Study on Lifting Pump Design Scheme based on System Engineering Method

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Abstract: ***Based on a system engineering method, macro design scheme of lifting pump is proposed. From the structure and function of lifting pump system, the paper analyzes the function of each part of lifting pump and mutual restriction and influence be-tween each other in the system. And establishes a mathematical model at work of the key functional part axial flow pump and turbine motor. The mathematical models of the axial pump and the turbine motor were then coupled together by flow differences to model the system according to the operating requirements of the lifting pump. According to the three functions of the system, the constraint conditions and the corresponding design variables for the design of axial flow pump and turbine motor are put forward. And the objective function of the optimization design is pointed out according to the working characteristics of axial flow pump and turbine motor. Finally, according to the system search and optimization design, the optimization design process is established. By using the system engineering method, a more complete design scheme for the macroscopic lifting pump system has been established, which provides some guidance for the efficient and fast design of lifting pumps.***

Keywords: lifting pump; system; functional part; design scheme

1. Introduction

Natural gas hydrate, which is a new type of clean energy and has a distribution under specific conditions on land and ocean, is a solid compound formed by natural gas and water under certain high-pressure and low-temperature conditions. It is estimated that 20.7% of the land and 90% of the deep seabed have favorable conditions for the formation of hydrates [1]. Among them, the reserves of marine hydrates are huge, and the conservative estimate is 2.83×1015m3, which is 100 times the amount of land resources, considered to be the most promising alternative energy source in the 21st century [2].

Marine hydrate areas are characterized by shallow burial depth, loose mineral deposits, weak or uncemented, unstable, no tight cap , and no well-developed reservoir-seal assemblage. The main methods used in the exploitation of Marine hydrate include the depression-reduction method, heat injection method, chemical injection method, CO_2 replacement method, etc.[3] At present, the main method of hydrate exploitation is the depression-reduction method. The depression-reduction method faces the risk of uncontrolled disintegration of hydrates, which may lead to blockage of production by mud, destabilization of production equipment, and collapse of hydrate bodies leading to subsea landslides.

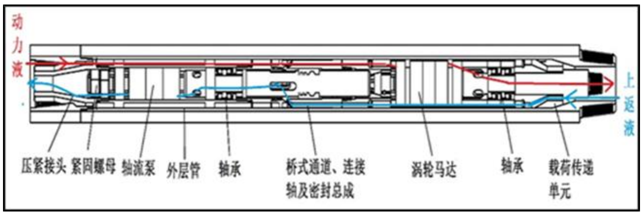
To solve these problems, Zhou ShouWei proposed a solid fluidization mining process of Marine gas hydrate based on the oil and gas drilling[1]. In production practice, the bit drills the pilot hole in the hydrate layer by the bottom hole motor, and then the solid hydrate layer around the pilot hole is broken by jet flow breaking. Finally, the mixed slurry of hydrate crushed material and sea water is pumped to the in-situ separator at the bottom of the well by the lifting pump device, after the sediment separation is completed, it is sent to the sea treatment equipment.

In the above-mentioned solid hydrate mining process, the lifting pump is the core component of this process, which plays a vital role in the safety production and daily output of natural gas. The lifting pump adopts hydraulic turbine to drive a multi-stage axial flow pump to pump the hydrate mixture slurry broken by jet to the separation device and the sea equipment. Its working performance index is mainly the flow and head of multistage pump. The flow and head of multi-stage pumps are controlled by adjusting the amount of power fluid pumped into the solid fluidized production system on the sea surface, which directly affect the output of natural gas, the bottom hole pressure on the production face, the erosion of various components in the flow process, the pressure loss and the safe fluctuation range of the operating parameters. It can be seen that the lifting pump is working in a more com-plex system. When the lifting pump is designed in detail, it is necessary to have a general evaluation of its design scheme and the change trend of its approximate parameters. Otherwise, it is difficult to get a satisfactory design scheme directly into the detailed design, and eventually the design work will be continuously repeated, resulting in a large amount of time waste and resource waste. Therefore, this paper studies the design scheme of lifting pump based on system engineering.

