Instability analysis of a model pump-turbine in vaneless space with different openings of guide vanes

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Abstract: Pump-turbines were always running at partial condition with the power grid changing. Flow separations and stall phenomena were obvious in the pump-turbine. Most of the RANS turbulence models solved the shear stress by linear difference scheme and they were isotropous, so they couldn't capture all kinds of vortexes in the pump-turbine well. At present, Partially-Averaged Navier-Stokes (PANS) has been found better than LES in simulating flow regions especially those with poor near-wall resolution. In this paper, a new nonlinear PANS turbulence model was proposed, which was modified from RNG $k$-$\varepsilon$ turbulence model and the shear stresses were solved by Ehrhard's nonlinear methods. The nonlinear PANS model was used to study the instability of "S" region of a model pump-turbine with misaligned guide vanes (MGV). The opening of pre-opened guide vanes had great influence on the "S" characteristics. Pressure fluctuations in the vaneless space for different opening of pre-opened guide vanes were analyzed. It is found that the "S" characteristics and instability can be improved when the relative pre-opening of MGV is 50%.

1. Introduction

Pump-turbine, which was one of the key components in a pumped storage power station, 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. At present, misaligned guide vanes (MGV) were used in some pumped
storage power stations to improve the “S” characteristics, but the influence of the relative opening of pre-opened guide vanes on the “S” characteristics was unclear.

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\cite{1-2}, 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 \cite{3-4}. Speziale\cite{5} proposed a new turbulence model that combines the advantages of RANS method with those of LES. Girimaji\cite{6} developed a bridging method inspired by the modeling paradigm proposed by Speziale \cite{5}. The method was given the name partially averaged Navier-Stokes (PANS) model and was purported for any filter width \cite{7}.

At present, Partially-Averaged Navier-Stokes (PANS) has been found to be better than that of the LES in flow regions where simulations suffered from poor near-wall resolution \cite{8}. Ji \cite{9,10} studied the unsteady cavitating turbulence flow around a highly skewed model marine propeller based on the PANS method. Wang \cite{11} and Huang \cite{12} 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 \cite{13}. Most of the RANS turbulence models solved the shear stress by linear difference scheme and they were isotropic models\cite{14}, 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:

\[
\frac{\partial U_i}{\partial t} + U_i \frac{\partial U_i}{\partial x_j} + \frac{\partial}{\partial x_j} \left( \frac{\partial V_i V_j}{\partial x_j} \right) = \frac{\partial \langle p \rangle}{\partial x_j} + \nu \frac{\partial^2 U_i}{\partial x_j \partial x_j} \tag{1}
\]

\[
- \frac{\partial^2 \langle p \rangle}{\partial x_j \partial x_j} = \frac{\partial U_i}{\partial x_j} \frac{\partial U_j}{\partial x_i} + \nu \frac{\partial V_i V_j}{\partial x_j \partial x_j} \tag{2}
\]

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

\[
V_i = U_i + u_i \tag{3}
\]

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.

Then we can obtain PANS model by the modification of RNG \( k-\epsilon \) turbulence model, and the equations are:
\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho U_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_1 \left( \mu + \frac{\mu_T}{\sigma_u} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \rho \varepsilon_v \quad (4)
\]

\[
\frac{\partial (\rho \varepsilon_v)}{\partial t} + \frac{\partial (\rho U_j \varepsilon_v)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_2 \left( \mu + \frac{\mu_T}{\sigma_u} \right) \frac{\partial \varepsilon_v}{\partial x_j} \right] + C_{\varepsilon}' \rho_k \varepsilon_v - C_{\varepsilon''} \rho \varepsilon_v^2 \quad (5)
\]

\( P_k \) denotes the production terms of turbulence kinetic energy. Equating the source terms, \( P_k \) in RNG \( k-\varepsilon \) turbulence mode has a relationship with \( P_{ku} \) in PANS model shown in equation 11.

\[
P_k = \frac{1}{f_k} (P_{ku} - \varepsilon_v) + \frac{\varepsilon_v}{f_k} \quad (6)
\]

Considering the non-linear turbulence flow in the pump-turbine, the shear stress was solved by non-linear turbulence model which was proposed by Ehrhard \cite{15}.

