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OPTIMIZATION DESIGN OF NEW-TYPE DEEP WELL PUMP BASED ON LATIN SQUARE TEST AND NUMERICAL SIMULATION

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ABSTRACT

In order to develop high efficiency and high head deep well pump of 150QJ20 type, a L_{18}（3^7）orthogonal experiment was performed with seven factors and three values including blades numbers, outlet angle, outlet width, etc.18 impellers were designed. The whole flow field of new-type two-stage deep well pump at the operating point for design was simulated by FLUENT using the standard model, SIMPLEC algorithm, second-order upwind scheme to solve, and analyze the independent of the number of the grid. 18 groups of the efficiency and head in design scheme were obtained. The effects of geometrical parameters on efficiency, head were researched using Latin square test method. The primary and secondary factors of the design parameters were acquired by way of variance analysis. According to the test result, an optimum program to further design was put forward. After manufactured and tested, the final optimal design model pump flow at rated efficiency of 66.59% point, single-stage head of 10.9m, match the motor as 5.5 kW, compared to the Chinese national standards（GB/T 2816-2002), which the rated flow point of the efficiency of 64% and matching motor 7.5 kW, the efficiency and head were significantly improved. The productions show good energy saving and material saving characters and can replace traditional pumps for deep well in the future, the comprehensive technical indicators achieve international advanced levels. The results would be instructive to the design of new-type deep well pump with the impeller head maximum approach.

INTRODUCTION

Due to the working conditions, the diameter of the submersible pump for deep well is limited by the well diameter when it works in the motor-pumped well, and the single-stage head can't be efficiently increased in the limited space with the traditional design methods. Therefore, the maximum diameter design method of impeller was created for the first time [Ref1, Ref2]. We have exploited the 100SJ8 submersible pump for deep well successfully with this method. The theoretical analysis, numerical simulation and experimental investigation show that the single-stage head of the deep-well pump improves greatly and the efficiency is also higher than before. This method is highly superior to the traditional design method in the hydraulic design of the deep-well pump.

According to the requirements of orthogonal test, 18 impeller of 150QJ20 new-type deep well submersible pump were designed with the maximum diameter method. For the manufacturing process of 18 groups requires high accuracy and long time, and the error was also inevitable in prototype test, two different models of new deep-well pump were simulated and tested. Comparing test to simulation results, the simulation efficiency at rated flow was higher than that of the prototype test by about 5%. The deviation was stable and the trends are similar. Then the impellers in 18 groups matching with distorted-reversed guide vane were simulated. The results were analyzed to explore the effects of the main geometric parameters on the head and efficiency. Finally, the optimal design program was presented.

NOMENCLATURE

Q_d Flow rate (m^3/s)
H Head (m)
n Rotation speed (r/min)
Q Flow (m^3/s)
n_s Specific speed= $\frac{3.65n\sqrt{Q}}{H^{3/4}} = \frac{3.65.r/\min\sqrt{m^3/s}}{m^{3/4}}$
P_i Total pressure at inlet (Pa)
P_0 Total pressure at outlet (Pa)
M Moment of force ($N \cdot m$)
η Efficiency (%)

THE ORTHOGONAL EXPERIMENTAL STUDY

Orthogonal test is a mathematical method used to arrange, calculate and analyze the experimental results in a standardized orthogonal table that has been made by principle of orthogonal, see reference [Ref3].

The Test Purpose

(1)Exploring the influence of the new deep-well pump geometric parameters on the efficiency and head at the nominal point.

(2)Choosing the best design program for the deep-well pump with the flow rate Q=20m^3/h, the head H=65m, the speed n=2850r/min, the specific speed n_s= 128.

Determination of the test factors and programs

The factors that affect the efficiency and head are mainly Z、β_2、D_2、b_2、u_2 etc. Based on professional knowledge and special design requirements of the new-type deep well, the following geometric parameters were taken into consideration: β_2、D_{2min} (the outside diameter of impeller back shroud), b_2、Z、D_1 (the diameter of the blade inlet), $\Delta\beta_1$ (the angle of the blade inlet), h (the axial spacing between the impeller back shroud and the bottom of the reversed guide vane),are shown in Fig.1. The influence law of the above factors on the efficiency and head can be obtained according the following test. Table 1 is the level of these factors. L_{18}（3^7）orthogonal table was selected and the test programs are determined in table 2.

