Research Article

Optimization Design and Internal Flow Field Study of Open-Design Vortex Pump

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To determine the influences of the main structural parameters on open-design vortex pump performance, optimize the vortex pump performance, and reduce the running vibration and improve stability, orthogonal testing method was introduced in this paper. The selected main factors included impeller outer diameter (D_2), impeller outlet width (b_2), outlet setting angle of impeller (β_2), and inlet setting angle of impeller (β_1), and the nine types of impellers were coded according to orthogonal table. After obtaining the preliminarily optimum value range for each factor through range analysis, comprehensive analysis was employed based on the orthogonal test to investigate the main factors and identify the primary and secondary influencing factors affecting the performance of the vortex pump. An optimization scheme was obtained for further design. The results show that the numerical calculation results of the optimization scheme pump are in good agreement with the test results, and it shows the feasibility of the numerical calculation method. The testing results showed that efficiency and head of the optimal model were 4.2% and 9 m higher than those of the prototype model, respectively. Improved efficiency and head met the design requirements. The orthogonal testing method proved the feasibility of performance optimization of the vortex pump. The backflow occurs at the pump entrance and rotates in the same direction with impeller. It moves along the pipe wall from the lateral cavity to the inlet and encounters with the approaching flow.

1. Introduction

The production and application of vortex pump are relatively late, and the continuous in-depth research and the improvement of its structure are gradually applied to the sewage pumping of domestic sewage treatment and aquaculture industry, as well as the pulp pumping of light chemical industry. However, this equipment exhibits disadvantages, which include low efficiency and high energy consumption [1, 2]. At present, there are few researches on the vortex pump. A new flow model was proposed by Schivley and Dussourd [3] to study the internal flow field of vortex pumps and pointed out that fluids can be sucked into pumps under the effects of atmospheric pressure when the static pressure is negative at the central region of the vortex. Chen [4] established the apparatus first for measuring the

velocity distribution on the axial vertical profile in the vortex pump impeller and pressure distribution on the blade surface. The measuring methods are explained in more detail, and the measured results are presented. Quan et al. [5] aimed at the problem of whole efficiency decline caused by the internal flow structure of the vortex pump and took a certain type of vortex pump as the research object, and six groups of forward-curved and backward-curved blades were designed as the control group. It was found that, for the efficiency of the vortex pump, the efficiency of forward-curved impeller was higher than that of backward-curved impeller before the design flow point, and after the design flow point, the efficiency of backward-curved impeller was higher than that of forward-curved impeller. Sha et al. [6] used rapeseed as solid particles and done experimental studies under different solid volume fractions and obtained

the external characteristic curve of clean water and different solid-phase volume fractions, and the basic principles and characteristics of the solid-liquid two-phase flow of the vortex pump are further analyzed. A comprehensive analysis of the qualitative causal relationship between the change of the external characteristics of the pump and the internal flow field and the improvement measures for the optimal design of the vortex pump are proposed. In order to develop an understanding of the flow conditions and maximize the efficiency and minimize the cavitation occurrence of vortex pumps, numerical and experimental investigations have been carried out by Alexander Steinmann [7]. The cavitation behavior was estimated using the volume-of-fluid Rayleigh-Plesset cavitation model. CFD results turned out to be appropriate to capture the qualitative flow conditions and therefore increase the knowledge about vortex pumps. Beyond that, the CFD results are not fully sufficient to calculate the quantitative values with required accuracy.

Peng et al. [8] used the particle model to study the wear of slurry pump under the condition of biased design. The results showed that the interface between the sheath and rotor near the tongue had a serious local wear under the condition of small flow. By removing the front and rear back blades, the wear degree at the junction can be effectively reduced, but the wear of the inner side of the front guard board will be intensified. Wang et al. [9] studied the effect of different turbulence models on the sound field of the pump. The results show that RNG $k-\epsilon$ model is the best one for the multistage centrifugal pump with vaned diffuser. In order to research the internal sound field characteristics and reduce the running noise of self-priming vortex pump at a low flow rate, Wang et al. [10] adopted the CFD + Lighthill sound analogy theory to solve the internal acoustic field and optimize the noise of pump. The results showed that, with the decrease of the flow rate, the SPL of the pump discharge was increased, and the sound source of the self-priming vortex pump was mainly distributed in the channel and the tongue close to the pump discharge. Liu et al. [11] obtained the influence of the special volute structure of the vortex self-priming pump on its self-priming performance through 2D PIV and numerical simulation. The results show that the combination of the guide wall and the impeller outlet promotes the gas-water mixing and the formation of bubble flow and improves the self-priming performance.

