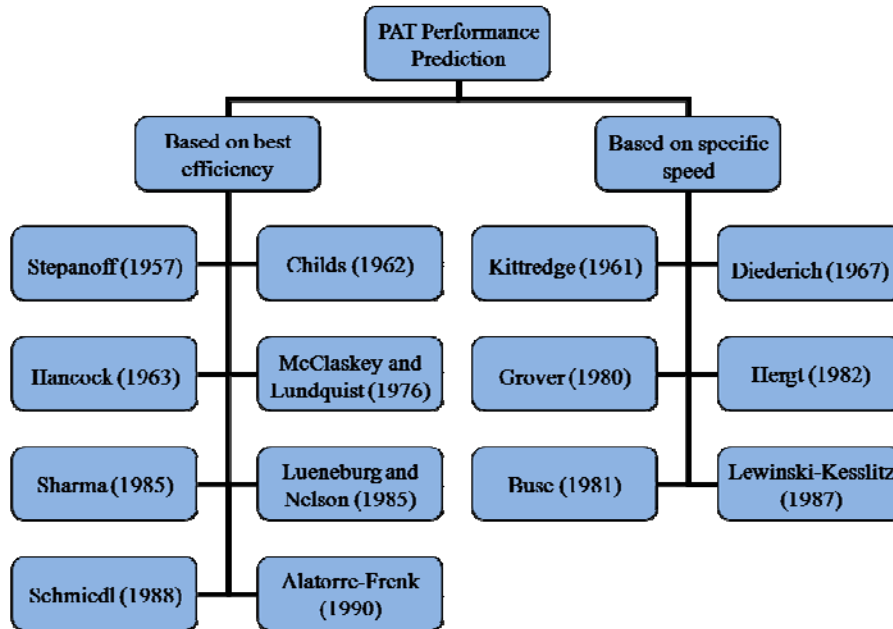

Fig. 1: PAT performance prediction methods proposed by researchers

2. EXPERIMENTAL SETUP

A complete schematic of the experimental set up is shown in Fig 2. The experimental set up was installed at Alternate Hydro Energy Centre, Indian Institute of Technology, Roorkee. A Kirloskar (KC 100-65-315) pump of specific speed 18 (m, m^3/s) was selected to operate in pump and turbine mode.   The maximum head, flow and input power for the pump are 32.8 m, 0.0148 m^3/s and 8.18 kW respectively. The discharge was measured using venturimeter. Apart from this manometer, pressure gauge, valves, tank and pipes are also installed in the experimentation set up. To measure the output power a synchronous generator of 12.5 kVA was coupled to the PAT. The pump/turbine generator set is shown in Fig 3. The PAT was operated at the speed of 1500 rpm. An auxiliary pump is also used to supply the water at desired head and flow rate. This auxiliary pump supply water from the tank to the inlet of the pump testing in reverse mode. To regulate the excessive flow a by pass pipe was connected between the upper pipe and tank. The components of the experimental setup are listed in Tab. 1.
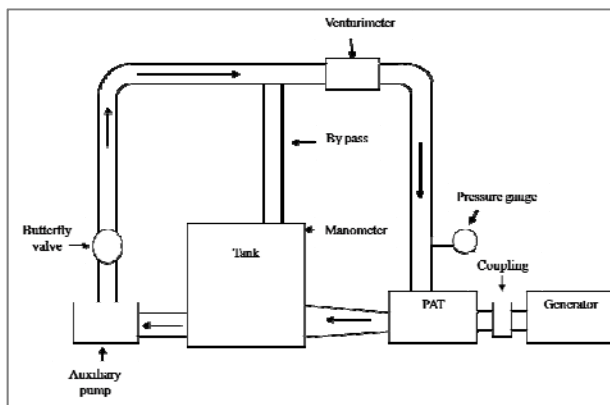

Fig. 2:   Schematic layout of experimental setup

Fig. 3:   Tested centrifugal pump coupled with generator

Tab. 1: Components for PAT experimental set-up

S. No.	Component	Specifications
1.	Metallic open tank	1.35m×1.35m×1.6m
2.	Constant speed motor pump	Head 32.8 m, flow 0.0148 m³/s
3.	Flow meter	Venturimeter
4.	Flow regulation/ Controls	Gate valve
5.	Auxiliary pump	21.8 kW, max. head 60 m, max. flow 0.035 m³/s
6.	Generator	Turbine-Generator set with electrical resistance load.

