**INTRODUCTION**

Achieving carbon peak and carbon neutrality will have an important and far-reaching impact on climate change.\(^1\) Under the guide of this goal, energy recycling is an important way to reduce carbon emissions. As a kind of energy recovery device, hydraulic turbine is widely used in the industry of high residual pressure recovery.\(^2,3\) By converting pressure energy of liquid into rotating energy of hydraulic turbine runner, hydraulic turbine realizes the goal of energy recovery. At present, reverse pump as turbines (PAT) are widely used because of its simple structure and low price. In previous researches, many scholars have conducted in-depth studies on selection,\(^4-6\) performance prediction,\(^7,10\) and optimization\(^11,12\) of PAT's. However, when PAT operating at off-design condition, it...
will become an energy consumption device, resulting in the secondary waste of energy. In addition, due to the low stage energy head, more stages of PAT are needed to reach the recovery head in the field of ultra-high pressure liquid recovery, which results in lower efficiency and poor stability of the whole machine.13-15 Hydraulic turbine in turbine mode (T-type turbine) designed by prime mover theory can avoid the shortcoming of PAT. Compared with conventional turbine, T-type turbine removes movable guide vane and control mechanism, which has higher efficiency and wider high-efficiency area than PAT.16 In addition, the stage energy head of T-type turbine with low specific speed is high, which effectively reduces the stage in the field of high residual pressure recovery.

As the core part of turbine, the runner directly affects the efficiency of whole machine. The number of blades, as the main design parameter of runner, has a more profound influence on the efficiency of turbine.17 Shi18 pointed out that runner has optimal blade numbers, and too many or too few blade numbers will lead to poor performance. Punit19 concludes that that choice of blade number should be carefully made because of the influence of blade number is dominating. Abdolkarim20 draws the conclusion through numerical simulation and test, the efficiency first increases and then decreases with the increase in blade numbers. Thiyagaraj21 finds that an appropriate number of blades can improve the power coefficient of turbine. Ahmed22 points out that passage and incidence losses decrease with the increase in blade numbers, which improving the swallowing capacity and efficiency. Ibnu23 finds out that the water discharge decreased with the increase in blade numbers. Young-Do24 figures out that the blade number of runner has remarkable influence on efficiency and power. Other studies indicate that the number of blades has a significant effect on the pressure pulsation of the whole machine.25,26 Yang27 studies the influence of blade number on pressure pulsation characteristics of PAT and points out that increasing blade numbers can reduce pressure pulsation amplitude of flow passage components. Whether this rule can be applied to T-type turbine needs to be further studied.

Considering the majority of references are related to the blade number of PAT, which take the study for blade number of T-type turbine or multistage turbine, this paper establishes the two-stage T-type turbine with the same blade numbers in primary and secondary runner, and different blade numbers in primary and secondary runner, respectively. Among them, the blade number of guide vane is 15, the blade number of runner in the original scheme is 12, and the designed blade number of runner is set as 10, 12, 14, and 16, respectively, in order to avoiding the resonance phenomenon of turbine unit.28 Further, the influence of blade numbers on hydraulic performance of T-type hydraulic turbine is studied.

The remainder of this paper is organized as follows. First, T-type turbine with low specific speed is analyzed by theoretical analysis in Section 2. Then, Section 3 verifies the accuracy of numerical simulation and determines the numerical simulation scheme. Section 4 analyzes the performance of turbine with same blade of primary and

