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Optimization design for reducing the axial force of a vaned mixed-flow pump

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\textbf{ABSTRACT}

Vaned mixed-flow pump is widely used for liquid transportation in industry and agriculture. The structural-asymmetricity induces large axial force which causes axial vibration and shaft system damage. In this study, impeller forces are analyzed and optimized based on numerical simulation and experiment. Balance holes and a balance plate are used in cooperation. Orthogonal experiment is conducted to analyze the effects of balance hole diameter, sealing clearance width on the hub, and balance plate clearance width on axial force at design condition. Then, BP-artificial neural network is established with Global Dynamic Criterion algorithm. Efficiency and axial force at design condition are chosen as optimization objectives. The results prove that optimized pump has lower axial force and higher efficiency at design condition. The head coefficient meets design requirement. Accordingly, a cooperative method named the ‘Combining Hole-Plate Pressure Balancing Method’, made by adding balance holes and installing a balance plate in the hub clearance chamber, is proposed. The method can effectively reduce axial force and improve the adverse effect of axial force on the pump without obvious influence on hydraulic performance. This study is significant as a means to reduce axial force, lessen damage to a shaft system, and improve operation stability.

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\textbf{KEYWORDS}

Vaned mixed-flow pump; axial force; pressure balance; orthogonal experiment; artificial neural network

\textbf{Nomenclature}

| Symbol | Definition |
|--------|------------|
| $B_d$  | bottom width of the balance plate |
| $b_i$  | impeller blade outlet width |
| $B_p$  | width of the balance plate clearance |
| $B_{ref}$ | reference clearance width |
| $B_s$  | width of sealing clearance on hub |
| $B_t$  | top width of the balance plate |
| $C_p$  | pressure coefficient |
| $C_{\omega d}$ | dimensionless design flow rate coefficient |
| $C_{\psi}$ | head coefficient |
| $C_{\psi d}$ | dimensionless head coefficient |
| $D_{gh1}$ | guide vane blade inlet diameter at hub |
| $D_{gh2}$ | guide vane blade outlet diameter at hub |
| $D_{gmv}$ | guide vane blade maximum diameter |
| $D_{gs1}$ | guide vane blade inlet diameter at shroud |
| $D_{gs2}$ | guide vane blade outlet diameter at shroud |
| $D_h$  | diameter of the balance hole |
| $D_h^*$ | diameter of the balance hole center |
| $D_{ih1}$ | impeller blade inlet diameter at hub |
| $D_{ih2}$ | impeller blade outlet diameter at hub |
| $D_{is1}$ | impeller blade inlet diameter at shroud |
| $D_{is2}$ | impeller blade outlet diameter at shroud |
| $D_{out}$ | guide vane outlet diameter |
| $D_{ref}$ | reference diameter |
| $F_a$  | axial force |
| $\bar{F}_{ANN}$ | average value of the artificial neural network |
| $\bar{F}_{in}$ | average value of numerical simulation |
| $g$    | acceleration of gravity |
| $H$    | head |
| $H_d$  | design head |
| $i$    | factor |
| $i_{ij}$ | experimental index value of factor $i$ and level $j$ |
| $j$    | level |
| $k_{ij}$ | average value of the sum of experimental index |
| $M$    | torque |
| $n_d$  | impeller rotation speed |
| $n_{ds}$ | design rotation speed $[r/min]$ |
| $n_s$  | specific speed |
| $p_{in}$ | pressure at the inlet |
| $p_{out}$ | pressure of domain outlet |
| $Q$    | flow rate |
| $Q_d$  | design flow rate |
| $R_{err}$ | relative error |
| $R_g$  | guide vane hub cone radius |
| $R_i$  | range of factor $i$ |
| $u_s$  | uncertainty |

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1. Introduction

Vaned mixed-flow pumps are extensively used in agriculture and industry. The fluid in the pump flows into the impeller along the axis, and then into the guide vane in the oblique direction (Bing et al., 2012; Y. Liu et al., 2013). The flow has strong asymmetry in the axial direction (Bing et al., 2013). In the impeller, the fluid changes the flow direction, and also exerts a reaction force on the impeller. The hub and shroud of the impeller are surrounded by water, which forms a certain pressure. But the stressed area is different, which makes the pressure on hub and shroud unequal. The axial force acting on the impeller points to the inlet (Du et al., 2016; Zhou et al., 2013). Excessive axial force will affect the safe operation of the impeller innersurface against the total axial force was more than 0.8 (Shi et al., 2009). Wang et al. simulated the flow field of three kinds of multi-stage pumps with different clearances. The axial force of the first impeller stage was greater than the second impeller stage, and pressure on the shroud surface of the impeller changed with the clearance, which led to the change of axial force (Wang et al., 2013). Gantar et al. analyzed the influence of rotating fluid in the clearance chamber as well as the increase of wear ring radial leakage on axial force. Their results showed that the wear ring radial clearance had a significant effect on axial force, which cannot be ignored (Gantar et al., 2002). For other types of pumps, such as axial piston and gear pumps (Bergada et al., 2008; Y. Liu et al., 2019), the axial force problem has also been studied by many scholars. The above analysis shows that the axial force in the pump is an important factor affecting operation stability.

