OPTIMIZATION OF HYDRAULIC PERFORMANCE OF AUTOMOBILE COOLING PUMP BASED ON ORTHOGONAL TEST

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ABSTRACT: *In this paper, an automobile cooling water pump with poor hydraulic performance is taken as the research objective. Orthogonal test optimization design method is adopted to determine the orthogonal design table of L9 (33) and design 9 sets of orthogonal test impeller model schemes. Using ProE software to the original model, 9 groups of test model, the optimized model of 3D modeling, impeller watershed model is set up, and then use pumplinx software of CFD numerical simulation calculation, and analyzes the results of simulation calculation, the geometric parameters of three kinds of factors affect the performance of hydraulic pump, and determine the final optimal combination. The test results show that the head of optimized model has an increase of 6.364 meters and the efficiency is increased by 0.02% than the original model. The excellent optimization results are obtained, and the feasibility of orthogonal test applied to the optimal design of pump hydraulic performance is verified.*

KEYWORDS: *Automotive water pump; Orthogonal test optimization; ProE 3D modeling; CFD numerical simulation technology; pumplinx*

1. INTRODUCTION

As the core part of the car engine cooling system, the car cooling water pump cools the engine by increasing the working pressure of the coolant in the car cooling circulation system and maintaining the cooling cycle of the system.(Li and Li and Shi,2016)

Due to the high working environment temperature of the automotive cooling water pump, compact working space, high speed, high stability and long life requirements, the design of the automotive water pump has higher requirements.(Noon and Kim,2016)

The relevant performance parameters of the engine cooling water pump studied in this paper are: speed n=6698r/min, flow rate $Q=160.5L/min=0.002675m^3/s$, head H = 11.6m, efficiency η=41%.The pump assembly is shown in Figure 1, and the assembly exploded view is shown in Figure 2.

Considering the compact structure of the pump, the large number of assembly parts, the complicated structure and the difficulty of hydraulic design, combined with the optimization direction of increasing head and improving efficiency.(Bing and Cao,2013; Wang and Li and Qi,2015)This paper intends to adopt the orthogonal test optimization design method, combined with CFD(computational fluid dynamics) simulation technology, through multi-plan impeller parameter optimization test analysis, the optimal combination scheme is found, and then CFD simulation are verified.(Wei and Wang,2017;Mittag and Gabi,2015).