| TABLE 1 Main parameters of hydraulic turbine in turbine mode |
|-------------|---------------|
| **Parameter** | **Value** |
| Runner      | Inlet diameter $D_1$/mm | 510 |
|             | Outlet diameter $D_2$/mm | 290 |
|             | Inlet width $B_1$/mm | 52 |
|             | Inlet placement angle $\beta_1$/° | 110 |
|             | Exit angle $\beta_2$/° | 31 |
|             | Blade wrap angle $\varphi$/° | 30 |
|             | Blade Number | 10, 12, 14, 16 |
| Volute      | Base circle diameter $D_3$/mm | 634 |
|             | Exit width $B_3$/mm | 60 |
|             | Inlet diameter $D_4$/mm | 285 |
| Stage guide vane | Outlet diameter of positive guide vane $D_5$/mm | 510 |
|             | Inlet diameter of positive guide vane $D_6$/mm | 732 |
|             | Outlet diameter of negative guide vane $D_7$/mm | 732 |
|             | Inlet diameter of negative guide vane $D_8$/mm | 360 |
2 | THEORETICAL ANALYSIS OF HYDRAULIC TURBINE IN TURBINE MODE

T-type turbine is mainly composed of six parts: volute, first-stage guide vane, primary runner, stage guide vane, secondary runner, and outlet chamber, the runner is designed according to the theory of hydraulic prime mover, and guide vane and volute are designed to meet the runner inlet circulation. The design parameters of turbine are shown in Table 1.

The basic equation of T-type turbine can be expressed as follows:

\[ H_e = H \eta_h = \frac{\omega}{2\pi g} \left( C_1 - C_2 \right) \]  

In the assumption of normal export, it can be expressed as follows:

\[ H_e = \frac{\omega}{2\pi g} C_1 \] \hspace{1cm} (2)
\[ C_1 = 2\pi r_1 v_{u1} \] \hspace{1cm} (3)
\[ C_2 = 2\pi r_2 v_{u2} \] \hspace{1cm} (4)

where \( H \) is the working head of turbine, \( H_e \) is the effective utilization head, \( \eta_h \) is the hydraulic efficiency of turbine, \( \omega \) is the rotational angular velocity, \( v_u \) is circumferential component of velocity, \( r_1 \) and \( r_2 \) are inlet and outlet radius of runner. Subscripts 1 and 2 represent inlet and outlet.

According to the velocity triangle:

\[ H_e = \frac{\omega}{2\pi g} C_1 = \frac{1}{g} u_1 v_{u1} \]
\[ = \frac{1}{g} u_1 ( u_1 - v_{m1} \cot \beta_1 ) \]
\[ = \frac{1}{g} u_1 \left( u_1 - \frac{Q}{\pi D_1 b_1} \cot \beta_1 \right) \]  

where \( u \) is circumferential velocity, \( v_m \) is meridian velocity, \( \beta_1 \) is setting angle of runner inlet, for low specific speed T-type turbine, its value range is 110°–150°, \( Q \) is flow rate, \( D_1 \) is inlet diameter of runner, and \( b_1 \) is inlet width of runner.

3 | NUMERICAL CALCULATION

3.1 | Mesh generation

According to the research goal of this paper, the performance of turbine with different blade numbers is studied by numerical simulation. First, the six parts of turbine are modeled by Pro.E. Then, ICEM is used to divide the structured grid of each flow passage component, as shown in Figure 2. In order to eliminate the influence of mesh number on the numerical simulation results, six mesh schemes with the ranging from 1.12 million to 10.51 million were tested for the model with 14 blades, which performance of head and efficiency are compared. It can be seen from Figure 3 that with the increase in mesh number, the head gradually decreases and the efficiency gradually increases, keep stability when mesh number reaches 5.91 million. According to the calculation time and calculation stability, 5.91 million was selected for final simulation. For comparison, there is little difference about grid number and grid quality of the other schemes. The inlet boundary is set as speed inlet, and the outlet boundary is set as pressure outlet. The frozen rotor-stator interface is set between runner and guide vane and between runner and outlet chamber. The solution method selected SIMPLE. The calculation convergence standard is $10^{-5}$.