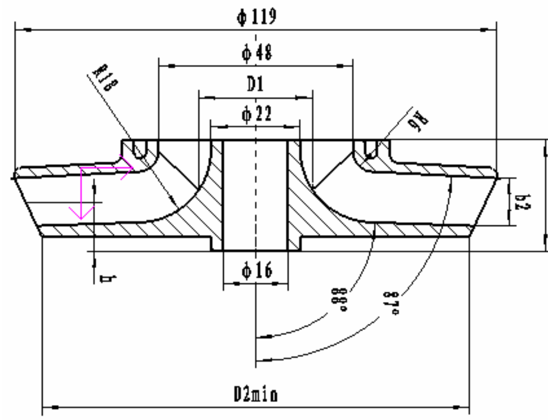


Fig.1 New well pump impeller

HYDRAULIC DESIGN

The design parameters

The basic parameters of this design, Q=20m^3/h, single-stage head H=11m, speed n=2850r/min. 18 impellers were designed according to orthogonal experiment.

The maximum diameter design method of impeller

To the 150QJ-type deep well pump, permitted outer diameter is 143mm and inner diameter 121mm according to the national standard. Supposing the diameter of the impeller front shroud to be 119mm, the impeller diameter can achieve maximum. This design of other parameters can be found in [Ref4].

Twisted return guide vane design method

The guide vane designed with twisted side of guide vane anti-missile imported leaf design method, face sub-3 flow line basis, the final space to be smooth synthetic surfaces. Through the Pro/E modeling, impeller and guide vane model are shown in Fig. 2 and Fig. 3.

Tab. 1 Orthogonal experimental factors

Levels	A D_{2min}/mm	B b_2/mm	C β_2(°)	D Z	E D_1/mm	F $\Delta\beta_1$	G h/mm
1	110	13	15	5	21	0	1.5
2	108	12	20	6	17	6	2
3	106	11	25	7	14	12	2.5

Tab .2 Test scheme

No.	A (D_{2min})	B(b_2)	C（β_2）	D（z）	E（D_1）	F（$\Delta\beta_1$）	G（h）
1	110	13	15	5	21	0	1.5
2	110	12	20	6	17	6	2

3	110	11	25	7	14	12	2.5
4	108	13	15	6	17	12	2.5
5	108	12	20	7	14	0	1.5
6	108	11	25	5	21	6	2
7	106	13	20	5	17	6	2.5
8	106	12	25	6	21	12	1.5
9	106	11	15	7	17	0	2
10	110	13	25	7	17	6	1.5
11	110	12	15	5	14	12	2
12	110	11	20	6	21	0	2.5
13	108	13	20	7	21	12	2
14	108	12	25	5	17	0	2.5
15	108	11	15	6	14	6	1.5
16	106	13	25	6	14	0	2
17	106	12	15	7	21	6	2.5
18	106	11	20	5	17	12	1.5

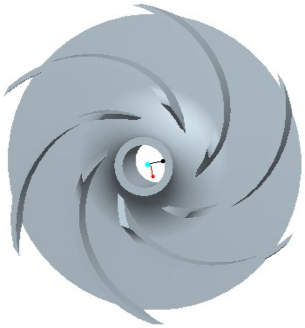


Fig. 2 Impeller without front shroud

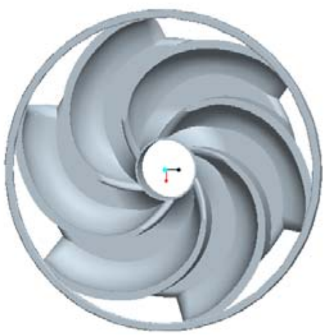


Fig. 3 Guide vane

NUMERICAL SIMULATIONS

Governing equations

The over-current components of deep well pump consist of the inlet, some stages impeller and guide vane. The relative reference system fixed to the rotor with speed 2850r/min was adopted. The whole flow field was assumed to be 3-D incompressible steady viscous turbulent flow field. The continuity equation and momentum equation, k-ε turbulence equation were introduced.

The properties of working media

The medium is water at room temperature and pressure with fixed density (ρ=998.2 kg/m^3) and dynamic viscosity (μ=0.001003kg/(m· s)).

Calculation of regional meshing and model selection

After modeled by Pro/E, the import section, impeller and guide vane were imported to the Gambit for a further processing. The two stages full-flow field was meshed with structured and unstructured grids, shown in Fig. 4 and Fig. 5. After comparing the five groups model grids between 1～2.5 million, the results showed that the efficiency fluctuates within 0.5%. We selected the grid number of 1.75 million according to the computer performance.

