Optimization of the impeller for hydraulic performance improvement of a high-speed magnetic drive pump

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Abstract
Magnetic drive centrifugal pumps have compact structure and lower efficiency than ordinary centrifugal pumps. The surrogate-based optimization technique was applied to improve the performance of a high-speed magnetic drive pump with the help of numerical simulations. Eight geometrical parameters of the impeller were considered as the design variable. About 290 samples of impeller were generated by optimal Latin hypercube sampling (OLHS) method, and the corresponding efficiencies of all the impeller samplings were obtained from numerical simulation. The performance test of the prototype pump was carried out, and the experimental results were in good agreement with the numerical simulation results. The hydraulic efficiency at 1.2 \textit{Qd} of the magnetic drive pump was set as the optimization objective. Using response surface methodology (RSM), surrogate models were established for the objective functions based on the numerical results. The multi-island genetic algorithm (MIGA) was used to optimize the impeller. The hydraulic efficiency of the optimal impeller at rated flow rate was 72.89\%, which was 6.23\% higher than the prototype impeller.

Keywords
High-speed magnetic drive pump, numerical simulation, optimization, response surface methodology, multi-island genetic algorithm

Introduction
The magnetic drive pump realizes non-contact torque transmission through the magnetic coupling, and changes the dynamic seal into the static seal, to make the pump completely leak free. It is often used to transport toxic, harmful, flammable, and explosive mediums. However, owing to the low transmission efficiency of magnetic coupling, the efficiency of the magnetic drive pump is lower than that of ordinary centrifugal pumps.\textsuperscript{1-3} Due to its high usage, the optimization of the magnetic drive pump and other turbomachinery is very important to reduce global energy consumption, and to increase their performance.\textsuperscript{4-9}

At present, optimization methods based on computational fluid dynamics (CFD) technology have been more often applied to enhance the performance of turbomachinery. Compared with the traditional trial-and-error method, the combination of an optimization method with numerical simulation for pump performance improvement can shorten the development cycle and reduce the experimental resources.\textsuperscript{10} Many

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researchers have carried out optimization of pumps and other turbomachinery with the help of numerical simulations. Some of them focus on improving the pump performance by optimizing impeller blade shapes.

Thakkar et al.\textsuperscript{11} presented an optimization approach for improving the performance of a sanitary centrifugal pump. The CFD technology, the response surface methodology (RSM), and a multi-objective optimization algorithm were integrated to optimize the blade angle and width. The results show that the pump head and efficiency are enhanced by 9.15\% and 10.15\%, respectively. Pei et al.\textsuperscript{12} adopted the artificial neural network and the particle swarm optimization algorithm to improve the blade angle distribution for enhancing the efficiency of a centrifugal pump. Kim et al.\textsuperscript{13} considered the hub and tip blade angles as variables to optimize the impeller of a counter-rotating-type pump-turbine, in order to improve the pump and turbine mode efficiencies. Suh et al.\textsuperscript{14} improved the efficiency and outlet static pressure of a multi-phase pump by optimizing the shape of the impeller and the diffuser blade.

Some researchers focus on the optimization of the impeller meridional plane to improve the performance of pumps. Sen-Chun et al.\textsuperscript{15} employed genetic algorithm and back propagation neural network algorithm to conduct the optimization of impeller meridional plane of a pump as turbine. Zhao et al.\textsuperscript{16} constructed an automatic simulation optimization platform based on ISIGHT software. The pump design software CFturbo, PumpLinx, and MATLAB were systematically integrated to conduct the optimization of the LBE-cooled main coolant pump. The multi-objective optimization of the meridian plane was carried out to improve the energy conversion capability of the main coolant pump. Shim and Kim\textsuperscript{17} hired surrogate-based optimization to improve the hydraulic performance and operating stability of the pump were improved. In addition, there are some literatures discuss the optimization of pump operation stability. Qian et al.\textsuperscript{18} conducted the optimization of blade thickness distribution to reduce the hydro-induced vibration. The vibration performance of the prototype impeller, the traditional splitter blades, and the optimized impeller were compared and analyzed. Based on three-dimensional unsteady RANS analysis, the pressure fluctuations in the volute under rated flow rate were compared between original and optimized designs.\textsuperscript{19} The surrogate-based optimization was performed to improve the hydraulic performance of the centrifugal pump. Shim et al.\textsuperscript{20} considered radial force as one of the optimization objective functions to improve the performance of a centrifugal pump with double volute. Both the radial force and the amplitude of the radial force fluctuation were reduced by the optimization of the geometric parameters. Further, Wang et al.\textsuperscript{21} compared the prediction accuracies of three surrogate models (Response surface model, Kriging model, and Radial basis neural network) on the optimization of centrifugal pump, and found the RSF model predicted the highest efficiency.