2. Systematic description of structure and principle of the lifting pump

2.1 Scientificity of scheme design research of lifting pump by system engineering method

A system is an organic whole with specific functions, which is composed of several components that interact and depend on each other, and this organic whole is part of the larger system to which it belongs. From the definition of the system, the components of the system interact and depend on each other, rather than being pieced together by several unrelated parts. Therefore, the specific functional components of the lifting pump and whether there is interaction between each component. Functional components refer to the mechanical structure assembly with relatively complete specific functions composed of a number of mechanical parts, such as bearing, which is composed of inner ring, outer ring, cage and rolling body. The structure of the lifting pump is shown in Figure 1.



(1) Compression joint (2) fastening nut (3) axial flow pump, outer tube, radial bearing, bridge passage, connecting shaft, seal assembly, turbine motor, thrust bearing, load transmission

Figure 1: Schematic diagram of lifting pump system

Axial flow pump, radial bearing, seal assembly, turbine motor, and thrust bearing can be considered according to functional components. The rest can be designed following the requirements of parts, and the design of these parts can be quickly determined according to the spatial and mechanical relations after the preliminary determination of the above functional components, so it is not within the scope of this paper. This paper mainly studies the design scheme of functional components that determine the performance and life of a lifting pump. The purpose is to reasonably control the parameter matching among the functional components so that the final design can meet the requirements of the performance, life, and operation convenience of gas hydrate fluidized solid mining.

The turbine motor is the power source which converts hydraulic energy of the power fluid pumped from the sea to the bottom of the well into mechanical energy and drives the axial flow pump to rotate. The axial flow pump works on the hydrate slurry entering the axial flow pump channel and pumps it to the next processing link. Both turbine motor and axial flow pump produce axial load and radial load when they work. Axial load is borne by thrust bearing and radial load is borne by the radial bearing. If there is no bearing in lifting pump system, the tur-bine motor and axial flow pump can not work normally. In addition, because there are two streams of liquid with different flow directions and pressure in the lifting pump, and they cross at the bridge channel, so it is necessary to use the total seal assembly to separate the two streams of fluid. The design of seal assembly is closely related to the operating speed of the turbine motor and axial flow pump, and the head of the pump. Rotary speed affects the life of the seal, and the pump head determines the design pressure of the seal.

Therefore, it can be seen from the above analysis that the five functional components of the lifting pump have mutual influence, and the research of the lifting pump should be studied as a system.

2.2Analysis of the interaction between the functional components in the lifting pump system

When analyzing the functional components, it should be noted that all the interrelations are ultimately derived from the functional requirements of the lifting pump system. In determining the design scheme, it is mainly to make parameter matching among the functional components according to the demand relationship. In order to match the parameters among functional components correctly or excellently, a parametric system model should be established, which we call system modeling. Before the system modeling, the specific mutual constraints among the functional components should be analyzed.

(1) Influence of axial flow pump on other components in the system

The axial flow pump is the most critical working part of the lifting pump, the quality of its design determines the performance of the lifting pump. The design scheme of the axial-flow pump is mainly based on the range of flow fluctuation, pressure fluctuation, and speed fluctuation. The main geometric parameters of an axial-flow pump are the number of blades, axial chord length, aspect ratio, inlet angle, outlet angle, maximum camber, and its position of blade airfoil, maximum thickness and position of the airfoil. If the three-dimensional design method is adopted, at least three design sections along the display direction are needed, and the design parameters of the runner will reach 27. At the same time, the guide vane parameters of the axial flow pump include the number of blades, axial chord length, aspect ratio, inlet angle, outlet angle, the maximum camber of blade airfoil and its position, and airfoil thickness. The guide vane is usually designed by a two-dimensional design method. Therefore, there are 35 complete design parameters for the axial flow pump. It is very difficult to coordinate the matching relationship between these 35 parameters reasonably. The traditional design methods rely on test and design experience to determine the values of relevant parameters subjectively.

In order to systematically analyze the influence of the axial flow pump on the whole lifting pump system, the relationship between the input and output parameters of the axial flow pump and its geometric parameters should be established. The basic working theory of axial flow pump [4].