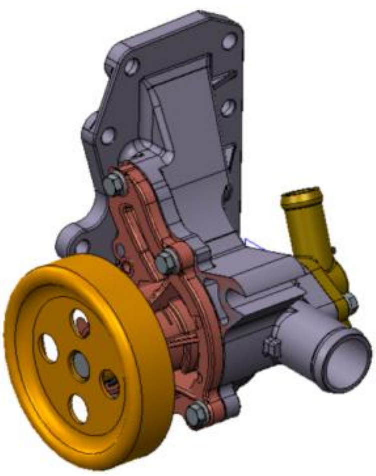

Fig. 1 Water pump assembly drawing

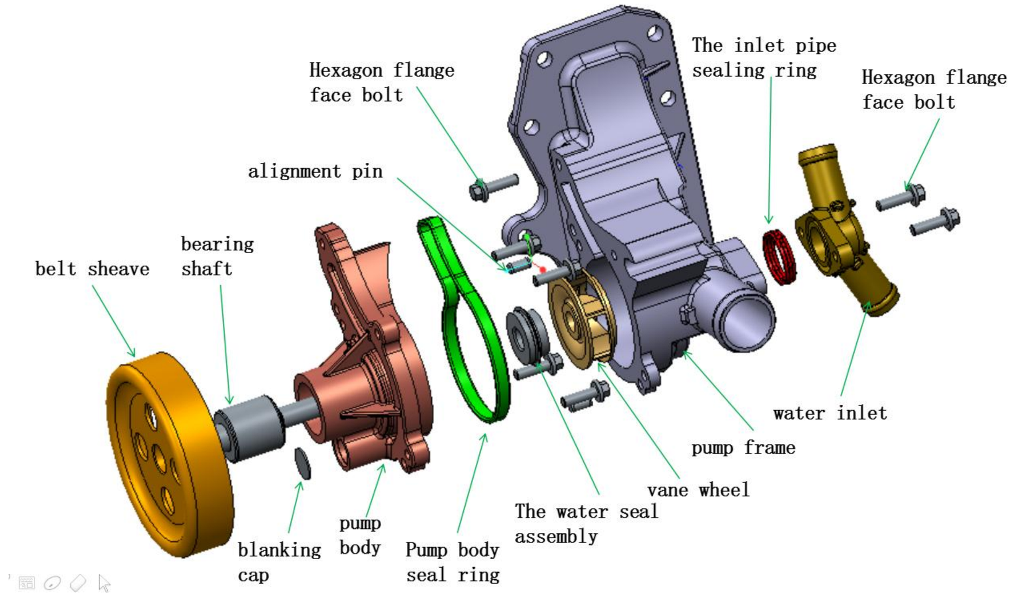

Fig.2 Explosive drawing of water pump assembly

2. ORTHOGONAL TEST DESIGN

According to the previous design experience, the design of the blade outlet angle β_2, the blade outlet width b_2, the blade number Z, the impeller outer diameter D_2, the inlet diameter D_j, and the blade wrap angle φ during the design process all have an influence on the hydraulic performance of the cooling water pump.(Yuan and Zhang and Zhang,2014)Here, only the three factors that have a great influence, β_2,b_2,Z are selected as the research objects, and the L_9 (3^3) three-factor and three-level orthogonal design test is carried out, and a total of nine experimental schemes are performed.

2.1 Original model analysis

1) The blade outlet width $\mathbf{b_2}$

Calculate the specific speed according to the original pump parameters $\mathbf{n_s}$:

$$n_s = \frac{3.65\, n\sqrt{Q}}{H^{3/4}} = \frac{3.65 \times 6698 \times \sqrt{0.002675}}{11.6^{3/4}} = 201.2 \quad (1)$$

Calculate the blade outlet width by the following formula $\mathbf{b_2}$:

$$b_2 = k_b \sqrt[3]{\frac{Q}{n}} \quad (2)$$

$$k_b = 0.64\, k_{b2} \left(\frac{n_s}{100}\right)^{5/6} \quad (3)$$

Where k_{b2} is the b_2 correction factor, refer to the relationship table of single-stage pump k_{b2} and specific speed in the technical manual.(Guan,2011)The data of the cut-out part table is shown in Table 1. Combined with the specific speed of the pump n_s=201.2, kb_2=0.88.

Table 1. Relationship between kb2 and specific speed of single-stage pump (excerpt)

$\mathbf{n_s}$	180	190	200	210	220
$\mathbf{k_{b2}}$	0.911	0.896	0.881	0.868	0.855

Substituting values into equation (3):

$$k_b = 0.64\, k_{b2} \left(\frac{n_s}{100}\right)^{5/6}$$

$$= 0.64 \times 0.88 \times \left(\frac{201.2}{100}\right)^{5/6} = 1.008 \quad (4)$$

$$b_2 = k_b \sqrt[3]{\frac{Q}{n}}$$

$$= 1.008 \times \sqrt[3]{\frac{0.002675}{6698}} = 0.0074\ m = 7.4\,mm \quad (5)$$

From the calculation results, the blade outlet width b_2=7.5mm can be determined initially.

2) the blade outlet angle β_2

The similarity algorithm is a design method often used in the design of the pump impeller.(Caiand Xiong and Fang, 2012)The model pump is designed to be equal to or close to the designed pump speed n_s, so that the size coefficient λ is determined according to the similarity of the pump, and the angle size directly selects the model pump size.(Wu,1999)The formula is as follows:

$$D = D_M \lambda \quad (6)$$

$$\beta = \beta_M \quad (7)$$

Thus determine the blade outlet angle β2 =40°

3) the blade number Z

The number of blades Z is calculated by the following formula:

$$Z = 6.5\frac{D_2 + D_1}{D_2 - D_1}\sin\left(\frac{\beta_2 + \beta_1}{2}\right) \tag{8}$$

Analytical formula (8) shows that the calculation of the number of blades is related to the diameter of the inlet and outlet and the angle of the inlet and outlet. Calculated in order:

a. Calculate the impeller outlet diameter D_2:

$$D_2 = k_D\sqrt[3]{\frac{Q}{n}} \tag{9}$$

$$k_D = 9.35\,k_{D2}\left(\frac{n_s}{100}\right)^{-1/2} \tag{10}$$

In the formula, k_{D2} is the D_2correction factor, similar to the above k_{b2}, and need to refer to the relationship table of single-stage pump k_{D2} and specific speed in the pump design technical manual,(Guan,2011)the section table data shown in Table 2. In combination with the specific speed of the pump n_s=201.2, take k_{D2}=1.006.