The hydraulic performance of the pump depends on the structural parameters of the impeller and volute. The hydraulic design is very important for the performance of the pump, but the conventional empirical design often cannot achieve the optimal matching of the geometric parameters of the impeller. Therefore, it is necessary to optimize the geometric parameters of the impeller to improve the hydraulic performance of the pump. Although there are many literatures on the optimization of pumps, there are few reports on the comprehensive optimization of impeller meridional plane and blade parameters to improve pump performance. In addition, the magnetic drive pump is usually compact in structure, and it is difficult to achieve the optimum matching among the geometrical parameters of the impeller due to the limitation of the size during the design, resulting in low pump efficiency. The optimization of the high-speed magnetic drive pump is not explored by the researchers. So, inspiring from these research gaps, this paper presents an efficient method for the optimization of a high-speed magnetic drive pump. To improve the pump performance, the multi-Island genetic algorithm (MIGA) combined with RSM, design of experiments (DOE), and CFD technology, have been applied comprehensively to get an optimal impeller.

**Prototype pump model**

A high-speed magnetic drive centrifugal pump with a shrouded impeller is studied in this paper. The design parameters volumetric flow rate, head, and rotational speed of the magnetic drive pump are 30 m\textsuperscript{3}/h, 130 m, and 8000 rpm, respectively. The parameters of the prototype impeller are listed in Table 1, and the 2-D diagram of the prototype impeller is shown in Figure 1. The pump impeller is a typical shrouded impeller with five cylindrical blades with a wrap angle of 120°. The shroud corner radius \( R_1 \) and hub corner radius \( R_2 \) are given in Figure 1. The prototype impeller is designed by the velocity coefficient method, and many geometric parameters are obtained by the designer based on experience, so there is a lot of space for optimization.

**Numerical methods**

**Governing equations**

In the simulation, the fluid is assumed to be incompressible and homogeneous. The Navier-Stokes equation based on the Newtonian fluid are as follows\textsuperscript{22}:
\[
\frac{\partial p}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}
\]

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

Where \(u\) is the velocity, \(p\) is the pressure, \(\rho\) is the mixture density, \(\mu\) is the viscosity, \(\tau_{ij} = -\rho u_i u_j\) is the Reynolds stress.

In this study, the commercial code CFX 19.2 was adopted for the flow field simulation. The k-epsilon (k-\(\varepsilon\)) turbulence model is usually used in rotating machinery, and it has the characteristics of fast convergence and high precision.\(^\text{23}\) Because a large number of impeller models need to be calculated in the process of optimization. In order to save time, the k-\(\varepsilon\) turbulence model which with faster convergence speed for our work was employed to conduct the steady-state CFD simulation of the optimization. In addition, the k-omega (k-\(\omega\)) turbulence model was used to conduct the CFD simulation of the optimized impeller.

### Mesh generation

The three-dimensional model of the computational domain was established according to the geometrical parameters of the pump. The whole computational domain of the magnetic drive pump was divided into four subdomains, suction pipe, impeller, volute, discharge pipe, as shown in Figure 2. The ICEM-CFD software was employed to mesh the computational domain. The suction pipe and the discharge pipe adopted hexahedral structured grids, while impeller and volute adopted unstructured grids.

Mesh independence analysis result is listed in Table 2. When the total number of grid cells reaches 4.8 million, the hydraulic efficiency of numerical simulation is stable. The whole computational mesh elements of the prototype impeller used in this study is approximately 4.8 million, as shown in Figure 3. The grid cells number of suction pipe, discharge pipe, impeller and volute are 0.22, 0.2, 0.66, and 3.75 million. The \(y^+\) value of the mesh in the impeller and volute domain are within 70. The uncertainly and error of the adopted grid scheme are shown in Table 3.\(^\text{24–26}\) Furthermore, the grid cells number of all the impeller models used for CFD analysis during the optimization process was greater than 0.6 million.

### Boundary conditions

The impeller was set as rotating domain and the speed is 8000 rpm, while the other domains were specified as

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**Table 1.** Design parameters of prototype impeller.

| Parameters                  | Value |
|-----------------------------|-------|
| Inlet diameter \((D_1/mm)\) | 60    |
| Outlet diameter \((D_2/mm)\)| 120   |
| No. of blades \((Z)\)       | 5     |
| Inlet blade angle \((\beta_1/)\) | 25 |
| Outlet blade angle \((\beta_2/)\) | 30 |
| Wrap angle \((\phi/)\)      | 120   |
| Shroud corner radius \((R_1/mm)\) | 13.2 |
| Hub corner radius \((R_2/mm)\) | 27.4 |
| Leading edge thickness \((T_1/mm)\) | 3   |
| Trailing edge thickness \((T_2/mm)\) | 3   |

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**Table 2.** Mesh independence test results.