At present, the research on this kind of vortex pump with special structure is scarce. The selection of main structural parameters of impeller, especially, is crucial for the performance of the vortex pump. This article took a typical open design vortex pump as the research object. Its optimization design was based on the orthogonal method, and the experimental verification is carried out. The optimal scheme of the vortex pump is further analyzed by the numerical simulation and the flow field is obtained, which also provided a reference for the design of the impeller.

2. Orthogonal Test

2.1. Pump Parameters. An open design vortex pump with a specific speed of 166 was selected as the research object. The design flow rate was $Q_{des} = 500 \text{ m}^3/\text{h}$; rated speed $n = 1485 \text{ r/min}$.

The main structural parameters are shown in Table 1. At the rated point, experimental efficiency is 42.36%, and the experimental head measures 20.24 m; these values do not satisfy the specified efficiency and head values. Figure 1 shows the open-design vortex pump two-dimensional (2D) assembly [2].

2.2. Numerical Calculation Methods

2.2.1. Calculation Model. The calculation model of vortex pump was created based on the original model pump. The whole flow domain of the CFD model consists of five components, namely, inlet section, lateral cavity, impeller, volute, and outlet section. The original model pump calculation domains are shown in Figure 2 [2].

2.2.2. Mesh Sensitivity Analysis. The entire mesh of the pump generation was carried out in Icem 13.0 software, as shown in Figure 2. In theory, with the increase in grid element number, errors caused by the grid will reduce gradually until they disappear [12–14]. However, considering computer configuration and computer time, the number of grid elements cannot be too large [15–17].

Table 2 shows results of grid sensitivity analysis. When grid sizes measure less than 5.0 and grid numbers are larger than 1.38×10^6 , efficiency and head only experience slight changes, indicating the stability of simulation results. Considering computer calculation capabilities, the grid size of 4.5 was used to carry out the following experiments (as shown in Figure 3). In this paper, the value of y^+ of the whole calculation model is lower than 50 [18, 19].

2.2.3. Boundary Condition. The CFD flow domain is divided into two types of subdomains: the first type of subdomain is the impeller, which is attached to the rotating frame and solved in a rotating framework via the multiple reference frame (MRF), and the rotational speed was set as 1485 r/min. Other fluid domains, except the impeller, are the second type of subdomain, which is solved in a stationary framework [20, 21]. The boundary conditions of the whole domain are shown in the Table 3.

2.2.4. Turbulence Model. In general, the selection of turbulence model is empirical in numerical simulation of hydraulic rotating machinery, and it is not clear which turbulence model calculation results of the vortex pump are more accurate. Therefore, in order to find a more accurate turbulence model to calculate the vortex pump, in this paper, five turbulence models are selected to simulate the vortex pump in the design condition and carefully validate it by comparing the numerical simulation results with the experimental data.

The five turbulence models are the standard $k-\epsilon$ model, RNG $k-\epsilon$ model, realizable $k-\epsilon$ model, standard $k-\omega$ model, and SST $k-\omega$ model. The numerical simulation results were compared with the experimental results with different turbulence models as shown in Table 4. On the whole, the

TABLE 1: Parameters of impeller and volute.

impeller inlet diameter (D_1)	140 mm	Impeller outlet diameter (D_2)	350 mm
Blade wrap angle of impeller (θ)	35°	Impeller outlet width (b_2)	60 mm
Inlet setting angle of impeller (β_1)	40°	Outlet setting angle of impeller (β_2)	35°
Blade number of impeller (z)	10	Lateral cavity width (L)	50 mm
Volute inlet diameter (D_3)	430 mm	Volute inlet width (b_3)	110 mm

FIGURE 1: Simple diagram of pump section model.

FIGURE 2: Calculation domains.

TABLE 2: Mesh independent analysis.

