Physical model experimental analysis of bidirectional shaft tubular pump device

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Abstract. Based on the two-way cross flow pump station of Dingbo water conservancy project in Jiangyin City, the energy characteristics, cavitation characteristics and runaway characteristics of the model device of the two-way shaft cross flow pump are tested, and the performance test data of the two-way shaft cross flow pump device in the forward and reverse operation are obtained. The results show that the flow pattern of the two-way cross flow pump is stable without vortex and the operation of the pump is stable. When the blade angle is -4°, the physical model efficiency of the two-way Shaft Tubular pump is 70.95% under the condition of 3.05 m forward design head and 57.39% under the condition of 1.43 m reverse design head; the cavitation performance of the pump near the forward and reverse design heads is the best, and the critical necessary NPSH is less than 8 m. When the maximum positive net head is 2.99 m, the maximum runaway speed of the prototype pump is 1.50 times of the rated speed of the pump; when the maximum negative net head is 2.57 m, the maximum runaway speed of the prototype pump is 1.13 times of the rated speed of the pump; the smaller the blade angle, the higher the unit runaway speed.

Keywords. Hydraulic Machinery; Model Test; Pump System; Tubular Pump

1. Introduction

Shaft tubular pump station, as a common low-head tubular pump station structure, is widely used in East Route Project of South-to-North Water Transfer and urban flood control and drainage works in China [1]. The Shaft Tubular pump device installs motor and gearbox in the shaft. The center axis of the flow channel is straight from the inlet to the outlet. The flow channel is smooth without bending, the flow pattern is smooth, the hydraulic loss is small and the pump device is efficient. It is not only small in civil work but also convenient in construction. Water flows through both sides of the shaft and the plant is low. Bidirectional Shaft Tubular pump device can be used in comprehensive operation conditions of a combination of suction, drainage and water diversion, and is suitable for practical application in low-lying areas.

Some scholars have carried out relevant research work on the device of the Shaft Tubular pump and obtained corresponding research results. Literature [2-4] adopted a physical model and numerical simulation technology to analyze and study the internal flow field and hydraulic performance of a unidirectional Shaft Tubular pump device. Literature [5-7] analyzed the evolution of shaft profile, structure of the outlet channel and standardization of shaft flow channel of the vertical tubular pump unit. Literature [8-9] has carried out research work on guide vane position and flow path optimization...
of the front shaft tubular pump device. Literature [10-11] proposed a new structure of the Shaft Tubular pump unit and analyzed its physical and numerical simulation and model test. Scholars mainly use numerical simulation methods to study the internal flow of the pump units in their research and analysis work of the Shaft Tubular pump unit, while the physical model test method is seldom used.

To ensure efficient, stable and safe operation of the pump station, according to section 9.1.4 of Code for Design of Pump Station (GB50265-2010), when the diameter of the impeller is greater than 1.60m, the physical model test of pump unit should be carried out. At present, the physical model test is a common method for checking the safe and efficient operation of the pump station. Based on the background of Dingbo Hydraulic Complex Project of Xicheng Canal, this paper carries out the physical model test on the bidirectional Shaft Tubular pump device, and obtains the test data of energy performance, cavitation performance and escape characteristics of the bidirectional Shaft Tubular pump device under the conditions of forward and reverse operation, to provide data reference for the selection and design of similar pumping stations.

2. Engineering overview

2.1. Engineering overview
Dingbo Hydraulic Complex Project of Xicheng Canal is an important part of the regulation works of Xicheng Canal. Located at the junction of the Xicheng Canal and Yangtze River, the project is an important part of the Tongjiang Estuary Gate of the regulation works of Xicheng Canal (Huangchang River to Yangtze River Section). The general layout of the Dingbo Hydraulic Complex Project is in the form of a gate station combined with a one-word layout. The control gate is located in the east of the river, and the pump station and management area are arranged in the west of the river. The control gate and pump station are arranged in a centralized and compact form. The layout schematic diagram of the complex project is shown in Figure 1. The main functions of Dingbo Water Conservancy Complex Project are to enlarge the capacity of regional flood discharge to the north of the Yangtze River, to improve the capacity of flood control and waterlogging control in Wucheng Xiyu District, to enhance the capacity of regional river diversion and water resource regulation, to enhance the hydrodynamic force of regional river network and to improve the regional water environment capacity.