\[
P_s = -\rho U_j \frac{\partial U_j}{\partial x_j} \quad (7)
\]

\[
\overline{U_j U_j'} = \frac{2}{3} k \delta_j - 2C_{\mu} \nu^2 TS_{ij} + C_i C_{\mu} \nu^2 T^2 \left( S_{ij} S_{ij} - \frac{1}{3} S_{ij} S_{ij} \delta_j \right) + C_2 C_{\mu} \nu^2 T^2 \left( \Omega_{ij} \Omega_{ij} - \frac{1}{3} \Omega_{ij} \Omega_{ij} \delta_j \right) + C_3 C_{\mu} \nu^2 T^3 \left( S_{ij} - \Omega_{ij} \Omega_{ij} \right) S_{ij} + C_4 C_{\mu} \nu^2 T^3 S_{ij} S_{ij} + C_5 C_{\mu} \nu^2 T^3 S_{ij} \Omega_{ij} \Omega_{ij} \quad (8)
\]

where \( C_{\mu} = \beta \frac{k^2}{\varepsilon} \), \( C_i = -0.2 \cdot C_2 = 0.4 \cdot C_3 = 2.0 \cdot \exp \left( -\left( S - \Omega \right)^2 \right) \cdot C_4 = -32.0 C_{\mu} \), \( C_5 = -16.0 C_{\mu} \), \( C_6 = 16.0 C_{\mu} \cdot T \) is turbulence time scale, \( \nu \) 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 and the structure of MGV is shown in Fig.1.
4. Verification of grid independence

The model’s grids, which were composed of 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.3.

| 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   |

5. Results and discussion

5.1 “S” characteristics with MGV

During the starting period, the “S” region with $Q_{11}>0$ determines the instability of the pump-turbine, and the relative opening of pre-opened guide vanes has a relationship with the “S” characteristics, so the “S” characteristics with $Q_{11}>0$ in different relative openings are calculated. The results of the “S” characteristics with different relative openings are shown in Fig.3. The relative openings are 33%, 50% and 70%, respectively. The existence of the “S” characteristic can be seen when the relative opening is less than 50%, but the increase of relative opening of guide vane improves the “S” region. The “S” characteristic disappears when the relative opening is 50%, and that is to say there is a one-to-one relationship between the $Q_{11}$ and $n_{11}$. The increase of relative opening of pre-opened guide vane will not make further improvement on the “S” region when the relative opening is larger than 50%. The increase of relative opening will cause flow instability in the pump-turbine, even leading to serious pressure fluctuation. The chosen of optimal relative opening should consider both the “S” characteristics and the flow instability.
5.2 Flow in the pump-turbine with MGV

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. Streamlines in the runner on blade to blade surface at no-load condition with different opening of pre-opened guide vanes are shown in Fig.4. MGV change the flow in the runner. Different flow rates in the 20 passages of guide vane lead to reverse flow in the runner. The flow rate in the passage of the runner against the pre-opened guide vanes is larger than others, which may be the reason for the change of the performance of the pump-turbine. When the relative opening of pre-opened guide is more than 50%, the change of the flow in the runner is more obvious.

Flow in the casing and the region of guide vanes is shown in Fig.5. When the relative opening of pre-opened guide vanes is small, the resistance is large enough for the fluid flowing through the space between two adjacent guide vanes so that a water ring swirling at a high speed is formed in the vaneless space. The increase opening of MGV will destroy the water ring, as can be found in Fig.14(b).

Vortex rope in the draft tube at the runaway point is shown in Fig.6. The vortex rope is calculated by the pressure contour. The volume of vortex rope in the draft tube increases as the relative opening of MGV increases. When the relative opening of MGV is 50%, the use of MGV changes the shape of vortex rope into foam-like structure.

5.3 Pressure fluctuation in the vaneless space

Pressure fluctuations in the vaneless space at runaway point are shown in Fig.7. The fluctuation of the pressure in the pump-turbine with SyV is consistent, and the pressure fluctuation only contains high frequency component. The use of MGV increases the amplitude of the pressure fluctuation. With the increase of the relative opening of pre-opened guide vane, the low frequency of pressure fluctuation is more and more obvious. Low frequency is very dangerous to the operation of the pump-turbine.

The spectrum characteristics of the pressure fluctuation are shown in Table 3. Results of pressure fluctuation in the vaneless space agree well with the experimental data. The amplitude of pressure fluctuation increases as the opening of pre-opened guide vane increase. At the runaway point, the dominant frequency (DF) of pressure fluctuation is the blade passing frequency. The second dominant frequency (SDF) of pressure fluctuation is two times of blade passing frequency when the pre-opening is small. The increase of the pre-opening makes the second dominant frequency lower and lower.
6. Conclusions
In this paper, a PANS model was proposed based on RNG $k$-$\varepsilon$ turbulence model. The shear stress was resolved by nonlinear model. The nonlinear PANS model was used to simulate the flow field in a
curved rectangular duct, and was proved to be accurate in the simulation of large curvature flow and second flow. The “S” characteristics of a pump-turbine with MGV were investigated and compared with the results of SyV. The MGV improved the “S” characteristics by the changing of the flow field in the pump-turbine. It destroyed the water ring in the vaneless space and changed the periodic flow in the runner. The use of MGV aggravated the pressure fluctuation in the vaneless space. The amplitude of pressure fluctuation increased with the pre-opening increased. The dominant frequency of pressure fluctuation in the vaneless space was the blade passing frequency, but the second dominant frequency gradually decreased as the pre-opening increased.

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