Turbine Mode Performance Evaluation of Centrifugal Pump

Ajit Singh Aidhen, Sandeep Malik, Chavan Dattatraya Kishanrao

ABSTRACT--- This paper presents the performance evaluation results of a radial discharge centrifugal pump obtained through experimentation and CFD simulation. The paper also presents a brief theory behind the difference in the performance of a centrifugal pump operated in pump mode and turbine mode. The pump CFD simulation is performed with Star CCM+ simulation software. The pumps studied are both mono block radial discharge centrifugal pumps with rated speed of rotation per minute as 1400 and 2800 and specific speeds 20.65 (m,m³/s) and 35.89 (m,m³/s) respectively. The CFD results were first validated for pump mode by comparing them with the manufacturer provided performance curves. The results of CFD simulation for turbine mode operation are then compared with experimentally obtained results. The paper also presents a brief theory about PAT concept.

Keywords: Centrifugal pump, PAT (Pump as turbine), B.E.P (Best efficiency point), Experimental study, Computational fluid dynamics..

I. INTRODUCTION

Depleting fossil fuels and environmental concerns due to usage of fossil fuels have shifted the focus of researchers towards harnessing more and more renewable energy [1]. Hydropower is one of the major renewable energy sources. There are concerns establishing large hydropower projects such as disturbance to the fresh water ecosystem, rehabilitation of people, cost etc. Small run off type hydropower plants are more suitable for high terrain villages where supply of electricity by grid is not feasible. This high terrain area having small flow streams of water can be provided electricity by stand alone hyro turbines. The custom designed hydro turbines are expensive as compared to using pump as turbine. Pump as turbine has many advantages as they are available in wide range, cheap, easy to maintain and handle. Selection of a pump to be used as hydro turbine however would require performance curves of these pumps in turbine mode. The pump manufacturers provide performance curves for the pump mode and not turbine mode. Many researchers have suggested methods for performance evaluation of pump in turbine mode based on pump mode B.E.P and based on specific speed [2–10]. Fig.1 shows the conceptual diagram of pump mode and PAT mode operation with direction of flow, torque and angular velocity. The main differences in pump and PAT are seen in Table-1.
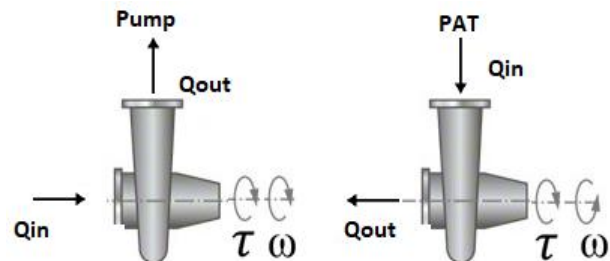

Fig.1. Pump and PAT conceptual diagram

The performance of rotodynamic machine such as pump when used in pump mode and turbine mode differs. In pump mode the energy transfer is affected by angle of vane at impeller output. In turbine mode the energy transfer gets affected by shape of volute. The pump manufacturers provide performance curves for pumps in pump mode. The same pump to be operated as hydro turbine requires research to develop standardized performance curve. Most of the prediction methods suggested by researchers so far however are uncertain and show scope of developing better prediction methods. Pump as turbine is more suitable for standalone electricity generation at constant load where electronic load controllers are used which balance the load between consumer load and dump loads. Addition of hydraulic control devices takes away the cost advantage. Fig.2 shows components of a monoblock radial discharge single stage centrifugal pump used in this study.
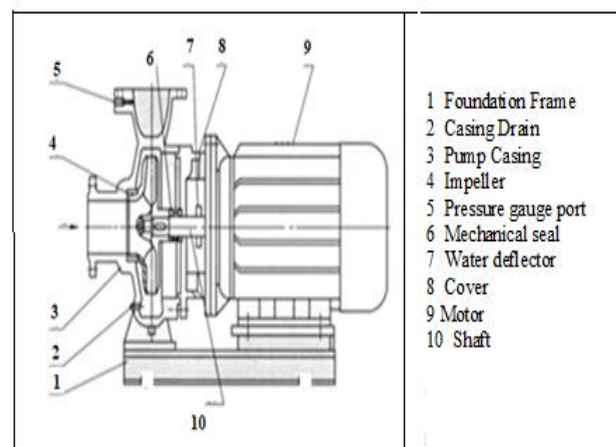

