

Optimization of a Centrifugal Pump Used as a Turbine Impeller By Means of an Orthogonal Test Approach

Peng Tian¹, Jun Huang¹, Weidong Shi^{1, 2, *} and Ling Zhou¹

Abstract: A prototype centrifugal pump with a specific speed of 110 is used to investigate and optimize the performances of a turbine for power generation. Particular attention is given to the design of the internal impeller. The internal flow field is simulated in the framework of a commercial computational fluid dynamics software (ANSYS). Four geometrical parameters of the impeller are considered, i.e., the inlet diameter, the inlet width, the blade number, and the blade angle. The optimization is carried out on the basis of a three-level approach relying on an orthogonal test method. The results of the numerical simulations show good agreement with the experimental tests under different flow conditions. In accordance with the L_9 (3^4) design table, the head and efficiency under the rated flow rate of the nine designed schemes are calculated and processed with the method of range analysis to obtain an optimized model.

Keywords: Pump as turbine, numerical calculation, orthogonal test, optimization design.

1 Introduction

In current small water recycling applications, a pump that works as a turbine is the main hydraulic equipment [Derakhshan and Nourbakhsh (2008); Barbarelli, Amelio, Florio et al. (2017); Ismail, Othman and Zen (2016); Arriaga (2010)]. A turbine integrates a shaft, and a motor has the advantages of a simple structure, convenient disassembly and assembly, low initial investment, little operation and maintenance cost, small space occupation, various types and specifications, large water head and flow rate, and wide application range. It is widely used in industrial liquid pressure energy recovery and development of small hydropower (less than 100 kW) [Du, Yang, Shen et al. (2017); Nautiyal, Varun, Kumar et al. (2011); Okot (2013); Elbatran, Yaakob, Ahmed et al. (2015); Singh and Nestmann (2011); Buono, Frosina, Mazzone et al. (2015); Jing and Lin (2017)].

The factors that affect the hydraulic performance of a turbine include blade wrap angle, impeller outlet diameter, and impeller outlet width. An orthogonal test is a scientific design method that is used to study a single objective that has multi-factors and multi-levels [Long, Zhu, Wang et al. (2016)]. In the orthogonal test, different factors and levels have their corresponding orthogonal chart. The best group can be found by conducting a multi-factor experiment. The orthogonal test chooses the best group that has the best

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performance and comprehensive comparability unlike all the possible groups in the chart. Through an analysis of the test results, the main parameters that affect the performance of the hydraulic turbine can be found, and a more reliable optimization model is designed.

At present, the turbine mostly adopts the reverse form of centrifugal pump and its performance is unpredictable [Raverdy, Mary, Sagaut et al. (2015); Giosio, Henderson, Walker et al. (2015); Wang, Kong, He et al. (2013)]. Thus, in terms of the prediction of the turbine's performance, domestic and foreign scholars usually infer the expression of the turbine's performance, which is based on pump's efficiency or specific speed, by using a test or theoretical method [Fecarotta, Carravetta and Ramos (2011); Sangal, Garg and Kumar (2013); Wang, Wang, Kong et al. (2017); Wang, Kong, Xia et al. (2017); Su, Huang, Zhang et al. (2016)]. Compared with the normal operating condition, the efficiency of the pump as a turbine is relatively low, and its working stability is deteriorated. To improve the performance of the turbine, Yang [Yang (2011)] optimized the geometry of the blades. Sheng, Singh et al. [Singh and Nestmann (2010)], Derakhshan et al. [Derakhshan and Kasaeian (2012); Yang, Derakhshan and Kong] proposed that rounding the turbine blade inlet edge and trimming the outer edge of the shroud and hub can increase the turbine efficiency by 1% to 4%. The current results of numerical simulations remain to be improved because of the complexity and variety of the turbine's geometry structures.

2 Physical and numerical models

2.1 Geometric model

A physical model of the turbine was built in Pro-E. As shown in Fig. 1, the inlet section and the draft pipe are extended to ensure fully developed flow in pipes, which can increase the simulation's accuracy and convergence. The main geometric parameters of the impeller and the volute are shown in Tab. 1.