Table 2. Relationship between kD2 and specific speed of single-stage pump (excerpt)

n_s	180	190	200	210	220
k_{b2}	1.002	1.004	1.006	1.007	1.009

Substituting values into equation (10):

$$k_D = 9.35\,k_{D2}\left(\frac{n_s}{100}\right)^{-1/2}$$

$$= 9.35 \times 1.006 \times \left(\frac{201.2}{100}\right)^{-1/2} = 6.63 \tag{11}$$

$$D_2 = k_D\sqrt[3]{\frac{Q}{n}}$$

$$= 6.63 \times \sqrt[3]{\frac{0.002675}{6698}} = 0.049\ m = 49\ mm \tag{12}$$

From the calculation results, take the impeller outlet diameter D_2=50mm

b. Calculate the impeller inlet diameter D_1:

$$D_j = \sqrt{(D_0^{\,2} + d_h^{\,2})} \tag{13}$$

Where the hub diameter d_h is determined by the structure d_h=20mm=0.02m

$$D_0 = K_0\sqrt[3]{\frac{Q}{n}} \tag{14}$$

Among them, k_0 is the inlet correction coefficient. For most pumps k_0=3.5~4.0, further increase of k_0 will improve the cavitation performance of the pump and improve the working conditions under high flow. Select k_0=4.0 here.

Substituting values into equation (14):

$$D_0 = K_0\sqrt[3]{\frac{Q}{n}} = 4 \times \sqrt[3]{\frac{0.002675}{6698}} = 0.029\ m = 29\ mm \tag{15}$$

$$D_j = \sqrt{(D_0^{\,2} + d_h^{\,2})} = \sqrt{(0.029^2 + 0.02^2)}$$

$$= 0.037\ m = 37\ mm \tag{16}$$

$$D_1 = K_1 D_j = 0.8 \times 37\ mm = 29.6\,mm \tag{17}$$

From the calculation results, take the impeller inlet diameter D_1=29mm

In summary, the diameter of the inlet and outlet are D_1=29mm, D_2=50mm,and the inlet and outlet angles are β_1=15°, β_2=40°, respectively.

Substituting values into equation (8):

$$Z = 6.5\frac{D_2 + D_1}{D_2 - D_1}\sin\left(\frac{\beta_2 + \beta_1}{2}\right)$$

$$= 6.5 \times \frac{50 + 29}{50 - 29}\sin\left(\frac{40° + 15°}{2}\right) = 9.29 \tag{18}$$

From the calculation results, take the number of blades Z = 7.

In summary, it is determined that the impeller parameters of the original model are the blade outlet width b_2=7.5 mm, the blade outlet angle β_2=40°, and the number of blades Z=7, thereby determining the original impeller model. A simplified 2D map of only the dimensions studied is shown in Figure 3, and a 3D image is shown in Figure 4.

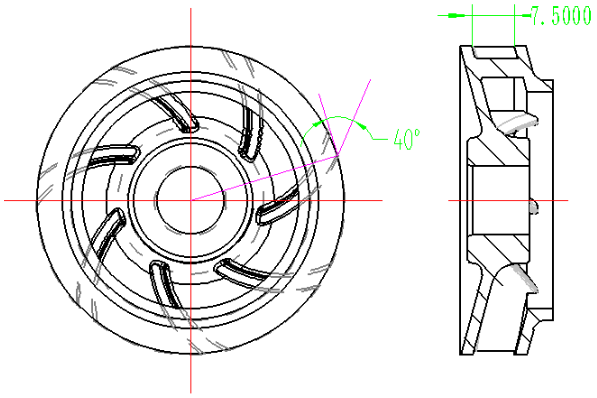

Fig. 3 Simplified 2D drawing of the original model impeller

Fig. 4 3D drawing of the original impeller model

After determining the impeller parameters, extract the original model computational domain as shown in Figure 5. The original impeller computational domain is shown in Figure 6. The two are assembled to obtain the overall computational domain used for CFD simulation analysis, and the 2D cross-sectional view of Figure 7 is used to explain the assembly relationship between the two internal watersheds.