| No. of grid cells (in million) | Hydraulic efficiency (%) |
|-------------------------------|--------------------------|
| 1.6                           | 68.62                    |
| 2.5                           | 69.96                    |
| 3.4                           | 71.83                    |
| 4.8                           | 71.76                    |

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stationary domain, as shown in Figure 2. The boundary conditions were set as follows: the total pressure was set to the inlet with a 5% turbulence intensity, the outlet was specified the mass flow rate. A no slip wall with a smooth surface was set in each domain. The interface between rotating domain and stationary domain is set as the frozen rotor type, and the association between interface grids adopts GGI mode. In addition, the root mean square residual error is used in the numerical simulation, and the residual error of convergence is less than $10^{-4}$.

**Optimization methods**

The process flowchart of the optimization is demonstrated in Figure 4. First of all, the steady-state CFD simulation of the prototype pump is conducted by CFX. The pump performance obtained by numerical simulation is verified by the experimental results. Then, the hydraulic efficiency is considered as the optimization objective, and eight geometric parameters of the impeller are selected as the optimization variables. The OLHS method is employed to conduct the design of experiments. Numerical simulation of the sample impeller models is conducted, and the pump performance is obtained. Since the functional relation between variables and objective function is difficult to be described by methods of algebra, an approximate model is established between the variables and pump performance by using RSM. Then, MIGA is selected in ISIGHT software for comprehensive optimization, and an optimal impeller is obtained. Finally, the inside flow characteristic of the optimal impeller is studied in detail to demonstrate the performance improvement of the pump.

**Optimization objective and design variables**

Since the pump requires high efficiency under large flow conditions, the hydraulic efficiency at $1.2Q_d$ ($Q_d$: design flow rate) was considered as the optimization objective. In order to ensure that the pump head meets the design requirements, the pump head was taken as the constraint condition, and the range was $128–132$ m$^3$. The efficiency and head of the pump can be defined as following:

$$\eta = \frac{\rho g Q H}{T_o \omega}$$  \hspace{1cm} (3)

$$H = \frac{P_{out} - P_{in}}{\rho g} + \Delta h$$  \hspace{1cm} (4)

Where $\eta$ is pump efficiency, $H$ is pump head, $\rho$ is the density of the fluid, $g$ is the gravitational acceleration, $Q$ is the flow rate, $T$ is the torque of impeller (including blades, hub, and shroud), $\omega$ is the angular velocity, $\Delta h$ is the height difference of inlet and outlet, $P_{out}$ and $P_{in}$ is the total pressure at the outlet and inlet, respectively.

The pump performance depends upon various geometrical parameters of the impeller. The shape of the blade$^{11–14}$ and the profile of the impeller meridional plane$^{15–17}$ have significant impact on the efficiency and head of the pump. In the present study, the eight geometrical parameters of impeller, namely blade inlet angle $\beta_1$, blade outlet angle $\beta_2$, blade wrap angle $\phi$, No. of blades $Z$, leading edge thickness of blade $T_l$, trailing edge thickness of blade $T_s$, shroud corner radius $R_1$, and hub corner radius $R_2$ were considered as the design variable. The shroud corner radius $R_1$ and hub corner radius $R_2$ control the profile of the impeller meridional plane, as shown in Figure 1, and the other six variables control the blade shape. The other geometrical parameters of the impeller are fixed to match the impeller with the pump casing. The ranges of the eight variables are specified, as listed in Table 4.

**Design of experiments**

The design of experiments needs to consider that samples can reflect the whole design space$^{27,28}$. The optimal Latin hypercube sampling (OLHS) method can distribute all experimental points in the design space with a good space filling property and homogeneity.$^{29,30}$ In this study, to ensure the homogeneity of the distribution of design points, the OLHS method was employed to
create the variable matrix. The ISIGHT software was used to generate 290 groups of experimental samples. Then a set of sample impeller models with different variables were built, and the objective function of the impellers was obtained by CFD simulation. Some of the sample impeller parameters and simulation results are listed in Table 5.

**Surrogate model and optimization algorithm**

Surrogate-based optimization techniques can significantly improve the efficiency of optimization. The RSM model is one of the widely used surrogate models in the process of experimental optimization. The RSM is a statistical method that explores the relationship between the objective function and the design variable. The methodology based on RSM is convenient and easy to implement, which leads to save time and cost of experimental work. There is a highly nonlinear relationship between the performance of high-speed magnetic drive pumps and geometrical parameters of impeller. In order to establish a reliable mathematical model, the RSM approximation model between hydraulic and design variables was constructed based on the results of DOE. The RSM approximate model was assumed to be a second-order polynomial, and it can be defined as following:

\[
Y(X) = C_0 + \sum_{i=1}^{n} C_iX_i + \sum_{i=1}^{n} C_iX_i^2 + \sum_{i<j} C_{ij}X_iX_j \quad (5)
\]

Where \(X\) represent variables and \(n\) is the number of design variables. From left to right, the model includes an intercept, linear terms, quadratic interaction terms, and squared terms. The accuracy of the coefficient set of the polynomial is evaluated using regression analysis with an analysis of variance.