3. RESULTS

The experiment shows that a centrifugal pump can certainly be used as turbine at various flow rates and head without any technical complications. From the experiment it shows clearly that reverse running centrifugal pump is a simple and easy method to produce power in small and micro hydropower schemes. The easier to way to present the experimental results in the form of non-dimensional parameters viz. head coefficient (ψ), discharge coefficient (ϕ) and power coefficient (π). These non-dimensional parameters are expressed as follows:

$$\psi = \frac{g.H}{n^2.D^2} \qquad (1)$$

$$\phi = \frac{Q}{n.D^3} \qquad (2)$$

$$\pi = \frac{P}{\rho.n^3.D^5} \qquad (3)$$

From the measurements and calculations, the head, power and efficiency curves of the tested pump in pump and turbine mode are drawn and shown in Fig 4-7.
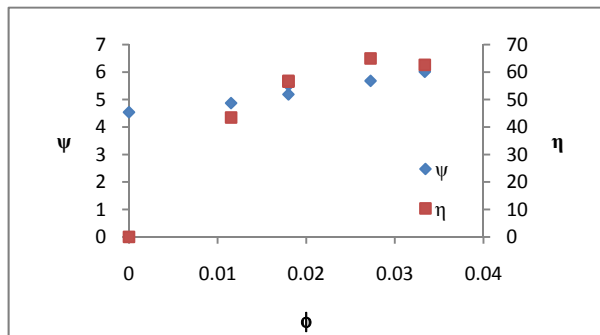

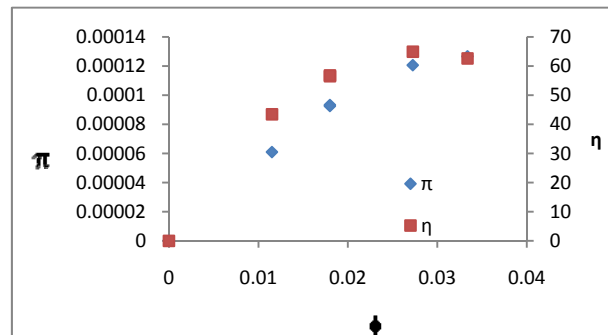**Fig. 4: Head number and efficiency curves of tested pump in pump mode**

Fig. 5: Power number and efficiency curves of tested pump in pump mode

As noted by previous researchers, the above tested pump operates at higher head and flow rates in turbine mode than in pump mode. The higher values of flow rate in turbine operation indicate large handling of water as compared to pump operation. The curves also show that BEP of the pump in turbine operation is lower than in pump operation. Since Efficiencies of pump in turbine operation are more often several percent (3 - 5%) lower [4]. Lower efficiency of above tested PAT indicates higher losses in turbine operation as compared to pump operation.
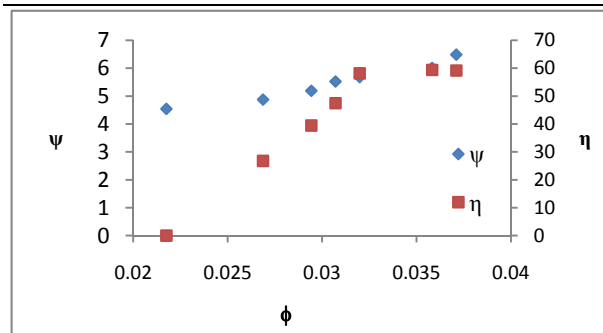
Fig. 6: **Head number and efficency curves of tested pump in turbine mode**
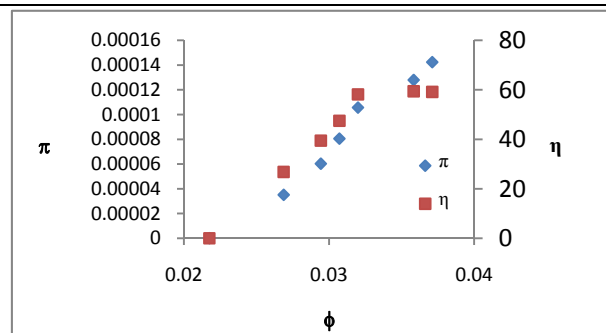