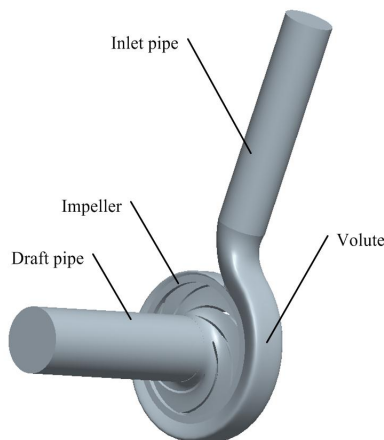


Figure 1: Model of turbine

Table 1: Parameters of the prototype turbine

Part	parameter	value
Impeller	Impeller inlet diameter D_2 /mm	180
	Impeller outlet diameter D_1 /mm	90
	Wrap angle of blade β_2 (°)	120
	Impeller inlet width b_2 /mm	20
	Blade inlet angle β_{b1} (°)	25
	Blade outlet angle β_{b2} (°)	29
	Rotation rate n /(r·min ⁻¹)	1450
	Blade number	6
Volute	Diameter of base circle D_3 /mm	200
	Outlet height b_0 /mm	180
	Volute inlet diameter D_5 /mm	70
	Shape of sections	round

2.2 Mesh independence study

The quality of mesh has great effects on the speed and accuracy of the simulation. In this model, structured meshes are used. The inlet and outlet of the impeller and volute are grid refinement and the minimum mesh quality is 0.4. The computational domain mesh and mesh quality are presented in Fig. 2.

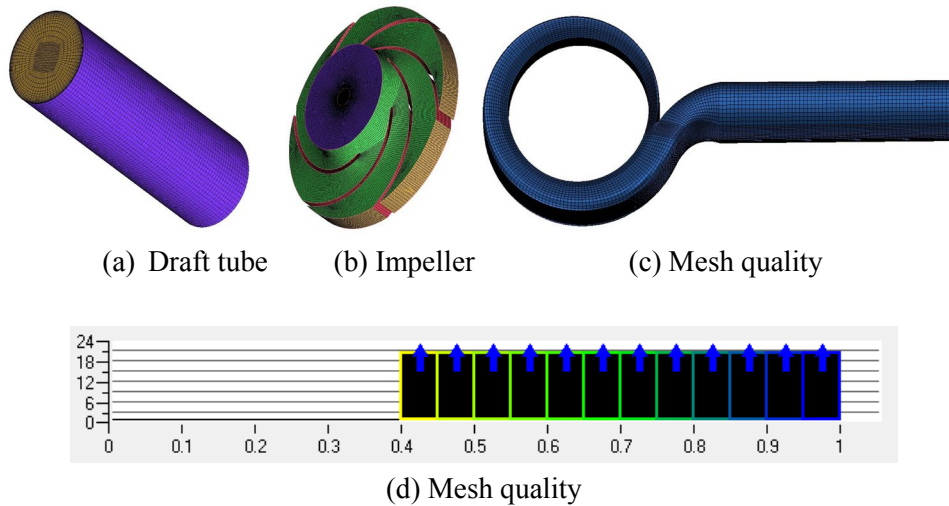


Figure 2: Computational domain meshes and mesh quality

The mesh independence of the model is studied. Given that efficiency is the final embodiment of comprehensive parameters, the efficiency is chosen as the evaluation index. As shown in Fig. 3, the number of grids is more than 1.2 million, and the change in efficiency is less than 0.5%. In this paper, the mesh number of the volute, impeller, and draft pipe are 659624, 409871, and 155677, respectively, and the total volume number is 1225176.