Fig. 4 Grid model

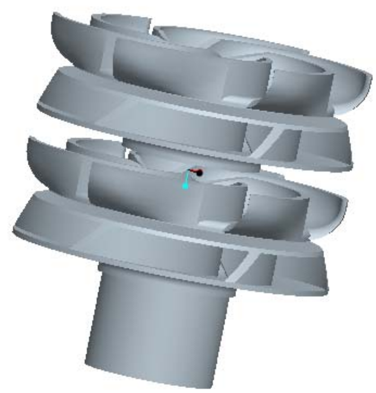


Fig. 5 3-D entity diagram

The two stages flow channel of deep well pumps were took as computing area, at the same time; the whole calculation area was divided into two parts. The first is the import section of the pump and the rotating part of the impeller chamber. The second is the distorted-reversed guide vane static area. The connection surface between the two sub-regions is the interface. The coupling between the rotator and stator was simulated using Multiple Reference Frame (MRF).

Boundary condition

Supposing that the impeller inlet is irrotational flow, inlet surface center as the pressure reference point [Ref5], the relative pressure is zero. The boundary condition of outlet is set to be outflow [Ref6]. Solid-wall is supposed to be no slip. The standard wall function was approached to the turbulent flow of near-wall.

Numerical algorithm and solution control parameters

The SIMPLEC algorithm and second-order upwind discrete difference equation were applied. The factor coefficients of sub-relaxation for algebraic equations are as follows, pressure coefficient of Asian Relaxation is 0.3, momentum sub-relaxation factor 0.7, turbulent kinetic energy sub-relaxation factor 0.8, turbulent kinetic energy dissipation rate 0.8. The convergence precision is set to be 10^{-5}.

ORTHOGONAL TEST RESULTS AND ANALYSIS

At the design point Q = 20m^3/h, numerical simulation results of the efficiency and head of 18 impellers is shown in Tab. 3. In order to evaluate the influence of the impeller's seven factors on pump performance and find the main factors and optimization solutions, orthogonal test results were analyzed, shown in Tab. 4 and Tab. 5. The primary and secondary orders of the geometric parameters were shown in tab. 6. The impact of factor A on the head was in order of $A_1A_3A_2$, on the efficiency $A_3A_1A_2$. The impact of factor B on the head was in order of $B_1B_2B_3$, on the efficiency $B_3B_2B_1$. The impact of factor C on the head was in order of $C_3C_2C_1$, on the efficiency $C_1C_2C_3$. The impact of factor D on the head is in order of $D_3D_2D_1$, on the efficiency $D_2D_3D_1$. The impact of factor E on the head was in order of $E_3E_2E_1$, on the efficiency $E_1E_2E_3$. The impact of factor F on the head was in order of $F_1F_3F_2$, on the efficiency $F_3F_2F_1$. The impact of factor G on the head was in order of $G_1G_2G_3$, on the efficiency $G_1G_3G_2$.

Tab. 3 Summary of test results

No.	1	2	3	4	5	6	7	8	9
H/m	12.25	12.87	13.65	11.83	13.72	11.69	11.91	13.6	11.79
η/%	65.7	65.6	63.03	66.88	64.58	63.92	62.7	65.36	67.63
No.	10	11	12	13	14	15	16	17	18
H/m	14.58	11.73	12.39	12.72	12.19	11.25	14.05	11.5	10.96
η/%	61.49	66.27	66.93	63.12	61.06	67.02	61.09	67.91	65.82

Tab. 4 Efficiency results analysis at the rated point

η/%	A	B	C	D	E	F	G
K_1	389.02	380.98	401.41	385.47	392.94	386.99	389.97
K_2	386.58	390.78	388.75	392.88	388.48	388.64	387.63
K_3	390.51	394.35	375.95	387.76	384.69	390.48	388.51
k_1	64.84	63.5	66.9	64.25	65.49	64.5	65
k_2	64.43	65.13	64.79	65.48	64.75	64.77	64.91
k_3	65.09	65.73	62.68	64.63	64.12	65.08	64.75
R	0.66	2.23	4.24	1.23	1.38	0.58	0.39

