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Flow Characteristics of Two-way Passage Vertical Submersible Pump System

Liu Chao[1] Zhou Qinglian[2] Qian Jun[3] Jin Yan[1] Xie Chuanliu[1]

(1. *School of Hydraulic Energy and Power Engineering, Yangzhou University, Yangzhou* 225127, *China*
2. *Lianyungang Water Resources Planning and Design Institute Ltd.*, *Lianyungang* 222006, *China*
3. *Management Division of Qinhuai River Hydraulic Engineering of Jiangsu Province*, *Nanjing* 210001, *China*)

Abstract: The submersible axial-flow pump system has been more widely applied in pumping projects. Usually the existing submersible pumps used for bi-directional pumping stations are the reversible pumps, the forward and reverse pumping of which is less efficient. The flow characteristics of a new bi-directional submersible pump system combined with the submersible pumps and two-way passages together were explored. By CFX software the full-flow numerical simulation of the system was made and the system flow field was obtained. Also the system hydraulic performance was predicted. The flow characteristics of suction passage with different measures were analyzed. The velocity distribution uniformity at outlet of suction passage showed the elliptical-type flow guide cone of the passage worked the best to prevent harmful vortex, which guaranteed the flow conditions of pump operation. Using specially designed discharge-chamber with unilateral angle of 18°, the flow separation was effectively inhibited, the hydraulic losses were reduced which ensured the overall efficiency of pump systems at high level. A model test was conducted in the high-precision test-bed of hydraulic machinery. The test results showed that under the pump head of 3.11 m, flow rate of 256 L/s, the pump system efficiency reached 71.9%, which was above the forward or backward pumping efficiency of reversible pump system by 7 and 13 percentage points, respectively. It was evident that the vertical submerged pump system with two-way passage was suitable for bi-directional pumping station. Experimental results and predicted model performance results were consistent in high efficiency area, and the numerical calculations were well verified.

Key words: pump system; submersible pump; two-way passage; numerical simulation

0 Introduction

Two-way passage pump system can meet the needs of water diversion and drainage, and minimize the use of land resource[1]. But the low efficiency of the system and the vibration of the pump unit are always difficult problems faced to the engineering and academia[2]. Usually, the efficiency of two-way passages pump system is generally around 65%, which is 13% ~ 18% lower than the one way passage pump system. The vibration of the pump unit caused by the vortex in the two-way passage is more common. The vibration can directly cause the damage of the unit components and that is unable to run. Since then, the two-way pump system is turned to use the reversible horizontal axis pump system in both directions. The reversible pump system operates forward or backward to change the direction in and out of the water, so as to realize the bidirectional pumping[3-4]. The flow passage of this pump system is the same as the unidirectional passage which has no vortex inside, but the forward and reverse running efficiency is low. The efficiency of the forward running system is about 65%, while the efficiency of the reverse running system is only about 58%. Until 1990s, the using of curve diffusion outlet structure improved the pump system efficiency by 8 ~ 10 percentage points, and the set of flow guide cone and the vortex grid effectively eliminated the vibration of the pump unit[5-6]. This form of pump system has been included in the relevant specifications, also

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Corresponding author: Liu Chao, Professor. E-mail: liuchao@yzu.edu.cn. Tel: +86-139-05274051.

applied in many projects[7-13]. In recent years, the submersible axial-flow pump system has been more widely applied in pumping projects, which was developed from small size of pump to the 2.5 m diameter of large axial flow pump system[14-18]. The submersible pump is submersible electric pump, submersible motor and pump impeller and guide vane installed as a whole, has the advantages of good integrity, convenient installation, maintenance standardization and high management efficiency. The existing submersible pumps used for two-way pumping stations are generally reversible pump which can not meet the requirements of users because of their low efficiency. If the submersible pumps and bi-directional passages are combined together to form a new two-way passage submersible pump system, it can not only improve the operation efficiency but also ensure the safety of operation. Although the application of submersible pump can simplify the structure, the submersible motor will change the outlet structure and inevitably increase the guide vane diffusion angle of the axial flow pump, which is easy to generate flow separation and increase flow resistance.

The CFX software was used to simulate the entire pump system, and the overall numerical analysis and optimization of hydraulic design was adopted, on this basis, the model of two-way passage submersible pump was made and the model test was carried out, the comprehensive performances of the pump system were obtained and a new form of the bi-directional pumping station projects was provided.

1 Flow characteristics of two-way passage pump system

1.1 Mathematical model of pump system numerical simulation

Flow in the pump is incompressible, the basic equations used to describe the fluid flow included the continuity equation and the momentum equation (N-S equation). The RANG Reynolds averaged equation of momentum equation was adopted. The formulas respectively are

$$\frac{\partial u_i}{\partial x_i}=0 \tag{1}$$

$$\frac{\partial(\rho u_i)}{\partial t}+\frac{\partial(\rho u_i u_j)}{\partial x_j}=F_i-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right] \tag{2}$$

where ρ is the fluid density; u_i and u_j are the velocity component; t is time; F_i is the volume force component in the i direction; p is the pressure; μ is the dynamic viscosity coefficient; x_i and x_j are coordinate components. The standard $k-\varepsilon$ model is used as the turbulence model.

1.2 Geometry modeling of the pump system

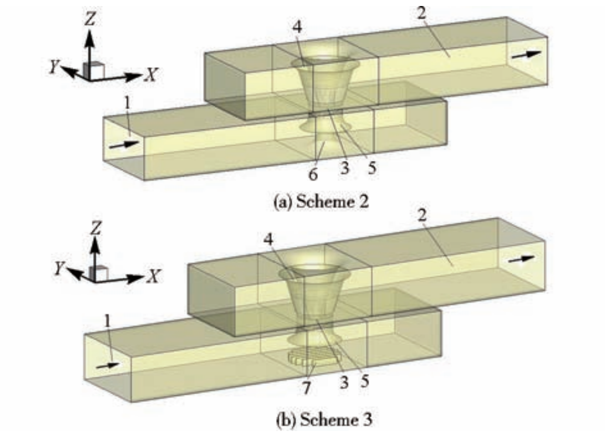
During the two-way passage vertical axial flow pump station has been built, below the bell of suction passage prone to vortex, which causes the vibration of the pump unit and reduces the efficiency of the pump system, even affects the safe operation of the pump unit. Therefore, it is necessary to take measures to prevent vortex in the suction passage. Curve shaped diversion cone and the eliminate vortex lattice are respectively set up in the suction passage, and the pump performance is compared. In engineering practice, a lower passage in two-way passage pump system is commonly used as artesian water diversion channels, under the bell a set of vortex lattice is installed with a certain height, well flow direction parallel arrangement and small water resistance, to compare the influence of different cases on the performance of pump system.

The three-dimensional model of the two-way passage submersible pump systems is composed of suction passage, outlet passage, impeller, guide vane and so on, and the calculation area is the whole pump system, as shown in Fig. 1 to Fig. 5. Impeller diameter of the pump is 300 mm. In order to facilitate the installation of submersible pump motor, the diffusion guide vane with unique large diffusion angle (the unilateral diffusion angle is 18°) and the curved diffusion water chamber are adopted. The guide vane diffusion angle is about double of the conventional guide vane diffusion angle, the number of blades is 7. There are three calculation schemes as follows: scheme 1 (without water diversion measures), scheme 2 (with water diversion cone) and scheme 3 (with vortex grid).

1.3 Computational grid

Structured grid is adopted in the inlet and outlet passage of the pump device, the same to the inlet and outlet chamber and impeller guide vanes whose structure and shape are more complex, and the grid density is appropriately increased, as shown in Fig. 6.