The most common way to reduce axial force is by the use of a balance hole or balance disc. Zhao et al. adopted balance holes to reduce axial force. The relation of a balance hole's radial position to axial force and hydraulic character was studied. They showed that the force along the axial direction in a centrifugal pump decreased when the balance hole radial position was decreased. The balance hole had more influence on axial force reduction, but at the same time, it led to decreasing head and efficiency (Zhao et al., 2011). Zhang et al. analyzed the influence of balance hole diameter on force along the axial direction in a partial-emission pump by numerically simulating seven kinds of pumps which have multiple sizes of balance hole. The results showed that the force along the axial direction decreased when the balance hole diameter increased in the operating range. The axial force was reduced to one third of the original after the balance hole was adopted. Balance hole size plays an important role in reducing the axial force (Zhang et al., 2014). Cao et al. studied the variation of the axial force of a low specific-speed centrifugal pump after changing its balance hole size. When the area ratio of a balance hole to a seal ring was about six, the improvement effect was better. At this ratio, the influence on head and efficiency was small (Cao et al., 2015). Zhao et al. deduced the mathematical relationships among the geometric parameters of the two clearances in the multistage pump, the clearance rate, and the balance force. Then the whole system, composed of balance disc and impeller, was simulated. The system achieved stability in a short time, and showed a periodic change rule. The dynamic balance characteristics of the balance disc were obtained (Zhao et al., 2008; Zhao et al., 2013).

At present, many researchers have applied a variety of optimization algorithms and strategies to the optimization design of various pieces of hydraulic machinery (Tao et al., 2014; Tao et al., 2018; Tao et al., 2019). Based on the artificial neural network and artificial bee colony algorithm, Derakhshan et al. optimized the geometric structure of a centrifugal pump impeller (Derakhshan et al., 2013). In order to verify the optimization results, CFD can be applied to predict the internal flow in a pump (Akbarian et al., 2018; Ghandari et al., 2019; Salih et al., 2019). By comparing the performance curve of the optimized pump with that of the original pump, it was found that the head was only slightly increased and the efficiency was effectively improved. The optimized pump had good hydraulic

\[
\begin{align*}
Z_g & \text{ guide vane number} \\
Z_i & \text{ impeller blade number} \\
\eta & \text{ design condition efficiency} \\
\eta_d & \text{ design condition efficiency} \\
\rho & \text{ water density} \\
\omega & \text{ impeller rotation angular speed}
\end{align*}
\]
performance. Zhu et al. established a multi-objective system for optimizing and designing, including three-dimensional inverse-design, CFD, experimental designing, responding-surface method, and multi-objective genetic-algorithm. The system was then used for designing a pump turbine runner with medium and high head. The test results showed that the optimized pump-turbine impeller has excellent energy performance (Zhu et al., 2015).

In summary, many researchers have recently used a balance hole or balance disc to reduce axial force. Research on design, analysis, and optimization has been carried out. How to combine various methods to reduce axial force is still relatively rare in the current research. Therefore, the method of combining a balance hole and a balance plate is innovatively used in this study to reduce axial force. The difficulty of the research method is using the comprehensive optimization strategy to explore the optimization scheme of the method. This study is applied to a vaned mixed-flow pump for improving the adverse effect of axial force. This research will be important for reducing axial force as well as damage to a pump shaft system. At the same time, it can also reduce vibration and noise due to excessive axial force, which is of great significance for improving the operational stability of the pump.

2. Research object and parameters

2.1. Pump object

The research object, which is a vaned mixed-flow type pump unit, includes a mixed-flow type impeller and a guide-vane with a narrow passage. During operation, the impeller rotates and centrifugal force pushes water flow from the impeller inlet to the guide vane. The flow direction changes from the axial direction to the direction between axial and radial. Then, the guide vane, which is stationary, guides the flow back to the axial direction. Figure 1(a) shows the impeller and guide-vane structures. Geometric dimensions of each part are shown in Figure 1(b) with the indication of flow direction. The diameter at the impeller inlet on the hub is defined as the reference diameter $D_{\text{ref}}$, $D_{\text{ref}} = 0.17$ m. Based on $D_{\text{ref}}$, the impeller’s and guide-vane’s geometric parameters will be nondimensionalized and listed in Table 1.