Tab. 5 Head results analysis at the rated point

H/m	A	B	C	D	E	F	G
K_1	77.46	77.33	70.34	70.72	74.15	76.38	76.35
K_2	73.39	75.6	74.56	75.98	74.21	73.79	74.85
K_3	73.81	71.73	79.76	77.95	76.30	74.48	73.46
k_1	12.91	12.89	11.72	11.79	12.36	12.73	12.73
k_2	12.23	12.60	12.43	12.66	12.37	12.30	12.47
k_3	12.30	11.95	13.29	12.99	12.72	12.41	12.24
R	0.68	0.93	1.57	1.21	0.36	0.43	0.48

Tab. 6 Influence order of geometric parameters on performance

			High$\rightarrow$ Low				
H	β_2	b_2	D_1	Z	$D_{2\,min}$	$\Delta\beta_1$	h
η	β_2	Z	b_2	$D_{2\,min}$	h	$\Delta\beta_1$	D_1

Tab. 7 Impeller performance parameters

Q (m^3/h)	P_i (Pa)	P_0 (Pa)	M (N.m)	P (kW)	H (m)	η (%)
12	2847	270624	4.9885	1.4884	27.32	59.97
16	4802	256614	5.694	1.6994	25.67	65.86
20	7326	232957	6.1851	1.8459	23	68.13
24	10480	197321	6.3849	1.9056	19.05	65.37
28	14179	152563	6.4653	1.929	14.12	55.79

Tab. 8 Impeller performance parameters

Q (m^3/h)	P (kW)	H (m)	η (%)
12	4.4652	81.9	59.97
16	5.0982	77.01	65.86
20	5.5377	69	68.13
24	5.7168	57.15	65.37
28	5.787	42.36	55.79

THE INNER FLOW DISTRIBUTION ANALYSIS OF OPTIMIZATION SCHEME AND EXPERIMENT

Performance prediction

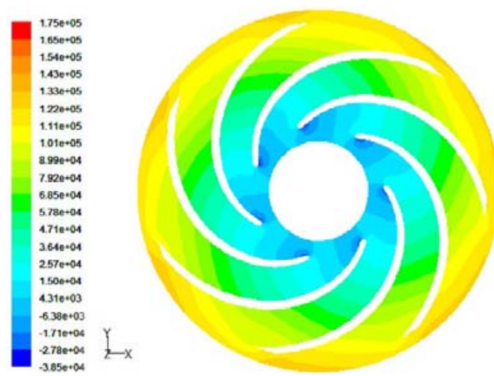
To analyze the steady flow character of multi-conditions in the optimization scheme ,the test has made the whole flow field of two-stage well pump on 5 conditions(Q=12, 16, 20, 24, 28 m^3/h) were simulated as a basis for the performance prediction. Tab. 7 shows the performance parameters of the two-stage impeller at 5 different conditions. In accordance with the efficiency of conversion, of such performance parameters of six−stage pump, such as Tab. 8.

The internal flow analyses at small, rated and large flow conditions

The whole flow field of the optimized new-type two-step well pump was simulated at the small flow 16m^3/h, rated flow 20m^3/h and large flow 24m^3/h. The total grid number is 1.7 million. The K-ε model and SIMPLEC algorithm are adopted. The surface roughness of impeller and guide vane is set to be 0.025 mm, and the axial clearance of the impeller inlet was sealed completely. The pressure distributions and velocity distributions at rated point of optimization program and two operating conditions near rated point were analyzed.

Pressure analysis

Fig.7 shows the static pressure at three operating conditions of the impeller middle section. From the figures, it can be seen that the static pressure increases uniformly from inlet to outlet of the impeller, and achieves maximum on the blade tail face. Near the inlet region, the pressure is lowest and appears negative pressure in the back of blades.



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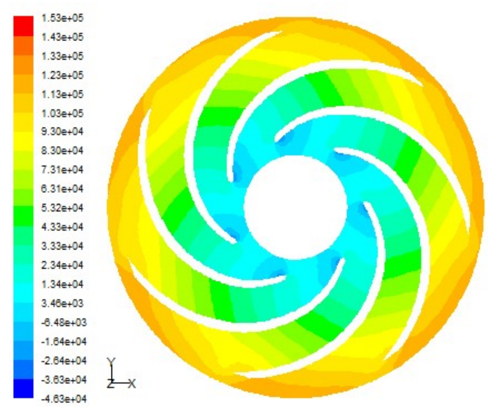