Genetic algorithm is often used to solve complex engineering problems. Multi-island genetic algorithm is improved on the traditional genetic algorithm and has better global solving ability and computational efficiency.
efficiency. MIGA divides each population into several subgroups, namely “islands,” which can suppress the phenomenon of prematurity in traditional genetic algorithms. In present study, the MIGA was employed to optimize the constructed RSM approximate model.

Results and discussions

Test setup and numerical simulation validation

In order to verify the numerical results, the performance experiment of the prototype pump was carried out on the closed-loop high-speed pump test rig, which was composed of a water tank, pipelines, valves, sensors, and control cabinet, as shown in Figure 5. The flow rate was adjusted by the outlet valve and measured by a turbine flowmeter, and the measurement accuracy was ±0.5%. The pump head was calculated by measuring the inlet and outlet pressure of the pump. The ranges of pressure sensors were −0.1 to 0.1 MPa at inlet and 0 to 2.5 MPa at outlet, and the measurement accuracy was ±0.5%. The rotational speed of the pump was measured by a photoelectric sensor with an error was 0.1%.

The validation results of CFD analysis with experiments are shown in Figure 6. The simulation efficiency and the test efficiency cannot be compared because the test efficiency including efficiency of the magnetic coupling, motor and frequency converter. Hence only the pump head of the test and the simulation are compared in Figure 6. The numerical results agreed well with the experimental results at all flow rates. The test head

| No. | $\varphi$ | $T_1$ | $T_2$ | $\beta_1$ | $\beta_2$ | $R_1$ | $R_2$ | $z$ | $\eta$ | $H$ |
|-----|----------|-------|-------|-----------|-----------|-------|-------|-----|-------|-----|
| 1   | 104.94   | 2.92  | 3.21  | 31.98     | 20.69     | 11.00 | 18.54 | 6   | 0.713 | 125.21 |
| 2   | 112.44   | 2.98  | 2.21  | 20.85     | 20.33     | 11.48 | 24.30 | 5   | 0.698 | 123.64 |
| 3   | 100.65   | 2.26  | 2.61  | 27.81     | 26.44     | 15.54 | 17.12 | 5   | 0.679 | 119.78 |
| 4   | 100.55   | 3.05  | 3.14  | 32.61     | 24.39     | 16.95 | 24.98 | 6   | 0.719 | 126.42 |
| 5   | 121.68   | 2.04  | 2.32  | 31.81     | 29.36     | 12.76 | 17.98 | 5   | 0.706 | 125.99 |
| 6   | 129.45   | 3.19  | 3.32  | 27.70     | 23.33     | 17.00 | 12.72 | 7   | 0.728 | 133.06 |
| 7   | 124.09   | 2.76  | 2.75  | 25.64     | 31.77     | 10.65 | 18.91 | 5   | 0.697 | 125.93 |
| 8   | 105.47   | 3.22  | 2.88  | 30.15     | 32.59     | 10.80 | 23.37 | 6   | 0.712 | 129.38 |
| 9   | 117.66   | 3.48  | 2.69  | 26.21     | 18.64     | 11.60 | 20.83 | 6   | 0.679 | 128.77 |
| 10  | 125.16   | 3.44  | 3.39  | 30.21     | 25.67     | 11.93 | 16.00 | 7   | 0.731 | 136.03 |
| 281 | 111.77   | 2.78  | 3.44  | 30.27     | 31.66     | 15.64 | 26.89 | 5   | 0.699 | 126.22 |
| 282 | 102.53   | 2.59  | 2.31  | 23.24     | 33.59     | 16.49 | 24.91 | 5   | 0.682 | 124.84 |
| 283 | 134.81   | 3.31  | 2.29  | 30.90     | 23.97     | 11.18 | 19.59 | 7   | 0.746 | 138.52 |
| 284 | 118.20   | 3.01  | 2.69  | 26.96     | 27.08     | 17.87 | 28.38 | 6   | 0.714 | 132.93 |
| 285 | 122.89   | 2.56  | 2.80  | 29.58     | 29.48     | 17.97 | 14.89 | 5   | 0.686 | 125.03 |
| 286 | 116.72   | 3.05  | 3.50  | 24.79     | 28.37     | 18.00 | 19.47 | 6   | 0.693 | 131.34 |
| 287 | 127.17   | 3.39  | 3.09  | 28.95     | 22.80     | 17.25 | 15.39 | 5   | 0.690 | 124.43 |
| 288 | 118.73   | 2.00  | 2.51  | 34.78     | 25.50     | 14.39 | 17.49 | 6   | 0.700 | 132.22 |
| 289 | 103.33   | 2.95  | 2.17  | 24.27     | 18.75     | 15.24 | 23.92 | 6   | 0.707 | 127.93 |
| 290 | 108.26   | 3.24  | 2.86  | 25.72     | 30.22     | 17.36 | 22.54 | 5   | 0.709 | 132.56 |

Figure 5. Test rig.