The relationship between the head produced by an axial flow pump and the geometric parameters and flow rate can be expressed as:

$$H_{t\infty}=\frac{u_2}{g}\left(u_2-\frac{Q_t}{F_2}\cot\beta_2\right)$$

The relationship between the torque consumed by an axial flow pump and the flow rate can be expressed as:

$$M=\frac{1}{\omega}\rho g Q_t H_t=\rho Q_t\left(v_{u2}R_2-v_{u1}R_1\right)$$

The relationship between axial force, motion parameters and flow area of axial flow pump during operation can be expressed as:

$$P_{x1}=C_{x1}\rho\frac{w_\infty^2}{2}F$$

Working characteristic curve of axial flow pump is shown in Figure 2.

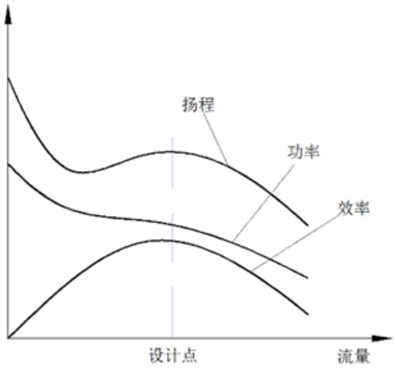


Figure 2: Working characteristic curve of axial flow pump

In the design of axial flow pump, the fluctuation range of inlet and outlet angles of blades can be determined according to the fluctuation range of flow, head and speed, but the blade profile parameters and blade shanks parameters cannot be directly determined. It needs to be determined by efficiency calculation.

The efficiency of an axial flow pump is defined as:

$$\eta = \frac{\rho g H Q}{M\omega}$$

From the working mechanism of axial flow pump, it can be seen that input parameters of the axial flow pump are the torque and speed transmitted by the turbine motor and the external output is flow and head. It works with the additional effects of centrifugal and axial forces. The magnitude of centrifugal force is affected by flow and rotation speed, and the magnitude of axial force is affected by flow. The changes of these parameters occur simultaneously. For example, in actual operation, if head of the pump is required to increase, the flow rate will decrease and the power consumption will increase. In this case, there is a certain impact on other functional components. The output power of the turbine motor is required to increase, the power load of the turbine motor increases and its speed decreases. Then it causes a reduction in the speed of the axial flow pump, which causes a reduction in the flow rate of the pump, and also a reduction in the centrifugal and axial forces. The decrease of axial force and centrifugal force is beneficial to the life of both types of bearings.

(2) Influence of turbine motor on other components in the system

The turbine motor is the power source of the lifting pump system, and the ability of the lifting pump to cope with the fluctuation of external conditions depends on the output characteristics of the turbine motor. When designing a turbine motor, the design parameters for the operating conditions depend on the requirements of the axial flow pump. The main parameters are the range of variation in output torque, the range of variation in output speed and the high efficiency adaptation zone. When customizing the design scheme of the turbine motor, the main objective is to determine the parameters related to the stator and rotor of the turbine motor. The main parameters are the number of blades, the axial chord length, the display ratio, the inlet angle, the outlet angle, the maximum curvature of the arc in the blade airfoil and its position, the maximum thickness of the airfoil and its position. Like the axial flow pump, there are at least 27 design parameters for the stator and rotor respectively when using the 3D design method. If the blade profile control parameters are considered, the design parameters of the turbine motor will reach more than 60. In the design of turbine motor with existing methods, some parameters are determined subjectively according to experience, and then some parameters are calculated according to the working theory of turbine motor.

To study the influence and restriction mechanism of the turbine motor on the whole lifting pump system, it is also necessary to establish the relationship between the input and output parameters of the turbine motor and the geometric parameters of the turbine motor. The following relationships are established from the basic working theory of the turbine motor[5、6].