Mesh software	ICEM			
Grid size	4.5	5.0	5.5	6.0
Grid elements	1636421	1383790	986575	733659
Efficiency η (%)	46.342	46.339	45.725	45.236
H (m)	23.541	23.536	23.203	22.924

SST $k-\omega$ and RNG $k-\epsilon$ models predict the highest efficiency and head of vortex pump, and standard $k-\epsilon$ model has the lowest and most close predicted value. Thus, the standard $k-\epsilon$ model is chosen for the following numerical calculation.

2.2.5. Numerical Algorithm and Other Parameters. The numerical simulation of vortex pump is carried out by ANSYS-CFX 13.0, and the finite volume method is used to solve the Reynolds-averaged Navier-Stokes equations.

Second-order upwind discretization was used for the convective and diffusive terms, and the second-order upwind format is selected. SIMPLEC algorithm is selected to calculate the pressure-velocity coupling [22–25], and the convergence precision was set to 10^{-5} .

2.3. Factors and Experimental Scheme. Vortex pump features a simple structure, but its internal flow is complex. Structural parameters are as follows: impeller diameter (D_2), inlet diameter (D_1), number of blades (Z), blade width (b), lateral cavity width (L), blade outlet angle (β_2), blade inlet angle (β_1), and volute inlet diameter (D_3).

In this paper, structural parameters of studied pump impeller are as follows: $D_2 = 340$ mm, $Z = 10$, $b = 60$ mm, $L = 50$ mm, $\beta_2 = 35^\circ$, and $\beta_1 = 40^\circ$. The impeller was completely exposed to the cavity, and the impeller has no front cover, as shown in Figure 1. Changing the blade width b is equivalent to changing the lateral cavity width L . The area of the impeller passage decreases with the increase of blade number, and the impeller passage is more easily plugged. Efficiency and head do not meet the predetermined values when blade number decreases. Therefore, the number of blades (Z) was not considered and was kept constant. D_2 , b , β_2 , and β_1 were selected as the orthogonal experimental factors in this study. Factor levels are shown in Table 5. Nine kinds of impeller structure parameters were selected based on the standard orthogonal table [26], as indicated in Table 4 (D_2 , b , β_2 , and β_1 are represented by A, B, C, and D, respectively). The hydraulic design of impeller was obtained by the structural parameters of each impeller. In rated flow, efficiency and head values of the nine sets of programs were obtained based on CFD, as shown in Table 6.

3. Orthogonal Results and Flow Field Analysis

3.1. Orthogonal Test Analysis. At rated flow condition, efficiency and head values of the nine sets of schemes were obtained through CFD, as shown in Table 4. Efficiency and head are the most important pump indicators. Therefore, efficiency and head are selected as evaluation indexes.

The two evaluated indexes in this research belong to multi-index orthogonal experiment. Intuitive analysis of overall balance method was used to determine the main factor and optimization program. Visual analysis for each indicator was carried out, and the results were studied by comprehensive comparative analysis.

Table 7 shows the results of range analysis, with each column representing a different range. The values reflect the influence level of indicators, which affects the factor level in turn. A large range indicates large influence of indicator on the factor level selection. Therefore, factors with the largest

FIGURE 3: Sketch of the structured mesh: (a) volute; (b) impeller.

TABLE 3: Boundary conditions.

Location	Boundary condition types	Settings
Inlet	Mass flow inlet	101,325 Pa, turbulence intensity
Outlet	Pressure outlet	Equal to the operating flow rate
Wall	Nonslip wall	Surface roughness 0.005 mm
Impellers	Rotating	1485 r/min
Volute	Stationary	0 r/min
Lateral cavity	Stationary	0 r/min

TABLE 4: Numerical results with different turbulence models at Q_{des} .

Turbulence model	Standard $k-\epsilon$	RNG $k-\epsilon$	Realizable $k-\epsilon$	Standard $k-\omega$	SST $k-\omega$	Test data
Efficiency η (%)	46.34	48.22	47.48	46.45	48.51	42.36
H (m)	23.54	24.64	24.16	23.74	24.89	20.24

TABLE 5: Orthogonal experimental factors.

Level	Factors			
	D_2 (mm)	b (mm)	β_2 (°)	β_1 (°)
1	340	70	35	60
2	350	60	40	50
3	360	50	45	40

TABLE 6: Test arrangements and scheme.