Figure 6. Comparison of CFD and experimental results.
under the rated flow is 134.4 m, while the CFD simulation head is 133.05 m, and the error is 1%. The maximum error between test and simulation under the off-design condition is 3.9%.

**Numerical simulation results of samples**

Steady-state CFD simulation of 290 sampling points generated by DOE was carried out, and the corresponding pump performance at 1.2 \( Q_d \) was obtained. The calculation results of the impeller schemes with different input parameters are shown in Figure 7. The maximum and minimum hydraulic efficiency are 75.5% and 63.9%, respectively. The maximum and minimum head of the pump is 140.4 and 119.27 m, respectively. It can be seen that the pump performance corresponding to the impeller with different parameters varies greatly. The maximum deviations of hydraulic efficiency and head for all impeller samples are 11.6% and 21.13 m, respectively.

**Results of optimization**

In the present work, the RSM model is performed based on the numerical sets generated by the OLHS method. In ISIGHT software, with design variables as input, pump efficiency, and head as response, the RSM approximation model was established between the objective function and design variables. The regression equation of the pump efficiency and pump head obtained by the RSM is given in equation (6).

where \( X_1 \sim X_8 \) represent \( \phi, T_1, T_2, \beta_1, \beta_2, R_1, R_2, Z \), respectively. In the formula, \( C_0 \sim C_{44} \) are the coefficients of the equation, which is attached in Tables 6 and 7.

\[
Y(X) = C_0 + C_1 \cdot X_1 + C_2 \cdot X_2 + C_3 \cdot X_3 + C_4 \cdot X_4 + C_5 \cdot X_5 + C_6 \cdot X_6 + C_7 \cdot X_7 + C_8 \cdot X_6 + C_9 \cdot X_7^2 + C_{10} \cdot X_2^3 + C_{11} \cdot X_3^2 + C_{12} \cdot X_4^2 + C_{13} \cdot X_5^2 + C_{14} \cdot X_6^2 + C_{15} \cdot X_7 + C_{16} \cdot X_2^2 + C_{17} \cdot X_1 \cdot X_2 + C_{18} \cdot X_1 \cdot X_3 + C_{19} \cdot X_1 \cdot X_4 + C_{20} \cdot X_1 \cdot X_5 + C_{21} \cdot X_1 \cdot X_6 + C_{22} \cdot X_1 \cdot X_7 + C_{23} \cdot X_1 \cdot X_8 + C_{24} \cdot X_2 \cdot X_3 + C_{25} \cdot X_2 \cdot X_4 + C_{26} \cdot X_2 \cdot X_5 + C_{27} \cdot X_2 \cdot X_6 + C_{28} \cdot X_2 \cdot X_7 + C_{29} \cdot X_2 \cdot X_8 + C_{30} \cdot X_3 \cdot X_4 + C_{31} \cdot X_3 \cdot X_5 + C_{32} \cdot X_3 \cdot X_6 + C_{33} \cdot X_3 \cdot X_7 + C_{34} \cdot X_3 \cdot X_8 + C_{35} \cdot X_4 \cdot X_5 + C_{36} \cdot X_4 \cdot X_6 + C_{37} \cdot X_4 \cdot X_7 + C_{38} \cdot X_4 \cdot X_8 + C_{39} \cdot X_5 \cdot X_6 + C_{40} \cdot X_5 \cdot X_7 + C_{41} \cdot X_5 \cdot X_8 + C_{42} \cdot X_6 \cdot X_7 + C_{43} \cdot X_6 \cdot X_8 + C_{44} \cdot X_7 \cdot X_8
\]  

(6)

Assessing the accuracy of the approximate model is very necessary before conducting the optimization with MIGA. The R-square error is used to analyze the deviation between the results of the numerical simulation and those predicted by the RSM approximate model. The closer the R-square value is to 1, the higher the model reliability is. Fifteen sample points were randomly selected for R-square error analysis, as shown in Figure 8. It can be seen that R-square value of efficiency and head are greater than the acceptable level of 0.9, which indicates that the RSM approximate model has faithful prediction accuracy. Therefore, the established approximate model can be used for the hydraulic optimization of the high-speed magnetic drive pump.

