Numerical prediction of pressure fluctuations in a prototype pump turbine base on PANS methods

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Abstract. Unsteady flow and pressure fluctuations within a prototype pump turbine are numerically studied using a nonlinear Partial Averaged Navier Stokes (PANS) model. Pump turbine operating at different conditions with guide vanes opening angle 6° is simulated. Results revealed that the predictions of performance and relative peak-to-peak amplitude by PANS approach agree well with the experimental data. The amplitude of the pressure fluctuation in the vaneless space at turbine mode on a “S” curve increases with the decrease of the flow rate, and it has maximum value when it runs close to runaway line at turbine braking mode. The amplitude of the pressure fluctuation in the vaneless space at turbine braking mode on a “S” curve decreases with the reduce of the flow rate. The above high pressure fluctuations should be avoided during the design of pump turbines especially those operating at high-head condition.

1. Introduction

Pump-turbine played an important role in the optimization of power network. The starting period and stopping period of a pump-turbine ran at a high frequency, leading to a lot of instability problems due to the existence of “S” characteristics. When a pump-turbine ran in the “S” region, pressure fluctuation was very dangerous, even leading to some serious accidents.

Pump-turbines were always running at partial condition with the power grid changing, leading to flow separations and stall phenomena in the pump-turbine. The study of instability of a pump-turbine should overcome the difficulties in the simulation of strong vortex and separation flow. Large-scale coherent structures play a crucial role in the internal instabilities. For predicting the fluctuation of the dominant scale of vortexes, the traditional Reynolds-averaged Navier-Stokes (RANS) method suffers from inherent physical limitations[1], which couldn’t capture the flow with all kinds of scales. Large eddy simulation (LES) may not be computationally viable due to the large computational cost. Instead, the mixed computational methods, which combined the desirable aspects of RANS and LES, were used in engineering computation [2]. Speziale [3] proposed a new turbulence model that combines the advantages of RANS method with those of LES.
Ji [4] studied the unsteady cavitating turbulence flow around a highly skewed model marine propeller based on the PANS method. Huang [5] used PANS model to investigate the cavitating flow around a hydrofoil. The PANS models mentioned above were all modified from standard $k-\varepsilon$ turbulence model, while the standard $k-\varepsilon$ turbulence model was poor in the simulation of strong swirling flows [6]. Most of the RANS turbulence models solved the shear stress by linear difference scheme and they were isotropic models[7], so they couldn't capture all kinds of vortexes in the pump-turbine well.

In this paper, a new nonlinear PANS turbulence model was proposed. The results based on the new nonlinear PANS turbulence model were compared with experimental results. Then the instability of a pump-turbine was studied by the nonlinear PANS model.

2. Nonlinear PANS model

For incompressible flow, the continuity equation and Reynolds averaged Navier-Stokes equations:

\[
\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} + \frac{\partial (p)}{\partial x_j} + \frac{\partial \tau (V_i, V_j)}{\partial x_j} = -\frac{\partial (p)}{\partial x_j} + \nu \frac{\partial^2 U_i}{\partial x_j \partial x_j}
\]

\[
\frac{\partial^2 (p)}{\partial x_j \partial x_j} = -\frac{\partial U_i}{\partial x_j} + \frac{\partial \tau (V_i, V_j)}{\partial x_j \partial x_j}
\]

\(V_i\) is partitioned into resolved and unresolved parts in the instantaneous velocity field, using an arbitrary homogeneous filter.

\[
V_i = U_i + u_i, \quad U_i = \langle V_i \rangle
\]

where \(U_i\) is the resolved velocity field; \(u_i\) is the unresolved field. It is used by equation 4 instead of the resolved field.

The additional non-linear term \(\tau (V_i, V_j)\), which is the generalized central second moment, is defined as:

\[
\tau (V_i, V_j) = (V_i V_j - \langle V_i \rangle \langle V_j \rangle)
\]

The PANS models are:

\[
\frac{\partial (p_k)}{\partial t} + \frac{\partial (\rho U_j k)}{\partial x_j} \alpha_i = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu}{\sigma_u} \right) \frac{\partial k}{\partial x_j} \right] + P_{in} - \rho e_u
\]

\[
\frac{\partial (\rho e_j)}{\partial t} + \frac{\partial (\rho U_j e_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu}{\sigma_u} \right) \frac{\partial e_j}{\partial x_j} \right] + C_{e1} \rho u_{*j} \frac{e_j}{k} - C_{e2} \rho \frac{e_j^2}{k}
\]

Considering the nonlinear turbulence flow in the pump-turbine, the shear stress was solved by nonlinear turbulence model which was proposed by Ehrhard [6].

\[
P_{s} = -\rho U_j \frac{\partial U_i}{\partial x_j}
\]

\[
\overline{U_j U_j} = \frac{2}{3} k \delta_{ji} - 2 C_{\mu \nu} \nu^2 T S_j + C_{\mu \nu} \nu^2 T^2 \left( S_j S_j - \frac{1}{3} S_i S_i \delta_{ji} \right) + C_{4} C_{\mu \nu} \nu^2 T^2 \left( S_i S_j - \Omega_j \Omega_i \right)
\]

\[
+ C_{4} C_{\mu \nu} \nu^2 T^2 \left( S_i \Omega_j - \Omega_i S_j \right) S_{ji} + C_{4} C_{\mu \nu} \nu^2 T^2 S_{ji} S_{ji} + C_{4} C_{\mu \nu} \nu^2 T^3 S_{ji} \Omega_{ji} + C_{4} C_{\mu \nu} \nu^2 T^3 \Omega_{ji} \Omega_{ji}
\]