![Diagram of Dingbo Water Conservancy Complex Project of Xicheng Canal](image)

**Figure 1.** Diagram of Dingbo Water Conservancy Complex Project of Xicheng Canal.

2.2. Main parameters of the pump station
The bidirectional Shaft Tubular Pump Station of Dingbo Hydraulic Complex Project has the functions of forwarding drainage and reverse water diversion. The total drainage flow of the pump station is 120 m³/s. Four vertical shaft tubular pump units are used with the single unit flow of 30 m³/s and impeller diameter of 3000 mm. Quick gate cut-off is used, the hydraulic hoist is hoisted and closed, and four 10 kV synchronous motors are equipped with single-unit power of 1600 kW and the total installed capacity of 6400 kW. The flow channel structure of the pump station adopts a shaft-type flow channel and the straight pipe flow channel. During forward operation, the two-way vertical shaft tubular pump device takes shaft flow channel as intake channel and straight pipe flow channel as an outlet channel. In reverse operation, the straight pipe flow channel is the intake channel and the shaft flow channel is an outlet channel. The three-dimensional model of the two-way vertical shaft tubular pump device is shown in
Figure 2. Operation water level and characteristic head of pump station of fixed wave hydraulic complex are shown in Table 1.

![Figure 2](image)

**Figure 2.** 3D flow path model.
1. Shaft Flow Path; 2. Impelle; 3. guide vane; 4. Outlet.

**Figure 3.** Model pump unit test material.

![Figure 3](image)

**Table 1.** Operation water level and characteristic head combination of the pump station

| Operating conditions     | Water level combination | Net lift / m | Total lift / m |
|-------------------------|-------------------------|--------------|---------------|
|                         | Yangtze River side/ m   | Inland river side/ m |              |              |
| Drainage conditions     | Design lift             | 6.40         | 3.65          | 2.75         | 3.05          |
| (Forward)               | Average lift            | 4.60         | 3.94          | 0.66         | 0.96          |
|                         | Maximum lift            | 6.59         | 3.60          | 2.99         | 3.29          |
| Diversion work          | Design lift             | 2.57         | 3.70          | 1.13         | 1.43          |
| (Reverse)               | Average lift            | 3.02         | 4.03          | 1.01         | 1.31          |
|                         | Maximum lift            | 1.43         | 4.00          | 2.57         | 2.87          |

3. **Model pump device experiment**

3.1. **Model pump and pump unit**

The model scale of the two-way vertical well tubular pump device is 1:10. The hydraulic model of the two-way pump is SZM35 developed by Jiangsu Key Laboratory of hydraulic power engineering. The nominal diameter of the impeller of the model pump is d=300 mm. The impeller model is shown in Figure 3 (a), the hub ratio is 0.40, and the number of blades of the impeller is 4. The guide vane body is shown in Figure 3 (b). The diameter of the hub of the guide vane body is 120 mm, and the number of blades of the guide vane body is 5. The model of the two-way Shaft Tubular pump device is shown in Figure 3 (c). After the installation and inspection of the model pump device, the axial runout of the positioning surface of the impeller chamber and guide vane body is 0.10 mm, the radial runout of the outer surface of the hub is 0.08 mm, and the blade top clearance is controlled within 0.20 mm.

3.2. **Test contents**

1) Positive and negative energy performance test of two-way Shaft Tubular pump device model when each blade angle is (-8°, -6°, -4°, -2°, 0°).
2) The cavitation performance test of a two-way Shaft Tubular pump device model under five characteristic lift conditions of positive and negative direction when each blade is placed at an angle of (-8°, -6°, -4°, -2°, 0°).

3) The forward and reverse runaway characteristic test of the two-way pump model device with 3 blade setting angles (-8°, -4°, 0°).

The physical model test shall be conducted in accordance with the code for hydraulic performance test of centrifugal pump, mixed flow pump and axial flow pump (precision level) (GB/T) 18149-2000 and sl140-2006, in which 6.1.3 of the acceptance test code for pump model and device model stipulates that the energy performance test points of pump device at each blade angle shall not be less than 15, and the critical NPSH shall be determined according to the flow holding constant, and the effective NPSH value shall be changed to 1% of the efficiency reduction Fixed.