The MIGA was used to optimize the approximate model and obtain the optimal impeller parameters. In the ISIGHT software, some parameters of the MIGA were set as follows: the sub-population size was 50, the number of islands was 10, the number of generations was 20, the rate of crossover was 0.9, the rate of mutation was 0.01, and rate of migration was 0.01. The optimal design scheme was obtained through 10,000 steps of iteration. The geometric parameters of the optimal and prototype impeller are listed in Table 8. Compared with the prototype impeller, the wrap angle \( \phi \) of the optimal impeller model is increased by nearly 20°, and the blade outlet angle \( \beta_2 \) decreased by about 4°, and other parameters changed slightly.

**Comparison of the pump performance**

Numerical simulation of the optimal impeller model under different flow rates was carried out, and the performance curve of the pump was obtained. The
Table 6. Coefficient values of RSM model of efficiency index.

| Coefficient value | Coefficient value | Coefficient value |
|-------------------|-------------------|-------------------|
| C0  | 0.554196512 | C15 | 4.52E-06 |
| C1  | -0.001520534 | C16 | 0.001378348 |
| C2  | 0.01372958 | C17 | -0.000101859 |
| C3  | 0.028259689 | C18 | 2.34E-05 |
| C4  | 0.00119637 | C19 | -3.91E-06 |
| C5  | 0.001721581 | C20 | 2.08E-05 |
| C6  | 0.002241768 | C21 | -3.14E-05 |
| C7  | 0.003951243 | C22 | -7.43E-07 |
| C8  | 0.00042658 | C23 | -4.22E-05 |
| C9  | 1.14E-05 | C24 | -0.000664501 |
| C10 | -0.002096946 | C25 | 0.00102806 |
| C11 | -0.001498435 | C26 | 0.000235665 |
| C12 | -6.76E-08 | C27 | -0.000430958 |
| C13 | -6.43E-05 | C28 | 2.23E-05 |
| C14 | -4.87E-05 | C29 | 0.001605373 |

Table 7. Coefficient values of RSM model of head index.

| Coefficient value | Coefficient value | Coefficient value |
|-------------------|-------------------|-------------------|
| C0  | -76.86051978 | C15 | -0.000405367 |
| C1  | 1.457277825 | C16 | -0.941996522 |
| C2  | 0.924534544 | C17 | -0.016252881 |
| C3  | 1.71346721 | C18 | 0.021842169 |
| C4  | 0.50170253 | C19 | 0.000993176 |
| C5  | 0.167198067 | C20 | 0.001228664 |
| C6  | -0.516693121 | C21 | -0.006662566 |
| C7  | 0.881111573 | C22 | -0.000374729 |
| C8  | 24.6403283 | C23 | -0.047364479 |
| C9  | -0.001478161 | C24 | 0.002351904 |
| C10 | 0.118635475 | C25 | 0.028948105 |
| C11 | -0.039463768 | C26 | -0.033029606 |
| C12 | -0.009936944 | C27 | -0.081436878 |
| C13 | 0.001363163 | C28 | 0.019514443 |
| C14 | 0.023882643 | C29 | 0.220807518 |

Performance comparison between the optimal impeller and the prototype impeller is demonstrated in Figure 9. At each flow rate, the hydraulic efficiency of the optimal impeller is obviously higher than those of the prototype impeller, while the pump head changes slightly. The hydraulic efficiency and pump head of two impeller models at 1.0 Q_d and 1.2 Q_d are listed in Table 9. The results indicate that hydraulic efficiency is significantly improved through optimization. The hydraulic efficiency of the optimal impeller is 6.23% and 4.44% higher than that of the prototype impeller at 1.0 Q_d and 1.2 Q_d, respectively. In addition, by comparing the predicted results and CFD simulation results of the optimal impeller at 1.2 Q_d, the deviations of hydraulic efficiency and pump head are 2.06% and 0.3%, respectively. It is indicated that RSM approximate model can accurately predict pump performance.

The losses in the pump include leakage, disk friction, hydraulic losses in impeller and volute, and mechanical loss. This paper only optimized the impeller without changing the volute and the magnetic coupling. Compared with the prototype impeller, the optimized impeller only changed the blade shape, the profile of hub and shroud, and did not change the impeller diameter. The changes of leakage, disk friction and mechanical losses in the pump can be ignored. Through the optimization of the impeller, the hydraulic loss in the pump is reduced and the hydraulic efficiency of the pump is improved.

**Comparison of the flow field**

The internal flow fields of the prototype pump and the optimal pump under different flow conditions are compared and analyzed. The pressure distribution at mid-span of the two impellers is shown in Figure 10. For the two impeller models, there is a negative pressure region at the impeller inlet under different flow rates. The
static pressure gradually increases from the impeller inlet to the volute outlet. The pressure in the optimal pump is significantly higher than that of the prototype pump at 0.7 Qd. The pressure in the two pumps is basically the same at 1.0 Qd and 1.2 Qd.