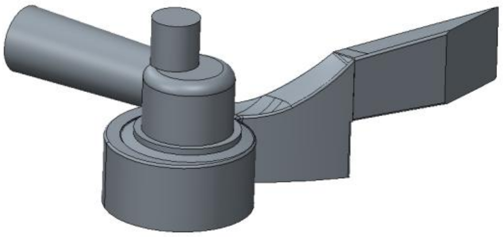

Fig. 5 The computational domain of Original model

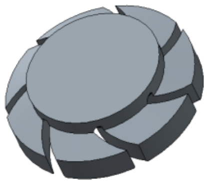

Fig. 6 The computational domain of the Original impeller model

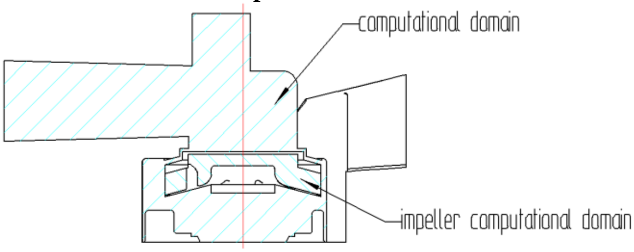

Fig. 7 The 2D sectional view of the total computational domain

After obtaining the total computational domain of the original impeller model, it was imported into PumpLinx software for CFD simulation analysis. PumpLinx software is hydraulic simulation CFD software developed by American Simerics for various types of pumps. PumpLinx is the core of a powerful CFD solver that solves compressible and incompressible fluid flows.(Ren and Jiang and Wang,2013)It is suitable for CFD simulation analysis of pump performance in this paper.

In the PumpLinx software, the computational domain is surface-split, set the interaction surface, set the inlet and outlet surface, mesh the model, set the boundary conditions, set the liquid performance, perform iterative calculation, and analyze the calculation result cloud diagram and curve data as shown in Figure 8.

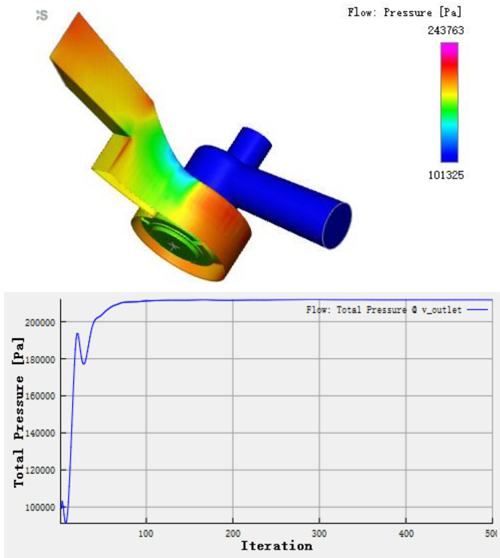

Fig. 8 PumpLinx total pressure cloud diagram of simulation results and total pressure curve at the outlet

The software simulation result data is extracted, and the average value of the final stable outlet pressure is calculated as P_2=211857.3Pa. The mean value of the final set of stable data impeller power is calculated as P=704.489W. The above two data are used to calculate the head H and efficiency η of the original model:

$$H = \frac{\Delta P}{\rho g} = \frac{P_2 - P_1}{\rho g} = \frac{211857.3 - 101325}{971.8 \times 9.8} = 11.606 \ m \tag{19}$$

$$\eta = \frac{\Delta P \times n}{P} = \frac{(211857.3 - 101325) \times 0.002675}{704.489} = 0.41 \tag{20}$$

2.2 Orthogonal table design

After analyzing the hydraulic performance of the original impeller model, determine the optimal direction for increasing the head and improve efficiency, Three factors the blade outlet angle β_2, the blade outlet width b_2 and the blade number Z which have a great influence on hydraulic performance, are selected as research objects. The factor level table is shown in Table 3.

Table 3. Factor level table

	Influencing factor A	Influencing factor B	Influencing factor C
	Number of blades Z	Outlet width b_2/mm	Outlet angle β_2/°
Level 1	6	7	40
Level 2	7	8.5	60
Level 3	8	10	80

Perform $L_9(3^3)$three-factor and three-level orthogonal design test, reasonably allocate nine sets of test plan orthogonal tables, as shown in Table 4, and sequentially perform the CFD simulation and analysis operations as described above, and record the test results into the table 4.

Table 4.L9(33) Orthogonal Test Plan and Test Results

Test number	**Factor**			**Lift Head (H/m)**	**Efficiency (η/%)**
	Number of blades Z	Outlet width b_2/mm	Outlet angle β_2/°		
1	1(6)	1(7)	1(40)	9.61	0.381
2	2(7)	1(7)	2(60)	12.98	0.417
3	3(8)	1(7)	3(80)	15.28	0.417
4	1(6)	2(8.5)	3(80)	13.46	0.390
5	2(7)	2(8.5)	1(40)	12.90	0.434
6	3(8)	2(8.5)	2(60)	14.91	0.429
7	1(6)	3(10)	2(60)	15.12	0.438
8	2(7)	3(10)	3(80)	17.10	0.451
9	3(8)	3(10)	1(40)	15.90	0.486

2.3 Test result data analysis

Orthogonal test method can be widely used in practice, not only because of its uniform and reasonable test distribution, but also because of its scientific method of analyzing test results and data, which clearly leads to many valuable conclusions.(Tang and Gong and Su,2016;Guo and Parsa,2008)The following two methods are used here to integrate the test results.