3.3. Model test system
The performance test of the model pump device is carried out on the high-precision hydraulic machinery test bench of Jiangsu Provincial Key Laboratory of water conservancy and power engineering. The comprehensive uncertainty of the test bench is ± 0.39%, which meets the accuracy requirements of the industrial standard of the people's Republic of China acceptance test code for pump model and device model (SL140-2006). The schematic diagram of the test bench is shown in Figure 4. The test-bed is a vertical closed-loop system, with a total length of 60.0 m and a main pipe diameter of 0.5 m. Only the 10 times straight pipe before and after the installation of electromagnetic flowmeter is 0.4m in diameter, and the volume of water in the whole system is 50m³.

![Figure 4. The high-precision hydraulic machinery test stand.](image)

Notes: 1. Intake tank 2. Test pump device and drive motor 3. Pressure outlet tank 4. Bifurcated water tank 5. Flow in situ calibration device 6. Flow in situ calibration device 7. Operating condition control gate valve 8. Steady regulator rectifier drum 9. Electromagnetic flowmeter 10. System forward and reverse operation control gate valve 11. Auxiliary pump unit

3.4. Test method
The flow of the model pump device is directly measured by DC400 electromagnetic flowmeter, and the speed of the test motor of the model pump device is adjusted by the DC rectifier. The rated speed of model test of pump device is 1440 r/min.

The head H of the pump unit is the total head difference of the two pressure measuring sections at the outlet and inlet of the pump unit. The pressure measuring section is shown in Figure 5. Section 1-1 is the inlet pressure measuring section and section 2-2 is the outlet pressure measuring section. The total head difference is equal to the algebraic sum of the static pressure difference and the dynamic pressure difference of the two sections:
(1)

where \((z_2 - z_1 + \frac{p_2}{\rho g} - \frac{p_1}{\rho g})\) is static pressure difference, \(m\); \((\frac{u_2^2}{2g} - \frac{u_1^2}{2g})\) is motive pressure difference, \(m\).

The cross-section area is almost the same, and the velocity is very small, so the dynamic pressure difference is almost zero.

\[
H = \left( z_2 - z_1 + \frac{p_2}{\rho g} - \frac{p_1}{\rho g} \right) + \left( \frac{u_2^2}{2g} - \frac{u_1^2}{2g} \right)
\]

Figure 5. Schematic diagram of pressure measurement section.

The speed and input torque of the pump shaft are measured directly by JC1A200 speed and torque sensor mounted between the drive motor and the pump shaft. Shaft power can be obtained from equation (2):

\[
N = \frac{\pi}{30} n (M - M')
\]

where \(M\) is the model pump input torque, \(N \cdot m\); \(M'\) is Mechanical loss torque of model pump, \(N \cdot m\); \(n\) is test speed of model pump, r/min.

Cavitation test keeps the flow unchanged. Cavitation occurs in the pump by gradually reducing the pressure of the system by vacuum extraction in the closed circulation system. Effective cavitation margin of pump unit under different system pressures is calculated by the following formula:

\[
NPSH_{av} = P_{av} + h - \frac{P_v}{\rho g}
\]

where \(NPSH_{av}\) is cavitation, \(m\); \(P_{av}\) is absolute pressure at pressure measuring point of intake tank of pump unit, \(Pa\); \(P_v\) is Saturated steam pressure of water at test water temperature, \(Pa\); \(h\) is Absolute pressure transmitter higher than pump vane rotation center line (pump shaft), \(m\).

Runaway characteristics can be expressed in terms of unit speed and unit flow as follows:

\[
n_{1,R}' = \frac{n_R D}{\sqrt{H}}
\]

\[
Q_{1,R}' = \frac{Q_R}{D^2 \sqrt{H}}
\]

Where \(n_{1,R}'\) is unit speed, r/min; \(Q_{1,R}'\) is unit Flow, m³/s; \(D\) is Nominal diameter of impeller, m; \(H\) is total lift difference between upstream and downstream, m; \(n_R\) isMeasured rotational speed at \(H\) value, r/min; \(Q_R\) is flow measured at \(H\) value, m³/s.

Take the value when the unit speed tends to stabilize as the unit escape speed \(n_{1,R}'\). The actual escape speed at different head points of the prototype pump can be determined by the following formula:

\[
n_{R,P} = n_{1,R}' \frac{\sqrt{H_P}}{D_p}
\]

Where \(n_{R,P}\) is Actual Runaway Speed of Prototype Pump, r/min; \(D_p\) is Diameter of prototype pump impeller, m; \(H_P\) is Head of working point of prototype pump, m.