Fig.2. Components of a monoblock centrifugal pump

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Table- 1: Difference in Pump and PAT

	Pump	Pump as Turbine
Head	Increasing flow increases head to be generated by the pump	Frictional losses decrease turbine head with increasing flow.
Energy flow	Input mechanical energy and output hydraulic energy	Input hydraulic energy and output mechanical energy
Torque direction	Same for both mode	Same for both mode
Rotational direction	Direction opposite to turbine mode direction	Direction opposite to Pump mode direction

II. THEORY

The specific speed is used to define the type of pump. The specific speed of an ideal pump geometrically similar to the actual pump, which while running at that speed will deliver a unit of volume, in unit time to a unit head.

$$Specific \ speed \ Ns = \frac{N\sqrt{Q}}{H^{3/4}}$$

Where, N is the pump speed (RPM)
Q is the discharge (m³/s)
H is the head (m)

The pumps are thus classified based on specific speed as Radial flow pumps (Specific speed range 10 to 50), mixed flow pumps (50 to 150), and axial flow pumps (135 to 320). Fig.3. shows the impeller types and the values of specific speeds.
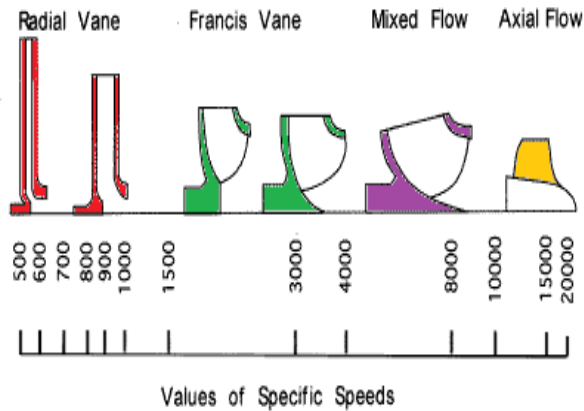

Fig. 3. Impeller type based on specific speed

Radial flow pumps are with specific speed range between 500 to 5000 where the ratio of impeller eye diameter to impeller outside diameter
D1/D2 > 1

Mixed flow pump**s** are with specific speed range between 5000 to 10000 where the ratio of impeller eye diameter to impeller outside diameter
D1/D2 < 1

Axial flow pumps are with specific speed range between 10000 to 15000 where the ratio of impeller eye diameter to impeller outside diameter
D1/D2=1.

Radial discharge pumps are suitable for sites with low flow and high head and axial discharge pumps are suitable for high flow and low head sites.
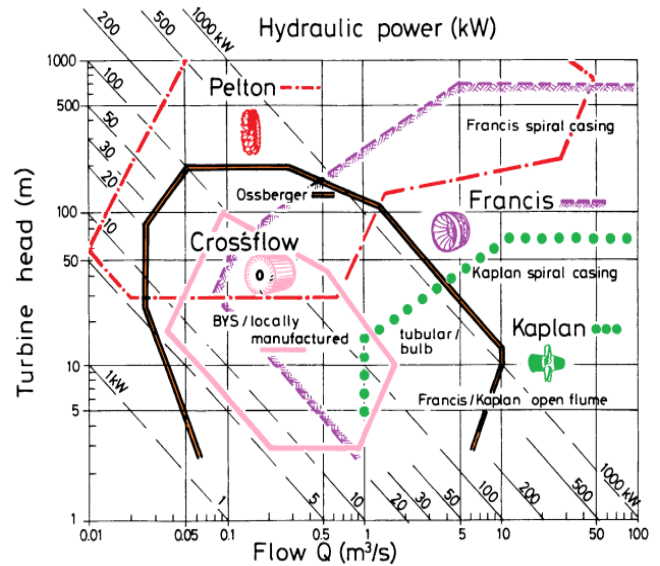

Fig. 4. Application range of different turbines [11]