The turbulence selected \( k-\varepsilon \) model, which is widely used in numerical simulation of turbomachine. When flow is incompressible, the model could be expressed as follows: where \( \mu_t \) is turbulence viscosity, \( u_i \) is time average viscosity, \( k \) and \( \varepsilon \) are turbulence kinetic and specific dissipation rate, \( G_k \) is production of turbulence, and \( C_{1k} \) and \( C_{2\varepsilon} \) are empirical constants.
In order to verify the accuracy of numerical simulation, the performance of single-stage runner with 14 blades is tested. High inlet pressure of T-type turbine was supplied by three booster pumps, as shown in Figure 4. The dynamometer was used to collect the torque and rotational speed. Pressure sensor was installed at inlet and outlet to collect the inlet and outlet pressure. T-type turbine was

\[ \frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho u_i k)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_1}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon \quad (6) \]

\[ \frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho u_i \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_1}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1k} \frac{\varepsilon}{k} G_k - C_{2k} \frac{\varepsilon^2}{k} \quad (7) \]

\[ G_k = \mu_t \left\{ 2 \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)^2 + \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right)^2 \right\} \quad (8) \]

3.2 | Test verification

In order to verify the accuracy of numerical simulation, the performance of single-stage runner with 14 blades is tested. High inlet pressure of T-type turbine was supplied by three booster pumps, as shown in Figure 4. The dynamometer was used to collect the torque and rotational speed. Pressure sensor was installed at inlet and outlet to collect the inlet and outlet pressure. T-type turbine was
tested at 1283 g. By measuring all data, the head, efficiency, and power of turbine were obtained.

It can be seen from Figure 5 that the trend of the performance curve obtained from test and numerical simulation is consistent. Because the numerical simulation did not consider mechanical loss and leakage, the efficiency is higher than test date. At the highest efficiency point, the head error is 2.35%, efficiency error is 3.08%, and shaft power error is 6.8%. The numerical calculation method used in this paper is suitable to predict T-type turbine’s performance.

4 | TURBINE WITH SAME BLADE NUMBERS OF PRIMARY AND SECONDARY RUNNER

4.1 | Performance analysis

According to traditional design of multistage turbine, the blade number of each stage runner is same. Therefore, for the two-stage T-type turbine studied in this paper, a study on blade number affecting T-type’s performance was carried out when the two-stage runner with the same blade number.

External characteristic curves of T-type turbine with blade number ranging from 10 to 16 were acquired and presented in Figure 6. Under small flow conditions, the head and shaft power change little with the increase in the blade number. The efficiency gradually increases as its blade number increases from 10 to 14, but as the blade number continues to increase, the efficiency begins to decrease. Under the design condition, the efficiency of two-stage runner with 14 blades is 5.5%, 3.8, and 1.7% higher than that of the two-stage runner with 10, 12, and 16 blades, reaching 80.24%. Figure 7 shows the velocity vectors of two-stage runner with blade number ranging from 10 to 16 when the flow $Q$ is 0.89 m$^3$/s. It can be seen that when blade number is small, there is a vortex rotating opposite to the direction of runner’s rotation. With the increase in blade number, the flow in primary runner is significantly improved. By comparing and analyzing the flow of primary runner and secondary runner, it is found that the flow in secondary runner is obviously more disordered than that in primary runner. Although the increase in blade number improves its internal flow, there is always a phenomenon of flow separation on the blade working face.

Hydraulic characteristics of primary runner and secondary runner with different blade numbers schemes are plotted in Figure 8. It can be seen that the head and shaft power of secondary runner are lower than that of primary runner, and the difference of head and shaft power increases with the increase in flow rate. Under small flow rate, the efficiency of primary runner is higher than that of secondary runner. With the increase in flow, the difference of efficiency between primary runner and secondary runner changes little. Figure 9 is the velocity vector distribution at the outlet of primary runner with different blade numbers (see Figure 2 for the specific location) when the flow rate $Q$ is 0.89 m$^3$/s. It can be seen that there is a circulation opposite to the rotation direction of primary runner (referred to as negative circulation). With the increase in blade number, the average velocity at the outlet of primary runner gradually increases, and the negative circulation also gradually increases with the increase in blade number.
FIGURE 6 Performance curve of hydraulic turbine with different blade numbers

FIGURE 7 Runner streamline distribution under design conditions
FIGURE 8  Primary and secondary runner performance with different blade numbers

FIGURE 9  Distribution of primary runner outlet streamline under design conditions
During the diversion process, the stage guide vane needs to eliminate the negative circulation firstly, then provide circulation to the secondary runner, resulting in the inlet circulation of secondary runner reduce, which is the main reason for the flow disorder and efficiency decline in the secondary runner.