The model efficiency of the pump unit referred to in this report is calculated after deducting the mechanical loss torque by the following formula:

\[
\eta = \frac{\rho g Q H}{N} \times 100\%
\]
Where $\eta$ is model efficiency of pump unit, %; $Q$ is flow rate of model pump unit, m$^3$/s; $H$ is model pump unit head, m; $\rho$ is real-time water density test, kg/m$^3$; $g$ is Local Gravity Acceleration, m/s$^2$.

4. Model test results

4.1. Energy characteristic test

The performance test of the model pump unit tested the energy performance of the bidirectional Shaft Tubular pump unit under positive and reverse operating conditions at 5 Blade placement angles (-8°, -6°, -4°, -2°, 0°). The optimum operating parameters of the bidirectional Shaft Tubular pump unit at each blade placement angle are shown in Table 2 and Table 3. According to the energy performance test results of the physical model of the bidirectional Shaft Tubular pump unit, the comprehensive characteristic curves of the bidirectional pump unit model in fixed wave pump station are obtained, and the positive and reverse comprehensive characteristic curves are shown in Figure 6 and Figure 7.

| Angle (°) | Flow (L/s) | Lift (m) | Shaft power (kW) | Efficiency (%) |
|-----------|------------|----------|------------------|----------------|
| -8        | 249.80     | 3.117    | 10.553           | 72.17          |
| -6        | 283.13     | 3.055    | 11.687           | 72.41          |
| -4        | 291.67     | 3.814    | 14.988           | 72.60          |
| -2        | 326.61     | 3.515    | 15.509           | 72.41          |
| 0         | 362.26     | 3.308    | 16.285           | 72.00          |

| Angle (°) | Flow (L/s) | Lift (m) | Shaft power (kW) | Efficiency (%) |
|-----------|------------|----------|------------------|----------------|
| -8        | 225.99     | 2.521    | 9.464            | 58.86          |
| -6        | 239.86     | 2.985    | 11.994           | 58.36          |
| -4        | 271.71     | 2.649    | 12.106           | 58.09          |
| -2        | 281.32     | 2.983    | 14.756           | 57.66          |
| 0         | 301.63     | 3.078    | 15.801           | 57.39          |

From the performance test results, it can be seen that when the blade placement angle is -4 degrees, the forward design head is 3.05 m, the model flow rate of the two-way Shaft Tubular pump unit is 312.62 L/s, and the efficiency of the pump unit is 70.95%. The operation range of high efficiency zone is wide. In the range of flow 220~330L/s, the efficiency of bidirectional Shaft Tubular pump device is higher than 67%. Conversion according to Article 7.2.1 of Code for Acceptance Test of Pump Model and
Device Model (SL140-2006), when the forward head of prototype pump unit is 3.05 m, the flow rate is 31.26 m$^3$/s, which is higher than the operation requirement of design flow rate 30 m$^3$/s.

When the reverse design head is 1.43 m, the flow rate of the model pump unit is 301.64 L/s and the efficiency of the pump unit reaches 57.39%. When the reverse design head of the prototype pump unit is 1.43 m, the flow rate is 30.16 m$^3$/s.

4.2. Cavitation performance test

The cavitation test of the pump unit model adopts the energy method with fixed flow rate. The effective cavitation margin of the intake pump unit model is 1% lower than its performance point as the critical cavitation margin (based on the impeller center).

When the blade placement angle is -4°, the prototype pump has the best cavitation performance near the positive and reverse design head of 3.05 m and 1.43 m, and the critical necessary cavitation margin is less than 8 m.

4.3. Uncertainty analysis of efficiency testing

4.3.1. System uncertainty of test stand. The primary sensor, differential pressure transmitter and torque speed sensor used for testing flow, head, torque, speed, etc. have been calibrated by the national approved metrology calibration department and the calibration time is within the validity period. The system uncertainty for the performance and efficiency test of the bench pump unit is the square and root of each individual system uncertainty, i.e.