The geometric model and mesh generation of pump



(a) Scheme 2

(b) Scheme 3

Fig. 1 Two-way flow submersible axial flow pump system

1. Suction passage 2. Outlet passage 3. Impeller and guide vane
4. Curved diffusion chamber 5. Bell mouth 6. Diversion cone
7. Eliminate vortex lattice

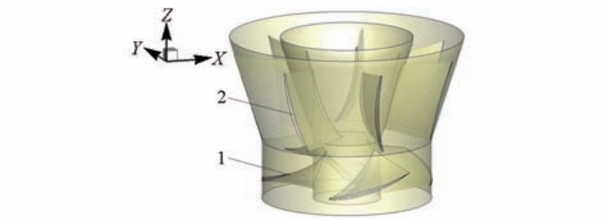


Fig. 2 Pump impeller and guide blade

1. Impller 2. Guide vane

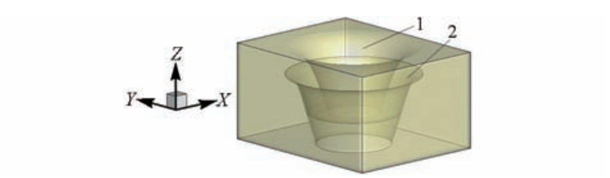


Fig. 3 Curved diffusion chamber

1. Outlet bell 2. Inner wall of chamber

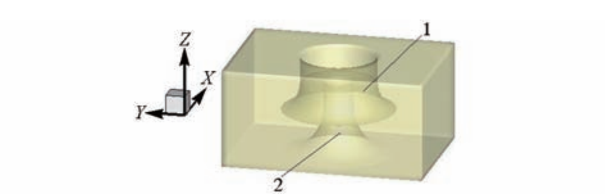


Fig. 4 Detail structure of diversion pier

1. Inlet bell 2. Flow diversion cone

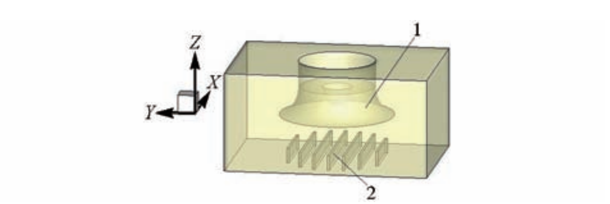
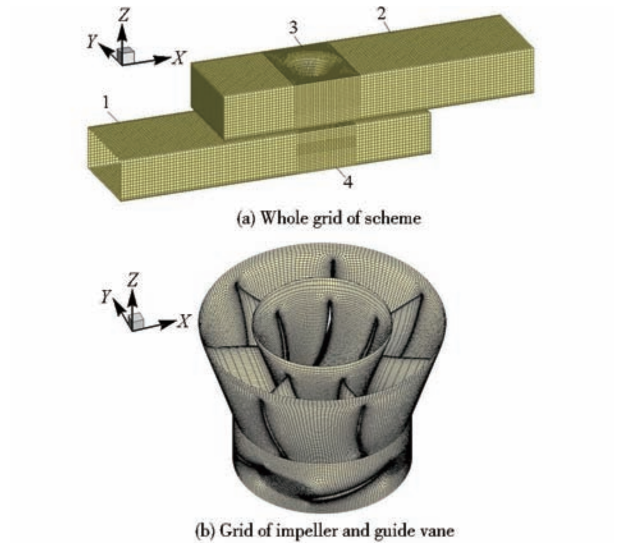


Fig. 5 Detail structure of anti-vortex grid

1. Inletbell 2. Anti-vortex grid

impeller diffuser vane are completed by TurboGrid, as shown in Fig. 6b. The blade numbers of guide vanes are 7 and 5. After the completion of the guide vane model, the ATM optimized method is used to mesh generation, the quality of the mesh is adjusted by



(a) Whole grid of scheme

(b) Grid of impeller and guide vane

Fig. 6 Computation grid of pump system and its main components

1. Suction passage 2. Outlet passage 3. Diffusion chamber
4. Intake chamber

adjusting the control points of the grid to meet the requirements of the calculation (mesh angle was between 15° and 165°).

Mesh quality of the two-way passage submersible axial-flow pump system meets the accuracy requirement of numerical simulation. The total number of grid nodes is 1 209 384 ~ 1 222 508, the total number of cells is 1 104 692 ~ 1 119 646, and the convergence precision is satisfied.

1.4 Calculation parameters and boundary conditions

The numerical calculation pump model coordinates the origin to the center line of the impeller and the bottom outlet node, the Z axis is the impeller center line pointing in the direction of the water chamber, the X axis is the flow direction from the inlet to the outlet. Standard $k-\varepsilon$ turbulent model is used in numerical simulation, and the discharge range is from $0.45Q_0$ to $1.26Q_0$ (Q_0 is the design discharge). The calculation speed is 1 250 r/min.

The inlet boundary of the pump system isset in the inlet of the suction passage, and the flow direction is perpendicular to the direction of the inlet section, mass flow inlet conditions are used.

The outlet boundary is set in the outlet of the outlet passage, and the flow direction is perpendicular to the direction of the outlet section, the pressure outlet conditions are used, pressure outlet is static, the relative pressure is 1 atm, the density of water is 1 000 kg/m^3.

No slip condition is selected for the wall boundary of the pump system.

2 Numerical simulation results of flow characteristics of pump system

2.1 Internal flow characteristics of pump system

2.1.1 Whole flow field

The whole flow field under the design condition of the two-way passage vertical submersible axial flow pump system with different measures in suction passage is obtained by numerical simulation calculation, and the inner flow line of the pump system is shown in Fig. 7. Clearly shown on the figure, the flow of the main stream area is relatively smooth, and the flow pattern is similar to the flow pattern in the unidirectional passage[19]. But large backflow zone is appeared in the blind end of the inlet and outlet passages. The blind end of suction passages is relatively long, the flow velocity is low, and the backflow intensity is small, which causes minimal influence to the flow of the pump inlet section and the hydraulic loss. While the blind end of the outlet passage is short, and the guide vane diffusion angle is large, which causes a large strength of backflow intensity in this area and water flow disturbance, leading to a higher hydraulic loss in the blind end.

Seen from the flow of the discharge-chamber, the velocity distribution is symmetrical and not to take off flow separation and vortex. Meanwhile, velocity is reduced to below 2 m/s in the outlet of the discharge-chamber, flow energy is recycled effectively, and the hydraulic loss is under control.

Diversion cone and anti-vortex grid are set up respectively in the suction passages of scheme 2 and scheme 3 which effectively prevent the vortex below the pump inlet. The scheme 1 has no diversion measures, which causes vortex on the bottom of flow passage easily. All of the three schemes have relatively smooth main stream in their suction passages. The distribution of the main stream on the exit section of the suction passage in scheme 2 is more symmetric and uniform.

At the exit section of the outlet passage, the flow pattern is more complex and the streamline has a larger swing. It is due to that the flow at the impeller exit with a large residual circulation, and the secondary reflow caused by changing 90° in flow direction[20]. The residual circulation at guide vane exit of the pump

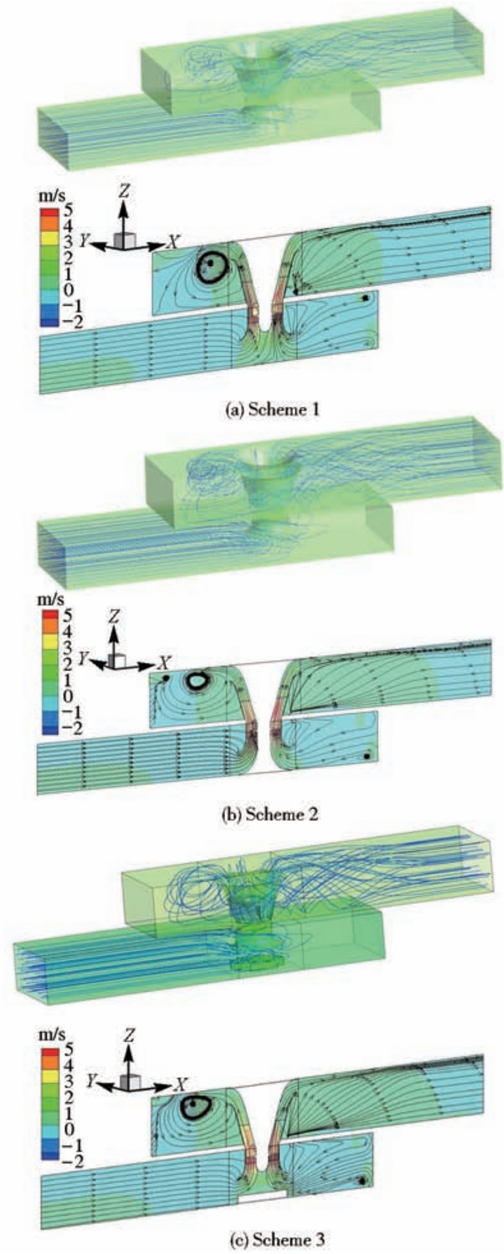


Fig. 7 Three dimensional streamlines of pump system and longitudinal profiles under design conditions for different schemes

causes a spiral flow in the outlet passage and the spiral flow shifts to the top wall of the outlet passage. If the spiral flow is improved, the recycling of kinetic energy will be increased, and the efficiency of the two-way passage pump system will be improved[21-23].