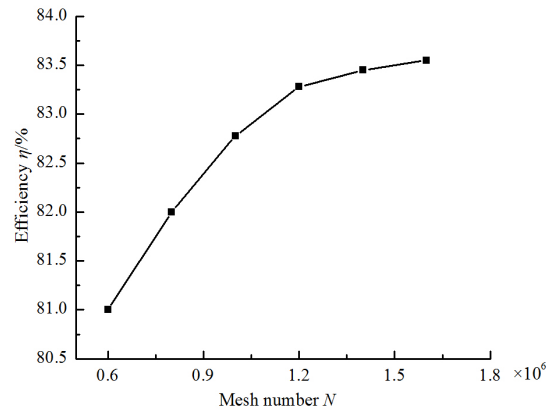


Figure 3: Efficiency and grid number

2.3 Numerical simulation and boundary condition

After the mesh is generated in ICEM, the computational fluid dynamics (CFD) software ANSYS CFX 15.0 is used for numerical simulation. The whole basin is set up as three-dimensional, incompressible, and turbulent flow with steady calculation. The standard k-epsilon turbulence model is adopted [Emma, Dario and Adolfo (2017); Wang, Kong, Yang et al. (2018); Shi, Liu and Luo (2018)]. The water domain in the volute and outlet pipe is set as stationary parts, whereas the water domain in the impeller is defined as the rotation part. The rotating speed is 1450 rpm. The inlet of the volute is set as the pressure inlet with the ambient pressure. The outlet boundary condition is set to flow rate to obtain the external characteristic curve, and different flow conditions (40, 45, 50, 55, and 60 m^3/h) are simulated. The shroud and the hub are treated as a rotary wall whose rotation direction and speed are consistent with the impeller. The interface between static and rotary parts is both “frozen rotor.” The others can be provided with a static wall surface. In the solve control, the max iteration number is 2000, and the convergence is assumed with all residues less than 10^{-5} .

3 Orthogonal test design

In turbine design, flow, head, and rotation rate are the three most important parameters. An optimal combination is required for the three indexes to obtain the most beneficial turbine design [Wei, Xue and Zhao (2010)]. In the hydraulic parts, the impeller has a greater influence on the hydraulic performance than the volute does. In this paper, a turbine with a specific speed of 110 is used as the original model, whose parameters are flow $Q=50 \text{ m}^3/\text{h}$, head $H=10 \text{ m}$, and rotation rate $n=1450 \text{ rpm}$. In accordance with the previous experience of

impeller design and related empirical knowledge, D_2 , b_2 , Z , β_2 are used as orthogonal experimental factors. The impeller is designed to match the existing volute structure, and three levels of each factor are selected, as shown in Tab. 2. The test was designed according to four factors and the three-level orthogonal table, as shown in Tab. 3.

Table 2: Factor and level graph

Level	Factor			
	A	B	C	D
	D_2/mm	b_2/mm	Z	$\beta_2/(\text{°})$
1	175	18	5	80
2	180	20	6	100
3	185	22	7	120

Table 3: Test scheme

Number	factor				Corresponding parameters			
	A	B	C	D	D_2	b_2	Z	β_2
1	A ₁	B ₁	C ₁	D ₁	175	18	5	80
2	A ₁	B ₂	C ₂	D ₂	175	20	6	100
3	A ₁	B ₃	C ₃	D ₃	175	22	7	120
4	A ₂	B ₁	C ₂	D ₃	180	18	6	120
5	A ₂	B ₂	C ₃	D ₁	180	20	7	80
6	A ₂	B ₃	C ₁	D ₂	180	22	5	100
7	A ₃	B ₁	C ₃	D ₂	185	18	7	100
8	A ₃	B ₂	C ₁	D ₃	185	20	5	120
9	A ₃	B ₃	C ₂	D ₁	185	22	6	80

4 Validation of the external characteristic test

4.1 External characteristic test

To verify the correctness of the numerical simulation, an open hydraulic turbine test rig is established. Its structure is shown in Fig. 4.

As shown in Fig. 4, pressure sensors 1 and 2 are installed at the inlet and outlet of the turbine, respectively. An electromagnetic flow meter is installed in the inlet pipe of the turbine to measure the flow rate, which cooperates with valves 1 and 2 to realize flow control. The flow rate, rotation rate, torque, and the pressure on the inlet and outlet measured by the sensors are collected by a computer to calculate the turbine’s efficiency, head, and shaft power.

