Numerical calculation and optimal design of a hot water circulation pump

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Abstract. The problem of hot water circulation pump head shortage is common. In this paper, numerical simulation technology, combined with orthogonal experimental design, was used to research optimal designs to improve the hot water circulation pump’s head. CFX software was used to calculate the flow field in the pump, and the head-flow rate curve could be achieved. The accuracy of CFD was validated through comparison between numerical and experimental data. According to the experience, the number of impeller blades, thickness and width of impeller outlet were changed to improve the hot water circulation pump’s head. A three factors and level values of model pump orthogonal experiment was designed, and numerical simulation of whole flow field based on CFX was adopted to implement the orthogonal experiment. Finally, the best designed scheme for model pump was obtained. The analysis of results indicates that the head of hot water circulation pump has increased by 7.77% at rated conditions. The distribution in impellers’ internal flow field is symmetrical, and accords with the law of fluids flow in the common centrifugal pump.

1. Introduction

Head is a vital parameter for evaluating the performance of pump, and is also an important criterion for users to choose pump. The hydraulic performance of the impeller, which is a rotating part in the pump, has an influence on the overall performance of pump. At present, many scholars [1-5] had already researched on the geometrical parameters of the impeller how to impact on pump performance, but most of them were focused on the influence of a single parameter. He X J et al [6] took use of grey theory analysis of synthetic influence of multiple geometrical parameters of centrifugal pump on its head. Cong X Q [7] et al, using orthogonal experimental method, designed no-overload sewage pumps.

With the advanced development of the discipline of computer technology concurrent with the maturation of computational fluid dynamics, numerical simulation, theoretical analysis and experimental research are the most important methods to research interior flow of the pump. At present, numerical simulation becomes the most popular method to research the pump’s interior flow field, because it is lower costs and more convenient to observe interior flow field than experimental research [8-9].

In our daily lives, hot water circulation pump is widely used in the process of pressurized hot water. The problem of pump head shortage usually appears when it’s used. This will decrease the range of its application and the users must re-select it. So, it is vital to improve the pump head for solving above mentioned problems. So an orthogonal experiment [10], combined with numerical simulation, was carried out to do synthetic parameters’ optimization for improving the pump head in this thesis. Then
distribution of total pressure and relative velocity in impeller’s interior flow field are obtained by comparing between the impeller before and after optimization.

2. Geometric modelling and meshing

2.1. Geometric modeling representation
In this paper, for the numerical investigation of the flow field of hot water circulation pump, the geometry of the pump model: Lpm125 is used. The pump has one single axial suction and vane less volute casing. The pump has equipped with five blades and is driven by a three-phase AC electric motor, whose speed is 2850 r/min under the rated conditions. Flow rate of this kind of hot water circulation pump is 5.0m³/h under the rated conditions. Head of it is 3.5m under the rated conditions.

Its full flow field geometry model is built in this paper, because full flow field can more accurately forecast the performance of the pump. Three-dimensional modelling software, such as Pro/E, was used to build its geometrical models, including the inlet segment, impeller, chamber and volute. In order to effectively build geometrical models and generate meshes, the space between impeller front and back shroud and volute should be integrated with the impeller and volute radial clearance, which is shown in Fig1. It was called a chamber. To get a relatively stable inlet and outlet flow, three and five times of pipe diameter had been extended in the pump inlet and outlet section respectively. The hot water circulation pump full flow field geometrical model is showed in Fig2.

![Two-dimensional chamber.](image1)

![Full flow field geometry model.](image2)

2.2. Meshing
ICEM-CFD software was used to generate mixed grids for the all models. For better conformity of the geometry with the computational domain, at a chamber domain, the structured mesh was selected, and at the other regions, the unstructured mesh configuration was employed to correctly cover the complex geometry. The head was selected as the monitoring parameters to test the grid independence, which is shown in Table.1. Ultimately the number of grids, which used to calculation, was determined. The mesh nodes of full flow field are about 372,596, which are shown in Fig.3.