Inlet circulation of primary and secondary runner is listed in Table 2. It can be seen that the inlet circulation of primary runner increases with the increase in flow. Since inlet circulation of primary runner is provided by volute and first-stage guide vane, the change in blade number has little impact on the inlet circulation. Under small flow rate, with the increase in blade number, the negative circulation at outlet of primary runner gradually decreases; therefore, the circulation at the inlet of secondary runner gradually increases, which is the reason why the head of secondary runner is slightly higher than that of primary runner under small flow conditions. Under design and large flow conditions, with the increase in blade number, the outlet circulation of primary runner increases and the inlet circulation of secondary runner decreases, resulting in water head of secondary runner gradually lower than that of primary runner.

### Table 2  Inlet and outlet circulation

| Blade Number | Primary runner inlet circulation | Primary runner outlet circulation | Secondary runner inlet circulation |
|--------------|----------------------------------|----------------------------------|-----------------------------------|
|              | Small flow | Design   | Large flow | Small flow | Design   | Large flow | Small flow | Design   | Large flow |
| 10           | 24.5825    | 54.9250  | 59.9357    | −8.4426    | −9.7673  | −22.0418   | 24.2841    | 54.4537  | 57.1422    |
| 12           | 24.1649    | 55.8316  | 60.0646    | −6.8289    | −11.574  | −23.1707   | 24.7244    | 54.0568  | 56.9153    |
| 14           | 23.9798    | 55.0218  | 60.0306    | −4.76938   | −12.437  | −27.3707   | 25.9182    | 53.1792  | 56.2454    |
| 16           | 24.0734    | 55.8600  | 59.1722    | −3.8720    | −15.472  | −28.6970   | 27.7399    | 52.9441  | 54.0582    |

Excessive pressure pulsation will cause vibration and noise in turbine, which will affect normal operation and reduce service life. Therefore, to study how the blade number affects the pulsation, unsteady simulations of T-type turbine were performed at the flow rate of 0.89 m³/s and rotational speed of 1500 rpm when the blade number of two-stage runner ranges from 10 to 16.

#### 4.2 Pressure pulsation analysis

As a rotating machine, the interaction of rotating runners and stationary guide vanes can cause strong pulsations.

#### 4.2.1 Unsteady numerical calculation

The monitoring points within volute, runner, guide vane, and outlet chamber are shown in Figure 10. The result of steady simulation is initial condition of unsteady simulation. Then, static pressure of each monitoring points was obtained from unsteady simulation. In order to compare amplitude and frequency of pulsation, pressure coefficient was defined as $C_P = \frac{(P - \bar{P})}{0.5 \rho u^2}$, Where $P$ is instantaneous pressure, $\bar{P}$ is average pressure in application time, $\rho$ is fluid density, kg/m³; and $U$ is circumferential speed of runner inlet, m/s. The runner rotated 7 cycles and rotated 2° in each time step. Time step was set as 0.0002222 s, and total time was set as 0.279972 s. The static pressure results of each monitoring point obtained in last cycle are selected for analysis. Rotational frequency of runner $f_n = n/60 = 25$ (Hz), blade frequency $f = f_n \times Z$ (blade number).