Fig.4. shows the application range of different types of hydro turbines [11]. The pump as turbine operation is similar to Francis turbine. The pumps although do not have guide vanes as compared with Francis turbine
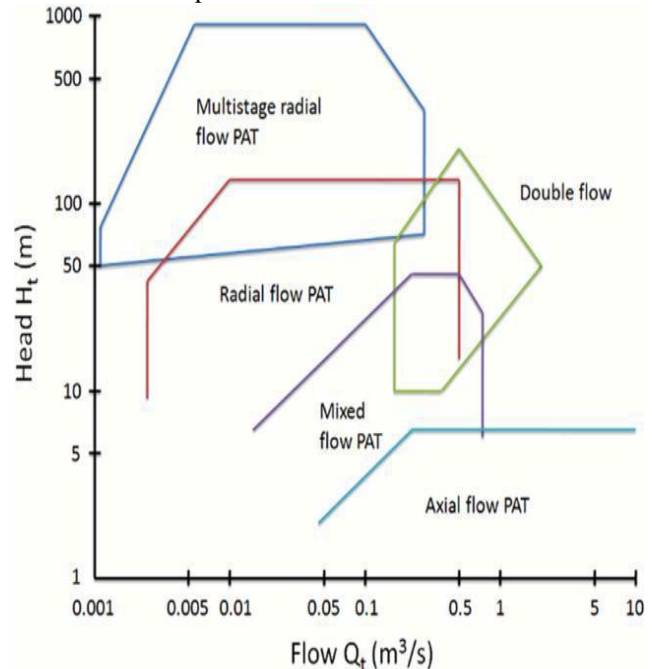

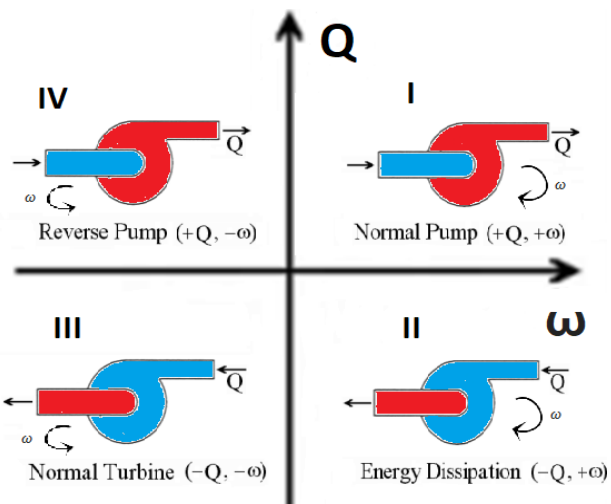**Fig.5. Application range of various types of PAT [11]**

Fig.6. Four quadrant operation of pump

The pump operation can be studied in four quadrants.Fig.6. shows the direction of pump rotation and flow direction in the four quadrants.

Quadrant –I : Normal pump mode (+ ω,+ Q)

Quadrant –II : – Energy dissipation mode (+ ω, – Q)

Quadrant –III : – Normal turbine mode (– ω, – Q)

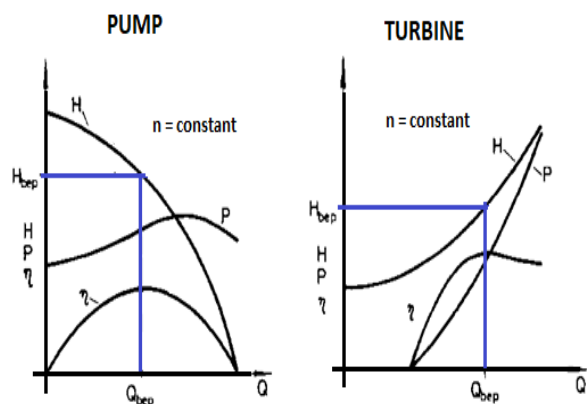Quadrant –IV: – Reverse pump mode (–ω,+ Q)

Fig.7.Typical Performance curves in pump mode operation and and PAT mode operation. [11]

The head in turbine mode operation at B.E.P is higher than pump mode and increases to the right hand side of pump mode B.E.P as seen in Fig.7.