2.1.2 Axial velocity distribution at outlet section of suction passage

The outlet of suction passage connects with inlet of impeller. Generally, pump impeller causes a cyclical impact to the axial velocity distribution of outlet section of suction passage. For the convenience of analysis, on the place which is 0.16 times of the impeller diameter below the inlet section of the pump, the horizontal circle cross section is selected as the outlet section of the suction passage. The axial velocity contours under design conditions of different schemes are shown in Fig. 8.

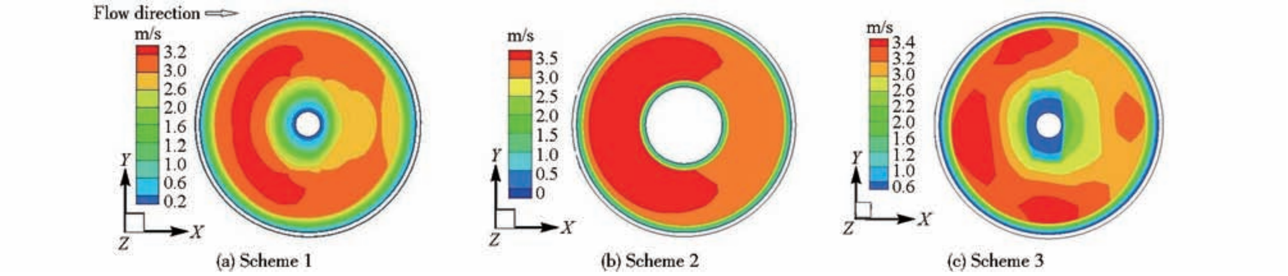


Fig. 8 Axial velocity contours under design conditions

Obviously, the velocity distribution of scheme 2 with diversion cone has a better uniformity than that of scheme 1 without any diversion measures and scheme 3 with anti-vortex grid. The regularity of the velocity distribution at the outlet section of the suction passage is that the water side velocity is higher than the other side and axial velocity distribution is rarely influenced by pump impeller rotation. According to the numerical simulation results, the uniformity of the axial velocity distribution on the outlet section of the suction passage is calculated. Results are shown in Fig. 9. Under design condition, in scheme 1, the axial velocity uniformity of the outlet section of suction passage is 92.0% and the velocity-weighted average angle is 78.3°. In scheme 2, the axial velocity uniformity of outlet section of suction passage is 96.2% and the velocity-weighted average angle is 79.7°. In scheme 3, the axial velocity uniformity of outlet section of suction passage is 88.2% and the velocity-weighted average angle is 75.4°. The axial velocity uniformity of suction passage and velocity-weighted average angle with diversion cone has better evenness than those of schemes without diversion measures and anti-vortex grid. The diversion cone limits the flow singular point on the bottom of the suction passage, improves the flow boundary, guides the water flow and maintains high evenness under different discharge conditions. Relationship curve of the axial velocity uniformity and the velocity-weighted average angle changing with discharge of pumping system is shown in Fig. 9.

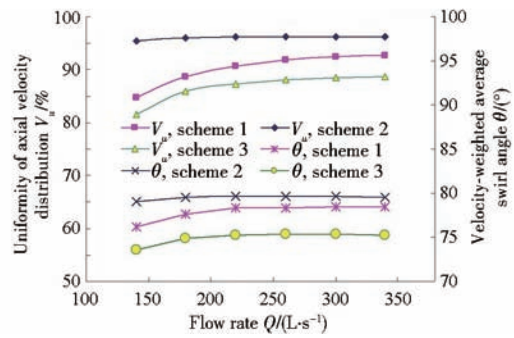


Fig. 9 Hydraulic characteristics of suction passage

2.1.3 Flow characteristics of inlet and outlet passage

In order to study the inner flow characteristics of inlet and outlet passage more clearly, the horizontal section which is −0.45 m away from the bottom of inlet passage in z direction is chosen as the characteristic section for analyzing under the design condition. Fig. 10 is the streamlined diagram at horizontal section of suction passage.

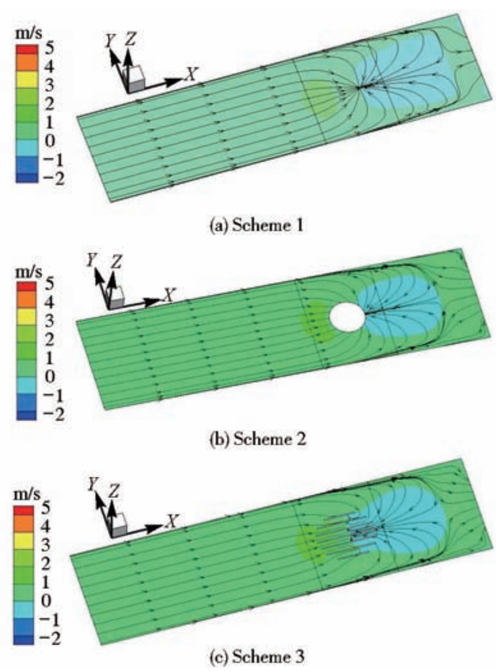


Fig. 10 Streamlines of inlet passage at horizontal section (z −0.45)

Seen from Fig. 10, in the scheme with diversion cone, water flow drains circularly from bell in the inlet passage, the pressure distribution is uneven. The local low pressure region on the water side shows that the velocity distribution is also uneven. Under the condition without diversion measure, the inlet flow status and the velocity distribution have almost the same principle. The flow which has the singular point under the impeller is different from the elbow inlet. The rotational energy of flow accumulated, once the rotational energy reaches the critical value, the vortex rope appears[24]. Vortex rope is the vortex with gas in the central region and has large rotational strength, it causes violent vibration of the pump system when the vortex passes through the pump impeller, and it has a great harm on the pumping station. Under the condition with anti-vortex grid, the regularities of inlet flow pattern and velocity distribution are almost the same. But the anti-vortex grid can eliminate the singular point, so vortex rope can be prevented, the local low pressure region on the water side is improved and the velocity distribution is more uniform compared with the above two cases.

As shown in Fig. 11, the velocity distribution at horizontal section which is +0.36 m away from the bell mouth of the outlet passage in z direction. The water flows out from the discharge chamber and becomes disordered at the back of the passage. This is the main part of hydraulic loss in such outlet passage. By reducing the outflow velocity of the discharge chamber, turbulence intensity can be controlled to some extent so that the hydraulic loss will be less and the pipeline efficiency will be improved. Water flow from the discharge chamber to the outlet of the passage is still affected by the residual circulation, and spread orderly, the whole flow is smooth.