The relationship between output torque produced by the turbine motor and geometric parameters of turbine motor can be expressed as:

$$M_i = \frac{Q_i \gamma_0}{g} R_0 C_z (\operatorname{ctg} \partial_1 - \operatorname{ctg} \partial_2)$$

The relationship between the rotational speed produced by the turbine motor and the geometric parameters can be expressed as:

$$n = \frac{60}{\pi D} C_z (\operatorname{ctg} \partial_1 + \operatorname{ctg}\beta_1) = \frac{60}{\pi D} C_z (\operatorname{ctg} \partial_2 + \operatorname{ctg}\beta_2)$$

The relationship between power produced by turbine motor and geometric parameters can be expressed as:

$$H_i = \frac{C_z^2}{g} (ctg\partial_1 - ctg\partial_2)(ctg\partial_1 + ctg\beta_1)$$

The relationship between axial thrust produced by the turbine motor and the geometric parameters can be expressed as:

$$T = \frac{\pi}{4} \left(\frac{d_1^2 + d_2^2}{2} \Delta p_z + d_3^2 \Delta p_b\right)$$

The performance characteristic curve of turbine motor is shown in Figure 3.

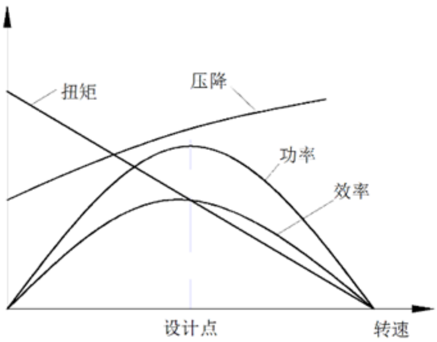


Figure 3: Performance characteristic curve of turbo motor

The restriction mechanism of turbine motor on other functional components in the lifting pump system is reflected in the following aspects. The output power of the turbine motor determines the power consumed by the lifting pump through the system, including the water power converted by the axial flow pump, the friction power consumed by the two types of bearings, and the friction power consumed by the seal assembly. The output torque of the turbine motor is balanced with the torque consumed by the axial flow pump, the two types of bearings and the seal assembly. In other words, changes in the working state of functional components that consume power and torque also affect output characteristics of turbine motor. The turbine motor also has some self coupling characteristics. For example, when the axial load and radial load of turbine motor change with the output, it also affects the torque consumed by the two types of bearings, and affects itself through the two types of bearings.

(3) The influence of two types of bearing and seal assemblies on other components in the system.
The working performance of thrust bearing, radial bearing and seal assembly does not directly affect the working performance of lifting pump, but affects the turbine motor and axial flow pump by changing the power consumption and torque.

Thrust bearing mainly bears axial loads, including axial load of axial pump and turbine. As a vulnerable part of the lifting pump, it mainly affects the safety and life of the lifting pump system. Safety and life time are both affected by the operating speed and the load. In the normal working range, the higher the speed, the greater the load, the greater the harm to safety and life.

The radial bearing mainly bears the radial load, including the radial load generated when the axial flow pump and turbine work, and the radial component force generated when the tool is in the same direction as the gravity. Compared with the thrust bearing, its working condition is much better, and its impact on the lifting pump system is mainly the life problem, that is, after the radial bearing is worn and fails, the lifting pump cannot work normally. Its life is related to the operating speed and the load. In the normal working range, the greater the load, the shorter the life.

The seal assembly is mainly used to divide the upper and lower fluids. There is friction on the contact surface of the seal. On the one hand, the friction consumes the torque generated by the turbine motor. On the other hand, it forms wear on the seal. The influence of seals on the system is mainly a life problem, and changing factors affecting its life are mainly the lift produced by the axial flow pump and the speed of system operation. When the lift generated by the axial flow pump is larger, the load on the sealing contact surface is larger, and the impact on the life is more unfavorable. The higher the operating speed of the system, the faster the wear rate, and the more unfavorable the impact on life.

3. Establishment of lifting pump system model based on coupling of key functional components

The design of the system revolves around the realization of system function. The function of the lifting pump system is mainly to pump the hydrate slurry broken by the jet to a specific functional link. It should be noted that the lifting pump injects liquid into the bottom of the well while pumping liquid from the bottom of the well. The relationship between injected and discharged fluids, and between injected and discharged fluids, has a direct impact on the volume of fluid in the wellbore and tool annulus. In other words, the difference between lifting liquid and injected liquid directly affects the pressure of the liquid column at the bottom of the well. The fluctuation of pressure cannot exceed the pressure fluctuation window that the formation can bear. Otherwise, it will cause bottom hole leakage if it is too large, and blowout if it is too small, which will seriously affect the safety of the solid-state fluidized mining process. Therefore, for the lifting pump, in addition to the amount of the upward return liquid, attention should also be paid to the difference between the upward return liquid and injected liquid. The upper return liquid is affected by the axial flow pump, and the injection liquid determines the output characteristics of the turbine motor. The injection liquid affects the upper return liquid. When designing lifting pump, the relationship between the upper return liquid and the injection liquid should be closely linked with the relationship between the turbine motor and the axial flow pump. The turbine motor and axial flow pump designed in this way can meet the needs of the lifting pump systems and even solid-state fluidized mining systems.