2.3.1 Range analysis

The range analysis method obtains the average range of each factor through the range calculation. According to the size comparison of the corresponding average range of each factor, the order of the influence of each factor on the test index within the test range is analyzed, so as to confirm the optimal factor level combination. Taking the horizontal of each factor as the abscissa and the average range of the test index as the ordinate, the relation line graph of each factor and the test index was drawn, and the influence of each factor on the test index was more intuitively seen, so the conclusion was further discussed.(Tang and Xu,2017)

Because the test has two test indicators of head and efficiency, it is necessary to perform two range analysis calculations for the two indicators separately. The results of the range analysis data for the head indicators are shown in Table 5. The relationship between each factor and the head indicator is shown in line chart Figure 9.

Combined with the range analysis and line graph analysis, we can get the conclusion that the biggest factor affecting the head index is the outlet width b_2, the optimal level is B3 (10mm). The second influencing factor is the number of blades, and the optimal level is A3 (8). The outlet angle β_2 has the least influence, and the optimal level is C3 (80°), and the degree of influence is ranked from big to small as B> A> C.

Table 5. Test result calculation (Lift head)

Parameter	**Lift Head (H/m)**		
	Number of blades Z	**Outlet width b_2/mm**	**Outlet angle β_2/°**
T1	38.18	37.86	38.39
T2	42.98	41.27	43.01
T3	46.08	48.11	45.83
t1	7.64	7.57	7.68
t2	8.60	8.25	9.17
t3	9.22	9.62	8.60
Range	1.58	2.05	1.49

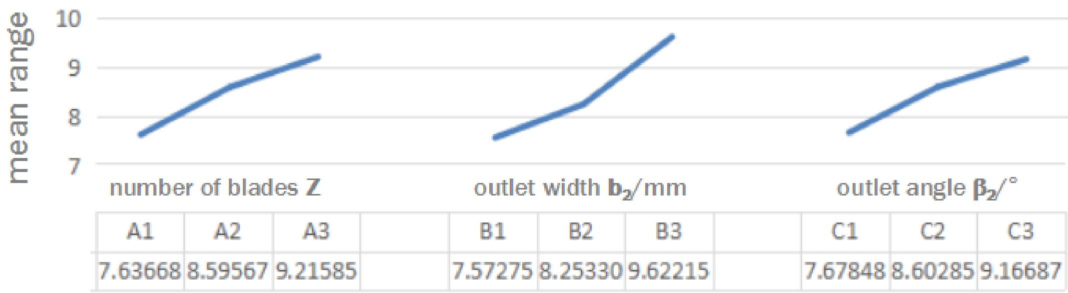

Fig. 9 The relationship between the level of each factor and the Lift head

Next, the data is analyzed for the efficiency indicators. The results of the analysis of the efficiency of the efficiency indicators are shown in Table 6. The relationship between the factors and the efficiency indicators is shown in Figure 10.

Table 6. Test result calculation (Efficiency)

Parameter	Efficiency(η/%)		
	Number of blades Z	**Outlet width b_2/mm**	**Outlet angle β_2/°**
T1	1.21	1.22	1.30
T2	1.30	1.37	1.28
T3	1.33	1.25	1.26
t1	0.24	0.24	0.26
t2	0.26	0.25	0.26
t3	0.27	0.27	0.25
Range	0.02	0.03	0.0086

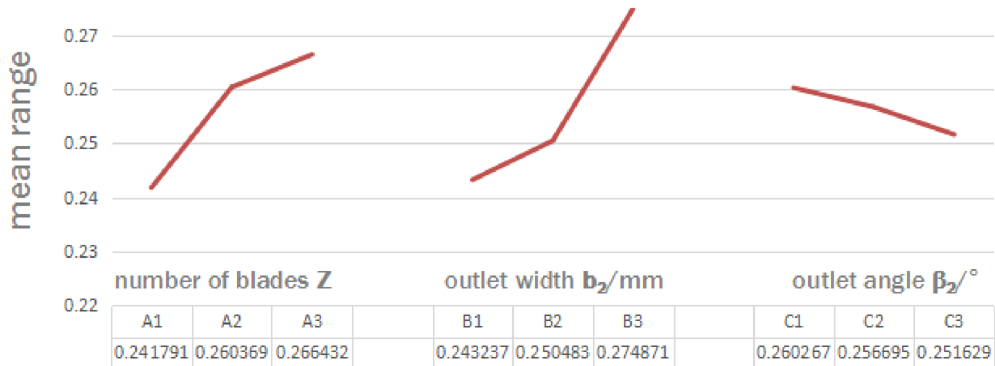

Fig. 10 The relationship between the level of each factor and the efficiency index

Combined with the range analysis and line graph analysis for the efficiency index data, the conclusion can be drawn that the factor with the greatest impact on efficiency is the outlet width b_2, the optimal level is B3 (10mm).The second influencing factor is the number of blades, and the optimal level is A3 (8). The outlet angle β_2 has the least influence, and the optimal level is C1 (40°), and the degree of influence is ranked from big to small as B> A> C.