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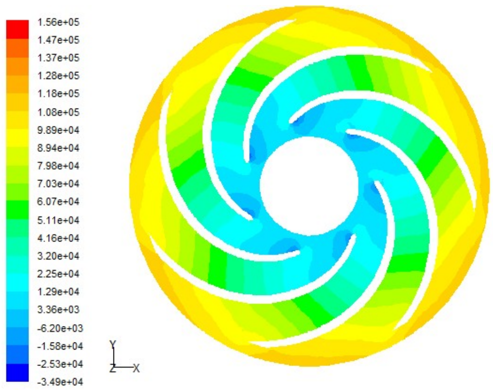
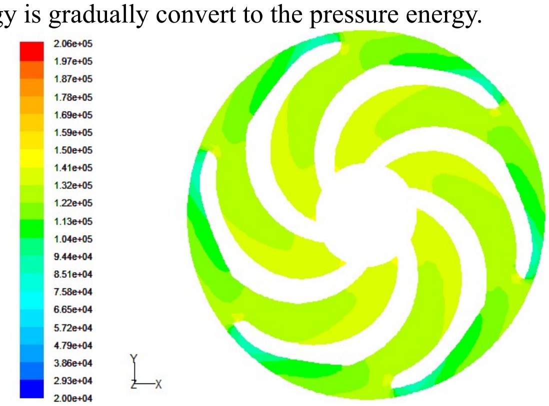


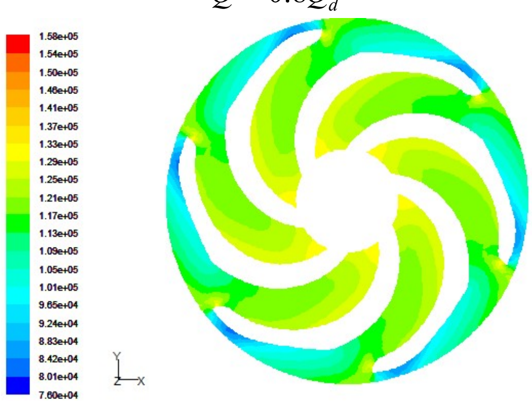
$Q = 1.2Q_d$

Fig. 6 Static pressure distributions on the middle face of impeller

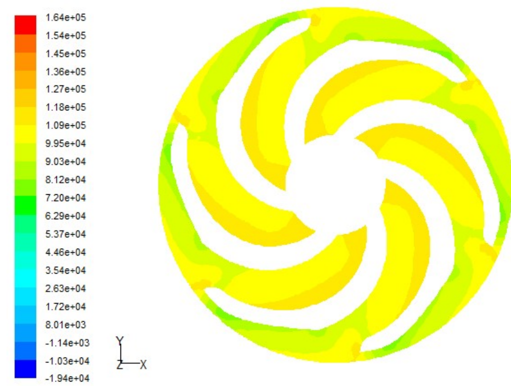
Fig. 7 shows the static pressure distributions of middle face of guide vane. It can be seen that static pressure of guide vane also increases from inlet to outlet, indicating that the kinetic energy is gradually convert to the pressure energy.



$Q = 0.8Q_d$



$Q_d = 1.0Q$

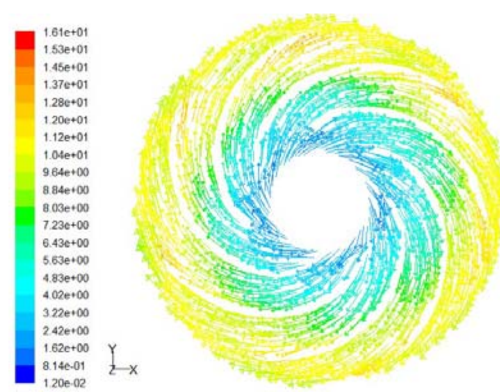


$Q = 1.2Q_d$

Fig.7 Static pressure distributions on the cross-section of guide vane

Velocity analysis

Fig. 8 shows the relative velocity vector in the cross-section of impeller. The turbulent kinetic energy distribution is shown in Fig. 9. The flow velocity in impeller distributes evenly without obvious return and whirlpool. The symmetry is obvious. The kinetic energy is relative large at the inlet, resulting in a certain impart which is not very obvious. The turbulent kinetic energy is very large at the impeller outlet and the flow, to some extent, affected by the pump shell is unstable.