#### 4.2.2 Analysis of pressure fluctuation characteristics

The pressure pulsations during last rotational cycle in volute were obtained after fast Fourier transform (FFT), which are plotted in Figure 11. Table 3 lists main pulsations frequency of monitoring points in volute. It can be seen that along fluid flow direction, pressure fluctuation in volute increases slightly at monitoring point 4 and 5. With increase in blade number, pressure fluctuation amplitude at each monitoring point in volute decreases gradually.
The pressure pulsations during last rotational cycle in primary runner after FFT are plotted in Figure 12. Table 4 lists main pulsations frequency of monitoring points in primary runner. It can be seen that the pressure pulsation in primary runner is stronger than that in volute. The pressure pulsation at monitoring point 6 is stronger than that at point 7 and point 8. When blade number is 14, pressure pulsation amplitude at point 6 is 1.33 times of point 7 and 2.2 times of point 8. With the increase in blade number, the main frequency amplitude of pressure pulsation at each monitoring point in primary runner gradually decreases. The main reason is increase blade number can
improve flow state, the vortex in primary runner gradually disappears, so the pressure pulsation gradually decreases.

The pressure pulsations during last rotational cycle in stage guide vane after FFT are plotted in Figure 13. Table 5 lists main pulsations frequency of monitoring points in stage guide vane. It can be seen that the pressure pulsation of monitoring points in stage guide vane gradually increases with the increase in blade number. Interaction has little influence on negative guide vane and transition section of guide vane; therefore, the main frequency at monitoring point 9 and point 10 is small. The monitoring point 11 at positive guide vane is affected by rotating secondary runner, so its pressure fluctuation is much greater than that at point 9 and point 10. When blade number is 14, the fluctuation at point 11 is 8.16 times of point 10 and 9 times of point 9.

The pressure pulsations during last rotational cycle in secondary runner after FFT are plotted in Figure 14. Table 6 lists main pulsations frequency of monitoring points in primary runner. It can be seen that the pressure pulsation in secondary runner is greater than that in primary runner. With the blade number increases, the difference of pressure pulsation between primary runner and secondary runner gradually increases. Comparing the monitoring points 7 and 13 at the middle place of two-stage runner, when the blade number is 10, the pressure pulsation of the two-stage runner changes little, but when the blade number increases to 16, the pressure pulsation of secondary runner is 2.46 times of primary runner. The pulsation amplitude

![FIGURE 13](image)

**TABLE 4** Main frequency amplitude of pressure pulsation in primary runner

| Z   | Dominant frequency (Hz) | 6    | 7    | 8    |
|-----|------------------------|------|------|------|
| 10  | 250                    | 0.2315 | 0.1901 | 0.1105 |
| 12  | 300                    | 0.2256 | 0.1809 | 0.0981 |
| 14  | 350                    | 0.1990 | 0.1496 | 0.0905 |
| 16  | 400                    | 0.1702 | 0.1263 | 0.0902 |

**TABLE 5** Main frequency amplitude of pressure pulsation in stage guide vane

| Z   | Dominant frequency (Hz) | Dominant frequency amplitude |
|-----|------------------------|-------------------------------|
| 10  | 250                    | 0.0037 | 0.0046 | 0.0777 |
| 12  | 300                    | 0.0052 | 0.0019 | 0.0923 |
| 14  | 350                    | 0.0067 | 0.0111 | 0.0905 |
| 16  | 400                    | 0.0093 | 0.0191 | 0.1221 |

**FIGURE 13** Frequency domain diagram of pressure fluctuation in stage guide vane
The frequency domain of pressure pulsation in outlet chamber is shown in Figure 15. Table 7 lists main pulsations frequency of monitoring points in outlet chamber. After liquid pushes two-stage runner to do work, most of energy is consumed when it reaches outlet chamber. Therefore, the pressure pulsation in outlet chamber is small, and pressure pulsation at each monitoring points of outlet chamber changes little. With the increase in blade number, pressure fluctuation changes differently at each monitoring points in outlet chamber.