Test serial number	A	B	C	D	Efficiency η (%)	H (m)
1	1	1	1	1	49.06	30.58
2	1	2	2	2	48.95	29.12
3	1	3	3	3	48.06	25.92
4	2	1	2	3	50.58	33.38
5	2	2	3	1	50.16	33.04
6	2	3	1	2	47.69	27.91
7	3	1	3	2	49.80	34.85
8	3	2	1	3	49.86	31.25
9	3	3	2	1	47.20	29.52

ranges are the most important factors; they are the most influential factors of indicators. The results show that B, A, D, and C are the major-minor sequences among factors influencing efficiency, and B, A, C, and D represent the major-minor sequences among factors on influencing head. $A_2B_1C_1D_3$ represents the optimum level obtained from single index analysis of efficiency, and $A_3B_1C_3D_1$ signifies the optimum level obtained from single index analysis of heads.

In general, when analyzing the influence of factors on assessment indicators, if it is a single factor, combination of the best factors is considered the best production condition when interaction of factors cannot be considered. However, this paper considered two indicators, which belonged to multi-index orthogonal experiment. The factors not only consider efficiency and head of primary and secondary indicators separately but also consider ultimate goals of primary and secondary factors according to pump performance.

TABLE 7: Variance analysis of efficiency and head.

	Efficiency η (%)				H (m)			
	A	B	C	D	A	B	C	D
K_1	146.070	149.439	146.610	146.421	85.620	98.811	89.739	93.141
K_2	148.431	148.971	146.730	146.439	94.329	93.411	92.019	91.881
K_3	146.859	142.950	148.250	148.500	95.619	83.349	93.810	90.549
k_1	48.690	49.813	48.870	48.807	28.540	32.937	29.913	31.047
k_2	49.477	49.657	48.910	48.813	31.443	31.137	30.673	30.627
k_3	48.953	47.650	49.340	49.500	31.873	27.783	31.270	30.183
R	0.787	2.163	0.470	0.693	3.333	5.154	1.357	0.864

The aim of this orthogonal test is to improve the efficiency and reach the specified head at the same time and also take into account the passing performance of the vortex pump. Therefore, the best combination of $A_2B_2C_3D_3$ was obtained by comprehensive analysis of the various factors with $D_2 = 350$ mm, $b = 60$, $\beta_2 = 45^\circ$, and $\beta_1 = 60^\circ$. Two-dimensional hydraulic design of impeller is carried out by using the optimal combination parameters. Then, 3D modeling, meshing, and numerical simulation are performed, and the efficiency and head value of the optimal scheme are obtained. The optimum program presented efficiency and head values of 51.02% and 32.03 m. The purpose of optimization design was achieved by comparing nine schemes of simulation results as shown in Table 6. The efficiency and head value of the optimal scheme pump are much higher than the original scheme pump.

3.2. Pump Performance Verification. Based on the two-dimensional hydraulic design drawing of the optimal impeller, the prototype was trial-produced, and the experimental research and verification were conducted.

Figure 4 shows prototype pump and optimization pump impeller. Optimization pump was tested by test rig, as shown in Figure 5. Experiments were conducted in an open-type pump system, with identification coming from Jiangsu Province of China [2]. Test rig comprises two parts, namely, data acquisition system and water circulation system [27, 28]. Data acquisition system changes all kinds of physical quantities at different conditions, whereas water circulation system supplies the necessary environment for vortex pumps. The test rig is shown in Figure 5.

Test facilities and measurement methods abide by measurement requirements described in [29]. A turbine flow meter was used to measure flow Q , and the precision of turbine flow meter was $\pm 0.3\%$. Speed n was measured by a tachometer. The pump outlet pressure was measured using a pressure transmitter with 0.1% measurement error [30]; the expanded uncertainty of efficiency was 0.5%.

In the pump performance test, pump efficiency is defined as follows [16]:

$$\eta = \frac{\rho g Q H}{P_s} \times 100\%, \quad (1)$$

where η is the pump efficiency, Q represents the flow (m^3/s), H is the pump head (m), and P_s is the output power of motor (W).

$$H = \frac{(P_{out} - P_{in})}{\rho g}, \quad (2)$$

where P_{in} is the inlet total pressure, and P_{out} is the outlet total pressure (unit of measuring P_{in} and P_{out} is Pa).

