Numerical simulation and experimental research the characteristic at rated guide vane opening on pump-turbine

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Abstract. The performance of a reversible pump turbine with S-shaped (dQ/dn>0) characteristics is of great harmful to the transient processes such as start-up and load rejection. When working point transition is required, then process in turbine mode must be fast, efficient and reliable. Especially during load rejection, the pressure increasing at spiral case and negative pressure taking place at draft tube. This phenomenon has great relation with the characteristic curve with pronounced S shape. In this paper, the characteristic curve is improved with runner optimization. And hydraulic performance results are compared with runner geometry parameters changing. The runner optimization is an effective method to solve this problem.

1. Introduction
The performance of a reversible pump turbine with S-shaped characteristics is of great crucial to the transient processes such as start-up and load rejection in figure 1. When working point transition is required, the process in turbine mode must be fast, efficient and reliable. Especially the S-characteristic curve takes place at large guide vane opening. When the single unite or mul-tiple unites reject full load at rated power at rated guide vane opening all units load rejection at the same time or in certain intervals such as delayed load rejection, it’s very dangerous to engineering. The negative pressure will take place if the one unite rejects the full load and after another unit is load rejected several seconds later. The transient study should find at which interval the delayed load rejection is the worst case and calculate the extreme pressure in the water passage to avoid any potential damage.

In recent years, there have been several severe accidents reported for the pumped storage projects (PSP) such as HuiZhou PSP(8x300MW) [1] and XiLongChi PSP(4x300MW). The safe operation during the transient is of significant importance for high head pumped storage projects.

The load case list should cover all the most critical load cases to ensure the safe operation of the whole plant. During the load rejection, the pressure in the spiral case is rising and the pressure after guide vane is dropping down after guide vane is closing, and the rotation speed of the machine is increasing or oscillating [2]. To ensure the safe operation of the whole plant, the maximum pressure in the water passage head water side, and the minimum pressure in draft tube, maximum over-speed should not exceed a certain limit, which is usually given by the customer.
Figure 1. The four-quadrant curve

The load case of delayed load rejection shows an extreme lower pressure or significant pressure drop at the entrance of draft tube has been paid high attention and was under huge discussion it is a potential risk for the safe operation of the complete power plant. With a clear picture of the root causes of this phenomenon, possible active and passive countermeasures from following different aspects were proposed, and carefully studied according to the Voith and EPFL research\textsuperscript{[3-5]}. So more attention to the characteristic curve should be paid in the design stage.

If this accident takes place on prototype pump-turbine, the runner, head cover, leak proof structure and bolted joint will be lifted and damaged. Even more serious, the tail water will flow back the water power building. How to solve this problem? How to solve this problem during the optimization stage with design the pump-turbine geometry and conduit system and tail water system. Now in this paper, the curve will be improved at large guide vane opening by numerical simulation and experimental method.

Priessmann method, Amein method, Vasiliev method and Strelkoff method\textsuperscript{[6-7]} mentioned in are unconditionally stable when computing the open-channel flow. However, when computing the mixed flow, due to the abrupt change of the wave speed and the flow surface width on the top of tunnels or pipes, these methods are unstable if the mixed flow occurs at any computing section.

Some working points are calculated by CFD. Many vortices are formed on S-shape working zone and runaway points because of the pumping effect, which will cause energy loss and discharge fluctuation. Some geometry parameters are compared to pay attention to the curve shape\textsuperscript{[8]}, such as diameter, geometry installed angle, wrap angle and hydrofoil shape. And some valued geometry parameters are validated to improve the curve shape.

The load case of delayed load rejection shows an extreme lower pressure or significant pressure drop at the entrance of draft tube has been paid high attention and was under huge discussion in the proposal stage of DFEM Hydro since it is a potential risk for the safe operation of the complete power plant. For solved these problems, some possible active and passive countermeasures from following four aspects were proposed:

A hill-chart with less pronounced S-shape in the turbine braking range was proposed and used for the final transient calculation. This may be achieved by an asynchronous closing of 2 guide vanes during the load rejections or emergency shut downs;

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Activate the closing of the spherical valves to change the track of the pump-turbine during the load rejections or emergency shut downs. A stepped closing law for the spherical valve were proposed, and studied:

- Limit the operation of the power-plant;
- Hill chart modification by the structural change of runner design during the new hydraulic development for the pump turbine

The new hydraulic design is applied in this paper. The different runner A and B are compared. The runner geometry parameters are modified with many attempts in this paper.

2. Pump turbine specifications
The optimization projects are accessed by CFD method and experimental results. The CFD method is an approximation method to calculate the characteristic curve. Because of the mesh number restricted the whole pump-turbine configuration only including spiral casing, stay vanes, guide vanes, runner and draft tube is considered in the physical model, as shown in figure 2. The integrated structure included the pressure pipe, pressure tank and pump-turbine. The computing domain enables to overcome the influence of boundary conditions, periodic interfaces, and pitch ratio of rotor-stator interface, and it especially considers the non-axisymmetric inflow from the spiral casing.

![Figure 2. Computation domain of the pump-turbine model.](image)

Specifications of the machine are listed in table 1.

| Table 1. Main specifications of the tested pump turbine |
|------------------------------------------------------|
| runner diameter at inlet (mm) | 4400 |
| runner diameter at outlet (mm) | 2100 |
| runner blade number Z₁ | 10 |
| guide vane number Z₀ | 16 |
| Maximum guide vane opening(°) α | 24 |
| rotational speed (rpm) | 500 |
| specific speed (m. kW) | 87 |

3. Mathematical model and computational details

3.1. Mathematical model
The continuity equation and momentum equation are adopted in simulation.

Continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0$$  \hspace{1cm} (1)

Momentum equation
\[
\frac{\rho Du}{Dt} = \rho f - \nabla p + \mu_{\text{eff}} \nabla^2 u + \left( \mu_{\text{eff}} + \lambda \right) \nabla (\nabla \cdot u)
\]  

(2)

The turbulence stress is taken into account with turbulence viscosity coefficient in a standard SST model proposed by Menter\(^9\).