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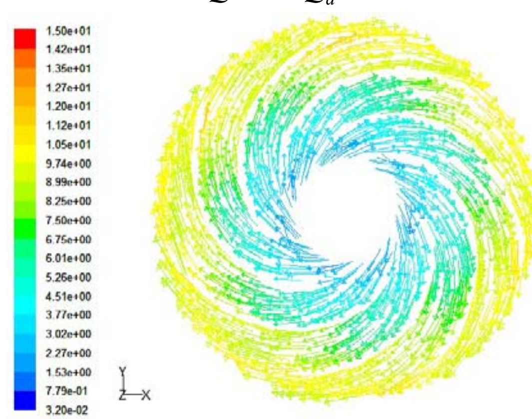

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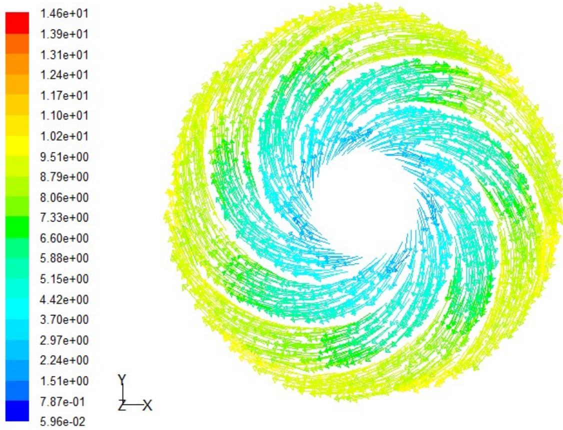
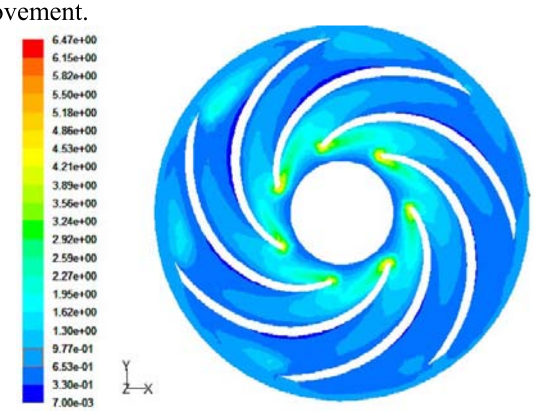


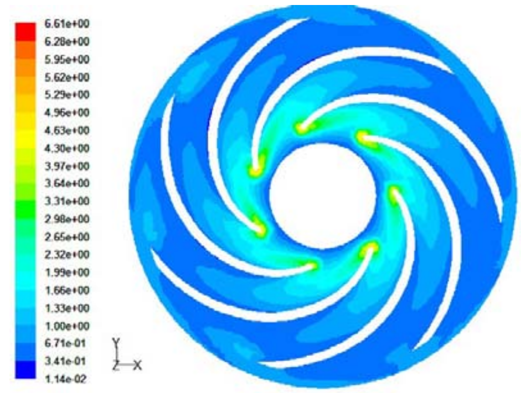
$Q = 1.2Q_d$

Fig. 8 Relative velocity distributions in the impeller

Fig. 10 shows the relative velocity vector on the cross-section of guide vane. Fig. 11 shows the turbulent kinetic energy distributions of guide vane. The flow in guide vane is stable, but at the inlet exist vortex obviously. And turbulent kinetic energy is also great. Backflow is not obvious at outlet. The turbulent kinetic energy distribution at the rated flow rate on X=0 cross-section is shown in Fig. 12. There are no obvious backflow and vortex in the whole flow passage. The velocity at impeller outlet is the greatest and decreases gradually after through the guide vane, which is in line with the actual situation. For the impeller outlet is oblique, the outlet velocity points to the back shroud and there is no significant impart on the pump wall. It shows that increasing diameter of the front shroud of impeller does not increase the impact loss at the impeller outlet. Some vortexes appear at the guide vane inlet. It indicates that the design of guide vane inlet needs a further improvement.