The comparison between the numerical and experimental results of the optimization model performance is shown in Figure 6. The numerical simulation yielded efficiency and head values that were higher than the experimental values, because volume loss and mechanical loss cannot be considered in tests. The numerical results were consistent with the experimental results, which also proved the feasibility of numerical simulation.

The comparison between the test results of the optimized model and prototype model is shown in Figure 7. The efficiency and head measured 46.92% and 29.36 m, respectively. Vortex pump efficiency was relatively low because the centrifugal and swirl forces produced by the rotating impeller caused a through flow and circulation flow, resulting in a significant hydraulic loss. In general, efficiency of vortex pump was 50%. Pump efficiency decreased evidently [27] when $n_s > 130$ and $n_s < 60$. n_s of the studied pump was 166, indicating low efficiency. However, efficiency value of the best efficiency point (BEP) was still 5.07% higher than the efficiency value of prototype pump. Head value of BEP was approximately 9 m higher than that of the prototype pump. Efficiency and head values were significantly improved and therefore met the design requirements.

3.3. Internal Flow Field Analysis

3.3.1. Characteristics of Inlet Rotating Reflux. Figure 8 shows the flow pattern analysis of the inlet pipe section of the numerical simulation of the vortex pump. It can be seen from the figure that part of the liquid flows back from the pump cavity to the inlet section, and the rotation direction of the reflux is consistent with that of the impeller. In this way, it enters into the inlet section in the form of a spiral from the lateral cavity, adheres to the wall of the inlet pipe, and continuously mixes with the inlet flow, resulting in the gradual weakening of the reflux kinetic energy, and finally reaches a balance with the inlet flow at a certain position. The spiral reflux in the inlet pipe is due to the special lateral cavity of the vortex pump, which has a strong circulation flow in the lateral cavity, which forms the reflux. This reflux leads to a large energy loss. Through the comparison of inlet flow patterns under three different working conditions in

FIGURE 4: The pump impeller in the original model and optimization model: (a) prototype impeller; (b) optimization impeller.

FIGURE 5: Experimental system. (1) Motor; (2) rotational speed meter; (3) test pump; (4) inlet pressure sensor; (5) inlet valve; (6) turbine flow meter; (7) outlet valve; (8) outlet pressure sensor.

FIGURE 6: Comparison between the numerical and experimental results of the optimization model performance.

Figure 8, it can be clearly seen that, under the small flow rate condition, the rotating reflux is strong, and the rotating reflux length is far away from the pump cavity, while, under

FIGURE 7: Comparison between the test results of the optimized model and prototype model.

the large flow rate condition, the reflux balance position moves to the pump cavity. This is mainly because, under the condition of a large flow rate, the inlet velocity is larger and the kinetic energy is stronger. When the impeller speed is

FIGURE 8: Swirl phenomenon in the inlet section of open-design vortex pump. (a) $0.6Q_{des}$ working condition, (b) $1.0Q_{des}$ working condition, and (c) $1.4Q_{des}$ working condition.

FIGURE 9: Streamline in a middle section: (a) prototype model; (b) optimization model.

FIGURE 10: A sketch of the impeller: (a) cross-sectional position; (b) blade code.

constant, the backflow intensity in the lateral cavity does not change, and the reflux strength entering the inlet is unchanged, so the rotating reflux length is close to the pump cavity under a large flow rate.

Figure 9 shows pump middle section streamline of prototype and optimal pump at rated flow conditions. Two different degrees of reflux phenomena exist. The loss was smaller in distant areas from the impeller, whereas the loss was larger near the cavity, where more significant influence was noted in the impeller inflow. The prototype pump exhibited a large recirculation zone between lateral cavity and pump inlet; refluxes between impeller outlet and volute inlet were also observed. Less reflux regions and less internal losses were noted in optimized pump model compared with prototype pump. Thus, the optimized pump demonstrated a better flow state than the prototype.

3.3.2. Analysis of Flow Line and Pressure Field in Impeller.

The flow field of impeller channel is significant to the special structure open-design vortex pump. The impeller channel of the optimized pump will be used to analyze the flow field characteristics. The cross-sectional position of the impeller and the blade number schematic is shown in Figure 10. “S” represents impeller cross-section, and “B” represents the blade of the impeller.