5 | TURBINE WITH DIFFERENT BLADE NUMBERS OF PRIMARY AND SECONDARY RUNNER

5.1 | Performance analysis

When the blade numbers of primary runner and secondary runner are same, the pressure pulsation in secondary runner is strong. In order to study whether there is pulsation superposition in secondary runner, turbine with different blade number of primary and secondary runner was studied in next section. The blade number of primary runner and secondary runner is ranging from 10 to 16, and 26 schemes were formed. The hydraulic efficiency of 12 schemes was calculated by removing the schemes with the same blade number of two-stage runner when the flow rate \( Q \) is 0.89 m\(^3\)/s, as shown in Table 8. \( Z_1 \) represents blade number of primary runner, and \( Z_2 \) represents blade number of secondary runner.

It can be seen from Table 8 that when blade number of primary runner is 12 and blade number of secondary runner is 16 (named scheme 12-16), hydraulic efficiency is the highest under design condition. By comparing with scheme 14-14, it can be seen (as shown in Figure 16) that the efficiency of turbine under design condition was increased by 0.929%, reached 81.169%. The optimal working condition of scheme 12-16 shifts to large flow (\( Q \) is 1.068 m\(^3\)/s), and its optimal efficiency is 81.9%.

Figure 17 shows the comparison of blade pressure of scheme 12-16 and scheme 14-14 under design conditions. It can be seen that the blade pressure of primary runner is higher than that of secondary runner. The variation law of blade pressure is basically same, which is gradually reduced from inlet to outlet. At the inlet of secondary runner, the pressure on blade working face is lower than that on back face, which is caused by insufficient circulation at

**TABLE 6** Main frequency amplitude of pressure pulsation in secondary runner

| \( Z \) | Dominant frequency amplitude |
| --- | --- | --- | --- |
| \( Z \) | Dominant frequency (Hz) | 12 | 13 | 14 |
| 10 | 250 | 0.1504 | 0.20081 | 0.1894 |
| 12 | 300 | 0.1694 | 0.2399 | 0.2332 |
| 14 | 350 | 0.2954 | 0.2878 | 0.4185 |
| 16 | 400 | 0.3666 | 0.3103 | 0.5913 |

**FIGURE 14** Frequency domain diagram of pressure pulsation in secondary runner
the inlet of the secondary runner. The pressure of scheme 14-14 is lower than that of scheme 12-16, and this difference is more obvious at the secondary runner.

Figure 18 shows the turbulent kinetic energy distribution of runner under design condition. It can be seen there is an obvious high turbulent kinetic energy area at the inlet of primary runner of scheme 14-14, while the turbulent kinetic energy distribution of scheme 12-16 is relatively average. The turbulence kinetic energy of secondary runner is higher than that of primary runner. For scheme 14-14, there is an obvious high turbulence region at the inlet and middle of the secondary runner, and the high turbulence region is relatively small of scheme 12-16.

5.2 Pressure pulsation analysis

The pressure pulsations during last rotational cycle in primary runner and secondary runner of scheme 14-14 and scheme 12-16 are plotted in Figure 19. By comparing pressure pulsation of two schemes, it can be seen that the pressure pulsation amplitude in primary runner of scheme 12-16 is slightly higher than that of scheme 14-14, and main frequency amplitude of pressure pulsation appears at multiple of passing frequency of primary runner blade number. The pressure pulsation amplitude in secondary runner has been significantly reduced in scheme 12-16,
and main frequency amplitude in secondary runner also appears at multiple of passing frequency of secondary runner blade number, but affected by primary runner, there is an obvious secondary frequency at multiple of passing frequency of primary runner blades, indicating that there is pulsation superposition in secondary runner. When blade number of each stage runner is same, pressure pulsation amplitude in secondary runner will be superimposed, resulting in enhancement of pressure pulsation in secondary runner. Therefore, when blade number of two-stage runner is different, pulsation superposition in secondary runner can be avoided and pressure pulsation in secondary runner can be reduced.