According to the range analysis of the test data, can initially draw conclusions that the influence of three factors on the hydraulic performance of the pump is ranked from B>A>C to large and small. When considering the impact on the head, the optimal combination is A3B3C3.When only consider the impact on efficiency, the optimal combination is A3B3C1,but due to the range of the outlet angle on the head, R_1 = 1.49, and the range in efficiency is R_2 = 0.0086,which is negligible in terms of efficiency. The optimal level is C3, and the tentative optimal combination is A3B3C3.

2.3.2 Variance analysis

Variance analysis method is a method to analyze and evaluate the reasons for the differences between data according to the test data, so as to judge the degree of influence of various related factors on the test results, and give the order of influence, which provides a reliable basis for determining the further direction of the test.(Wang,2015)

As with the range analysis, it is also necessary to analyze the two assessment indicators separately. First analyze the data of the head indicator. According to the main calculation formula of variance analysis:

$$W = \sum_{i=1}^{p}\sum_{j=1}^{r} x_{ij}^2 = 1837.51 \quad (21)$$

$$P = \frac{1}{pr}\left(\sum_{i=1}^{p}\sum_{j=1}^{r} x_{ij}\right)^2 = 1798.92 \quad (22)$$

$$Q_A = \frac{1}{r}\sum_{i=1}^{p}\left(\sum_{j=1}^{r} x_{ij}\right)^2 = 1809.47 \quad (23)$$

Then: Total variation (reflecting the difference between the results of the whole test):

$$S_T=W-P=38.59 \quad (24)$$

condition variation (reflecting differences in levels of factor A):

$$S_A=Q_A-P=10.55 \quad (25)$$

Error (reflecting differences within the same level):

$$S_E=W-Q=S_T-S_A=0.47 \quad (26)$$

Total degrees of freedom:

$$f_T=pr-1=n-1=9-1=8 \quad (27)$$

Degree of freedom of condition variation:

$$f_A=f_B=f_C=p-1=2 \quad (28)$$

Degree of freedom of error:

$$f_E=f_T-f_A=n-p=9-3=6 \quad (29)$$

Calculate the mean square and:

$$V_A=S_A/f_A=5.28 \quad (30)$$

$$V_E=S_E/f_E=0.23 \quad (31)$$

The F_0 value is calculated and compared to $F_\alpha(f_A,f_E)$ to determine whether each level of factor A has a significant effect on the test results.

$$F_0 = \frac{V_A}{V_E} =22.60 \quad (32)$$

When the F value is compared with the critical value, it can be divided into four cases:

1) $F > F_{0.01}(f_A,f_E)$,indicating that factor A is very significant;

2) $F_{0.01}(f_A,f_E) \geq F \geq F_{0.05}(f_A,f_E)$, it means that factor A has a significant influence on the evaluation index;

3) $F_{0.05}(f_A,f_E) \geq F \geq F_{0.10}(f_A,f_E)$, it means that factor A has a certain influence;

4) $F < F_{0.10}(f_A,f_E)$, it means that the factor A is not affected.

For the B and C factors, the calculation is as above, and all calculation results are listed in the Table 7 Variance analysis table, and conduct analysis and evaluation.

Table 7.Variance analysis table for Lift Head

Source of variance	sum of square	Degree of freedom	Mean square	F value		Threshold
A	10.55	2	5.28	22.60	Significantly affected	$F_{0.01}(2,2)=99.0$
B	18.16	2	9.08	38.89	Significantly affected	$F_{0.05}(2,2)=19.0$
C	9.41	2	4.71	20.16	Significantly affected	$F_{0.10}(2,2)=9.0$
Error E	0.47	2	0.23			
Sum T	38.59	8				

According to Table 7, it can be concluded that the degree of influence of the three factors on the head is ranked from big to small as B> A> C, and all three have significant effects.

The variance of the efficiency index is calculated in the same way, and the calculation results are listed in the variance analysis table in Table 8.