Figure 11 shows pressure (Pa) and relative velocity vectors in the average cross-sectional plane of the impeller at rated condition. As shown in S1, static pressure gradually increases from import to export inside the impeller. Static pressure of suction face B2–B5 is lower than those of other blades. The lowest static pressure is observed between B3 and B4. The largest vortex flow occurs between B8 and B9. Both sides of the blade

FIGURE 11: Pressure (Pa) and relative velocity vectors in the average cross-sectional plane of the impeller: (a) S1; (b) S2; (c) S3; (d) S4; (e) S5.

exhibited reduced vortex strength. This vortex mainly occurs at the suction face near the blade entrance. However, only part of the blade operates at the vortex. The main reasons are as follows: (1) the flow field under the

influence of volute tongue was coupled with the vortex volute for nonsymmetry; B7–B9 channels represent high-pressure volute areas; (2) from the cross-section to the front end of the blade, the distance is only 2 mm, and the

impeller is without a cover board; liquid flows from the front face to the back of the blade and to the direction of inlet. Liquid flow joins the flow through the blade inlet, causing a generation of the vortex flow.

With the blade flow field in the back cover, the liquid mainly comes from the inlet edge of the blade and across the blade of front face. After the rotation of the impeller, the liquid comes out from the front face of blade outlet and exhibits no influence on the inside channel of the impeller blade. S2–S5 illustrations clearly demonstrate this process. However, the suction face of the impeller inlet is assumed to appear in different degrees of backflow phenomenon in channel 8 of S2 and channels 7 and 8 of S5. This phenomenon differs from that of S1 in the backflow phenomenon. Vortex flow appears in S1 because of the broad volute of pump chamber in vortex pumps. Vortex flow was observed in S2 and S5 images; such an appearance is similar to that of the ordinary centrifugal pumps. Inlet setting angle of blade is due to the backflow in the suction face of the impeller inlet.

4. Conclusions

In this paper, a type of open-design vortex pump has been optimally designed and experimentally verified, and the optimized model pump has been numerically simulated under different flow conditions. Through numerical simulation and experiments, performance curves and internal flow field results have been obtained. The conclusions include the following aspects.

- (1) An optimization scheme was obtained for the orthogonal experiment method. The experimental results showed that efficiency and head of optimized pump were 4.2% and 9 m higher than those of the prototype model pump, respectively, and the numerical calculation results of the optimization scheme pump are in good agreement with the test results.
- (2) Due to the special open-design vortex pump structure, there is a broad lateral cavity, resulting in part of the liquid flowing back from the pump cavity to the inlet section. The rotation direction of the reflux is consistent with that of the impeller, and it enters into the inlet section in the form of spiral from the lateral cavity. Under the small flow rate condition, the rotating reflux is strong, and the rotating reflux is far away from the pump cavity, and the rotating reflux length (L) decreases with the increase of the flow rate condition.
- (3) In the open-design vortex pump front end of impeller and lateral cavity junction area, the fluid flows into the impeller from the lateral cavity and then flows out from the back half area of impeller to the lateral cavity, and the fluid in the lateral cavity flows back to the impeller channel from the whole suction face. The results of this study can provide a reference for the impeller design and engineering application.

Abbreviations

D_2 :	Impeller outer diameter;
b_2 :	Width of blade;
β_1 :	Impeller inlet blade angle;
β_2 :	Impeller outlet blade angle;
Q :	Flow rate;
H :	Head;
n :	Liquid density;
n_s :	Specific speed;
L :	Lateral cavity width;
z :	Number of blade;
η :	Pump efficiency;
p_{out} :	Total pressure at impeller outlet;
p_{in} :	Total pressure at impeller inlet;
ρ :	Liquid density;
g :	Gravity acceleration;
P_s :	Output power of motor.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare no conflicts of interest.

Authors' Contributions

X. G. and W. S. conceptualized the study and developed the methodology. R. Z. provided the software. R. Z. and W. S. performed validation. X. G. performed formal analysis, provided the resources, wrote the original draft, and is responsible for funding acquisition. H. W. performed investigation. T. Z. performed data curation and visualization. W. S. and R. Z. reviewed and edited the article. W. S. is responsible for the supervision and project's administration. All authors have read and agreed to the published version of the manuscript.

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