6 CONCLUSIONS

By studying the influence of blade number on the performance of hydraulic turbine in turbine mode, the following conclusions are drawn:

1. According to the conventional design method, T-type turbine has an optimal blade number to make the efficiency higher when the blade numbers of primary and secondary runner are same. However, affected by pressure pulsation of primary runner, pressure pulsation amplitude in secondary runner will be superimposed. The pressure pulsation amplitude at the middle monitoring point of secondary runner is 2.46 times of the primary runner. As a result, internal pressure pulsation of secondary runner is much higher than that of other overflow components, which will produce strong vibration and reduce service life of turbine.

2. The different blade numbers of two-stage runner can significantly improve performance of T-type turbine. The blade pressure of scheme 12-16 is higher than that of scheme 14-14, and the turbulent kinetic energy is lower, which obviously improves the performance of the runner. Under the design condition, the efficiency of scheme 12-16 is 2.07% higher than that of scheme 14-14, reaching 81.9%.

3. When the blade numbers of primary and secondary runner are different, the superposition of pulsation can be effectively avoided, reducing pressure pulsation of secondary runner; then, the pressure pulsation of whole machine was reduced. Compared with the scheme 14-14, which pulsation is 0.288, pulsation of scheme 12-16 decreases to 0.118. If the design cycle of
multistage turbine allows, it is recommended to set the blade number of each stage runner to be different.

ACKNOWLEDGMENT
This work is supported by the National Natural Science Foundation of China (Grant no. 51579104 and Grant no. 51909094).

ORCID
Lihong Zhang © https://orcid.org/0000-0003-2531-5538

REFERENCES
1. Hongliang WU, Daoxin P, Ling W. Model for sustainable development based on system dynamics and energy–economy–environment coordination: a case study of Beijing, China. *Energy Sci Eng*. 2021;9(6):1-15.
2. Li D, Wang H, Chen J, Nielsen T, Qin D, Wei X. Hysteresis characteristic in the hump region of a pump-turbine model. *Energies*. 2016;9(8):620. doi:10.3390/en9080620
3. Jiyun DU, Hongxing Y, Zhicheng S. Study on the impact of blades wrap angle on the performance of pumps as turbines used in water supply system of high-rise buildings. *Int J Low-Carbon Technol*. 2018;13(1):102-108.
4. Band Shahab S, Pezhman TG, bin Wan Yusof K, Hossein Ahmadi M, Nabipour N, Chau K-W. Evaluation of the accuracy of soft computing learning algorithms in performance prediction of tidal turbine. *Energy Sci Eng*. 2020;9(5):633-644.
5. Faraji J, Hashemi H, Dezaki AK. Optimal probabilistic scenario-based operation and scheduling of prosumer microgrids considering uncertainties of renewable energy sources. *Energy Sci Eng*. 2020;8(3):1-19.
6. Derakhshan S, Nourbakhsh A. Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds. *Exp Thermal Fluid Sci*. 2008;32(3):800-807.
7. Cao Z, Deng J, Zhao L, Lu L. Numerical research of pump as turbine performance with synergy analysis. *Processes*. 2021;9:1031. doi:10.3390/pr9061031
8. Zariatin DL, Zariatin DL, Risdianto, et al. Experimental analysis of semi-open impeller pump as turbine. *IOP Conf Ser: Mater Sci Eng*. 2020;852:012069.
9. Maleki A, Ghorani MM, Haghhighi MHS, Riasi A. Numerical study on the effect of viscosity on a multistage pump running in reverse mode. *Renewable Energy*. 2020;150(C):234-254.
10. Fontanella S, Facirotta O, Molino B, Cozzolino L, Della Morte R. A performance prediction model for pumps as turbines (PATs). Water. 2020;12:1175. doi:10.3390/w12041175

11. Miao SC, Zhang HB, Wang TT, et al. Optimal design of blade in pump as turbine based on multidisciplinary feasible method. Sci Prog. 2020;103(4):1-17.