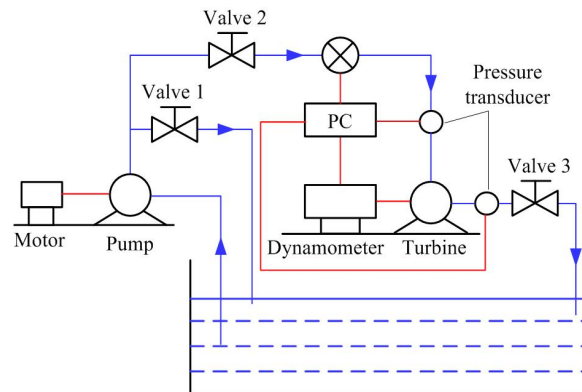


Figure 4: Test bed

4.2 Comparison of external characteristics

Fig. 5 shows the experimental results of the head and efficiency compared with the simulation results working in different flows.

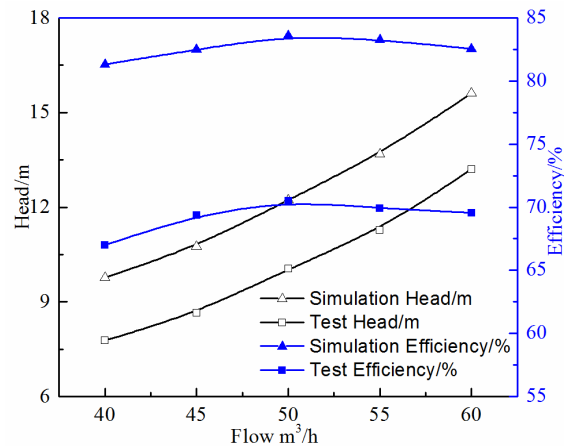


Figure 5: Comparison of external characteristic curves

Fig. 5 shows the experimental and computational relationships. The computational results indicate that the peak of the head is 12.23 m and its corresponding efficiency is 83.28% when the flow rate is 50 m³/h; these values are higher than the experimental results, which are 10.05 m and 70.48%, respectively. The difference between them is within 15%. The discrepancies between two cases are affected by wall roughness, import and export boundary conditions, 3D modeling, and mesh quality. Thus, this model is validated and can provide useful guidelines to optimize the turbine's performance.

Numerical simulation is suggested as a valid tool to predict the performance of a pump as a turbine, and it is also a convenient way to optimize the turbine's performance.

5 Result of the orthogonal test

5.1 Result of the orthogonal test

For turbine design, head and efficiency are usually used as the evaluation index of the orthogonal test. The head and efficiency under different designs can be achieved through the same numerical simulation used for the previous model. Results are shown in Tab. 4.

Table 4: Results of orthogonal test

Number	<i>H/m</i>	$\eta/\%$
1	10.71	67.70
2	9.66	81.27
3	10.31	84.51
4	14.14	82.05
5	12.85	81.61
6	14.64	77.96
7	14.71	83.48
8	16.39	80.20
9	15.42	79.42

Table 5: Range analysis of head

	<i>H/m</i>			
	A	B	C	D
K ₁	30.68	39.56	41.73	38.98
K ₂	41.63	38.9	39.23	39.02
K ₃	46.52	40.37	37.87	40.83
k ₁	10.23	13.19	13.91	12.99
k ₂	13.88	12.97	13.08	13.01
k ₃	15.51	13.46	12.62	13.61
R	5.28	0.49	1.29	0.62

Table 6: Range analysis of efficiency

	$\eta/\%$			
	A	B	C	D
K ₁	233.49	233.23	225.85	228.73
K ₂	241.61	243.07	242.75	242.7
K ₃	243.09	241.89	249.6	246.75
k ₁	77.83	77.74	75.28	76.24
k ₂	80.54	81.02	80.92	80.90
k ₃	81.03	80.63	83.20	82.25
R	3.20	3.28	7.92	6.01

After range analysis on the data of the orthogonal test, the result is shown in Tabs. 5 and 6. K_i is the sum of i levels of each factor. R_i is the range of factor i , which is the difference between the maximum value and the minimum value of the average index value at each level of the i_{th} factor.

$$k_i = \frac{K_i}{N_i} = \frac{1}{N_i} \sum_{j=1}^{N_i} y_{i,j} \tag{1}$$

$$R = \max(k_1, k_2, \dots, k_i) - \min(k_1, k_2, \dots, k_i) \tag{2}$$

R_i reflects the magnitude of change of the i_{th} index when the level of i_{th} factor changes. Greater effects will be generated on the experimental index when R_i becomes greater.