\[
C_{\mu \nu} = \min \left( \frac{1}{0.9 S_{ij} + 0.4 \Omega^2 + 3.5 S_{ij}} \right), \quad \Omega_j = 1 - \left( \frac{1}{2} \frac{\partial U_j}{\partial x_i} + \frac{\partial U_i}{\partial x_j} \right), \quad S = \frac{k}{\varepsilon} \sqrt{2 \gamma S_{ij}} \Omega_j, \quad \Omega = \frac{k}{\varepsilon} \sqrt{2 \gamma \Omega_j \Omega_j}
\]
where \( C_{\mu\nu} = \beta \frac{k^2}{\varepsilon} \), \( C_1 = -0.2 \), \( C_2 = 0.4 \), \( C_3 = 2.0 - \exp\left(-\left(S - \Omega\right)^2\right) \), \( C_4 = -32.0C_{\mu\nu}^2 \), \( C_5 = -16.0C_{\mu\nu}^2 \), \( C_6 = 16.0C_{\mu\nu}^2 \), \( T \) is turbulence time scale, \( v \) is the turbulence velocity scale.

3. Pump-turbine geometry

Parameters of the model pump-turbine are shown in Tab.1. \( D_1 \) denotes the runner inlet diameter in pump mode; \( Z_S, Z_G \) and \( Z \) are the numbers of stay vanes, guide vanes and runner blades, respectively; \( H_d \) denotes the rated head; \( n \) denotes the rotational speed of the runner; \( Q_d \) denotes the rated discharge. The pump-turbine’s structure is shown in Fig.1.

### Table 1. Parameters of the model pump-turbine

| \( H_d \) (m) | \( Q_d \) (m³/s) | \( n \) (rpm) | \( D_1 \) (m) | \( Z \) | \( Z_S \) | \( Z_G \) |
|---------------|-----------------|---------------|--------------|-------|--------|--------|
| 52.4          | 0.45            | 500           | 0.3          | 9     | 20     | 20     |

![Profile of Pump-turbine](image1.png)

![Mesh of the runner](image2.png)

The model’s grids, which were composed of an unstructured hexahedron and tetrahedron, were developed using ICEM, which is a commercial software package used for CFD discretization. Hexahedral grids were used for the runner and draft tube, and mixed grids were used for the other components. The mesh of the runner is shown in Fig.2. A mesh with about 9 million cells in total was chosen for the simulations. The number of nodes and elements in each part is shown in Tab.2.

### Table 2. Mesh of different hydraulic region

| Hydraulic region | Runner | Guide vanes | Stay vanes | Draft tube | Casing | Total         |
|------------------|--------|-------------|------------|------------|--------|---------------|
| Cells            | 2,734,700 | 1,734,700    | 1,455,324  | 1,173,888  | 1,840,506 | 8,939,118     |
| Nodes            | 2,842,386 | 1,904,000    | 1,585,040  | 1,196,517  | 1,952,039 | 9,479,982     |

4. Results and discussion

4.1. Flow in the pump-turbine with MGV

Streamline in the runner on blade to blade surface at no-load condition is shown in Fig. 3. No reverse flow can be seen at turbine mode(G1 and G2). Reverse flow appears in the runner at run-away point (G3). The runaway point is the point for starting load during the start-up process, it is very important for the instability of a pump-turbine. The appearance of reverse flow may cause large pressure
fluctuation in the pump-turbine. When the pump-turbine runs at turbine braking mode (G4, G5), the stall phenomenon can be seen in each passage of the runner. At zero flow rate point, the stall flow will block all the passages of the runner, and no flow can pass the passage of the runner.

4.2. Pressure fluctuation in the vaneless space
Pressure fluctuations in the vaneless space at different points on the S curve are shown in Fig. 4. When the pump-turbine runs at turbine mode, the amplitude of the pressure fluctuation in the vaneless space is small, and there is no low frequency. As the flow rate decreases, a low frequency signal can be observed. When the flow rate is less than the point G5, the low frequency signal disappears.

The pressure fluctuations at the frequency domain are calculated by fast Fourier transform (FFT), which are shown in Fig. 5. The amplitude of pressure fluctuation increases as the flow rate decreases when the flow rate is larger than the run-away point (G3). The dominant frequency (DF) of pressure fluctuation at each point on the S curve is the blade passing frequency. The second dominant frequency (SDF) is twice of the blade passing frequency at G1 and G2, while it is rotating frequency at G3 and G6.

![Streamlines in the runner](image)

*Figure. 3 Streamlines in the runner*
Figure 4. Pressure fluctuation at time domain
Figure 5. Pressure fluctuation at frequency domain

Table 3. Pressure fluctuation at the vaneless space

|    | \( \Delta H/H \) (%) | DF \( (x_f)_n \) | SDF \( (x_f)_n \) |
|----|----------------------|-----------------|-----------------|
| G1 | 3.6                  | 9               | 18              |
| Cal. | 11.9                | 9               | 18              |
| G2 | Exp.                 | 11.3            | 9               |
| G3 | 15.4                 | 9               | 1               |
| G4 | 14.8                 | 9               | 0.45            |
| G5 | 8.6                  | 9               | 18              |
| G6 | 13.1                 | 9               | 1               |

5. Conclusions
In this paper, pressure fluctuations in the vaneless space at each point of the “S” curve of a pump-turbine were investigated. Results revealed that the predictions of performance and relative peak-to-peak amplitude by PANS approach agree well with the experimental data. The amplitude of the pressure fluctuation in the vaneless space at turbine mode on a “S” curve increases with the decrease of the flow rate, and it has maximum value when it runs close to runaway line at turbine braking mode. The amplitude of the pressure fluctuation in the vaneless space at turbine braking mode on a “S” curve decreases with the reduce of the flow rate. The above high pressure fluctuations should be avoided during the design of pump turbines especially those operating at high-head condition.

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