3.2. Grid generation
Rotating and stationary domains are matched using rotor stator interfaces. The global mesh encompassed 9 million nodes with smallest skew angles of 22 degrees. As displayed in figure 3, special refinement is applied in the runner and in the guide vane domain. The mesh independent is validated before the scheme comparison. When the mesh number is 9,000,000, the comparison between simulation efficiency 0.930 and experimental value 0.928 has shown good agreement in figure 3 (a) and the efficiency value has shown the independence to mesh number. The mesh detail is shown in the Table 2. The mesh number can satisfy the calculation requirement and confirm the ability of the model to simulate flow characteristic of model pump-turbine. The \( y^+ \) on guide vane and runner blade is shown in figure 3(b) and (c). The whole mesh on pump-turbine is shown in figure 3(d).
Figure 3. Mesh independent and $y^+$ distribution of the reduced pump-turbine

Table 2. Mesh detail of the tested pump-turbine

| Mesh number ($\times 10^6$) | $y^+$ | Minimum Skew angle (°) |
|-----------------------------|-------|------------------------|
| Spiral case                 | 0.5   | 150                    | 22                |
| Stay vane                   | 0.6   | 50                     | 32                |
| Guide vane                  | 2.4   | 20                     | 54                |
| runner                      | 5.2   | 15                     | 52                |
| Draft tube                  | 0.3   | 60                     | 35                |

3.3. Boundary conditions
- Inlet condition. The mass flow inlet at the spiral casing (Turbine mode) is used, whose profile is assumed to be uniform.
- Outlet conditions. Pressure outlet is used at the draft tube (Turbine mode), static pressure ($p=0$ Pa) is specified for all cases.
- Other conditions. No-slip condition is assumed on all the solid walls, and standard wall function is used to calculate the turbulence kinetic energy and turbulence dissipation frequency near the wall. The interaction between rotation of runner and stagnant zones are dealt with by means of rotator and stator method.

These boundary condition can assure the numerical simulation stability.

3.4. Solution strategy
The software CFX13 is used to carry out the numerical simulation. In the calculation, the second order upwind scheme is used for discretization of convective term and the second order central scheme for discretization of diffusion term. The coupled solver is used and numerical convergence is set to a maximum of $10^{-4}$.

4. Optimized result and analysis
The optimized object is the $\Delta Q_{11}/\Delta n_{11}$ in figure 4, the value is bigger and S characteristic of scheme is better. And the constraint condition is hydraulic characteristic such as efficiency and pressure should be keep at the high level.

Many hydraulic schemes are designed and tested. Finally, the Runner structure is very important to efficiency, pressure fluctuation and S characteristic curve. The runner scheme A are designed and tested. The runner B are optimized base on the runner A. The leading edges are compared between runner A and B in figure 5. The characteristic parameters are tested and recorded. The runner design is based on the CFD calculation on DFEM optimization platform. Finally, the runner B is designed and tested. The

(d) Mesh distribution
different structure of runner A and B are shown in table 3. The diameter and leading edge shape are different. The leading edge shape is rounding edge and slash shape on the runner A and B, respectively.

![Figure 4. n_{11}-Q_{11} curve](image)

**Figure 4.** $n_{11}$-$Q_{11}$ curve

![Figure 5. Runner A and B](image)

**Figure 5.** Runner A and B

| Table 3. Runner A and B |
|-------------------------|
| Runner A(mm) | Runner B(mm) |
| D1 | 540 | D1 | 542 |
| D2 | 250 | D2 | 250 |
| b0 | 38 | b0 | 38 |

According to the tested results, the characteristic curve is compared. Compared scheme A and B, the characteristic is improved. The S characteristic of scheme B is less pronounced than runner A in figure 6. It’s benefit to transient processes such as start-up and load rejection especially interval the delayed load rejection. Because of the characteristic curve improved, the minimum pressure on the draft tube and maximum pressure on spiral case inlet are improved. The minimum pressure is larger than zero Pa and the cavitation volume cannot taking place on the draft tube. The so-called water column separation cannot appear. The maximum pressure on spiral case cannot exceed the guaranteed value which is determined by the iron tensile yield limit.

The hydraulic performance parameters are compared such as turbine efficiency, pump efficiency, pressure fluctuation and head between runner A and B in figure 7 and 8. The turbine efficiency is the
best efficiency on united speed $n_{11} = 36\text{r/min}$ at the guide vane opening 12 degree. The pressure fluctuation is the peak to peak value on vane-less area between guide vane and runner. The pump efficiency is the best efficiency on guide vane opening 16 degrees. The efficiency is the relative efficiency. The $H$-$Q$ curve is the combination curve on guide vane opening 13 degrees. Compared the efficiency and pressure fluctuation amplitude, the runner B has better hydraulic performance and characteristic curve.

Figure 6. Characteristic curve of runner A and B (Left is the full figure and right is the enlarge figure)

Figure 7. Turbine efficiency and pressure fluctuation between A and B at turbine mode

($Q_{11}$ is the discharge coefficient $Q/ (D^2H^{0.5})$)
Figure 8. Pump efficiency and head between A and B
(Pressure coefficient $2gH/u^2$ is the head and the flow coefficient $Q/(\pi/4D^2u)$ is the discharge at pump mode)

5. Conclusions
The characteristic curves are improved by geometry parameters modified. The efficiency, head and pressure fluctuation are compared. The runner geometry parameters are very important to pump turbine.

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