$Q = 0.8Q_d$



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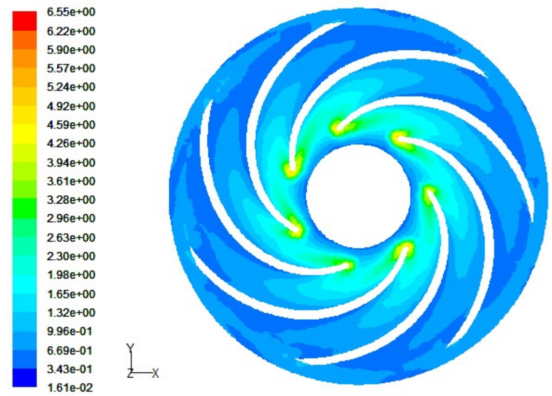
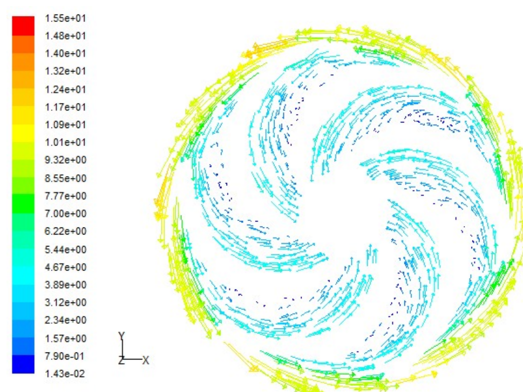


$Q = 1.2Q_d$

Fig. 9 Distributions of impeller turbulent energy



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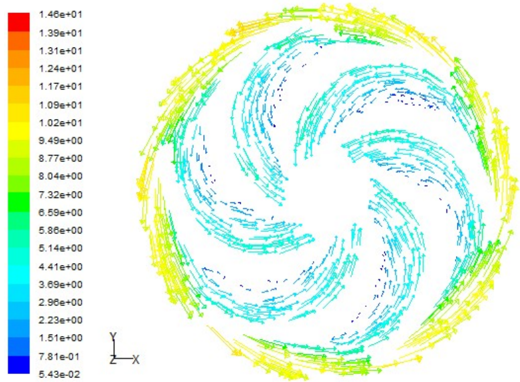
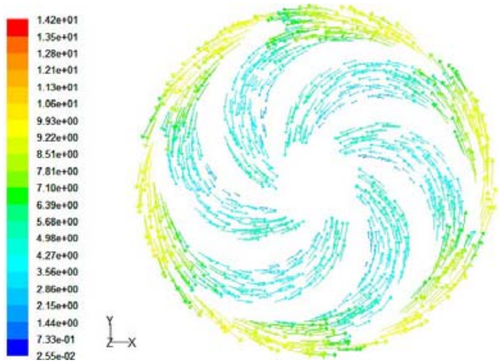
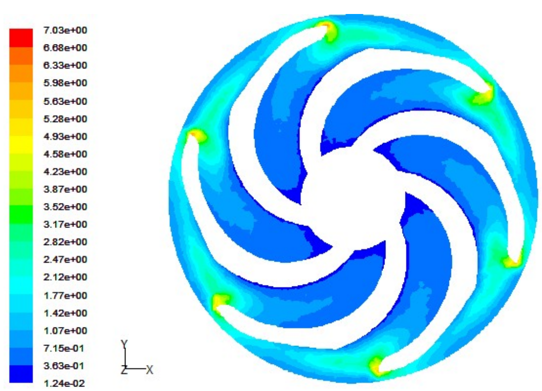


$Q = 1.0Q_d$

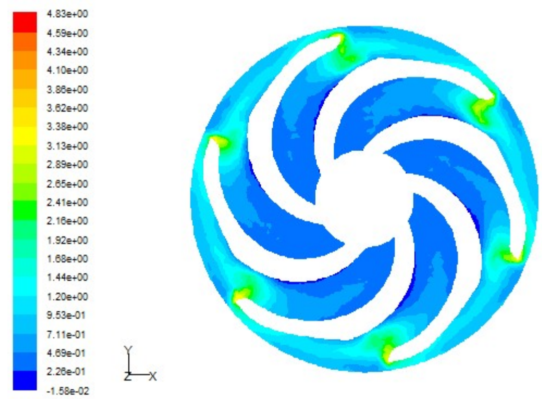


$Q = 1.2Q_d$

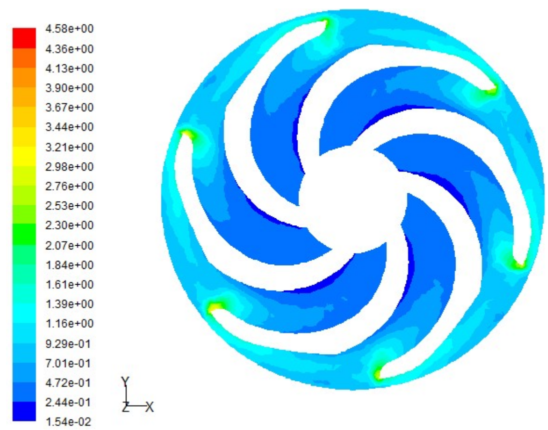
Fig. 10 Relative velocity distributions in the guide vane