As shown in Figure 2, the red curves represent the impeller clearance. Among them, the clearance of the inlet sealing ring and hub sealing are shown in the local enlargement diagram. The sealing clearance width on the hub in the initial design is defined as the reference clearance width $B_{\text{ref}}$, which is 1 mm.

The parameters of performance design in this study are listed in Table 2. The performance parameters are also nondimensionalized. The flow rate coefficient and head coefficient ($C_\phi$ and $C_\psi$) are expressed as:

\[
C_\phi = \frac{Q}{\pi \omega \rho \left(D_{ih2} + D_{is2}/2\right)^3}
\]

\[
C_\psi = \frac{2gH}{\omega^2(D_{ih2} + D_{is2}/2)^2}
\]

Table 1. Dimensionless design parameters of impeller and guide vane

| Parameter  | Definition                          | Value/$D_{\text{ref}}$ |
|------------|------------------------------------|-------------------------|
| $D_{ih1}$  | Impeller blade inlet diameter on hub| 1.260                   |
| $D_{ih2}$  | Impeller blade outlet diameter on hub| 2.941                   |
| $D_{hs1}$  | Impeller blade inlet diameter on shroud| 2.353                   |
| $D_{hs2}$  | Impeller blade outlet diameter on shroud| 3.294                   |
| $D_{gh1}$  | Guide vane blade inlet diameter on hub| 3.174                   |
| $D_{gh2}$  | Guide vane blade outlet diameter on hub| 3.174                   |
| $D_{gh1}$  | Guide vane blade inlet diameter on shroud| 3.992                   |
| $D_{gh2}$  | Guide vane blade outlet diameter on shroud| 4.500                   |
| $D_{gm}$   | Guide vane blade maximum diameter | 4.150                   |
| $R_g$      | Guide vane hub cone radius         | 2.706                   |
| $b_i$      | Impeller blade outlet width         | 0.558                   |

Figure 1. (a). The model and geometry of the vaned mixed-flow pump: structural diagram. (b). The model and geometry of the vaned mixed-flow pump: meridional shape
where Q denotes flow rate \( [m^3 \cdot s^{-1}] \), \( \omega \) denotes the angular speed of impeller-rotation \( [s^{-1}] \), \( \rho \) denotes the density of fluid medium \( [kg \cdot m^{-3}] \), \( g \) denotes the acceleration of gravity \( [m \cdot s^{-2}] \), and \( H \) denotes the pump head \( [m] \). The specific speed \( n_s \) can be expressed as:

\[
ns = \frac{3.65n_d \sqrt{Q_d}}{H_d^{3/4}}
\]  

where \( n_d \) denotes the rotating speed \( [r \cdot min^{-1}] \), \( Q_d \) denotes the flow rate value under design condition \( [m^3 \cdot s^{-1}] \), and \( H_d \) denotes the pump head under design condition \( [m] \).

### 2.2. Mesh generation

CFD is used in this study with the computational domain including the inlet-extension part, impeller, and the clearance around the impeller, guide vane, and outlet-extension part. The discretization of the computational domain is based on both the structural and unstructured mesh elements. The mesh independence check is done to eliminate the influence of mesh size on the accuracy of the numerical simulation. As shown in Table 3, five different grid nodes from about 0.8 million to about 2.9 million are selected and calculated under the same conditions. According to the wall function requirement of the turbulence model, the \( y^+ \) is controlled in the range of 30–300. Based on the criterion that the residual of pump head between different schemes is less than 0.5%, the mesh independence check is carried out. The total mesh node number is about \( 1.76 \times 10^6 \). The mesh of each computational domain is listed in Figure 3.

### 2.3. Numerical simulation settings

The Shear Stress Transport (SST) model (Spalart, 2000, 2009; Menter et al., 2003) is used as the turbulence model. The reference frame model is the multiple reference frame (MRF) model. The impeller domain is rotating with a speed of 1470 r/min. The other domains are in a stationary frame without any rotation. The conditions on domain boundaries are set: (a) The velocity inlet boundary is given on inlet based on the flow rate condition. The pressure at the velocity inlet is zero gradient; (b) The pressure outlet boundary is given on the outlet with the static pressure value of 0 Pa. The velocity at the pressure outlet is also zero gradient; (c) All the walls are no-slip wall boundaries. Each two domains are connected by the interface. A frozen rotor is used as the interface model. The simulation converges after the RMS residual of the continuity equation and momentum equation is less than \( 1 \times 10^{-5} \).