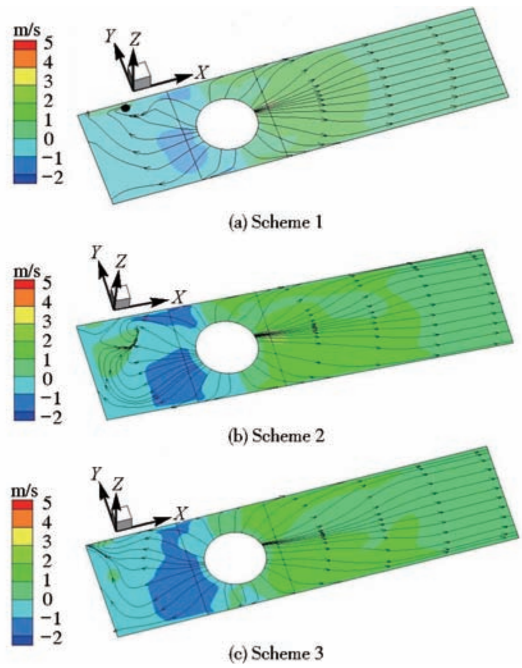


Fig. 11 Streamlines of outlet passage at horizontal section (z +0.36)

2.2 Performance prediction of pumping system

According to the numerical simulation of the pump system, the performance of two-way passage vertical submersible axial flow pump is predicted. The external characteristic curves are shown in Fig. 12. As seen in Fig. 12, there are no obvious changes on the head-discharge curve and efficiency-discharge curve with different diversion measures. The performance curves of scheme 1 and scheme 3 are almost coincident; in scheme 2, the head is slightly higher than those of the other two schemes with the same discharge; when the discharge is less than that of the optimal working condition, the efficiency of scheme 2 is slightly lower than those of the other two schemes, but when the discharge is more than that of the optimal working condition, the contrary is the case; the optimal point corresponding discharge and efficiency of scheme 2 are 270 L/s and 71.25%, respectively. The optimal point corresponding discharge and efficiency of scheme 1 are 270 L/s and 72.38%, respectively. The optimal point corresponding discharge and efficiency of scheme 3 are 270 L/s and 71.53%, respectively. In general, the efficiency among the three schemes is slightly different. The main purpose of install diversion vortex elimination measures is to eliminate or prevent the vortex under the bell mouth in suction passage, so as to avoid the hydraulic vibration. Engineering practice shows the hydraulic vibration can cause quite large damage to stable operation of the pump units. The performance prediction results of the pump system shows that setting vortex elimination measures have no obvious influence on the performance of pump system.

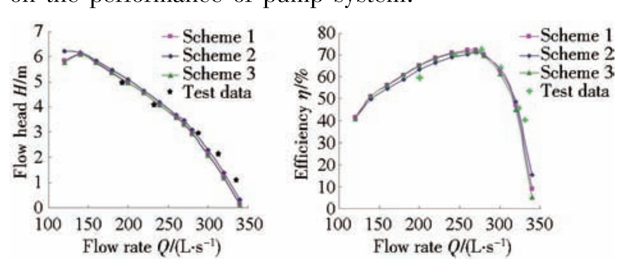


Fig. 12 Predicted pump performance curves of system with different measures

In order to analyze the influence on hydraulic loss of the suction passages caused by setting diversion measures, the hydraulic loss of suction passages with different diversion measures is calculated. Fig. 13 shows the hydraulic loss curves of three different suction passages.

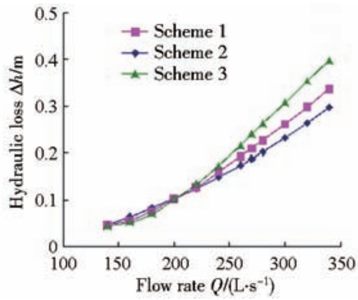


Fig. 13 Hydraulic losses of different suction passages

As shown in Fig. 13, under the small discharge condition, the hydraulic loss of the suction passage in scheme 2 is larger than that in scheme 1, and the hydraulic loss in scheme 1 is larger than that in scheme 3. It confirms that under small discharge condition the efficiency of scheme 2 is lower than that of scheme 1, which illustrates when the discharge is small, the flow resistance loss in suction passage caused by adding diversion cone and anti-vortex grids is larger than the energy loss caused by vortex without any measures. While under the large discharge condition, the hydraulic loss caused by vortex without any measures is larger than the flow resistance loss in suction passage caused by adding diversion cone, but smaller than that of adding anti-vortex grid, which means the hydraulic loss of scheme 2 is less than that of scheme1, but the hydraulic loss of scheme 1 is less than that of scheme 3. The water flow resistance of anti-vortex grids is the largest may be related to the head shape of the anti-vortex grids, which means the further optimization and improvement of the anti-vortex grids' structure and shape are necessary.

3 Model test of pump system

3.1 Model pump and pump system

In order to further obtain the comprehensive performance of the pump system and verify the result of the numerical simulation, a corresponding model pump system is made and the test is carried out on the high precision hydraulic machinery test bench. ZM4 - K model pump system is adopted in the test, which consists of ZM4 impeller and DYK guide vane. Impeller diameter is 300 mm. The hub ratio is 0.4. The number of the impeller blades is 4. The impeller is produced by numerical control process with brass material. The number of DYK guide vane blades is 7, produced by welding fabrication with steel materials. The inlet and outlet passages produced by the steel plate welding, there are observation windows in the central section of inlet and outlet passages, which can easily observe the flow pattern in the flow passage. The model pump system is shown in Fig. 14. The test velocity is 1 250 r/min.

Fig. 14 Test model of pump system

Test is implemented according to the standard of GB/T 18149—2000 "centrifugal pump, mixed flow pump and axial flow pump hydraulic performance test specification (precision)" and SL140—2006 "water pump model and device model acceptance test procedure".

3.2 Model test results

Energy performance and cavitation performance of the model pump system are tested under four different blade angle conditions (−4°, −2°, 0°, 2°). According to the test results, the comprehensive performance curve of a two-way passage vertical submersible axial flow pump model are obtained, as shown in Fig. 15.

The best efficiency point of the two-way passage vertical submersible axial flow pump system at different blade angles (−4°, −2°, 0°, 2°) are 70.94%, 71.90%, 71.54% and 69.25%, respectively. The maximum value occurred at blade angle of −2°. At this point, the corresponding head is 3.113 m, the discharge is 256.6 L/s, and the efficiency is 71.90%, which are 7 and 13 percentage points higher than those of reversing two-way submersible pump device on forward and reverse operations. The maximum operating head is larger than 5 m, and the ratio between the maximum head and design head is more than 1.5, which can meet the practical requirement of engineering. No harmful vortex in the suction passage is observed after adding diversion cone and so on.

As shown in the results of the cavitation performance test, the NPSH of pump system at the high efficiency point is less than 6.5 m within four different blade angles. Cavitation specific speed is larger than 1 100 that meets the requirements of pump operation.

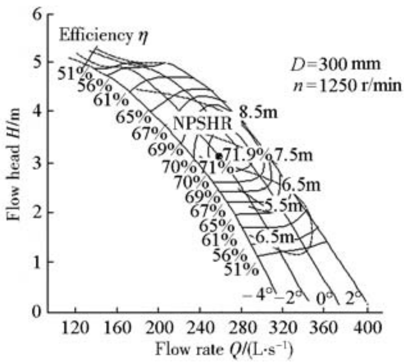


Fig. 15 Comprehensive performance curve of model pump

The data of model test verifies the pump system performance acquired by numerical calculating prediction. Both results are in good agreement on the high efficiency area near the maximum efficiency point.

4 Conclusions

(1) Adding diversion cone in the suction passage of two-way passage vertical submersible axial flow pump system, under the optimal conditions, the axial velocity uniformity on the outlet section of the suction passage is 96.2%, the velocity-weighted average angle is 79.7°. The diversion cone can effectively eliminate the vortex in the suction passages, which also can maintain a good inflow conditions.

(2) The results of pump performance test show that adopting the diffusion guide vane with 18° unilateral diffusion angle, and adjusting the blade angle at −2°, the head of pump system is 3.113 m, discharge is 256.6 L/s, and efficiency is 71.90%. The maximum operating head is larger than 5 m. The performance is similar to the two-way passage conventional axial flow pump system, which means the diffusion guide vane is feasible for the two-way passage vertical submersible axial flow pump system, and it is of great value in engineering application.

(3) Comparing with the results of performance test and numerical simulation of pump system, both of them are in good agreement on the high efficiency area, indicating that the numerical simulation results are reliable.

