Theoretical, numerical and experimental prediction of pump as turbine performance

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Abstract

Insufficient understanding of the correlation between pump and pump as turbine (PAT) performance is a major problem encountered in the PAT selection and design. Therefore, establishment of accurate PAT performance prediction methods is necessary. In this paper, theoretical analysis of the relationship between pump and PAT performance was first performed. A theoretical method of predicting PAT performance is developed using theoretical analysis and empirical correlation. In the next step, computational fluid dynamics (CFD) was adopted in the direct and reverse modes performance prediction of a single stage centrifugal pump. To give a more accurate CFD result, all domains within the PAT control volume were modeled and hexahedral structured mesh was generated during CFD simulation. Complete performance curves of its pump and turbine modes were acquired. To verify the accuracy of theoretical and numerical prediction methods, the pump was manufactured and tested on a PAT open test rig. Results comparison and discussion of the theoretical, numerical and some other methods with experimental data were carried out. Eventually, relatively accurate theoretical and numerical PAT performance prediction methods were developed.

1. Introduction

Small hydroelectric power stations became attractive for generating electrical energy after the oil crisis of the seventies. However cost per kW energy produced by these stations is higher than the hydroelectric power plants with large capacity. Numerous publications in recent years emphasize the importance of using simple turbine in order to reduce the cost of produced electrical energy.

We considered the idea of using pumps as hydraulic turbines an attractive and important alternative. Pumps are relatively simple machine, are easy to maintain and are readily available in most developing countries. From the economical point of view, it is often stated that pumps working as turbines in the range of 1 to 500 kW allow capital payback periods of two years or less which is considerably less than that of a conventional turbine.

Pump manufactures do not normally provide the characteristic curves of their pumps in reverse operation. Therefore, establishing a correlation enabling the passage from the "pump" characteristics to the "turbine" characteristics is the main challenge in using a pump as a turbine. The hydraulic behavior of a pump when rotates as a turbine will be changed. In general a pump will operate in turbine mode with higher head and discharge in the same rotational speed. Many researchers have presented some theoretical and empirical relations for predicting the PAT characteristics in the best efficiency point (BEP). A good literature review has been done by Nautiyal and Anoop Kumar [1]. But the results predicted by these methods are not reliable for all pumps with different specific speeds and capacities.

Most recent attempts to predict performance of PAT, have made using CFD [2–4]. However, without verifying the CFD results by experimental data, they are not reliable. Besides, also all of these simulations included only hydraulic losses.

In the present paper, a simple theoretical method to predict the BEP of PAT using theoretical analysis and empirical correlation on the basis of its pump performance is developed. The method has been compared with two other methods for some reference data. The PAT behavior is very complex and it is difficult to find just a relation to cover all pumps behavior in reverse mode. One idea is using full computational fluid dynamics to simulate PAT performance. In the next step, using commercial flow solver, all geometry of pump including inlet, impeller, chambers and volute has been simulated in modes of direct and reverse operation.
To verify numerical results, a complete micro hydropower test rig was established at Jiangsu University and was used for experimental verification of theoretical and numerical results. The simulated centrifugal pump was made and tested as turbine using this test rig. All required parameters were measured for obtaining complete characteristic curves of the PAT. Finally all theoretical, numerical and experimental results were compared and discussed.

2. Theoretical analysis

The pump inlet and outlet velocity triangles in direct and reverse modes have been shown in Fig. 1. Considerations show that the water inlet angle to the impeller in reverse mode is equal with volute angle. In fact, the volute operates as a guide channel. The water outlet angle from the impeller in turbine operation is equal with the impeller inlet angle (assume: no whirl at exit). So we can consider the same Euler heads for turbine and pump modes:

$$H_{p\text{ Euler}} = H_{t\text{ Euler}}$$  \hspace{2cm} (1)

Due to slip of finite blade number, pump and turbine theoretical head can be written as:

$$H_{p\text{ th}} = \mu H_{p\text{ Euler}}$$  \hspace{2cm} (2)

$$H_{t\text{ th}} = H_{t\text{ Euler}} / \lambda$$  \hspace{2cm} (3)

Where $\mu$ is slip factor for pump operation $\mu < 1$, $\lambda$ is slip factor for turbine operation. The slip factor for reverse mode is approximately equal to 1.0 [5].

Considering its two modes' hydraulic efficiency, Eqs. (2) and (3) can be written as:

$$H_{p} = H_{p\text{ th}} / \eta_{ph} = \mu H_{p\text{ Euler}} \eta_{ph}$$  \hspace{2cm} (4)

$$H_{t} = H_{t\text{ th}} / \eta_{th} = H_{t\text{ Euler}} / (\lambda \eta_{th})$$  \hspace{2cm} (5)

Therefore, we have:

$$h = \frac{H_{t}}{H_{p}} = \frac{b}{\eta_{p}} \left( \eta_{th} = \eta_{ph} \eta_{th}, b = 1 / \lambda \mu \right)$$  \hspace{2cm} (6)

Where $a$ and $b$ greater than 1.0

The reverse and direct operation leakages ratio is [6]:

Fig. 1. a) Pump impeller outlet velocity triangle. b) Turbine impeller inlet and outlet velocity triangles.
The reverse and direct operation theoretical flow rate ratio is [7]:

\[ \frac{Q_{lt}}{Q_{op}} = \sqrt{\frac{H_t}{H_p}} = \frac{b^{0.5}}{\eta_p^{0.5/2}} \]  \hfill (7) 

The reverse and direct operation theoretical flow rate ratio is [7]:

\[ \frac{Q''_t}{Q''_p} = \sqrt{\frac{H_t}{H_p}} = \frac{b^{0.5}}{\eta_p^{0.5/2}} \]  \hfill (8) 

\[ q = \frac{Q_t}{Q_p} = \frac{Q''_t + Q_{lt}}{Q''_p - Q_{lt}} = \left( \frac{Q''_p + Q_{lt}}{Q''_p - Q_{lt}} \right) \frac{b^{0.5}}{\eta_p^{0.5/2}} = c \frac{b^{0.5}}{\eta_p^{0.5/2}} \]  \hfill (9) 

where \( c > 1.0 \).

**Table 1**

<table>
<thead>
<tr>
<th>( D_1 )</th>
<th>( z )</th>
<th>( e )</th>
<th>( \beta_1 )</th>
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The values of $a$, $b$ and $c$ may be changed for various pumps. They depend on pump specific speed, pump size, mechanical tolerances and manufacturing accuracies.

In the next step we try to find the average and the best values of above $a$, $b$ and $c$.

3. Empirical correlation

Figs. 2–5 show some data reported by other researchers for PAT head ratio ($h$) and discharge ratio ($q$) based on pump head and discharge respectively [6,8–10]. In these figures, $h$ and $q$ of PATs are shown versus pump specific speed and maximum efficiency. It can be observed that two pumps with the same specific speeds have different $h$ and $q$. The best data for $a$, $b$ and $c$ can be chosen from Figs. 2 and 3: $a = c = 1.1$ and $b = 1.2$, therefore:

$$h = \frac{1.2}{\eta_p}$$

(10)

$$q = \frac{1.2}{\eta_p^{0.25}}$$

(11)

The comparison between new formula, Stepanoff [7] and Sharma [11] methods are shown in Figs. 4 and 5.

From Fig. 6, the pump specific speed can be selected using turbine specific speed with the following estimated relation:

$$N_{sp} = 1.125 N_{st} + 1.73$$

(12)

4. Experimental setup

Fig. 7(a) an open pump test bed used for pump mode test. It was established at Fluid machinery quality and technical inspection center of Jiangsu University. Its accuracy class could reach first grade accuracy.

A complete laboratory model of PAT open test rig, as shown in Fig. 7(b) was set up at Jiangsu University. High pressure fluid required for PAT energy recovery is supplied by a feed pump. An
energy dissipation pump is installed to consume energy generated by PAT and regulate PAT’s rotational speed. A torque meter with an uncertainty of $\pm 0.2\%$ is put between PAT and energy dissipation pump to measure torque and rotational speed of the PAT. The discharge was measured by a turbine flow meter with an uncertainty of $\pm 0.5\%$. PAT inlet and outlet pressure were measured by pressure gauge with an uncertainty of 0.4. After measuring all parameters, pump two modes’ head, shaft power, efficiency and performance curves were obtained.

5. Pump main geometric parameters

The focus of the investigation is a single stage centrifugal pump with a rotational speed of 1500 rpm. Table 1 lists the main geometric parameters of the designed pump. Fig. 8 shows the tested pump.

6. Numerical investigation

ANSYS-CFX is a commercial 3D Navier-stokes CFD code that utilizes a finite-element based finite-volume method to discrete the transport equations. It is a fully-implicit solver, thus it creates no time step limitation and is considered easy to implement. It is also a coupled solver meaning that the momentum and continuity equations are solved simultaneously. This approach reduces the number of iterations required to obtain convergence and no pressure correction term is required to retain mass conversion, leading to a more robust and accurate solver.

The fluid was split into five component parts; they were pump inlet pipe, front and back chambers, impeller and volute as shown in Fig. 8(d). This separation allows each mesh to be generated individually and tailored to the flow requirements in that particular component. To get a relatively stable inlet and outlet flow, four times of the pipe diameter have been extended in the PAT inlet and outlet section.

6.1. Mesh generation

ICEM-CFD was used to generate structured hexahedral grid for each component part. A grid independent test of turbine mode’s performance was performed; it is found that when mesh numbers are around 1 million, the variation of efficiency is within 0.5% as is indicated in Fig. 9. The final mesh number of volute, impeller, front chamber, back chamber, outlet pipe and total number are 479710, 378222, 134956, 129700, 174720, and 1297308 respectively.