### 3. Analysis of force characteristics in initial design

To analyze the force characteristics, axial and radial forces of the vaned mixed-flow pump taking clearance into account are calculated under various working conditions, as shown in Figure 4. Compared with axial force, the radial force is less than 1000 N in all conditions. The axial force is about 54500 N ~ 55500 N at the conditions of \( C_p = 0.056 \), \( C_p = 0.067 \), and \( C_p = 0.078 \). The axial force at the design condition is 26412 N. When the flow rate continues increasing to \( C_p = 0.133 \), the axial force decreases to 8830 N. Therefore, the axial force decreases

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**Table 2. Parameters for the performance design of pump.**

| Parameter | Definition | Value  |
|-----------|------------|--------|
| \( C_{\phi d} \) | Dimensionless design flow rate coefficient | 0.111 |
| \( C_{\psi d} \) | Dimensionless design head coefficient | 0.636 |
| \( \eta \) | Efficiency under design condition | \( \geq 85\% \) |
| \( n_d \) | Rotation speed under design condition \( [r/min] \) | 1470 |
| \( n_s \) | Specific speed | 266 |
| \( Z_i \) | Impeller blade number | 6 |
| \( Z_g \) | Guide vane number | 7 |

**Table 3. Details of the mesh independence check.**

| Mesh node \( \times 10^6 \) | 0.87 | 1.11 | 1.38 | 1.76 | 2.29 | 2.90 |
|-----------------------------|------|------|------|------|------|------|
| Residual [%]                | –    | 1.19 | 0.73 | 0.08 | 0.30 | 0.38 |
when the flow rate is rising. If the flow rate is less than the design condition, the axial force rises up especially quickly. This vaned mixed-flow pump is found to be rotational-periodic in a circumferential direction. When the impeller rotates, the forces on each part are almost symmetric. Therefore, the radial force will be low. Relatively speaking, the vaned mixed-flow pump has asymmetry along its axis. Greater axial force is produced when the impeller rotates at high speed.

In order to analyze the causes of the axial force, three conditions are selected to compare the internal pressure in the pump. Figure 5 provides the pressure coefficient distribution in the inlet section, impeller, guide vane, and clearance in the meridional view. The pressure coefficient $C_p$ is defined as follows:

$$C_p = \frac{p - P_{\text{in}}}{\rho g H} \quad (4)$$

where $P_{\text{in}}$ denotes the pressure value on the inlet boundary.

The pressure in the hub clearance is greatest in the partial load condition. The pressure difference between the hub clearance and the impeller inlet is also large. On the contrary, the pressure in the hub clearance is the lowest at over load condition. The pressure difference between the hub clearance and the impeller inlet is also lower. The pressure in the clearance increases with the increasing of the impeller outlet pressure. Thus, the pressure in the clearance strongly exceeds the impeller inlet. The impeller has a large axial force from the clearance to the inlet direction. It may cause adverse effects such as impeller axial movement and stationary component damage. In summary, in order to ensure stability in operation, it is necessary to reduce the excessive axial force in a vaned mixed-flow pump. Therefore, an optimization scheme for axial force reduction will be further discussed.

4. Parameterization of the optimization scheme

Based on the previous analysis of axial force, it can be concluded that pressure increasing in the hub clearance leads to the axial force increasing. Therefore, not only are the balance holes opened on the hub, but a balance plate is also added in the hub clearance to reduce the axial force. The combination of balance holes and a balance plate is used to reduce the large axial force. Figure 6 shows the schematic diagram of the position of balance holes and a balance plate. First, several variable parameters are defined. They are the dimensions of the balance hole, the sealing clearance on the hub, and balance plate clearance. Among them, the number of balance holes is the same as the number of blades, which is six. A balance hole is located in the middle of two blades. The distance from the balance hole’s center to the impeller’s center remains unchanged. The diameter of the balance hole’s center can be denoted as $D_h^*$. The top width $B_t$ and bottom width $B_d$ of the balance plate keeps constant. Taking $B_{\text{ref}}$ as the reference size, Table 4 lists the three constants’ relative value.

In order to analyze the cooperation between balance hole and balance plate, the diameter of balance hole $D_h$, the width of the sealing clearance on hub $B_{\text{sc}}$, and the width of balance plate clearance $B_p$ are selected as variables. The influence of the cooperation between balance
Figure 5. Pressure distribution at different conditions. Note: (a) $C_\phi = 0.078$ (b) $C_\phi = 0.111$ (c) $C_\phi = 0.133$

Figure 6. Schematic diagram of the position of balance holes and balance plate.