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双向流道立轴潜水泵系统流动特性研究

刘 超[1] 周庆连[2] 钱 均[3] 金 燕[1] 谢传流[1]

(1. 扬州大学水利与能源动力工程学院, 扬州 225127; 2. 连云港市水利规划设计院有限公司, 连云港 222006;
3. 江苏省秦淮河水利工程管理处, 南京 210001)

摘要: 为探索将潜水电泵和双向流道泵装置结合在一起的双向流道潜水电泵系统,通过 CFX 软件对该系统进行全流道数值模拟,获得了系统内的流动特性,并预测了泵装置的水力性能。对进水流道内加设不同导流措施的水流特性进行了分析,结果表明,加设椭圆线导流锥的进水流道出口流速分布均匀度效果最好,能够防止有害旋涡,保证水泵运行的进水条件。应用特别设计的、单边 18°的大扩散角出水室,有效地抑制了脱流和水力损失,确保水泵系统整体效率水平。在高精度水力机械试验台进行了模型试验。试验结果表明,泵装置扬程为 3.11 m,流量为 256 L/s,泵装置效率达到 71.9%,正、反向分别高于可逆式双向潜水泵装置 7 和 13 个百分点。说明双向流道配立轴潜水泵装置具有良好的工程应用价值。模型试验结果和性能预测结果在高效区范围内吻合,数值计算得到较好的验证。

关键词: 泵系统; 潜水泵; 双向流道; 数值模拟

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Flow Characteristics of Two-way Passage Vertical Submersible Pump System

Liu Chao[1] Zhou Qinglian[2] Qian Jun[3] Jin Yan[1] Xie Chuanliu[1]

(1. *School of Hydraulic Energy and Power Engineering*, *Yangzhou University*, *Yangzhou* 225127, *China*
2. *Lianyungang Water Resources Planning and Design Institute Ltd.*, *Lianyungang* 222006, *China*
3. *Management Division of Qinhuai River Hydraulic Engineering of Jiangsu Province*, *Nanjing* 210001, *China*)

Abstract: The submersible axial-flow pump system has been more widely applied in pumping projects. Usually the existing submersible pumps used for bi-directional pumping stations are the reversible pumps, the forward and reverse pumping of which is less efficient. The flow characteristics of a new bi-directional submersible pump system combined with the submersible pumps and two-way passages together were explored. By CFX software the full-flow numerical simulation of the system was made and the system flow field was obtained. Also the system hydraulic performance was predicted. The flow characteristics of suction passage with different measures were analyzed. The velocity distribution uniformity at outlet of suction passage showed the elliptical-type flow guide cone of the passage worked the best to prevent harmful vortex, which guaranteed the flow conditions of pump operation. Using specially designed discharge-chamber with unilateral angle of 18°, the flow separation was effectively inhibited, the hydraulic losses were reduced which ensured the overall efficiency of pump systems at high level. A model test was conducted in the high-precision test-bed of hydraulic machinery. The test results showed that under the pump head of 3.11 m, flow rate of 256 L/s, the pump system efficiency reached 71.9%, which was above the forward or backward pumping efficiency of reversible pump system by 7 and 13 percentage points, respectively. It was evident that the vertical submerged pump system with two-way

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作者简介: 刘超(1950—),男,教授,主要从事泵站工程理论与应用技术研究,E-mail: liuchao@yzu.edu.cn

passage was suitable for bi-directional pumping station. Experimental results and predicted model performance results were consistent in high efficiency area, and the numerical calculations were well verified.

Key words: pump system; submersible pump; two-way passage; numerical simulation

引言

双向流道泵系统是满足引排水需要,最大程度减少土地资源使用的泵系统形式[1]。而双向流道泵系统效率低和机组振动一直是工程和学术界面临的难题[2]。通常的双向流道泵系统效率一般在65%左右,比单向流道泵系统低13~18个百分点。双向进水流道内的涡带所引发的机组振动则较为普遍,振动可以直接导致机组部件的损坏而无法运行。此后,双向泵系统转向采用双向可逆式水平轴泵系统。可逆式泵系统通过水泵正反向运转来改变进出水方向,从而实现双向抽水[3-4]。这种泵系统的流道与单向流道相同,流道内没有旋涡问题,但是正反向运行效率比较低。正向运行装置效率为65%左右,反向运行装置效率仅为58%左右。直到20世纪90年代,曲线扩散出水结构的采用使泵系统效率提高了8~10个百分点,导流锥和防涡栅的设置有效地消除了机组的振动[5-6]。这种泵系统的形式已经被列入有关的规范中,也在较多的工程中应用[7-13]。近年来,潜水轴流泵系统在工程中应用愈来愈多,从尺寸较小的泵型,发展到2.5 m口径的大型轴流泵系统[14-18]。潜水泵实质是潜水电泵,潜水电动机和泵叶轮、导叶联成整体,具有整体性好、安装便捷、维修标准化、管理效能高等优点。现有的潜水泵用于双向泵站通常均为可逆式泵,其系统的效率较低,不能满足用户要求。若将潜水电泵和双向流道泵系统结合在一起形成新双向流道潜水电泵系统,则既能提高运行效率又能保证运行安全。应用潜水泵可以简化系统结构,但潜水电动机不可避免地改变了出水结构,使轴流泵的出水导叶扩散角增大,容易产生脱流,增加水流阻力。

本文应用CFX软件对该泵装置进行整体数值模拟分析和水力设计优化,在此基础上制作双向流道潜水泵装置模型进行模型试验,获得泵装置综合性能,为双向泵站工程提供新的的装置形式。

1 双向流道潜水泵系统内流特性

1.1 泵系统数值模拟的数学模型

泵装置内水流的流动为不可压缩紊流,用于描述流体流动状态的基本方程包括连续性方程和动量方程(N-S方程),本文动量方程采用RANG雷诺平均方程,公式分别为

$$\frac{\partial u_i}{\partial x_i}=0 \tag{1}$$

$$\frac{\partial(\rho u_i)}{\partial t}+\frac{\partial(\rho u_i u_j)}{\partial x_j}=F_i-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right] \tag{2}$$

式中 ρ——流体密度

u_i、u_j——流体在 i、j 方向的速度分量

t——时间 p——压力

F_i——i 方向的体积力分量

μ——动力粘性系数

x_i、x_j——坐标分量

紊流模型采用标准 $k-\varepsilon$ 模型。

1.2 泵系统几何建模

已建成的双向流道立式轴流泵泵站中,进水流道内喇叭管下方容易产生旋涡,引发机组的振动,降低泵装置效率,严重时影响水泵机组的安全运行。因此,有必要在进水流道内采取防涡消涡措施。本文分别在进水流道内加设曲线型导流锥和消涡栅,并对两者泵装置性能进行对比。在工程实际当中,双向流道的下层进水流道常用作自流引水通道,故本研究在喇叭管正下方增加具有一定高度、顺水流方向平行排列、引水阻力较小的一组消涡栅。

双向流道潜水泵系统三维模型由进水流道、出水流道、叶轮、导叶等几部分组成,计算区域为泵装置的全部,如图1~5所示。泵直径为300 mm。为便于潜水泵电动机安装,采用独特的大扩散角(单边扩散角度为18°)的扩散导叶及曲线扩散出水室。导叶扩散角为常规导叶扩散角的2倍左右,扩散导叶的叶片数为7片。共3个计算方案:方案1(进水无导流措施)、方案2(进水设导流锥)和方案3(进水设消涡栅)。

1.3 计算网格

对泵装置的进水流道和出水流道采用结构化网格,对结构形状比较复杂的进水室、出水室和叶轮导叶同样采用结构化网格,并适当进行加密。如图6所示。

泵叶轮和扩散导叶的几何模型建立和网格剖分均运用TurboGrid完成,如图6b所示。扩散导叶叶片数为7片和5片,导叶的模型生成后选用ATM Optimized方法划分网格,生成网格后通过调整网格

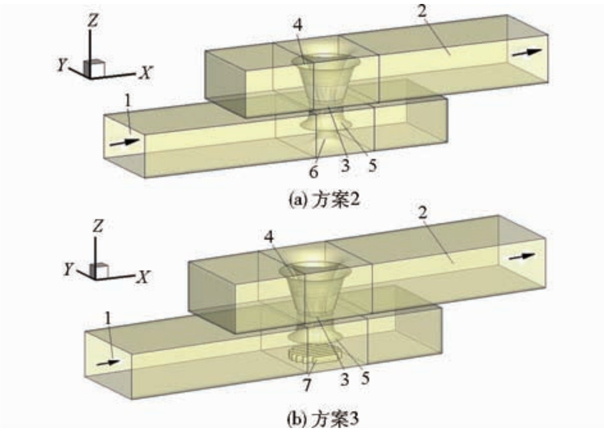

图 1　双向流道潜水轴流泵系统

Fig. 1　Two-way flow submersible axial flow pump system

1. 进水流道　2. 出水流道　3. 泵叶轮和导叶　4. 曲线扩散出水室　5. 喇叭管　6. 导流锥　7. 导流消涡栅

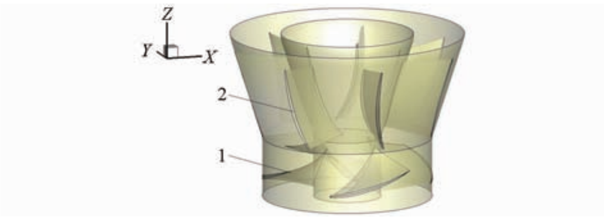

图 2　泵叶轮和导叶(大扩散角)