According to Table 8, it can be concluded that the degree of influence of the three factors on the efficiency is ranked from big to small as B> A> C, and the B factor has a significant effects, the A factor has a certain impact, and the C factor has a very small impact.

Comparing the analysis results of the two analytical methods, consistent with the results of the range analysis, the degree of influence of the three factors on the hydraulic performance of the pump is ranked from big to small as B> A> C, and the optimal combination is A3B3C3.

Table 8.Variance analysis table for Efficiency

Source of variance	sum of square	Degree of freedom	Mean square	F value		Threshold
A	0.0027	2	0.0014	13.23	Have a certain influence	$F_{0.01}(2,2)=99.0$
B	0.0046	2	0.0023	22.04	Significantly affected	$F_{0.05}(2,2)=19.0$
C	0.0003	2	0.0002	1.51	Can't see the impact	$F_{0.10}(2,2)=9.0$
Error E	0.0002	2	0.0001			
Sum T	0.0078	8				

3. COMPARED OPTIMIZED WITH ORTHOGONAL MODEL

3.1 Optimized model analysis

According to orthogonal test analysis above, the impeller parameters of the optimized model are determined as the blade outlet width b_2=10mm, the blade outlet angle β_2=80°, and the number of blades Z=8. The optimized impeller model was established and imported into pumplinx software for CFD simulation test. A simplified 2D drawing with only the dimensions studied is shown in Figure 11. The 3D drawing is shown in Figure 12.

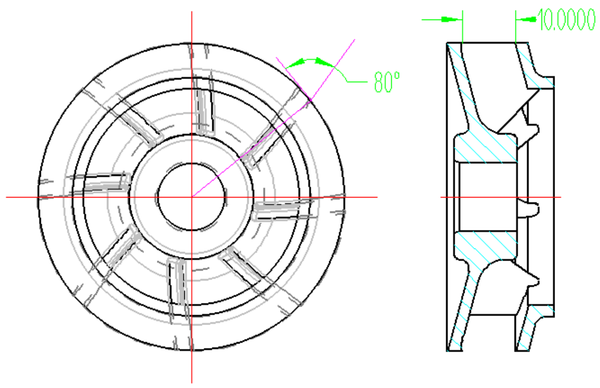

Fig. 11 Simplified 2D drawing of the impeller model after optimization

Fig. 12 3D drawing of the impeller model after optimization

After determining the impeller parameters, extract the optimized impeller computational domain as shown in Figure 13, and assemble it with the pump computational domain to obtain the overall computational domain for CFD simulation analysis. The assembly relationship between the two in the overall computational domain is the same as the original model, and will not be repeated here.

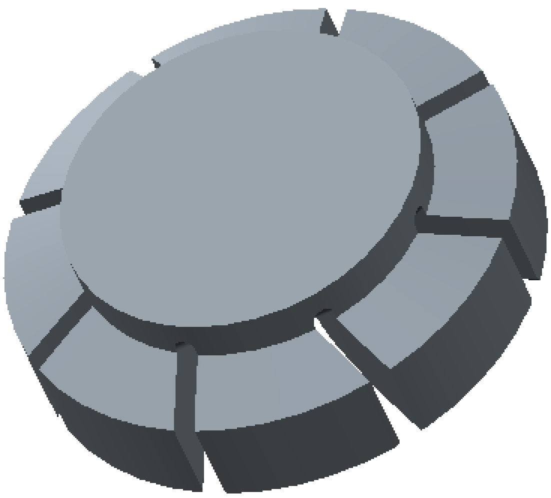

Fig. 13 The computational domain of the impeller model after optimization

Similar to the original model analysis process, the optimized total computational domain is imported into pumplinx software for CFD simulation analysis. The boundary parameters of the simulation process are the same as those in Chapter 2, and the simulation results are shown in Figure 14.