Table 4. Balance hole and balance plate size constants.

| Parameter | Definition                              | Value/Ref |
|-----------|-----------------------------------------|-----------|
| $D_h^*$   | Diameter of the balance hole center     | 97.3      |
| $B_t$     | Top width of the balance plate          | 5.0       |
| $B_d$     | Bottom width of the balance plate       | 57.5      |

Table 5. Definitions and ranges of the parameters.

| Parameter | Definition                              | Value/Ref |
|-----------|-----------------------------------------|-----------|
| $D_h$     | Diameter of the balance hole            | 2.5~12.5  |
| $B_s$     | Width of sealing clearance on hub       | 1~5       |
| $B_p$     | Width of the balance plate clearance    | 1~5       |

Therefore, considering the balance between computational resources and experimental accuracy, an orthogonal experiment is used in this study. A small number of representative experiments instead of comprehensive experiments can reduce the number of experiments and achieve the same experimental accuracy (Taguchi & Phadke, 1984; Xiao et al., 2014). For analyzing the effects of the three factors on axial force, five different levels were selected to design the orthogonal experiment in respective ranges. The different levels of each factor are as follows:

5. Orthogonal experiment

5.1. Orthogonal experiment design

A comprehensive experiment comparing the schemes would consume a lot of computing time and resources.
(1) $D_h$: $L_{11}$ is 2.5, $L_{12}$ is 5.0, $L_{13}$ is 7.5, $L_{14}$ is 10.0, $L_{15}$ is 12.5;
(2) $B_s$: $L_{21}$ is 1.0, $L_{22}$ is 2.0, $L_{23}$ is 3.0, $L_{24}$ is 4.0, $L_{25}$ is 5.0;
(3) $B_p$: $L_{31}$ is 1.0, $L_{32}$ is 2.0, $L_{33}$ is 3.0, $L_{34}$ is 4.0, $L_{35}$ is 5.0.

Table 6 shows the 25 orthogonal experiments with three factors and five levels. The axial force $F_a$ of a vaned mixed-flow pump under the design flow rate condition is selected as the experimental index. The influence of three factors on the axial force is analyzed in different schemes. Based on Table 6, the computational domain is established corresponding to each parameter combination. The original domain is modified according to the impeller clearance section shown in Figure 7. The balance hole regions and the balance plate structure are added. The $D_h$, $B_s$, and $B_p$ are adjusted according to the different orthogonal experiment parameters of each group. The numerical simulation software, mesh generation software, mesh topology and scale, turbulence model, boundary conditions, convergence criteria, and other settings are the same as those of previous simulations. The total number of mesh nodes of 25 orthogonal experiments are between $1.65 \times 10^6$ and $1.78 \times 10^6$.

### 5.2. Analysis of orthogonal experiment results

As shown in Table 7, the axial forces of each experiment at design conditions are calculated. Range analysis is carried out according to the orthogonal experiment results. The average value of the sum of the experimental index is defined as $k_{ij}$:

$$k_{ij} = \sum_{j=1}^{j_{\text{max}}} \frac{I_{ij}}{j_{\text{max}}}$$

where $I_{ij}$ is the experimental index value of factor $i$ and level $j$. Because the orthogonal experiment has three factors and five levels, so $j_{\text{max}} = 5$. $R_i$ is the range of factor $i$. It is an important index for judging the primary–secondary relationship of the influence of factors. The definition is as follows:

$$R_i = \max(k_{ij}) - \min(k_{ij})$$

The results of the orthogonal experiment are calculated and the effects of $D_h$, $B_s$, and $B_p$ on $F_a$ are analyzed. Figure 7 shows the comparison of $k_{ij}$ and $R_i$ under different factors and levels at design condition. As can be seen from the figure, $F_a$ decreases with the increase of $D_h$. When $L_{11} = 2.5$, $F_a$ is the greatest. When $L_{14} = 10$, $F_a$ is the lowest. When $D_h$ continues increasing, $F_a$ increases slightly, but the range is very small. $F_a$ increases evenly with the increasing of $B_s$. When $L_{21} = 1$, $F_a$ is the lowest. When $L_{25} = 5$, $F_a$ is the greatest. $F_a$ drops and rises with the increasing of $B_p$. When $L_{33} = 3$, $F_a$ is the lowest. When $L_{35} = 5$, $F_a$ is the greatest. From the range histogram, we can see that $D_h$ is the most important factor affecting $F_a$, and its range is much higher than that of the other two factors. The second is $B_s$; the range is about 21% of $D_h$. $B_p$ is the lowest factor affecting $F_a$, and its range is only 7% of $D_h$.