Fig. 2　Pump impeller and guide blade

1. 叶轮　2. 导叶

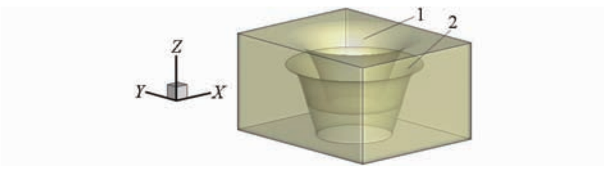

图 3　曲线扩散出水室

Fig. 3　Curved diffusion chamber

1. 出水喇叭管　2. 出水室内壁

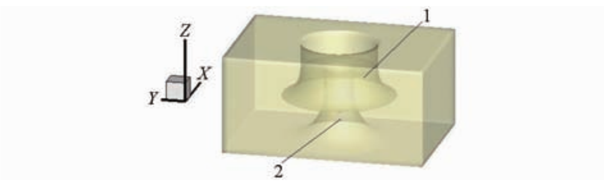

图 4　进水导流锥细部结构

Fig. 4　Detail structure of diversion pier

1. 进水喇叭管　2. 导流锥

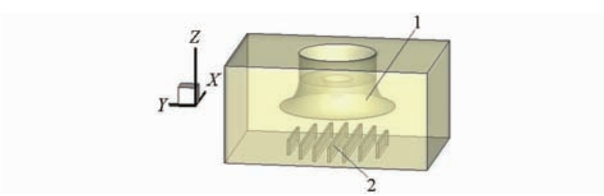

图 5　进水消涡栅细部结构

Fig. 5　Detail structure of anti-vortex grid

1. 进水喇叭管　2. 导流消涡栅

控制点的方法调整网格的质量,以达到计算要求

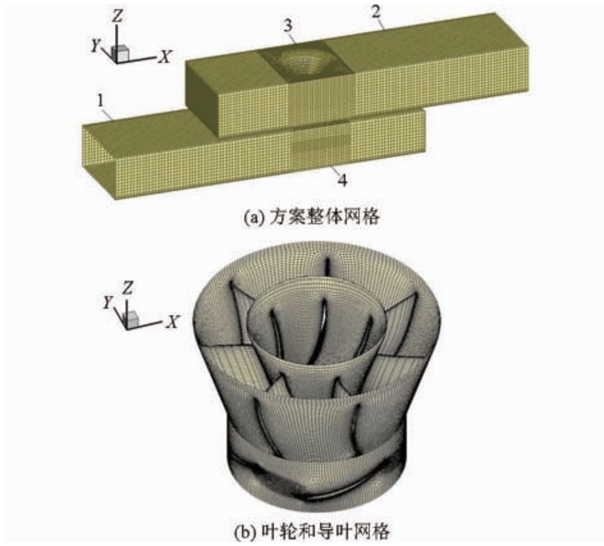

图 6　泵系统整体及主要部件计算网格

Fig. 6　Computation grid of pump system and its main components

1. 进水流道　2. 出水流道　3. 扩散出水室　4. 进水室

(角度在 15°~165°区间内)。

双向流道立式潜水轴流泵装置的网格质量满足数值模拟计算求解的精度要求,计算网格结点总数为 1 209 384 ~ 1 222 508,单元总数为 1 104 692 ~ 1 119 646,计算收敛精度满足要求。

1.4　计算参数和边界条件

泵装置坐标模型数值计算原点为叶轮中心线与出水流道底面交点,Z 轴正向为叶轮中心线指向出水室方向,X 轴正向为进出口水流方向。选用标准 $k-\varepsilon$ 紊流模型对 $0.45Q_0 \sim 1.26Q_0$(Q_0为设计流量)流量范围内泵装置内水流流态进行数值模拟。计算转速为 1 250 r/min。

泵装置进口边界设置在进水流道的进口处,水流方向为垂直于进口断面方向,采用质量流量进口条件。

出口边界设置在出水流道出口处,水流方向为垂直于出口面方向,采用压力出口条件,静压出口,相对压力为 1 个大气压,水的密度为 1 000 kg/m^3。

泵装置的壁面边界选择无滑移条件。

2　泵装置流动特性数值模拟结果

2.1　泵装置内部流动特性

2.1.1　整体流场

通过数值模拟计算,获得了进水流道不同措施下的设计工况(Q_0)双向流道立式潜水轴流泵装置内的全流场,泵装置内部流线如图 7 所示。由图中可以明显看到,泵装置流道内主流区的水流较为平顺,流动形态与单向进出水流道内的流动形态相近[19]。但在进、出水流道的盲端均出现较大的回流

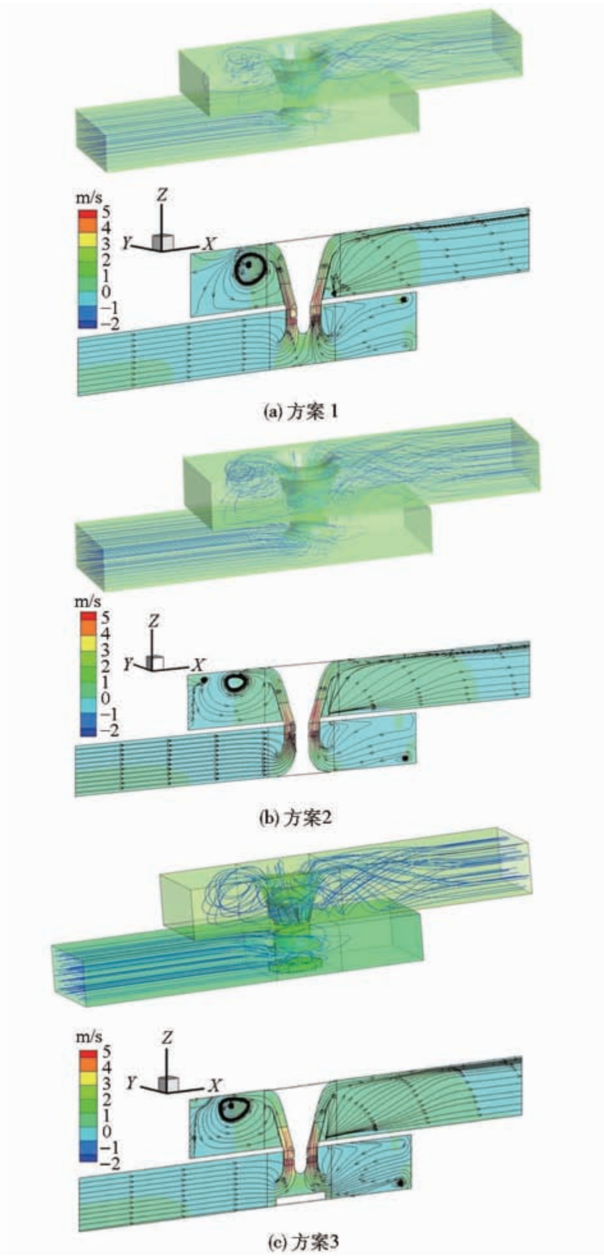

图 7　不同方案设计工况三维流线和纵断面流线图

Fig. 7　Three dimensional streamlines of pump system and longitudinal profiles under design conditions for different schemes

区。进水流道盲端相对较长，水流流速低，回流强度很小，对泵进口断面的水流及水力损失的影响微小；而出水流道的盲端较短，且导叶扩散角度较大，使得出水流道的盲端回流强度明显增大，水流紊乱，导致盲端的水力损失加大。