The charts of head efficiency are shown in Fig. 6, where the factors that greatly influence the head and efficiency can be found. The optimal values can be obtained after the analysis of the result.

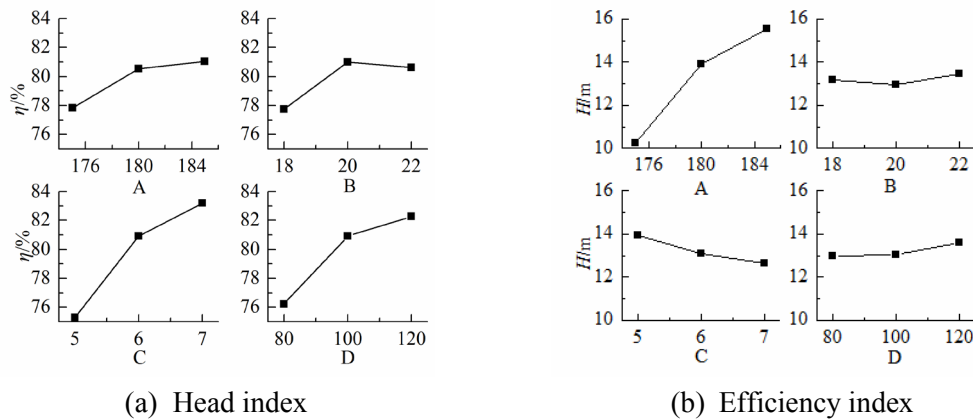


Figure 6: Performance index line chart

An analysis of the results shows that the main factor that affects the head is the diameter of the impeller, and the order of influencing level is ACBD. The main factor that affects the turbine’s efficiency is the number of the impellers. Also, the wrap of the blade has a great effect on it. The order of these factors, which is ranked by influence level, is CDBA. With the increase in the impeller’s diameter, the theoretical head and the friction loss of the impeller become greater. In summary, the actual head of the impeller increases. Thus, the head and efficiency can be raised by increasing the diameter of the impeller within a certain range. With the increase of the impeller’s number, the turbine’s capacity to recover energy increases gradually, thereby also increasing the head and efficiency. However, the exclusion of flow between the impellers and the area of the impeller’s surface become larger if more blades are added. As a result, the flow loss increases, and finally, the efficiency decreases. In conclusion, the ratio of various factors should be considered comprehensively according to the demand of the design.

5.2 Simulation analysis of optimization model

Comprehensively considering all four factors that affect the turbine’s head and efficiency, the best group of the head is $A_3B_3C_1D_3$, and corresponding to the efficiency is $A_3B_2C_3D_3$. Considering that the outlet width of the impeller has a small effect on the head while the number of blades has more influence on efficiency, the final combination after optimization is $A_3B_2C_3D_3$. Therefore, the inlet diameter of the impeller is 185 mm, the width of the impeller’s inlet is 20 mm, the number of blades is 7, and the wrap angle of the blade is 120° .

The same numerical simulation method is used for the optimized model. Fig. 6 shows the external characteristic curve after the comparison of the simulation result between the previous model and the model after optimization.

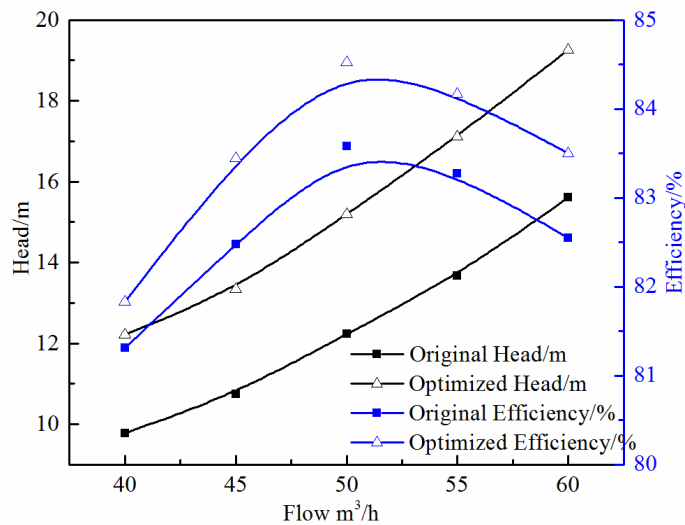


Figure 7: Comparison of the external characteristics of the optimized model with the original model