![Figure 7](image-url)
6. Optimization process

6.1. Establishment and training of a BP artificial neural network

In order to select the best combination of parameters to best reduce axial force, a BP artificial neural network is established based on the orthogonal experiment, taking the experiment parameters in Table 6 as input data and the experiment indexes in Table 7 as output data, so as to improve the prediction speed in the optimization process. Seventy-five per cent of the 25 samples are selected as training samples, 15% as validation samples, and 10% as test samples. During training, the hidden layer number can determine the performance of the artificial neural network. In this study, the number of hidden layers is chosen as 10. The Bayesian Regularization algorithm is selected as the training algorithm for the BP artificial neural network. In the process of optimization, the same settings are used for the improved training of the artificial neural network. Through continuous learning and training, a BP artificial neural network is established which can be used to reduce the axial force, ensure high efficiency, and meet the head coefficient requirement.

6.2. BP-GDC comprehensive optimization strategy

The Global Dynamic-Criterion (GDC) algorithm (Zhu et al., 2018) is applied to the optimization of axial force. Combining GDC with a BP artificial neural network, a comprehensive optimization strategy of BP-GDC is established. In the optimization process, the GDC algorithm is used to search the optimized samples based on the result-predictors of a numerical simulation and an artificial neural network. The results of numerical simulation are used for continuously training the artificial neural network. The artificial neural network is used instead of numerical simulation when the results’ difference between it and numerical simulation is less than 1%, improving optimization speed by guaranteeing prediction accuracy. Aiming at the three factors of $D_h$, $B_s$, and $B_p$, the optimum design sample can be obtained by taking $F_a$ and $\eta_d$ as the optimization objective at design condition.

In the optimization process, the artificial neural network is continuously trained and the difference between an artificial neural network and numerical simulation is gradually reduced. The relative error between the average value of the artificial neural network $F_{in}^{ANN}$ and the average value of numerical simulation $F_{in}$ obtained by 10 sets of parameters in each round is recorded as $R_{err}$:

$$R_{err} = \frac{|F_{in}^{ANN} - F_{in}|}{F_{in}}$$  (7)

After 11 iterations, $R_{err}$ has been reduced to less than 1%. The artificial neural network becomes accurate enough to replace the numerical simulation as the axial force predictor. Therefore, the artificial neural network is used in the subsequent prediction. Because the efficiency changes in a small range, the results of an artificial neural network are similar to those of numerical simulation in an initial prediction. The artificial neural network can successfully improve optimization speed and guarantee prediction accuracy.

6.3. Optimized scheme selection

Based on the BP-GDC comprehensive optimization strategy, the optimization scheme for axial force is obtained step by step. Samples meeting the design head coefficient requirement are selected to draw Figure 8. The objective of the optimization is to reduce the axial force and ensure the high efficiency of the pump. Therefore, both the axial force and the efficiency should be taken into account in the selection of optimized samples. For the final optimized sample, the axial force ranks 1.2% of all samples, and the efficiency ranks 0.3% of all samples. The parameters of the optimized samples are shown in Table 8. The optimized pump has a lower axial force and higher efficiency at design condition. The head coefficient of the design condition meets the design requirements. The objective of the optimization design of the axial force is achieved.

Fitting all the rules presented by the samples in Figure 9, the scatter distribution of these samples satisfies the fitting curve of a cubic polynomial as follows:

$$Y = a_0 + a_1 f + a_2 f^2 + a_3 f^3$$  (8)
where $a_0$, $a_1$, $a_2$, and $a_3$ are constants that $a_0$ is $9.074 \times 10^{-1}$, $a_1$ is $-5.432 \times 10^{-4}$, $a_2$ is $2.984 \times 10^{-8}$, and $a_3$ is $-5.264 \times 10^{-13}$.

7. Performance analysis after optimization

7.1. Force characteristics comparison

As shown in Figure 9, the changes in axial and radial forces before and after optimization are analyzed. The variation trend of axial force before and after optimization is approximately the same. The axial force decreases gradually with the increase in flow rate. The comparison shows that the axial force at all the conditions decreases significantly. The reduction of each condition is shown in Table 9. The minimum reduction of axial force is 27.28%, and the maximum reduction is 61.12% at design condition. The significant reduction of axial force proved that the optimization scheme is reasonable. Relatively speaking, the radial force of each condition after optimization is roughly the same as that before optimization, and does not change regularly with the increase in flow rate. In summary, the optimization scheme can effectively reduce the axial force without significantly affecting the radial force. The BP-GDC comprehensive optimization strategy can not only ensure optimization accuracy, but also shorten the optimization time and improve the optimization speed.