$$ (E_\eta)_s = \pm \sqrt{E_Q^2 + E_H^2 + E_M^2 + E_n^2} = \pm 0.3390 \% $$

Where $E_Q$ is System Uncertainty in Flow Measurement; $E_H$ is System Uncertainty for Static Head Measurement; $E_M$ is System Uncertainty for Torque Measurement; $E_n$ is System Uncertainty in Speed Measurement.
4.3.2. Random Uncertainty in Efficiency Testing. Uncertainty estimates for performance tests based on the degree of dispersion of efficiency measurements (as shown in Table 4) under stable design head conditions are given as follows:

\[
(E_\eta)_r = \pm t_{0.95(N-1)} \times \frac{S_\eta}{\bar{\eta}} \times 100\%
\]  
(9)

Where \(S_\eta\) is Standard deviation of average efficiency:

\[
S_\eta = \sqrt{\frac{\sum (\eta_i-\bar{\eta})^2}{N(N-1)}}
\]  
(10)

Where, \(N\) is number of measurements; \(\eta_i\) is \(i\)-th Efficiency Measurement; \(\bar{\eta}\) is average efficiency; \(t_{0.95(N-1)}\) is T-distribution values corresponding to 0.95 confidence rate and \((N-1)\) degrees of freedom, \(t=2.26\).

Based on the measured data in Table 4, the calculation results are as follows:

\[
\bar{\eta} = 69.752\% \quad S_\eta = 0.027494\% \quad (E_\eta)_r = 0.0891\% \leq 0.3390\%
\]

Data is reliable.

| Table 4. Measured efficiency data table. |
|----------------------------------------|
| Q(L/s) | H(m) | P(kW) | \(\eta(\%)\) |
| 294.19 | 3.671 | 15.157 | 69.70 |
| 294.36 | 3.660 | 15.090 | 69.85 |
| 294.46 | 3.647 | 15.047 | 69.82 |
| 294.64 | 3.662 | 15.097 | 69.92 |
| 294.52 | 3.664 | 15.128 | 69.78 |
| 294.51 | 3.678 | 15.208 | 69.68 |
| 295.20 | 3.665 | 15.187 | 69.69 |
| 294.50 | 3.684 | 15.231 | 69.68 |
| 294.67 | 3.689 | 15.261 | 69.69 |
| 295.17 | 3.669 | 15.200 | 69.70 |

4.3.3. Total Uncertainty in Efficiency Testing. The total uncertainty of efficiency testing is the sum and root of the system uncertainty and the random uncertainty, i.e.

\[
E_\eta = \pm \sqrt{(E_\eta)_s^2 + (E_\eta)_r^2} = \pm 0.3504\%
\]

The comprehensive uncertainty of efficiency for model test of bidirectional Shaft Tubular pump unit meets the requirements of Code for Acceptance Test of Pump Model and Device Model (SL140-2006).

4.4. Runaway characteristic test

By switching the test system of high-precision hydraulic machinery test stand, adjusting the auxiliary pump to make the pump running system run in the opposite direction without force on the torquemeter, the speed of the model pump is tested at three vane placement angles(-8°, -4°, 0°) at different head.

Under positive operating conditions, the unit run-off speed of the bidirectional Shaft Tubular pump unit at each blade placement angle is shown in Table 5, and the run-off speed of the original model pump at each angle is shown in Table 6. According to the test results, the forward escape characteristic curve of prototype pump of Dingbo Hydraulic Complex can be obtained, as shown in Figure 10.

Under reversed operating conditions, the unit run-off speed of the bidirectional Shaft Tubular pump unit at each blade placement angle is shown in Table 7, and the run-off speed of the original model pump at each angle is shown in Table 8. According to the test results, the forward escape characteristic curve of prototype pump of Dingbo Hydraulic Complex can be obtained, as shown in Figure 11.
Table 5. Runaway speed of prototype pump at positioning Blade Positioning Angles.

| Angles (°) | -8  | -4  | 0   |
|-----------|-----|-----|-----|
| Unit Runway Speed (r/min) | 400.52 | 374.76 | 353.18 |

Table 7. Unit runaway speed at reverse blade positioning angles.

| Angles (°) | -8 | -4 | 0   |
|-----------|----|----|-----|
| Unit Runway Speed (r/min) | 338.72 | 304.89 | 282.45 |

Table 6. Runaway speed of prototype pump at positioning angle of each vane.
(Maximum net lift 2.99m)

| Angle (°) | Runaway speed (r/min) | Ratio to motor speed rating |
|----------|------------------------|-----------------------------|
| -8       | 230.85                 | 1.60                        |
| -4       | 216.01                 | 1.50                        |
| 0        | 203.57                 | 1.41                        |

Table 8. Runaway speed of prototype pump at opposite vane placement angles.
(Maximum net lift 2.99m)

| Angle (°) | Runaway speed (r/min) | Ratio to motor speed rating |
|----------|------------------------|-----------------------------|
| -8       | 181.00                 | 1.26                        |
| -4       | 162.93                 | 1.13                        |
| 0        | 150.93                 | 1.05                        |

Figure 10. Forward runaway characteristic curve of prototype pump.