12. Wang X, Kuang K, Wu Z, Yang J. Numerical simulation of axial vortex in a centrifugal pump as turbine with S-blade impeller. Processes. 2020;8(9):1192.

13. Shiels S. Locating the greatest centrifugal pump energy savings. World Pumps. 1998;384:56-59.

14. Jain SV, Patel RN. Investigations on pump running in turbine mode: a review of the state-of-the-art. Renew Sustain Energy Rev. 2014;30:841-868.

15. Su X, Huang S, Li Y, Zhu Z, Li Z. Numerical and experimental research on multi-stage pump as turbine system. Int J Green Energy. 2021;11(9):1961-1972.

16. Chen J, Xiao Z, Liu D, Hu X, Ren G, Zhang H. Nonlinear modeling of hydro turbine dynamic characteristics using LSTM neural network with feedback. Energy Sustain Eng. 2021;14(2):142-149.

17. Byeon S-S, Kim Y-I. Influence of blade number on the flow characteristics in the vertical axis propeller hydro turbine. Int J Fluid Mach Syst. 2013;6(3):144-151.

18. Shi FX, Yang JH, et al. Numerical prediction for effects of guide vane blade numbers on hydraulic turbine performance. IOP Conf Ser: Mater Sci Eng. 2013;52(5):052029. doi:10.1088/1757-9746/52/5/052029

19. Singh P, Nestmann F. Experimental investigation of the influence of blade height and blade number on the performance of low head axial flow turbines. Renewable Energy. 2010;36(1):272-281.

20. Payambarpour SA, Najafi AF, Magagnato F. Investigation of blade number effect on hydraulic performance of in-pipe hydro savonius turbine. Int J Rotating Mach. 2019;2019:1-14.

21. Thyagaraj J, Rahamathullah I, Anbuchezhiyan G, Barathiraja R, Ponshamugakumar A. Influence of blade numbers, overlap ratio and modified blades on performance characteristics of the savonius hydro-kinetic turbine. Mater Today: Proc. 2021;46(P9):4047-4053.

22. Ketata A, Driss Z, Abd MS. Impact of blade number on performance, loss and flow characteristics of one mixed flow turbine. Energy. 2020;203:117914. doi:10.1016/j.energy.2020.117914

23. Asrafi I, Yerizam M, Effendi S, Mataram A. Micro hydro electric power plant (MHEP) prototype a study of the effect of blade numbers toward turbine rotational velocity. J Phys: Conf Ser. 2019;1198(4):042001.

24. Choi Y-D, Jia CF, Lim J-I, Kim Y-T, Lee Y-H. Performance analysis of a cross flow hydro turbine by runner blade number. J Kor Soc Mar Eng. 2008;32(5):698-706.

25. Shi FX, Ma DD, Zhao WY, Peng HT, Liang YS. Effect of different blade numbers on the radial force of pump as turbine in transient process: original papers. Int J Fluid Mach Syst. 2021;14(2):142-149.

26. Zhang J, Appiah D, Zhang F, Yuan S, Gu Y, Asomani SN. Experimental and numerical investigations on pressure pulsation in a pump mode operation of a pump as turbine. Energy Sustain Eng. 2019;7(4):1264-1279.

27. Yang SS, Kong FY, Qu XY, Jiang W-M. Influence of blade number on the performance and pressure pulsations in a pump used as a turbine. J Fluids Eng. 2012;134(12):124503-1-124503-10.

28. Shi GT, Liu XB, Wei WJ, et al. Pressure pulsation characteristics in pump as hydraulic turbine with guide. J Drain Irrigat Mach Eng. 2017;35(001):6-12.

How to cite this article: Zhang L, Li Y, Zhang Z, Chen J, Chen D. Influence of blade number on performance of multistage hydraulic turbine in turbine mode. Energy Sustain Eng. 2022;10:903-917. doi:10.1002/ese3.1070