7.2. Hydraulic performance comparison

To analyze the effect of the optimization scheme on the pump, hydraulic performance before and after optimization is compared in Figure 10. For the $C_{\phi}$-$\eta$ curve, the variation trend of head coefficient is the same before and after the optimization of axial force. The difference in head coefficient is small, which indicates that the optimization scheme has little effect on the head. For the $C_{\psi}$-$\eta$ curve, the efficiency at each condition after optimization is basically consistent with before. After optimization, the maximum difference of $C_{\psi}$ is 2.9%. The maximum difference of $\eta$ is 1.6%. The results show that the optimization scheme has little effect on the efficiency while greatly improving the axial force. In summary, the optimization scheme can reduce the axial force, ensure higher efficiency, and help the head coefficient meet the design requirement at the same time.

7.3. Analysis of the axial force improvement mechanism

In order to clarify the improvement mechanism of the axial force, the internal flow analysis of a vaned mixed-flow pump is conducted. Firstly, the pressure distribution in the meridional view is compared in Figure 11. The balance hole connects the fluid at the impeller inlet with the fluid in the hub clearance, which reduces the pressure in the hub clearance chamber. The sealing clearance on the hub and the balance plate clearance make the high pressure reduce after passing through several clearances. The pressure in the hub clearance chamber decreases. By comparing the pressure distribution at different conditions, the high pressure in the hub clearance chamber decreases greatly after optimization at partial load condition. With the increase in the flow rate value, pressure in the hub clearance chamber increases substantially before the optimization, and increases slightly after the optimization. This is also consistent with the trend of axial force in Figure 10.

To analyze the flow pattern in clearance, Figure 12 shows the comparison of velocity vectors. It can be seen that in a partial load condition, the high-pressure fluid at the impeller outlet flows into the hub clearance chamber through the sealing clearance on the hub, which makes the water flow in the clearance chamber to form a high-pressure vortex flow. After optimization, the high-pressure fluid is connected with the low-pressure fluid at the impeller inlet via the balance hole. The fluid in the clearance chamber flows back to the impeller inlet via the balance hole. The balance plate gradually reduces the fluid pressure entering the clearance chamber. Both of them can reduce the axial force of a mixed-flow pump impeller. At the design flow rate condition and over load condition, an unstable flow still exists in the clearance chamber before the optimization. After optimization, the low-pressure fluid at the impeller inlet flows into the hub clearance chamber through the balance hole due to the
Figure 10. (a). Comparison of hydraulic performance curves: $C_\psi - C_\phi$. (b). Comparison of hydraulic performance curves: $C_\psi$-\(\eta\).

Figure 11. (a). Comparisons of pressure distribution at different conditions: initial. Note: (a) $C_\phi = 0.078$ (b) $C_\phi = 0.111$ (c) $C_\phi = 0.133$. (b). Comparisons of pressure distribution at different conditions: optimized. Note: (a) $C_\phi = 0.078$ (b) $C_\phi = 0.111$ (c) $C_\phi = 0.133$. 
increase in flow rate. The fluid flows out of the clearance chamber through the sealing clearance on the hub with the impeller rotating, which also reduces the pressure in the clearance chamber.

8. Experimental verification

The experiment is conducted using the hydraulic test rig for pumps. Figure 13 is the test rig schematic. Figure 14 shows the pump model installed on this test rig. Hydraulic performance is calculated by measuring pressure, flow rate, and shaft power. The pressure is measured by a pressure transducer and the accuracy is within ±0.05%. The flow rate is measured by an electromagnetic flow meter and the accuracy is within ±0.18%. The shaft power is measured by a power meter and the accuracy is within ±0.10%. The uncertainty of two or more measured data can be calculated by a synthetization equation (Bo & Chen, 2004). The high accuracy of the instrument ensures the accuracy of the experiment. The pump head can be calculated by:

\[ H = \frac{p_{out} - p_{in}}{\rho g} \tag{9} \]

where \( p_{out} \) is the outlet pressure and \( p_{in} \) is the inlet pressure. The pump efficiency can be calculated by:

\[ \eta = \frac{\rho gQH}{Mr_{iso}} \tag{10} \]

where \( Q \) is the inlet flow rate and \( M \) is the torque on shaft.

The head coefficient and efficiency after optimization are obtained as shown in Figure 15. The simulated and test values of \( C_\psi \cdot C_\psi \) curves and \( C_\psi \cdot \eta \) curves are compared after optimization. For the \( C_\psi \cdot C_\psi \) curve, the optimized simulation value has the same trend as the test value and the head coefficient is similar. At design condition, the head coefficient \( C_\psi = 0.643 \) obtained from the test meets the design requirement \( C_\psi = 0.636 \). For the \( C_\psi \cdot \eta \) curve, the simulated value of efficiency is close to the test value at partial load condition. The test value of efficiency at design condition is 85.8%. For optimization and test, the maximum difference of \( C_\psi \) is 3.0%. The maximum difference of \( \eta \) is 5.7%. Because of the mechanical loss, friction loss, and other losses, the test data are usually lower than the simulated results. The difference
Figure 15. (a) Comparison of simulated and experimental values after optimization: $C_\psi - C_\phi$. (b) Comparison of simulated and experimental values after optimization: $C_\psi - \eta$.