At same speed the performance of pump in pumping mode and reverse mode i.e turbine mode are not similar. The ideal performance curves in both modes can be generated using Euler equation. The design head and flow for both modes can be assumed identical considering ideal conditions. It is important to consider the pump geometry and hydraulic losses for realistic fluids and machines. In the pump mode operation whirl free inlet achieves most favorable operation, at this the head is obtained through outlet velocity triangle with the angle of the vane at outlet of the impeller as an important factor.

The Euler theoretical approach considers infinite blades. However actual pump impeller has finite number of vanes and encounter circulation loss as a result of secondary flow as liquid flows through the impeller passage. As a result of circulation loss the actual head generated is lower than the theoretical head. In the turbine mode operation the performance depends on the volute casing angle and is analyzed through the inlet velocity triangle. The circulation

losses in the reverse operation of the pump are negligible as the impeller diameter on the pump suction side is smaller. This results in higher head and flow at B.E.P in reverse mode as compared to the pump mode. The other losses that are encountered in both pump and reverse mode operation include frictional losses, shock losses and leakage losses.
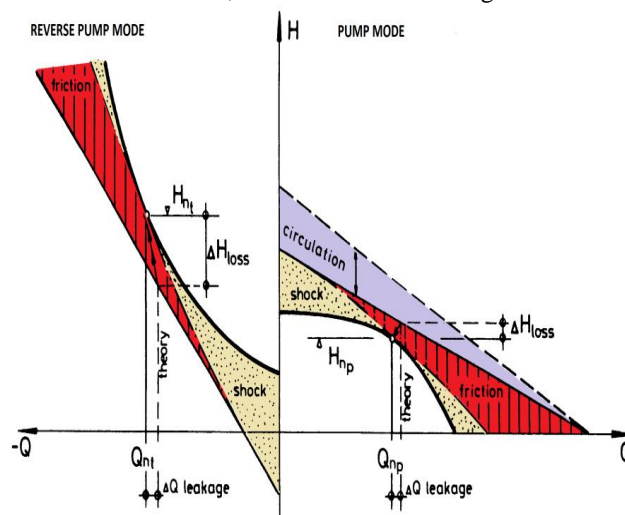

Fig.8. Pump mode and PAT mode friction and hydraulic losses

If

H_P = Actual Head in pump mode

H_E = Euler's head (Theoretical head)

H_T = Turbine head

η_P = Pump efficiency

η_T = Turbine efficiency

Considering no circulation loss following can be expressed

$$H_P = H_E * \eta_P \quad ---- (1)$$
$$H_T = \frac{H_E}{\eta_T} \quad ---- (2)$$

Since theoretical head in both modes is same from the above expressions (1) and (2) it can be derived that

$$\frac{H_P}{H_T} = \eta_P * \eta_T \quad ---- (3)$$

Considering efficiency as close to 80 % in both modes without taking account of geometric effect then it can be expressed that

$$\frac{H_T}{H_P} = \frac{1}{\eta_P^2} = \frac{1}{0.64} = 1.56 \quad ----- (4)$$

This suggests that for a pump to operate in turbine mode its B.E.P would require net head between 30% to 150% that of the pump mode. Many researchers have presented methods and correlations for prediction of turbine mode performance of pump, based on pump mode B.E.P, Pump specific speed and even combined correlations based on both. Although no correlation or method proposed so far is accurate in predicting the turbine mode performance of the pump for an entire range of pumps. The best method hence to predict the turbine mode performance of a pump is experimental evaluation from the manufacturer. This adds to the cost of the pump to be used as turbine and a part of the

cost advantage as compared to the conventional turbine would be lost. It is not practical to conduct experiment on each and every pump for its performance prediction in turbine mode. The development of CFD software has been of great importance in performance prediction of turbo machinery. The flow through the turbo machinery such as pump is complex and three dimensional. The results of simulation are approximate and not too accurate. The results of simulation mainly depend on the accuracy of geometry, accuracy of meshing, selection of appropriate turbulence model and also on boundary conditions applied. The result of CFD simulation hence needs to be validated through experimentation. In the present study two pumps were simulated with STAR CCM+ and the results were then compared with those obtained through experimentation. The details of the two pumps studied are provided in Table 2.