| Types     | Mesh nodes | Mesh elements | steps of convergence | Head/m | Deviation/% |
|-----------|------------|---------------|----------------------|--------|-------------|
| mesh-001  | 266225     | 657056        | 557                  | 4.11983| ---         |
| mesh-002  | 291165     | 801007        | 587                  | 4.12349| 0.0883862  |
| mesh-003  | 330319     | 1027656       | 648                  | 4.13591| 0.39030737 |
| mesh-004  | 338557     | 1073303       | 588                  | 4.11094| -0.2157856 |
| mesh-005  | 372596     | 1270532       | 566                  | 4.08235| -0.9097463 |
| mesh-006  | 411150     | 1494964       | 646                  | 4.13659| 0.4068129  |
| mesh-007  | 462341     | 1792924       | 1118                 | 4.12213| 0.05582755 |
| mesh-008  | 532885     | 2205130       | 1677                 | 4.16933| 1.20150589 |
3. Pump characteristic curves simulation

3.1. Boundary conditions and parameters settings
The internal flow of the centrifugal pump is three-dimensional, viscous, unsteady turbulent flows, and the law of its motion accords with three-dimensional Reynolds-averaged Navier-Stokes equations. The hot water circulation pump is a kind of pipeline centrifugal pump. At room temperature, the pump performance test was implemented, which water at 25°C was selected as medium. So the deviation of temperature between inside and outside of the pump is smaller. The internal thermal transfer can be ignored to guarantee similarity of numerical simulation. Therefore, the governing equations only contained mass conservation equation and momentum conservation equation.

ANSYS-CFX12.1 software was used to calculate and analyze the flow field in it. The inlet and outlet boundary conditions were set to mass flow rate inlet and static pressure outlet respectively. The pump performance curve would be obtained when the mass flow rate in the inlet was changed. Wall surface roughness within the volume of impeller was set to 25 \( \mu \text{m} \), and others’ was set to 50 \( \mu \text{m} \). The medium selected was water at 25°C. Considering the factors of industry standard, convergence property and accuracy during the simulation process [11], a standard \( k-\varepsilon \) turbulence model and a scalable wall function were selected. The convergence criterion was \( 10^{-5} \). The simulation type used in this paper was steady state simulation. Multiple reference frames was selected. The volute, inlet segment and outlet pipe were set in stationary frame. However, the impeller was set in rotary frame. The volute and impeller were related to each other through the “Frozen Rotor”.

3.2. Boundary conditions and parameters setting
According to the definitional equations of head, shaft power and hydraulic efficiency in centrifugal pump, the performance characters, including \((0.4~1.1)\) times the rated conditions, were calculated. The obtained numerical results were compared with the experimental ones, which are shown in Fig.4.

![Figure 3. Full flow field meshing.](image)

![Figure 4. Comparing simulated results with the experimental ones.](image)

The deviation between simulated results and experimental ones was achieved, which was shown in Fig.4. The maximal relative deviation of simulated head and hydraulic efficiency to experimental data were 3.1% and 10% respectively, when 0.6 to 0.9 times the rated conditions were calculated. The relative deviation of the head and hydraulic efficiency, which were calculated under the rated conditions, were 5.2% and 9.5% respectively. These over deviation may attribute to the neglecting of several volumes when the full flow field geometrical models were created. The real power of motor was not tested but achieved through the empirical value, which was set to 51% according to its power.

It may explain to the bigger deviation between simulated hydraulic efficiency and the experimental one. The selected turbulence models were not the same under each different operating condition. So the different relative deviations among many different operating conditions were generated. Overall, the fluctuating trend of head, shaft power and hydraulic efficiency, which obtained from the numerical simulated results, was conformed to the experimental data. It is reasonable to believe that CFD can be used to forecast the hot water circulation pump’s hydraulic performance.

4. Orthogonal experimental design
4.1. Purpose of orthogonal experimental design

- The relationships between the selected geometry parameters of impeller and head, shaft power and hydraulic efficiency under the rated conditions are explored in this paper. Influence of the order, which various factors impact on the performance of the pump, is identified.
- The optimal designed parameters of pump model are determined. Based on the results of orthogonal experimental design, many new models will be designed to prove the affects of parameters. Finally, the optimal designed parameters may be confirmed.