从出水室内的流动情况来看，流速分布较对称，未发生脱流和旋涡。同时，在出水室的出口处，水流速度已经降低到 2 m/s 以下，有效地回收了水流动能，控制了出水室的水力损失。

方案 2 和方案 3 的进水流道分别加设了导流锥和消涡栅，有效地防止了泵进口下方的旋涡。方案 1 无导流措施，流道底部则容易产生旋涡。3 种方案进水流道主流都比较平顺，方案 2 主流在进水流道出口（泵叶轮进口）断面的分布对称性好，更均匀。

在出水流道的出口段，流动形态比较复杂，水流流线有较大的摆动。这是由于叶轮出口水流较大的剩余环量，再叠加水流方向改变 90°所导致的二次回流的结果[20]。水泵后导叶出口水流的剩余环量使得出水流道内的水流流线呈现螺旋状，并向出水流道顶壁偏移。这种螺旋流状况如能得到改善，无疑会加大水流动能回收，提高双向流道泵装置的效率[21-23]。

2.1.2　进水流道出口断面轴向流速分布

进水流道出口与泵叶轮进口衔接，通常水泵叶轮对进水流道出口轴向速度分布有周期性影响。为便于分析，在水泵进口断面下方 0.16 倍叶轮直径的位置，选取喇叭管内的水平圆环截面，作为进水流道出口断面。3 个工况下不同方案的断面轴向速度云图如图 8 所示。

不难看出，在 3 种工况下进水流道出口断面轴向速度的分布情况，设导流锥的方案 2 流速分布均匀度明显优于无导流措施的方案 1 和设消涡栅的方案 3。进水流道出口断面速度分布规律是来水侧速度明显高于另外一侧，轴向速度分布受水泵叶轮旋转水流影响较小。根据数值模拟的结果对进水流道出口的轴向流速均匀度进行计算，结果如图 9 所示。在设计工况下，方案 1 进水流道出口断面轴向速度分布均匀度为 92.0%，速度加权平均角为 78.3°；方案 2 进水流道出口断面轴向速度分布均匀度为 96.2%，速度加权平均角为 79.7°；方案 3 进水流道

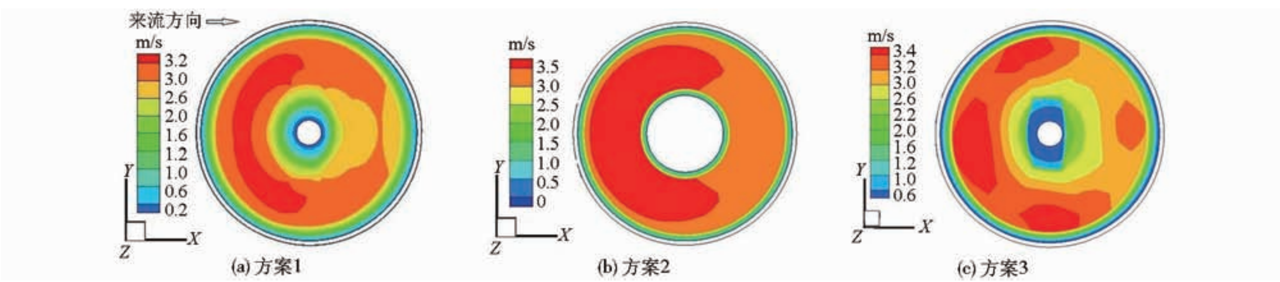

图 8　设计工况进水流道出口断面轴向速度云图

Fig. 8　Axial velocity contours under design conditions

出口断面轴向速度分布均匀度为 88.2%，速度加权平均角为 75.4°。设导流锥的进水流道出口断面的轴向速度分布均匀度和速度加权平均角明显优于无导流和设消涡栅的方案。导流锥的设置消除了进水流道底部流动奇点，改善了流动边界，对水流起引导作用，在不同流量下均保持较高的均匀度。图 9 中分别列出了 3 种方案不同工况的轴向速度分布均匀度和速度加权平均角随泵装置流量变化的关系曲线。

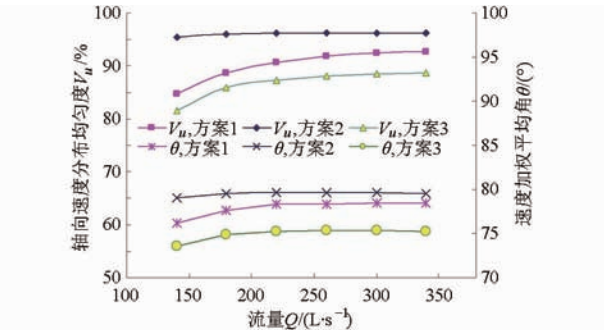

图 9　进水流道出口断面流速分布特性

Fig. 9　Hydraulic characteristics of suction passage

2.1.3　进出水流道流动特性

为了能够更清楚地了解进水流道内部流动特性，选取设计工况 Q_0 时，截取靠近进水流道底部 z 方向 -0.45 m 的水平截面（简称 $z-0.45$）作为特征断面进行分析，图 10 为进水流道水平断面流线图。

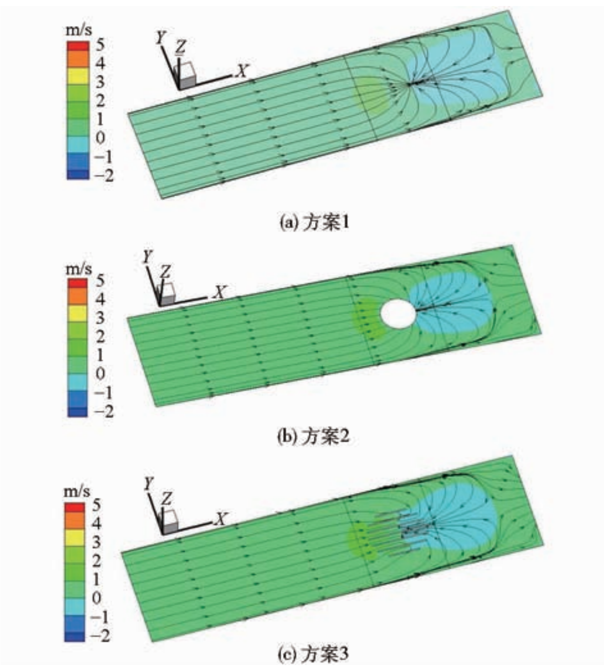

图 10　进水流道水平断面流线（$z-0.45$）

Fig. 10　Streamlines of inlet passage at horizontal section（$z-0.45$）

由图 10 可知，设导流锥时，水流从进水喇叭管四周环向汇入，压力分布不均，来水侧有明显的局部低压区，说明流速分布也不均匀；无导流措施时进水流态和流速分布情况的规律性基本相同，只是在叶轮的下方水流与弯管进水不同，有奇点存在，这部分水流的旋转能量不断积累，一旦达到临界数值就会产生涡带[24]。涡带是中心区域为气体的旋涡，旋转强度较大，旋涡进入水泵叶轮后引起水泵机组的剧烈振动，对泵站运行有很大的危害；设消涡栅时进水流态和流速分布情况的规律性也基本相同，但是消涡栅能够消除奇点，达到防止涡带产生的目的，来水侧局部低压区有所改善，流速分布比上面 2 种情况均匀。

出水流道出水喇叭口处 z 方向 0.36 m 的水平截面（简称 $z+0.36$）的流速分布如图 11 所示。可见水流从出水室流出后在流道后部有较大的紊乱混掺，这正是此类出水流道水力损失的主要部分。通过降低出水室出口流速可以在一定程度上控制出水流道后部的紊动强度，从而减少水力损失，提高管路效率。从出水室到流道出口的水流仍然受剩余环量影响，呈有序扩散，总体较为平顺。