Table- 2. Specifications of the pump used

Parameter	PUMP - I	PUMP - II
Rated R.P.M	1400 R.P.M	2800 R.P.M
Impeller dia.	225 mm	145 mm
Specific Speed	20.65 $(m,m^3/s)$	35.89 $(m,m^3/s)$
Discharge	Radial	Radial
Stage	Single	Single
Type	Mono block	Mono block
Head (B.E.P)	13 m	7.7 lps
Flow (B.E.P)	14 m	11.4 lps
Power	3 hp	2 hp

The experimental setup included 5 h.p centrifugal pump as feed pump with 32 m head and 7 lps flow at B.E.P. The description of the two pumps tested is as mentioned in Table-2. Piping system included valves for flow control and a bypass valve. The entire piping was done using uPVC (polyvinyl chloride).Instrumentation included pressure gauges, electromagnetic flow meter. VFD drive was incorporated for feed pump to increase R.PM and thus the flow. Belt brake dynamometer with digital display spring balance is used for braking system. The performance curves were obtained by running the pump in pump mode and then in turbine mode while maintaining the R.P.M constant. The experimental results were plotted with curves for flow vs head, flow vs power, flow vs efficiency both in pump mode and turbine mode. The readings were taken after obtaining steady state by running the pump in both modes for 10 minutes.
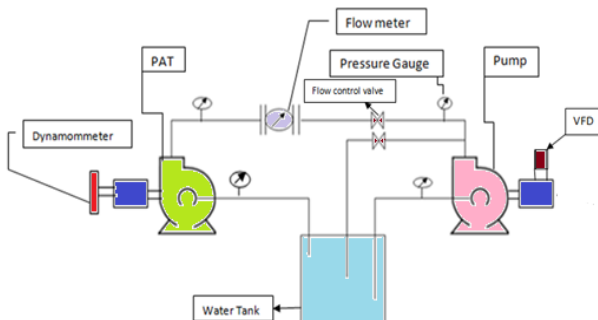

Fig.9. Experimental set up

III. NUMERICAL INVESTIGATION

CFD simulation software has been of great advantage in studying the internal flow of turbo machinery. The solid geometry of the pumps studied are imported to the simulation software and meshing is performed. The meshing details of the two pumps are shown in Fig.10. and Fig.13. Grid independence is carried out. The smallest mesh that gives mesh independent solution should be used as this reduces simulation run time. The results of simulation mainly depend on boundary conditions, physics used, meshing and turbulence model selected. In this study Realizable k-ε turbulence model is selected and the simulation is performed using Star CCM+ CFD software.

Type of Mesh: Polyhedral + Prisms
First Prism Layer Height: 0.005 mm
Total Height: 1 mm
Number of Layers: 10
Meshing Pump - I: Total Mesh Count – 1,380,220
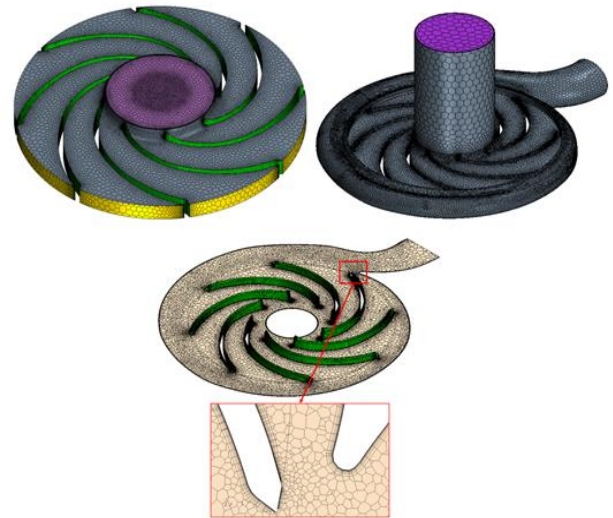**Meshing Pump -II:** Total Mesh Count – 1,886,400