4.2. Criterion of orthogonal experimental design

This experimental design includes multi-factor, so multi-factor analysis will be selected. The main investigational parameters show as follows: flow rate and head of this kind of hot water circulation pump is 5.0m³/h and 3.5m respectively under the rated conditions. The rated speed of selected three-phase AC electric motor is 2850 r/min. The pump performance test was implemented, which water at 25℃ was selected as the medium, and the head of results from experiment was 3.9m. The material of fluid domain is water at 25℃ to guarantee similarity of numerical simulation. From the results of simulation under the rated conditions, the head of selected pump is 4.11m. It is regarded as the criterion of orthogonal experimental design.

4.3. Determination of factors orthogonal experiment

According to the professional knowledge and designed experience, the three influence factors of pump’s performance were selected: thickness of the outlet in the impeller (S), number of blades (Z), and width of the outlet in the impeller (b₂). Each influence factors takes three levels. The three factors and three level values are shown in Table 2.

| Table 2. Factor level. |
|------------------------|
| Level values | S/mm | b₂/mm | Z |
| 1 | 8.1 | 5.4 | 4 |
| 2 | 9.9 | 5.7 | 5 |
| 3 | 7.9 | 6.0 | 6 |

Choosing orthogonal table is the most important issue in orthogonal design. After the factors and their levels are determined, an appropriate orthogonal table can be chosen by considering the number of interaction between the factors from different levels. A common rule of choosing an orthogonal table is the smaller orthogonal as much as possible in order to reduce the number of tests in the premise of the factors and their interaction can be arranged. Generally speaking, the number of levels in experimental factors should be equal to the number of levels in orthogonal table. In this paper, an orthogonal experimental design with three factors and three level values was implemented. The number of orthogonal design was nine.

5. Results and analysis

5.1. Data collection and analysis of the orthogonal experimental design

With orthogonal design and extreme difference analysis method, the optimum parameters combination and influence of the order are determined. Selection of optimal case is determined by the research index which in this paper is head under the rated conditions. The results of orthogonal experimental design are shown in Table 3. This paper did not present the calculation processes of mean and range, and only listed results in the table. \( I_i (\text{II}, \text{III}) \) is the sum of the head under the rated conditions in the \( i \) level, and \( K_i \) is the average of \( I_i (\text{II}, \text{III}) \). The values of \( R_i \) is the extreme difference for each level.

The order of the factors is listed according to the size of the ranges (\( R_i \)). The larger the range is, the more influence on test result the level change of this factor has. Factor that has the biggest range is the uppermost one. Calculation results in the orthogonal table show that the order of range
is $R_C > R_B > R_A$. The order of factors influence level is $Z > b_2 > S$. Number of blades is the uppermost, while thickness of the outlet in the impeller is the unimportant.

From the results that $K_3 < K_1 < K_4$ for A, $K_1 < K_2 < K_3$ for B, and $K_1 < K_2 < K_3$ for C, so the optimal case A3B3C3 may be obtained, namely thickness of the outlet in the impeller $(S)$ is 7.9mm, number of blades $(Z)$ is six, and width of the outlet in the impeller $(b_2)$ is 6.0mm. Head under the rated conditions is 4.42921m in the optimal case, which relatively increases by 7.77% comparing with before optimization, which the head is 4.11m.

Table 3. The results of orthogonal experimental design.

| Trial number | factors | S/mm | $b_2$/mm | Z | Trial results |
|--------------|---------|------|----------|---|--------------|
|              | Level values |       |          |   |              |
| 1            | 1       | 8.1  | 5.4      | 4 | 3.84351      |
| 2            | 2       | 8.1  | 5.7      | 5 | 4.12777      |
| 3            | 3       | 8.1  | 6.0      | 6 | 4.38454      |
| 4            | 4       | 9.9  | 5.4      | 5 | 4.06026      |
| 5            | 5       | 9.9  | 5.7      | 6 | 4.32371      |
| 6            | 6       | 9.9  | 6.0      | 4 | 3.92614      |
| 7            | 7       | 7.9  | 5.4      | 6 | 4.28925      |
| 8            | 8       | 7.9  | 5.7      | 4 | 3.93814      |
| 9            | 9       | 7.9  | 6.0      | 5 | 4.20811      |

| I            | 12.35582 | 12.19302 | 11.70779 |
| II           | 12.31011 | 12.38962 | 12.39614 |
| III          | 12.4355  | 12.51879 | 12.9975  |
| K1           | 4.1186   | 4.0643   | 3.9026   |
| K2           | 4.1034   | 4.1299   | 4.1320   |
| K3           | 4.1452   | 4.1729   | 4.3325   |
| R            | 0.0418   | 0.1086   | 0.4299   |