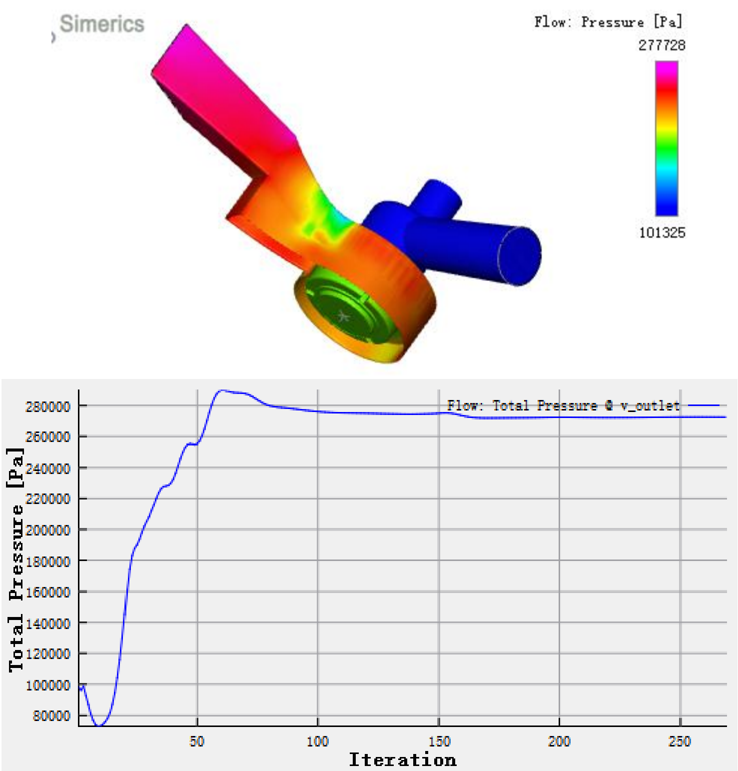

Fig. 14 Optimized total pressure nephogram and outlet total pressure curve

The software simulation result data is extracted, and the average value of the final stable outlet pressure is calculated as P_2=272509.9Pa.

Extract the software simulation result data and calculate the average value of the final stable outlet pressure as P_2=272509.9Pa. The average value of the final set of stable data impeller power is calculated to be P=1060.526W. The above two data are used to calculate the head H and efficiency η:

$$H = \frac{\Delta P}{\rho g} = \frac{P_2 - P_1}{\rho g} = \frac{272509.9 - 101325}{971.8 \times 9.8} = 17.97\ m \tag{33}$$

$$\eta = \frac{\Delta P \times n}{P} = \frac{(272509.9 - 101325) \times 0.002675}{1060.526} = 0.43 \tag{34}$$

3.2 Comparison of CFD simulation results

The original and optimized export pressure curve data are collected and summarized in the same 2D line chart, as shown in Figure 15.The parameters of the test results before and after optimization are compared, as shown in Table 9.

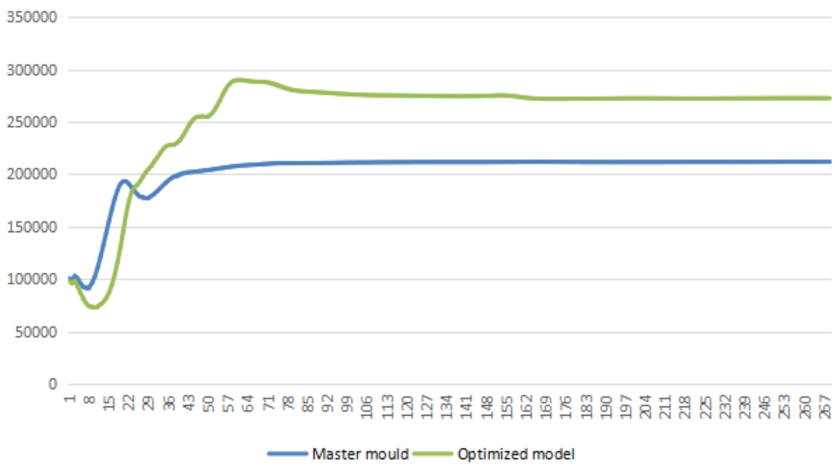

Fig. 15 Comparison chart of outlet pressure curves before and after optimization

Table 9.Comparison of Parameters before and after Optimization

	Original	Optimized
Outlet Pressure P_2(Pa)	211857.3	272509.9
Impeller power P(W)	704.489	1060.526
Lift Head H(m)	11.606	17.97
Efficiency η	0.41	0.43

4. CONCLUSION

1) In this orthogonal experiment, the degree of influence of three factors on efficiency is ranked from the largest to the smallest, the blade outlet width, the number of blades, and the blade outlet angle. The order of influence on the head is the same as the efficiency.

2) By analyzing the results of orthogonal test data by range analysis and variance analysis, the optimal combination impeller parameters are blade outlet width b_2=10mm, blade outlet angle β_2=80°, and blade number Z = 8.

3) Using ProE software to establish the optimized 3D model. Compare the results of CFD numerical simulation analysis with the results of the original model data. The optimized model is 6.364m higher than the original model, and the efficiency is improved by 0.02%. 4) Excellent optimization results were obtained, and the feasibility of orthogonal test applied to the optimal design of pump hydraulic performance was verified.

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