Fig. 7: Power number and efficiency curves of tested pump in turbine mode

4.　PERFORMANCE PREDICTION METHOD FOR PAT

As discussed earlier several methods have been proposed for prediction of reverse operation of centrifugal pump. But error of more than ±20% has been obtained, when compared with experimental results [13]. No method is 100% reliable to predict accurate turbine performance.　Therefore, these methods are confined to preliminary selection of pumps to be used as turbines. Also, methods to predict turbine mode performance from pump mode data have much importance because pump manufacturers provide characteristics curves for pump mode only. However, the preliminary selection of PAT is important to obtain a rough estimation of turbine mode characteristics from pump mode characteristics.　The simplest way to select a pump for a micro hydropower is prediction of turbine characteristics of the pump from its pump mode characteristics. But prediction of turbine characteristics of a pump from pump characteristics is always a big challenge because this gives an approximate turbine mode performance characteristic. Accurate prediction of turbine performance from pump data can make the preliminary selection of PAT for a particular micro hydropower site more easy and quick. There are many factors which affect the predicted performance of a centrifugal pump as listed in Tab. 2. The previous PAT performance prediction methods were based on either efficiency or specific speed of pump. Pump efficiency and specific speed are most useful parameters to define hydraulic performance of a centrifugal pump. Specific speed is a non-dimensional parameter which contains head, discharge and speed of the pump. It acts as tool to compare pumps and select appropriate pump for a particular situation. Therefore, the specific speed parameter cannot be neglected to develop a more accurate method for determining the performance of a PAT. However, there is no parameter which can alone relate to all aspects of final pump design therefore considering both pump efficiency and specific speed can help to predict accurate prediction of reverse operation of centrifugal pump. Before moving further, first it is important to relate pump efficiency with specific speed. Certainly, there is a particular specific speed at which maximum efficiency is obtained. In fact efficiency and specific speed relation is very useful to find approximate relative impeller shapes (centrifugal, axial flow, mixed flow) and average coefficients of centrifugal pumps. In addition to this, it tells about the probable limit of economical operation of pump. The head and discharge values at BEP in turbine mode are higher than in pump mode. This increase in head and flow rate in turbine mode varies according to the specific speed [14].

The above tested pump and four other pumps of specific speed 14.6 [15], 24.5 [16], 35.3[16], 46.1[17] are selected for further analysis. The regression analysis has been carried out for the estimation of relation between the efficiency at BEP and specific speed as it is also shown in Fig 8.

$$\chi = \frac{\eta_p - 0.212}{\ln(N_s)} \quad (4)$$

The term 'χ' gives relation between best efficiency and specific speed in pump mode.

Tab. 2: Factors affecting the predicted performance of a pump

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Mechanical Considerations	High horse power
	High suction pressure
	Operating speed
	Operating temperature
	Running clearances
Pump liquid	Slurries
	Abrasives
	High Viscosity
	Dissolved gasses
System Considerations	Net positive suction head
	Suction and discharge piping arrangement
	Shape of Head–Discharge curve
	Run off conditions
	Vibration and Noise limits
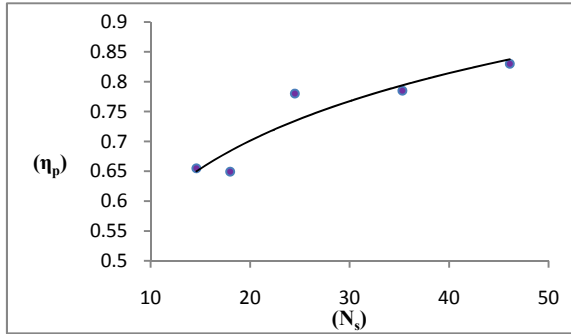
Fig. 8: Variation of efficiency in pump mode with specific speed
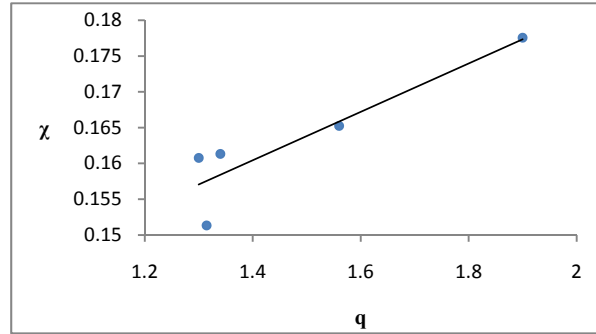