5.2. Comparing the optimization results
The interior flow field of pumps, which include the original and the optimized models, is compared in this paper. Then the conclusions, which include the flow in impeller interior of the optimized model accords with the fluid flow motions in the impeller interior of general centrifugal pump, and the optimized model complies with the designed requirements of the centrifugal pump, are obtained.

The effective flow area in outlet of the impeller increases, when the backside of impeller outlet is trimmed, number of blades and width of the outlet in the impeller are increased. The effective flow area in the original model is $1006.62 \text{ mm}^2$, while it is $1073.85 \text{ mm}^2$ in the optimized model. Therefore, axial plane component of velocity ($v_{mz}$) in impeller outlet decreases when the flow rate keeps the constant. Based on the velocity of the triangle theory, peripheral component of velocity ($v_{uz}$) increases because of decreasing axial plane component of velocity ($v_{mz}$). Based on the basic equation for pump, increasing peripheral component of velocity ($v_{uz}$) may benefit for promoting the head. The
deviation of flow and non-uniform distribution of the velocity in pump can be improved, because the angle located in the backside of impeller outlet and numbers of blades are added.

6. Conclusions
According to the results from simulating the performance characters of the hot water circulation pump, which (0.4~1.1) times the rated conditions were calculated, the fluctuating trend of head, shaft power and hydraulic efficiency was conformed to the experimental data. So the feasibility and reliability of the numerical simulation in the structural optimized design process is verified.

According to the results from the orthogonal design and numerical simulation, the order of factors influence level is $Z > b_2 > S$ for the head of the pump. According to the results of extreme difference analysis, the optimal case is A3B3C3, namely thickness of the outlet in the impeller(S) is 7.9mm, number of blades (Z) is six, and width of the outlet in the impeller ($b_2$) is 6.0mm. The head of the optimized model increases by 7.77% under the rated conditions. Comparing the fluid flow distribution of relative velocity and total pressure in the impeller interior, the conclusion, which the flow field in impeller interior of the optimized pump is more uniform and obeys to the law of fluid flow in the passages of impeller, can be obtained. The optimized model complies with the designed requirements of the centrifugal pump.

It is feasible for the method of optimize design of impeller, combining the orthogonal experimental design with numerical simulation. That method provides theoretical foundation for the actual designed process of the pump.

References
[1] Tan M G, Liu H L and Wang Y 2009 J. Drainage and Irrigation Machinery 27(5) 314-8
[2] Liu J R, Wang D M and Su Q Q 2010 Journal of Drainage and Irrigation Machinery Engineering 28(1) 22-30
[3] Pan Z Y, Xie R and Cao Y J 2009 Drainage and Irrigation Machinery 27(5) 319-22
[4] Shojaeeafard M H, Tahani M and Ehghaghi M B 2012 Computers & Fluids 60 61-70
[5] Wiesner F J 1967 J. ASME Journal of Engineering for Power 89(4) 558-72
[6] He X J and Lao X S 2007 Journal of HeBei engineering and technical college 3(1) 1-3
[7] Cong X Q, Yuan S Q and Yuan D Q 2005 Journal of agricultural machinery 36(10) 66-9
[8] Jafarzadeh B, Hajari A and Alishahi M M 2011 Applied Mathematical Modeling 35 242-9
[9] Fan H M 2010 Journal of Hydrodynamics 22(4) 518-25
[10] Zhu J J, Chew D A S and Lv S 2013 Habitat International 37 148-54
[11] Help Navigator ANSYS CFX. Release 12.1 CFX-Solver modelling Guide 115-119