Figure 13. Schematic of the test rig.

Figure 14. The pump model test rig.

between simulated and experimental values in this study is within a reasonable range.

9. Summary of the axial force reduction method

In this study, balance holes and a balance plate are used to reduce axial force. The influence of three factors – $D_h$, $B_s$, and $B_p$ – on axial force is discussed by the orthogonal experiment. By establishing a BP-GDC comprehensive optimization strategy, the optimization process is completed, and the optimization scheme of axial force is obtained. While reducing the axial force of each condition, it also ensures the high efficiency of the pump and the head coefficient to meet the requirements at design condition.

Based on the studies above, a combined method of adding balance holes and installing a balance plate in a hub clearance chamber is proposed, and named the ‘Combining Hole-Plate Pressure Balancing Method’. The method aims to reduce larger axial forces by adopting balance holes and a balance plate. Specifically, the method has the following characteristics:

1. The balance holes are evenly distributed on the hub and located between the leading edges of the two blades. The number of balance holes is the same as that of the blades. The balance hole makes the fluid in the hub clearance chamber connect with that at the impeller inlet. The high-pressure vortex in the hub clearance chamber is improved, and the pressure is reduced.

2. Controlling the width of the sealing clearance on the hub can effectively prevent the high-pressure fluid at the impeller outlet from entering the hub clearance chamber. The pressure gradient distribution from the impeller outlet to the clearance can be
improved. The pressure of the fluid decreases after passing through the clearance. Finally, the axial force caused by high pressure inside the hub clearance chamber is reduced.

(3) By installing the balance plate and controlling the clearance, the high-pressure fluid at the impeller outlet can be blocked. The clearance width needs a reasonable value. Adjusting the high pressure at the impeller outlet and the low pressure in the balance hole can balance the pressure in the hub clearance chamber and reduce the axial force.

In summary, the ‘Combining Hole-Plate Pressure Balancing Method’ relies on the interaction of the balance holes and the balance plate. Between them, the balance holes play a significant role in pressure regulation, while the balance plate performs fine-tuning of pressure balance on both sides. While keeping the original hub sealing, the pressure inside the hub clearance chamber is uniform to the greatest extent. The axial force of a vaned mixed-flow pump is significantly reduced and hydraulic performance is not greatly affected. This method is simple to use and suitable for axial force reduction in similar pumps.

10. Conclusion

(1) Aiming at the force characteristics of a vaned mixed-flow pump, the computational domain with clearance is established and numerical simulation is carried out. By the analysis of force characteristics, it is found that a vaned mixed-flow pump has the characteristics of lower radial force and greater axial force.

(2) According to the characteristics of pressure distribution in a flow field, the cooperation of balance holes and a balance plate is proposed. The parameters include the diameter of the balance hole \(D_h\), Width of sealing clearance on hub \(B_s\), and Width of the balance plate clearance \(B_p\). The orthogonal experiment is carried out based on the three parameters within their respective ranges. The relationship among the three parameters and the axial force at design condition is obtained.

(3) The parameter combination scheme is optimized based on the results of the orthogonal experiment. The BP-GDC comprehensive optimization strategy is established. The optimization scheme is finally obtained after a series of optimizations. The force characteristics and hydraulic performance of the optimized pump are analyzed and verified. The results show that the optimized pump has both low axial force and high efficiency at design condition, and the head coefficient of the design condition meets the design requirement, thus achieving the objective of the optimal design of the axial force.

(4) Based on the above, a cooperative method of adding balance holes and installing a balance plate is proposed, named the ‘Combining Hole-Plate Pressure Balancing Method’. In this method, the balance holes play a significant role in regulating the pressure, while the balance plate performs the fine-tuning of pressure balance on both sides. The pressure in the hub clearance is uniform to the greatest extent. The axial force of the vaned mixed-flow pump is significantly reduced. The hydraulic performance is not significantly affected. This method provides a reference for axial force reduction in similar pumps.

The axial force of a vaned mixed-flow pump can be effectively reduced by the ‘Combining Hole-Plate Pressure Balancing Method’. In this study, only three important parameters are discussed. It is suggested that the influence of other variable parameters in the combination scheme on axial force reduction can be further discussed in further research.

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