Fig. 9: Variation of correlation coefficient with discharge ratio

From regression analysis, the variation of χ with discharge ratio and head ratio has been presented in Fig 9 and 10 respectively. The relations for head and discharge ratio has been obtained are as:

$$\mathbf{q} = 30.303\,\chi - 3.424 \tag{5}$$

$$h = 41.667\,\chi - 5.042 \tag{6}$$

So, using pump specific speed and best efficiency (which are the basic specifications of a centrifugal pump) the turbine mode prediction is done.
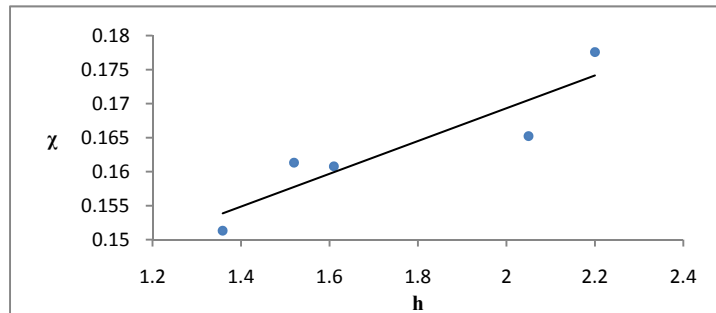

Fig. 10: Variation of correlation coefficient with head ratio

5. COMPARISON

The values of head and discharge ratios for the above selected pumps obtained from methods viz. Stepanoff [18], Childs [5], Alatorre-Frenk [19], Sharma [20], and Grover [9] are compared with experimental results to calculate the errors. The rated errors for head and discharge ratios are plotted against specific speed shown in Fig 11 and 12. The plots clearly indicate that the error has been considerably reduced for newly proposed method. The results obtained from the proposed relations shows almost ±11% deviations. Large deviations are shown in methods proposed by Alatorre-Frenk and Grover. Childs assumed pump best efficiency equal to turbine best efficiency. Therefore, the head ratio values obtained from Stepanoff and Childs are same as shown in Fig 12.
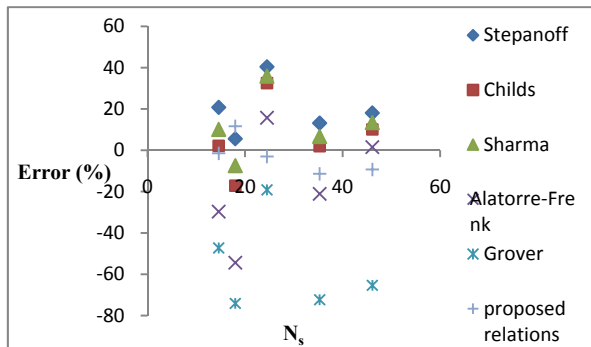

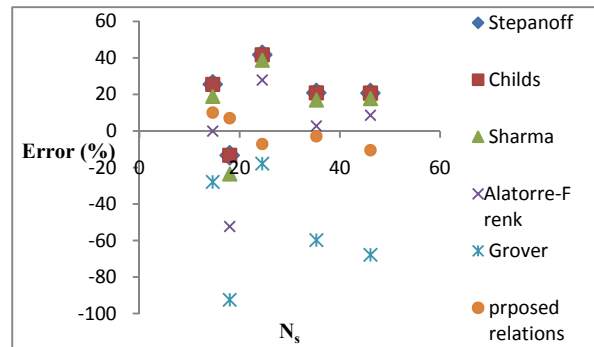**Fig. 11: Comparison of errors for discharge ratio from experiment and various methods**

Fig. 12: Comparison of errors for head ratio from experiment and various methods

6. CONCLUSIONS

A centrifugal pump of specific speed 18 (m, m^3/s) was experimentally tested to study the pump and turbine mode characteristics. Experiment shows that a centrifugal pump can satisfactorily be operated as turbine without any mechanical and technical problem. As compared to pump operation, the pump operates at higher head and discharge values in turbine mode. However, the best efficiency in turbine mode was found 8.53% lower than best efficiency in pump mode. Using experimental data of the tested pump and pumps of some previous researchers, relations are developed to obtain turbine mode characteristics of pump from pump mode characteristics. The developed relations were based on best efficiency and specific speed in pump mode. As compared to other methods, the deviation between experimental results and results obtained by proposed relations is low. This makes the performance prediction of turbine operation from pump mode operation characteristics simpler and closer to accuracy. However some uncertainties are still remains in prediction of turbine mode characteristics using pump operation data.

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