The streamlines at the mid-span of the optimal and the prototype impeller under various flow rates are illustrated in Figure 11. For the two impeller models, the streamlines in the pump are smooth, and the flow condition is good at 1.2 Qd. However, several vortexes exist in the blade channel near the tongue due to rotor-stator interaction at 1.0 Qd and 0.7 Qd, especially in the low flow rate. The flow condition of the optimal impeller is better than that of the prototype impeller. The flow field of optimal impeller at 0.7 Qd is significantly worse than that of 1.0 Qd and 1.2 Qd. This may be owing to the maximum efficiency of 1.2 Qd is considered as the optimization objective in this paper, which makes the best efficiency point of the optimal impeller shift to the large flow rate and cannot guarantee the efficiency of the low flow rate.

**Conclusions**

In this study, a set of impeller samples of the high-speed magnetic drive pump were designed based on the OLHS method. Through steady-state
numerical simulation of impeller samples, the hydraulic efficiency and head at 1.2 $Q_d$ were obtained. The surrogate models were constructed for the objective functions based on the numerical results. The MIGA was hired to optimize the impeller. The conclusions are as follows:

**Figure 10.** Pressure distribution of pump: (a) prototype impeller and (b) optimal impeller.

**Figure 11.** Streamline of the pump: (a) prototype impeller and (b) optimal impeller.
1. The RSM approximate model between the objective function and the design variables was established. The deviations of hydraulic efficiency and pump head between the predicted and the numerical simulation results at 1.2 $Q_d$ were 2.06% and 0.3%, respectively.

2. The hydraulic efficiency of the optimal impeller under rated flow rate was 72.89%, which was 6.23% higher than that of prototype impeller.

3. The streamline distribution in the optimized impeller was smooth, and the internal flow was improved under the design condition.

In the current work, eight design variables are selected to optimize the impeller without considering the blade width. In the future work, the parameters which have a significant impact on the pump efficiency should be considered as variables to optimize the impeller, and the geometric parameters of the volute should be optimized to further improve the pump efficiency. In addition, the performance under design flow rate shall be considered as the objective function. Further, the number of samples can be appropriately reduced to shorten the optimization cycle.