$Q = 0.8Q_d$



$Q = 1.0Q_d$



$Q = 1.2Q_d$

Fig. 11 Distributions of guide vane turbulent energy

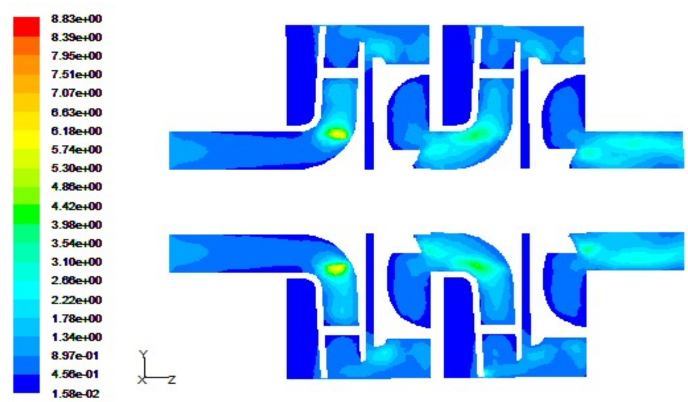


Fig. 12 Distributions of rated flow turbulent energy

OPTIMIZATION PROGRAM DESIGN SUMMARY

The internal flow analysis of performance prediction of the two-stage optimal model at small, rated and large flow conditions shows that the optimal model designed with orthogonal method has the virtues of high efficiency and stable flow. Finally the design goal of a new-type well pump hydraulic optimization was achieved. The prototype of the optimal was processed and manufactured by precision casting. The test result is shown in Tab. 9.

Tab. 9 Impeller performance parameters

NO.	Q (m^3/h)	H (m)	P (W)	η (%)
1	0	91.72	2498.71	0
2	2.55	90.81	3031.2	20.81
3	6.05	87.79	3588.29	40.33
4	10.17	81.99	4150.69	54.71
5	13.99	77.23	4776.44	61.64
6	15.99	74.86	5035.96	64.74
7	18.23	70.59	5282.97	66.35
8	19.88	66.52	5454.79	66.04
9	22.38	57.99	5362.19	65.93
10	24.19	50.73	5257.8	63.58
11	25.87	43.04	5170.02	58.68
12	28.35	31.34	4900.44	49.4
13	29.93	22.63	4719.98	39.08

Prototype test results show that the pump efficiency is 66.59% at rated point and the single-stage head is 10.9m. From the performance curve diagram, it can be seen that the new full-well pump is not overload and the maximum efficiency and maximum shaft power are taken at rated point. Compared to the national standard 7.5 KW, the shaft power of motor matched with the pump is 5.5 KW. Single-stage head has reached about 11m. It achieves energy-saving and material saving, and also meets the design requirements. In the next year, the “863 Program” project team will check the new-type well pump.

6 CONCLUSIONS

(1) The pump performance can be improved if a set of best parameter is combined. The results of orthogogonal experiment have certain reference for the hydraulic design of new-type well pump which adopts design method with a great head and performance improvement of deep-well pump.

(2) Using the orthogonal test design method, a deep research was taken, which described the influence of the main impeller geometric parameters on the new deep well pump characteristics. Eighteen design schemes were given by the L_{18} （7^3） orthogonal test method, obtaining gradation influences of the several impeller design parameters on centrifugal pump performance: β_2 (outlet angle), D_{2min} (impeller back shroud outer diameter), b_2 (outlet width), Z (blade number), D_1 (blade inlet diameter), $\Delta\beta_1$ (blade inlet incident angle) and the axial distance between impeller back shroud and the bottom of counter-guide vane. The important order of the seven parameters affecting the index of performance and the optimum of orthogonal test were also obtained through the analysis of range diagrams. And finally the new-type well pump's optimization design was realized.

(3) The new-type deep well pump was simulated using the Fluent software, forecasting its hydraulic performance. Compared with test results, it shows that the numerical simulation can well forecast its hydraulic performance, which provides a basis to the subsequent design and research.

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