Figure 11. Reverse runaway characteristic curve of prototype pump.

5. Conclusion

(1) When the blade angle is -4°, the forward design head of the two-way vertical well tubular pump device model is 3.05 m, the flow is 312.62 L/s, and the efficiency of the pump device reaches 70.95%; the maximum operation head of the model pump is more than 4.0 m, which meets the operational requirements of 3.29 m forward maximum head of the two-way pump in Dingbo water conservancy project. When the reverse design head of the model pump unit is 1.43 m, the flow is 301.64 L/s, and the efficiency of the pump unit reaches 57.39%; when the reverse head of the corresponding prototype pump unit is 1.43 m, the flow is 30.16 m³/s, which meets the operational requirements of the pump station; when the maximum operation head of the model pump exceeds 4.0 m, it meets the operational requirements of the two-way pump reverse maximum head of the fixed wave water conservancy project, which is 2.87 m.

(2) The prototype pump unit has the best cavitation performance near the positive and reverse design head at the blade placement angle of -4°, and the critical necessary cavitation margin is less than 8 m, which meets the requirements of the bidirectional pump of fixed wave hydraulic complex for the critical necessary cavitation margin during the forward and reverse operation.

(3) The maximum escape speed of the prototype pump under the condition of maximum net head of 2.99 m under forwarding operation is 1.50 times the rated speed of the pump at the blade placement angle of -4°. The maximum escape speed of prototype pump under the condition of maximum net head of 2.57 m under reverse operation is 1.13 times the rated speed of pump. The smaller the blade placement angle, the higher the unit escape speed.
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References
[1] Liu C 2015 Researches and developments of axial-flow pump system Trans. Chin. Soci. Agri. Machin. 46(06) 49-59
[2] Chen H X, Zhou D Q, Zhang L G and Li L Y 2013 Hydraulic performance improvement of bidirectional shaft tubular pump system based on CFD Water Resour. Power. 31 1183-7
[3] Yang F, Liu C, Tang F P, Cheng L and Lv D W 2014 Numerical simulation of 3D internal flow and performance analysis of the shaft tubular pump system. J. Hydroelectr. Engin. 33(01) 178-84
[4] Xia Y, Tang F P, Shi L J, Xie C L and Zhang W P 2017 Numerical simulation and experimental analysis of bidirectional shaft tubular pump device China Rural Water Hydropower 7 149-53
[5] Zhang R T, Zhu H G and Yao L B 2014 Comparison of shaft-type tubular pump systems with different outflow structures J. Hydroelectr. Engin. 33(1) 197-201
[6] Yang F, Liu C, Tang F P and Zhou J R 2014 Shaft shape evolution and analysis of its effect on the pumping system hydraulic performance J. Basic Sci. Engin. 22(1) 129-39
[7] J Chen Q, Zhu Q R, Su Z M, Z Zhou F and Chen S S 2019 Shaft tubular pump units based on regularized design J. Hydroelectr. Engin. 38(02) 101-11
[8] Fan M, Pei J, Li Y J, Yuan S Q and Chen J 2017 Effect of guide vane position on hydraulic performance of two-direction tubular pump device Trans. Chin. Soci. Agri. Machin. 48(02) 135-40
[9] Xie R S, Wu Z, He Y, Tang F P and Xie C L 2015 Optimization research on passage of bidirectional shaft tubular pump Trans. Chin. Soci. Agri. Machin. 46(10) 68-74
[10] Zhu H G, Dai L Y, Zhang R T, Zhu G X, Lv S Y and Fei H R 2011 Development and numerical analysis of new-type shaft tubular pumping system J. Drain. Irriga. Machin. Engin. 29(05) 418-22
[11] Zhu H G, Zhang R T and Zhou J R 2012 Optimal hydraulic design of new-type shaft tubular pumping system IOP Conf. Ser.: Earth Environ. Sci. 15 052026