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References
1. Kong F, Shen X, Wang W, et al. Performance study based on inner flow field numerical simulation of magnetic drive pumps with different rotate speeds. Chin J Mech Eng 2012; 25: 137–143.
2. Zhenjun G, Chaoqun H, Jianrui L, et al. The study on internal flow characteristics of magnetic drive pump. IEEE Access 2019; 7: 100003–100013.
3. Gao Z, Zhang J, Li W, et al. The flow-heat coupling study of cooling circulating channel of magnetic drive pump. Mechanics 2021; 27: 285–294.
4. Liu H, Wang K, Yuan S, et al. Multicondition optimization and experimental measurements of a double-blade centrifugal pump impeller. J Fluid Eng 2013; 135: 111031–111031.
5. Kumar PM, Seo J, Seok W, et al. Multi-fidelity optimization of blade thickness parameters for a horizontal axis tidal stream turbine. Renew Energy 2019; 135: 277–287.
6. Zhao F, Kong F, Zhou Y, et al. Optimization design of the impeller based on orthogonal test in an ultra-low specific speed magnetic drive pump. Energies 2019; 12: 4767.
7. Qiu N, Zhou W, Che B, et al. Effects of microvortex generators on cavitation erosion by changing periodic shedding into new structures. Phys Fluids 2020; 32: 104108.
8. Kim JH, Cho BM, Kim YS, et al. Optimization of a single-channel pump impeller for wastewater treatment. Int J Fluid Machinery Syst 2016; 9: 370–381.
9. Kim S, Choi YS, Lee KY, et al. Design optimization of mixed-flow pump in a fixed meridional shape. Int J Fluid Machinery Syst 2011; 4: 14–24.
10. De Donno R, Ghidonii A, Novaenta G, et al. Shape optimization of the ERCOFTAC centrifugal pump impeller using open-source software. Optim Eng 2019; 20: 929–953.
11. Thakkar S, Vala H, Patel VK, et al. Performance improvement of the sanitary centrifugal pump through an integrated approach based on response surface methodology, multi-objective optimization and CFD. J Braz Soc Mech Sci Eng 2021; 43: 24.
12. Pei J, Wang W, Osman MK, et al. Multiparameter optimization for the nonlinear performance improvement of centrifugal pumps using a multilayer neural network. J Mech Sci Technol 2019; 33: 2681–2691.
13. Kim JW, Suh JW, Choi YS, et al. Optimized blade design of counter-rotating-type pump-turbine unit operating in pump and turbine modes. Int J Rotating Machinery 2018; 2018: 1–12.
14. Suh JW, Kim JW, Choi YS, et al. Multi-objective optimization of the hydrodynamic performance of the second stage of a multi-phase pump. Energies 2017; 10: 1334.
15. Sen-Chun M, Zhi-Xiao S, Xiao-Hui W, et al. Impeller meridional plane optimization of pump as turbine. Sci Prog 2020; 103: 3685041987642.
16. Zhao Y, Lu Y, Zhu R, et al. MDO strategy for meridian plane design to improve energy conversion capability of LFR main coolant pump. Ann Nucl Energy 2020; 148: 107763.
17. Shim HS and Kim KY. Design optimization of the impeller and volute of a centrifugal pump to improve the hydraulic performance and flow stability. J Fluid Eng 2020; 142: 101211.
18. Qian B, Wu P, Huang B, et al. Optimization of a centrifugal impeller on blade thickness distribution to reduce hydro-induced vibration. J Fluid Eng 2020; 142: 021202.
19. Heo MW, Ma SB, Shim HS, et al. High-efficiency design optimization of a centrifugal pump. J Mech Sci Technol 2016; 30: 3917–3927.
20. Shim HS, Afzal A, Kim KY, et al. Three-objective optimization of a centrifugal pump with double volute to minimize radial thrust at off-design conditions. Proc IMechE, Part A: J Power and Energy 2016; 230: 598–615.
21. Wang W, Pei J, Yuan S, et al. Application of different surrogate models on the optimization of centrifugal pump. *J Mech Sci Technol* 2016; 30: 567–574.
22. Versteeg HK and Malalasekera W. *An Introduction to computational fluid dynamics*. New York, NY: Wiley, 1995.
23. Xu Z, Kong F, Zhang H, et al. Research on visualization of inducer cavitation of high-speed centrifugal pump in low flow conditions. *J Mar Sci Eng* 2021; 9: 1240.
24. Ismail BC, Urmila GHIA, Patrick JR, et al. Procedure for estimation and reporting of uncertainty due to discretization in CFD applications. *J Fluid Eng* 2008; 130: 078001.
25. Zhang N, Shen HC and Yao HZ. Uncertainty analysis in CFD for resistance and flow field. *J Ship Mech* 2008; 12: 211–224.
26. Cheng XK. Investigation into grid generation and its uncertainty. *J Ship Des* 2016; 6: 35–40.
27. Luo J, Ji Y, Lu W, et al. Optimal Latin hypercube sampling-based surrogate model in napls contaminated groundwater remediation optimization process. *Water Supply* 2018; 18: 333–346.
28. Zhang F, Cheng L, Wu M, et al. Performance analysis of two-stage thermoelectric generator model based on Latin hypercube sampling. *Energy Convers Manag* 2020; 221: 113159.
29. Jin R, Chen W and Sudjianto A. An efficient algorithm for constructing optimal design of computer experiments. *J Stat Plan Inference* 2005; 134: 268–287.
30. Shang X, Chao T, Ma P, et al. An efficient local search-based genetic algorithm for constructing optimal Latin hypercube design. *Eng Optim* 2020; 52: 271–287.
31. Gunst RF, Myers RH and Montgomery DC. Response surface methodology: process and product optimization using designed experiments. *Technometrics* 1996; 38: 285.
32. Samad A and Kim KY. Surrogate based optimization techniques for aerodynamic design of turbomachinery. *Int J Fluid Machinery Syst* 2009; 2: 179–188.

**Appendix**

**Notation**

**Symbols**

- $\eta$: hydraulic efficiency (%)
- $H$: Pump head (m)
- $Q$: Volumetric flow rate (m$^3$/s)
- $Q_d$: Rated flow rate (m$^3$/h)
- $\rho$: Density (kg/m$^3$)
- $\beta_1$: Blade inlet angle (°)
- $\beta_2$: Blade outlet angle (°)
- $Z$: No. of blades
- $\Phi$: Wrap angle (°)
- $r$: Density (kg/m$^3$)
- $b_1$: Blade inlet angle (°)
- $b_2$: Blade outlet angle (°)
- $\theta$: Angle (°)
- $\eta$: Hydraulic efficiency (%)
- $H$: Pump head (m)
- $Q$: Volumetric flow rate (m$^3$/s)
- $Q_d$: Rated flow rate (m$^3$/h)
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- $r$: Density (kg/m$^3$)
- $b_1$: Blade inlet angle (°)
- $b_2$: Blade outlet angle (°)
- $\theta$: Angle (°)
- $\eta$: Hydraulic efficiency (%)

**Abbreviations**

- CFD: Computational fluid dynamics
- OLHS: Optimal Latin hypercube sampling
- DOE: Design of experiment
- RSM: Response Surface Methodology
- MIGA: Multi-island genetic algorithm