Theoretically, the turbine motor and axial flow pump are similar in their operation under variable service conditions. However, the presence of multiple sets of thrust bearings and radial bearings in the lifting pump system changes the similarity between the two types of vane fluid machinery. Therefore, the variable conditions of the lifting pump are dissimilar to the axial flow pump and the turbine motor.

The relationship between the operating speed of the turbine motor and its geometrical parameters is as follows:

$$n = \frac{60Q_{in}}{\pi D_T F_T}(cotA_1 + cotB_1)$$

The relationship between output torque of turbine motor and its geometric parameters is as follows:

$$M = \frac{\rho D_\tau {Q^2}_{in}}{2F_T}(cotA_1 - cotA_2)$$

The relationship between the power of turbine motor and its geometric parameters is as follows:

$$N_T = \frac{\rho Q_{in}^3}{{F^2}_p}(cotA_1 - cotA_2)(cotA_1 + cotB_1)$$

For axial flow pumps under non-similar conditions, the relationship between the circumferential speed and its geometric parameters is as follows:

$$u = \frac{Q_{out}}{F_p}(cotC_2 + cotD_2)$$

The relationship between the conversion power and its geometric parameters is as follows:

$$N_p = \frac{\rho {Q^3}_{out}}{{F^2}_p}(cotC_2 + cotD_2)(cotD_2 - cotD_1)$$

According to the relationship between the power transmitted from the turbine motor to the axial flow pump $N_P = \eta N_T$, It can be deduced that the ratio of injection flow and discharge flow has the following relationship with geometric parameters:

$$\frac{Q_{out}^3}{Q_{in}^3} = \frac{\eta \frac{\rho}{{F^2}_T}(cotA_1 - cotA_2)(cotA_1 + cotB_1)}{\frac{\rho}{{F^2}_p}(cotC_2 + cotD_2)(cotD_2 - cotD_1)}$$

Therefore, the difference between the injection flow and the discharge flow has the following relationship:

$$\Delta Q = (1 - \sqrt[3]{\frac{\eta \frac{\rho}{{F^2}_T}(cotA_1 - cotA_2)(cotA_1 + cotB_1)}{\frac{\rho}{{F^2}_p}(cotC_2 + cotD_2)(cotD_2 - cotD_1)}} Q_{in}$$

According to the inlet and outlet velocity triangles of the turbine motor and the axial flow pump, the outlet angle A1 of the liquid flow relative to the turbine stator does not change with the change of the injection flow rate. There is an inverse tangential function between the turbine stator blade inlet angle A2, the rotor blade inlet angle B1 and the pump-in flow rate, but the change relationship is unknown. There is a functional relationship of the arctangent function between the liquid flow inlet angle C2 of the guide vane of the pump, the liquid flow inlet angle D1 of the runner vane, the liquid flow outlet angle D2 and the backflow rate. The changing relationship is unknown. Expressed as:

$$\begin{aligned} A_2 &= f_1[arctan(Q_{in})] \\ B_1 &= f_2[arctan(Q_{in})] \\ C_2 &= f_3[arctan(Q_{out})] \\ D_1 &= f_4[arctan(Q_{out})] \\ D_2 &= f_5[arctan(Q_{out})] \end{aligned}$$

Therefore, the difference between injection flow and flowback liquid flow is:

$$\Delta Q = (1 - \sqrt[3]{\frac{\eta \frac{\rho}{{F^2}_T}(cotA_1 - cot(f_1[arctan(Q_{in})]))(cotA_1 + cot(f_2[arctan(Q_{in})]))}{\frac{\rho}{{F^2}_p}(cot(f_3[arctan(Q_{out})]) + cot(f_5[arctan(Q_{out})]))(cot(f_5[arctan(Q_{ous})]) - cot(f_4[arctan(Q_{out})]))}}) * Q_{in}$$