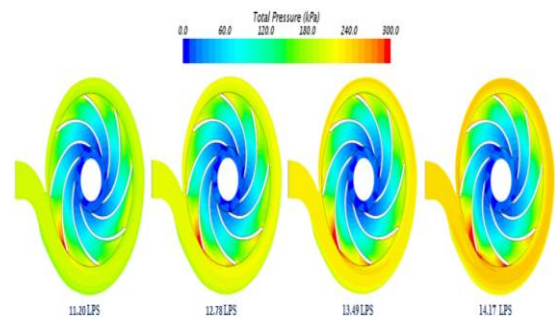**Fig.10. Meshing details Pump**

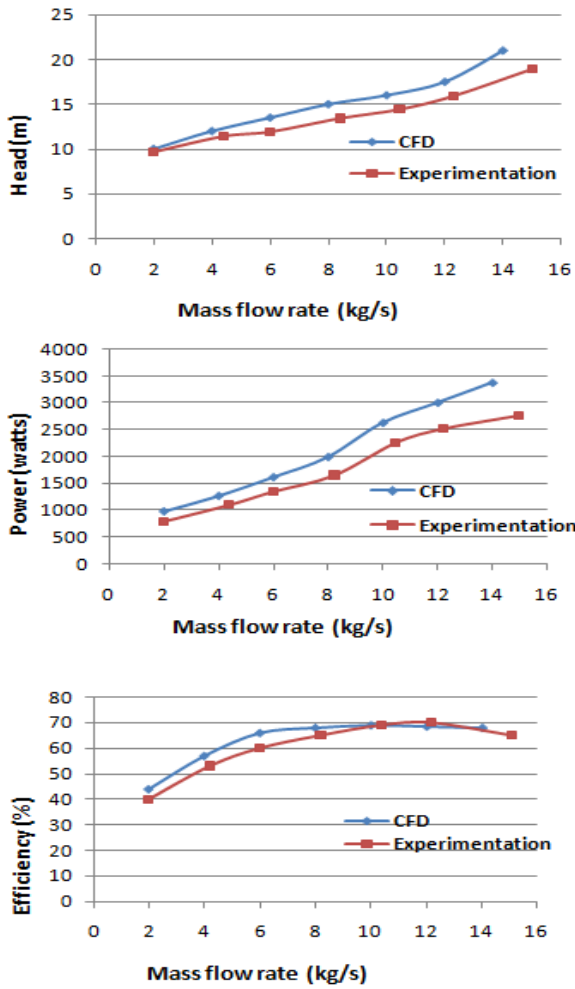**Fig.11. Velocity color plot Turbine mode pump–I (1400 R.P.M)**

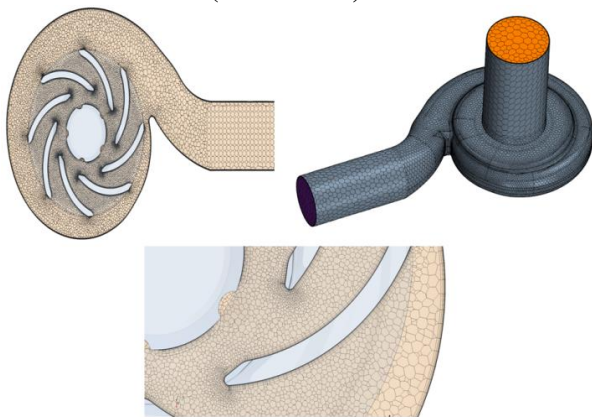**Fig.12. CFD and Experimentation curves for Pump – I (1400 R.P.M)**