Care was taken in the pump inlet pipe mesh generation in order that mesh between the interface of inlet pipe and front chamber leakage is almost the same. The $y^+$ near the boundary wall was around 40. Due to the complexity of generating a structured mesh based on geometry, great efforts had been taken in the mesh

| $Q/Q_{BEP}$ | 0.70 | 0.84 | 1.00 | 1.11 | 1.33 |
| EXP (%)     | 58.27 | 62.68 | 63.08 | 62.59 | 58.82 |
| CFD (%)     | 56.70 | 60.25 | 61.74 | 61.69 | 58.45 |
| Error (%)   | $-2.69$ | $-3.88$ | $-2.12$ | $-1.44$ | $-0.63$ |

Table 2

Pump experimental and numerical efficiency.

| $Q/Q_{BEP}$ | 0.70 | 0.84 | 1.00 | 1.11 | 1.33 |
| EXP (m)     | 19.31 | 18.47 | 17.17 | 16.03 | 14.13 |
| CFD (m)     | 18.15 | 17.53 | 16.65 | 15.89 | 13.76 |
| Error (%)   | $-6.01$ | $-5.09$ | $-3.03$ | $-0.87$ | $-2.62$ |

Table 3

Pump experimental and numerical head.

| $Q/Q_{BEP}$ | 0.70 | 0.84 | 1.00 | 1.11 | 1.33 |
| EXP (kW)    | 3.23  | 3.48  | 3.83  | 4.05  | 4.49  |
| CFD (kW)    | 3.15  | 3.46  | 3.80  | 4.05  | 4.41  |
| Error (%)   | $-2.48$ | $-0.57$ | $-0.78$ | 0.00  | $-1.78$ |

Table 4

Pump experimental and numerical shaft power.
generation of volute. Fig. 10 gives a general view of the generated meshes.

6.2. Solution parameter

The turbulence selected was k-ε model. The advection scheme was set to high resolution. The convergence criterion was 10^-15. The fluid selected was ideal water at 25 °C. All the wall surface roughness within the control volume was set to 100 µm. The inlet and outlet boundary conditions were set to static pressure inlet and mass flow rate outlet [12]. By changing the mass flow rate, the performance curves of PAT were acquired.

As the motion of the impeller blades relative to the stationary volute was central to the investigation, the analysis must involve multiple frames of reference. The volute and outlet pipe were set in stationary frame and the impeller was set in rotating frame. The interfaces between two stationary components, rotary and stationary components were set to general grid and rotor stator interface respectively.

7. Comparison of theoretical, experimental and numerical results

To validate the accuracy of CFD results, comparison between pump two modes’ experimental results and CFD results are presented. In the process of numerical simulation, all domains except the leakage through the balancing holes and mechanical seals are included, so the simulated efficiency includes volumetric loss through the wear ring and mechanical loss caused by disc friction. The volumetric loss caused by balancing holes and mechanical loss resulted from mechanical seal and bearings are neglected.

The investigated pump mode’s performance data are presented in Fig. 11. Tables 2–4 list its experimental and numerical results. Fig. 11 indicates that the tendency of numerical predicted pump performance curves are in agreement with those of experimental data. Tables 2–4 illustrate the maximal division of efficiency, head and shaft power within the scope of 0.7–1.33Q_BEP are −3.88%, −6.01% and −2.48% respectively. The relative error of predicted η, H and P_shaft to experimental data at BEP is −3.03%, −0.78%, −0.78% respectively. This demonstrates that CFD could predict pump performance with an acceptable accuracy.

The turbine mode’s performance data are presented in Fig. 12. Tables 5–7 list its experimental and numerical results. As is shown in Fig. 12, the tendency of PAT numerical predicted performance curves are in agreement with those of experimental data. Tables 5–7 indicate that the maximal divisions of efficiency, head and shaft power within the scope of 0.81–1.27Q_BEP are 3.33%, 3.09% and 5.71% respectively. The relative errors of predicted η, H and P_shaft to experimental data at BEP is 0.55%, 3.09% and 3.51% respectively. Numerical predicted efficiency, pressure head and shaft power are higher than those of experimental results. The agreement of numerical and experimental results shows that CFD could also predict PAT performance with an acceptable accuracy.

Table 7

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<th>Q/Q_BEP</th>
<th>0.81</th>
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Table 8

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8. Conclusions

Theoretical, numerical and experimental investigation into prediction methods of pump as turbine performance was carried out.

On the basis of former research results, a theoretical method of predicting BEP of PATs performance was developed through theoretical analysis and empirical correlation. Meanwhile two other predictions methods were presented for the purpose of comparison.

In the next step, a centrifugal pump was simulated using CFD in direct and reverse modes. In the process of numerical simulation, all domains except the leakage through the balancing holes and mechanical seals are included, so the simulated efficiency includes volumetric loss through the wear ring and mechanical loss caused by disc friction. To verify numerical results, the simulated pump was made and tested at the established test rig. CFD results were in good agreement with its two modes’ experimental data. The slight difference between experimental and numerical results may be caused by the simplification of the experimental setup.
attribute to the neglection of leakage loss through balancing holes, mechanical loss caused by mechanical seal and bearings and the surface roughness value set on the machine's surface.

BEP characteristics predicted by the theoretical method and CFD were more accurate than other two methods for our case study, but still there is minor difference with experimental data. Future works on theoretical methods and more experimental data with more case studies can improve the prediction of PATs performance.

Acknowledgment

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References