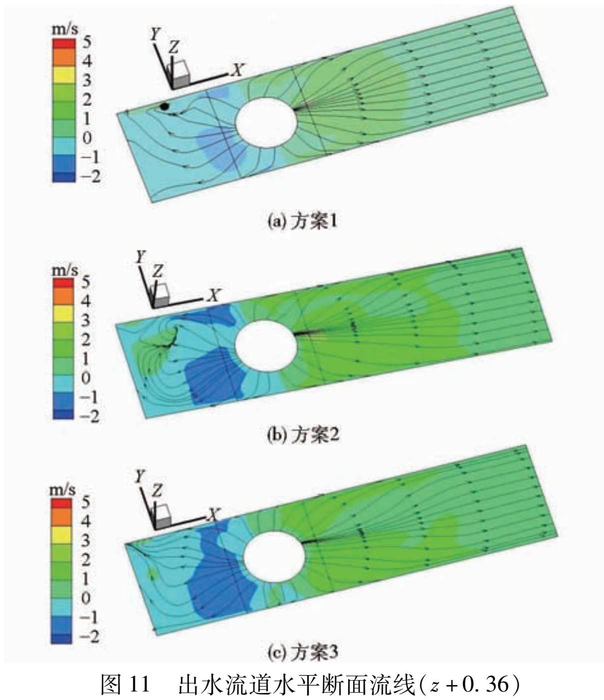

图 11　出水流道水平断面流线（$z+0.36$）

Fig. 11　Streamlines of outlet passage at horizontal section（$z+0.36$）

2.2　泵装置性能预测

根据泵装置数值模拟的数据计算，预测了双向流道立式潜水轴流泵装置性能，其外特性曲线如图 12 所示。从图中可知，进水流道采用不同消涡措施时，泵装置的扬程-流量曲线、效率-流量曲线的基本趋势未发生明显的改变。无导流措施的方案 1 和设消涡栅的方案 3 的扬程、效率曲线基本重合；方案 2 的扬程在相同流量下略高于其他 2 种方案；在小于最优工况流量时，方案 2 的效率略低于其他 2 种

方案,大于最优工况流量时则相反;方案 2 的最优工况点流量为 270 L/s,最高效率为 71.25%;方案 1 的最优工况点流量为 270 L/s,最高效率为 72.38%;方案 3 的最优工况点流量为 270 L/s,最高效率为 71.53%。总地来说,3 种方案的装置效率相差不大。设置导流消涡措施的主要目的是消除或防止进水喇叭管下方的旋涡,以免产生水力振动。工程实践表明,这种水力振动对水泵机组稳定运行危害极大。泵装置性能预测的结果显示加设防涡消涡措施对泵装置的性能并没有明显的影响。

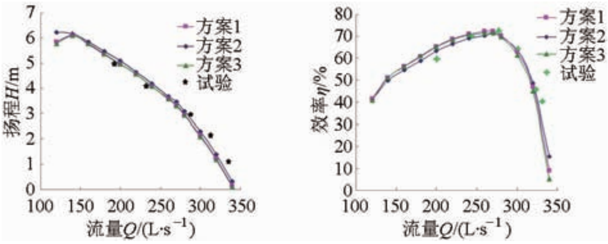

图 12 不同进水措施泵装置性能预测曲线

Fig. 12 Predicted pump performance curves of system with different measures

为了分析增设导流措施对进水流道水力损失的影响,对不同导流措施的进水流道水力损失进行了计算。图 13 为 3 种进水措施进水流道的水力损失曲线。

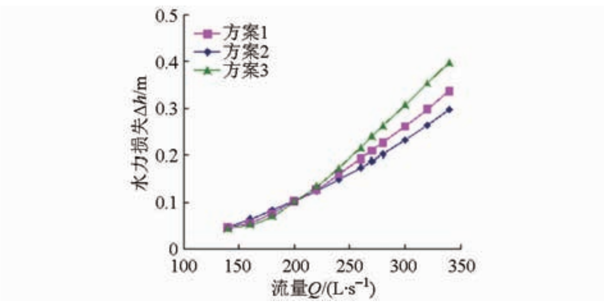

图 13 不同进水流道水力损失

Fig. 13 Hydraulic losses of different suction passages

从图中可以看出,在小流量工况下方案 2 进水流道的水力损失大于方案 1,方案 1 大于方案 3,这也印证了图 12 中小流量工况时方案 2 的效率低于方案 1,说明流量较小时,因增设导流锥、消涡栅等措施而增加的进水流道内水流阻力损失比无措施产生旋涡损失的能量还要多一些。在大流量工况时,情况则发生改变,无导流措施时旋涡产生的水力损失比增设导流锥产生的水力损失大,但是比加设消涡栅产生的水力损失小,即:方案 2 水力损失小于方案 1,方案 1 的水力损失小于方案 3。说明此时消涡栅对水流的阻力较大。这也许与消涡栅头部形状有一定关系,需要对消涡栅的结构及形状进一步优化改进。

3 泵装置模型试验

3.1 模型水泵与泵装置

为了进一步获得泵装置的综合性能并验证数值模拟结果,制作了相应的模型泵装置,在高精度水力机械试验台进行了模型试验。采用 ZM4 - K 型模型泵,由 ZM4 型叶轮和 DYK 型导叶组成。叶轮直径 $D = 300$ mm。轮毂比为 0.4,叶片数为 4,用黄铜材料经数控加工成型。DYK 型导叶叶片数为 7,用钢质材料焊接成型。进出水流道采用钢板焊接制作,进出水流道中部均开有观察窗,便于观测流道内的水流形态,模型泵装置如图 14 所示。试验转速为 1 250 r/min。

图 14 双向流道潜水轴流泵模型装置

Fig. 14 Test model of pump system

试验按照 GB/T 18149—2000《离心泵、混流泵和轴流泵水力性能试验规范(精密级)》和 SL140—2006《水泵模型及装置模型验收试验规程》标准执行。

3.2 模型试验结果

对模型泵装置分别测试了 4 个叶片安放角度(-4°、-2°、0°、2°)的能量性能和汽蚀性能。根据试验结果整理得到立式双向潜水泵模型装置综合特性曲线,如图 15 所示。

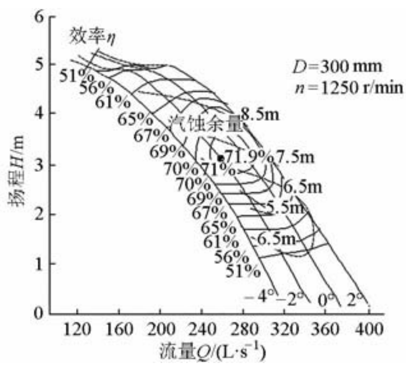

图 15 模型泵装置综合性能曲线

Fig. 15 Comprehensive performance curve of model pump

双向流道立轴潜水泵装置在不同叶片角度(-4°、-2°、0°、2°)的最高装置效率(BEP)分别达到 70.94%、71.90%、71.54%、69.25%;其中最大

值出现在水泵叶片角 －2°，此时泵装置扬程为 3.113 m，流量为 256.6 L/s，泵装置效率达到 71.90%，分别高于可逆式双向潜水泵装置正反向 BEP 工况 7 和 13 个百分点。泵装置的最大运行扬程高于 5 m，最大扬程/设计扬程值达到 1.5 以上，能够满足工程的实际需要。在对进水流道加设导流锥等措施后，通过观察未发现有害的进水旋涡。

汽蚀试验结果表明该泵装置性能在 4 个叶片角度下高效率点泵装置效率汽蚀余量为 6.5 m 以下，汽蚀比转数大于 1 100，满足水泵运行要求。

模型试验数据验证了数值计算预测的泵装置性能，在最高效率点附近的高效区二者吻合。

4　结论

（1）双向流道立轴潜水泵装置加设导流锥的进水流道，在最优工况下，出口断面轴向速度分布均匀度为 96.2%，速度加权平均角为 79.7°。导流锥能够有效地消除进水流道内的旋涡，保持良好的进水条件。

（2）泵装置性能试验结果表明：采用单边扩散角度 18°的扩散导叶，双向流道立轴潜水泵装置在叶片角 －2°下，泵装置扬程为 3.113 m，流量为 256.6 L/s，泵装置效率达到 71.90%；最大运行扬程均超过 5 m。这与双向流道常规轴流泵装置性能相近，说明此扩散导叶用于双向流道潜水泵装置可行，具有较好的工程应用价值。

（3）比较模型泵装置性能试验结果和数值模拟性能预测结果，在高效区范围内二者符合性较好，说明数值模拟计算结果是可信的。

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