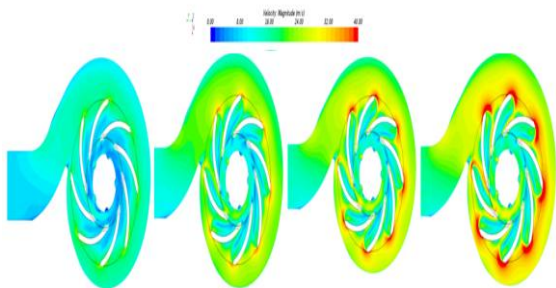**Fig.13. Meshing details pump – II**

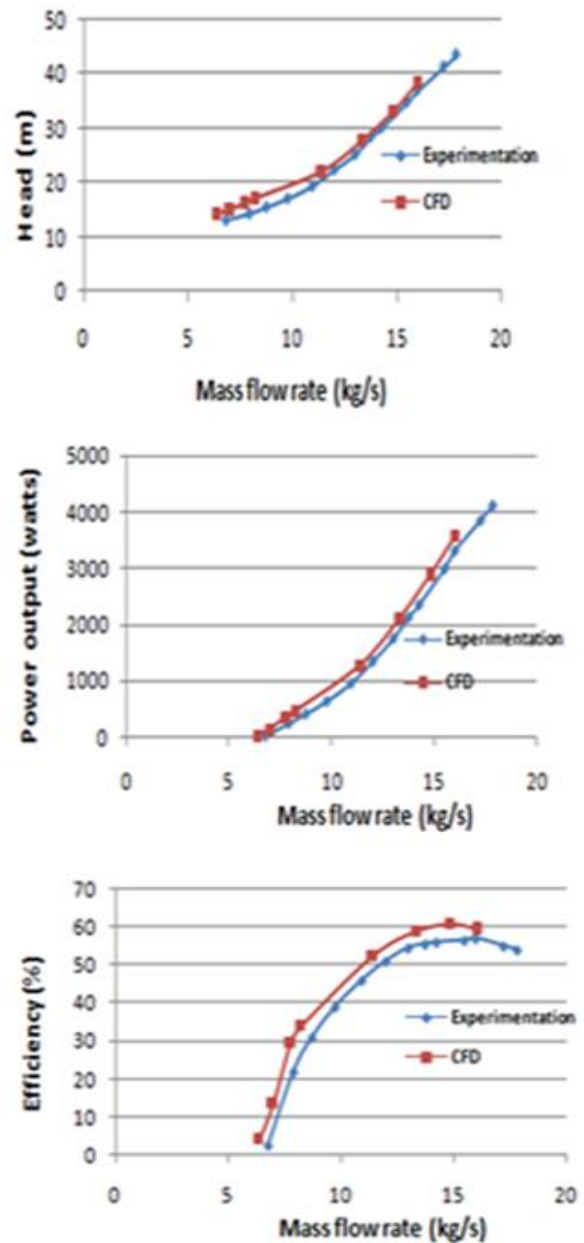**Fig.14. Velocity color plot Turbine mode pump-II (2800 R.P.M)**

Fig.15. CFD and Experimentation curves for Pump–II (2800 R.P.M)

IV. RESULT AND DISCUSSION

Centrifugal pumps are similar in many aspects as Francis turbine. Pump adds energy to the water flowing through the impeller and the same pump extracts energy from flowing water in turbine mode operation. The results reveal that the flow in the turbine mode is higher as compared to pump mode which accompanies higher head increasing the power output. It is important that the output of PAT is well regulated and kept within safe range for the bearings, shaft and couplings. The B.E.P in turbine mode is towards the right of the pump mode B.E.P and is attained at a higher head and flow compared to turbine mode. Higher R.P.M pump face high pressures at the vane tips. The CFD simulation to evaluate pump performance in turbine mode is an effective tool. The results of CFD simulation however

need validation through experimentation. In the present study two centrifugal pumps with specific speed 20.65 (m, m³/s) and 35.89 (m, m³/s) respectively were simulated in Star CCM+ and were also analyzed experimentally. The results of CFD simulation were in agreement with the experimentally obtained results with an error of less than 10%.

V. CONCLUSION

The presented study on pumps performance in turbine mode operation suggests that with certain care the pumps can be used as hydro turbines for small runoff river type hydro power projects. It is important that the output of PAT is well regulated and kept within safe range for the bearings, shaft and couplings. The cost advantage of using pump as turbine over slight drop in efficiency in turbine mode operation of pump makes it a worth considering option. The selection of pump for a suitable site with a certain head and flow is of utmost importance. It is not feasible to simulate a large number of pumps and also not feasible to experimentally study the wide range of pumps in turbine mode. The presented study adds up to the existing literature available

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