As shown in the Fig. 7, when the rated flow condition is $50 \text{ m}^3/\text{h}$, the head of original model is 12.23 m, while the head of the optimized model is 15.18 m. The efficiency of the original model is 83.28%, while the efficiency after optimization is 84.52%. The head and efficiency under other working conditions are also better than those of the original model. The increased performance of the optimized model shows the feasibility of the orthogonal test.

The comparison of the static pressure distribution and velocity vector between the original model and the optimized model is shown in Fig. 8.

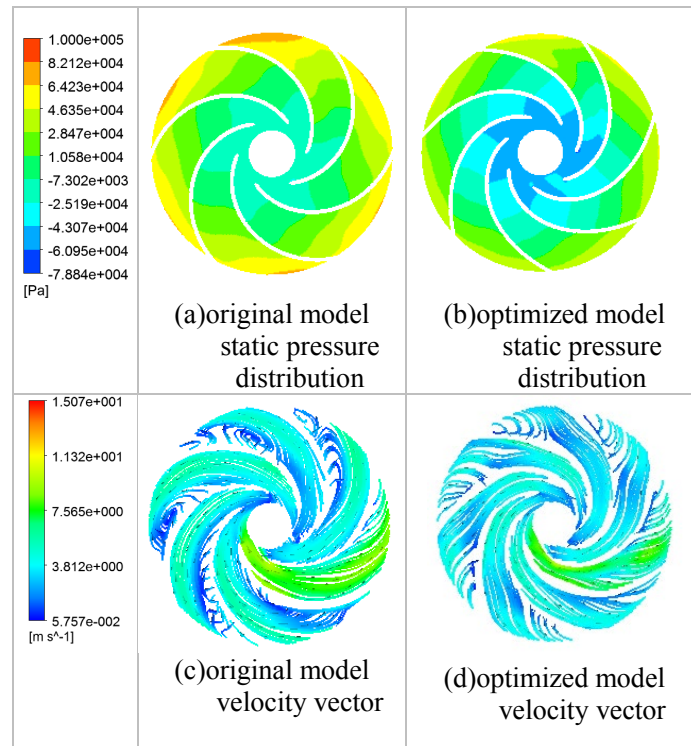


Figure 8: Static pressure distribution and velocity vector of impeller

In Figs. 8 (a) and 8(b), the water works on the impeller continuously, and the pressure in the impeller inlet to the outlet decreases gradually. The changes in the pressure gradient in the optimized impeller are more obvious than those in the original model. This result illustrates that the impeller working capacity is considerably enhanced. In Figs. 8(c) and 8(d), a large vortex is present at the entrance of the impeller in the original model, and the vortex at the inlet of the optimized impeller is relatively small. The recirculating flow is gradually reduced and the velocity vector distribution is relatively uniform after the optimization, resulting in reduced hydraulic loss in the optimized design.

6 Conclusion

1) In this paper, the influence of impeller inlet diameter, impeller inlet width, blade number, and blade wrap angle on the performance of a turbine for power generation is investigated by conducting an orthogonal test. CFD is used to simulate the original model and each optimization model. The results of the numerical simulation of the pump as turbine are close to the experimental results. Thus, numerical simulation can reliably predict the hydraulic performance of the model, which is used for the orthogonal test.

2) The same numerical simulation, which is used on the previous model, is used to simulate the nine groups of models in the orthogonal test to study the effects of the impeller's geometric parameters on the model's efficiency and head. After the optimization, the impeller's inlet diameter is 185 mm, the inlet width is 20 mm, and the

blade angle is 120°. The head and efficiency of the model after optimization obviously improved, thereby proving the feasibility of the orthogonal test. The results provide a certain reference for the optimization of the impeller in the pump as turbine.

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