According to the previous discussion and analysis, it can be found that the liquid flow angle and flow of the pump are mutually restricted and not independent. In the above formula, the flow area of the turbine in the lifting pump is almost the same as that of the axial flow pump, so the area can be reduced. The flow difference can be rewritten as:

$$\Delta Q = (1 - A) * Q_{in}$$

$$A = \sqrt[3]{\frac{(cotA_1 - cot(f_1[arctan(Q_{in})]))(cotA_1 + cot(f_2[arctan(Q_{in})]])}{(cot(f_3[arctan(Q_{out})]) + cot(f_5[arctan(Q_{out})]))(cot(f_5[arctan(Q_{out})]) - cot(f_4[arctan(Q_{out})]))}}$$

Among them, A is a trigonometric function operation. Moreover, in turbine motors and axial flow pumps, the change range of each angle is within 0 ~ 150 °, so the value of A will always be greater than 1, equal to 1, and less than 1. Therefore, the value of flow difference can be greater than 0, equal to 0, and less than 0. That is to say, the lifting pump can increase, decrease, and maintain the fluid flow in the wellbore..

4. Research on the model optimization scheme of lifting pump system

In the previous section, this paper established the mathematical relationship of the parameter flow difference for the performance evaluation of the lifting pump system. In the design, the blade parameters of the axial flow pump and turbine motor will be calculated according to the flow difference Δ Q, which is greater than 0, equal to 0 and less than 0. There will be many groups of actual blade parameters that can meet the combination of different flow differences. In the final design, what kind of geometric parameter combination should be further optimized.

In the optimization design, the constraint condition is the range of geometric parameters calculated according to the flow difference. The calculated geometrical parameters are the inlet and outlet angles of the turbine cascade, 12 in total for the stator and rotor, and the inlet and outlet angles of the axial pump impellers, 8 in total for the runner and guide vane. The objective function is the axial load acting on the thrust bearing and the energy conversion efficiency of the lifting pump system. Therefore, it can be found that the optimization of the lifting pump system is a multi-objective and multi-variable optimization problem. For this type of optimisation problem, the orthogonal test method is first used in the design analysis of the lifting pump. The sensitivity of each parameter is then analysed in conjunction with CFD simulation software. The dimensionality of the design variables is reduced using an orthogonal design table combined with the use of the response surface method to determine the way in which the samples are generated. Finally a sample database was constructed [12]. Through the sample database, the neural network is used to train and learn the samples. Finally, the genetic algorithm is used to optimize the objective function. The optimized design flow of the system model is shown in Figure 4.

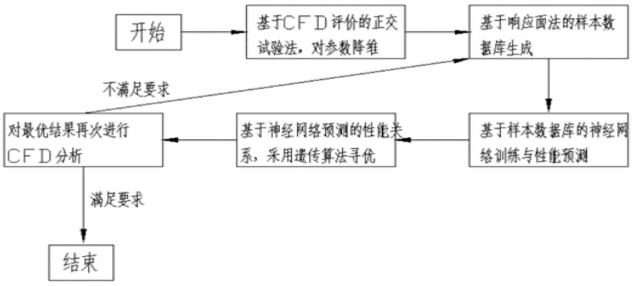


Figure 4: Optimization process of system model

5. Conclusion

From the perspective of system engineering, this paper analyzes how to determine the flow of the design plan at the beginning of the design of a lifting pump and establish a relatively complete macro design plan of the lifting pump system, which has certain guiding significance for the efficient and quick design of the lifting pump. The main works completed are as follows:

1) According to the definition of the system, starting from the structure and function of the lifting pump, the function of each functional component of the lifting pump and the mutual restriction and influence relationship of each functional component in the system are analyzed.
2) The mathematical model of the axial pump and turbine motor is established for the key functional com-ponents of the lifting pump, and then the mathematical model is coupled through the flow difference according to the performance requirements of the lifting pump, so as to build a model of the system.
3) According to the three functions that can be achieved by the system, the paper proposed the constraint conditions and corresponding design variables for the design of axial flow pump and turbine motor, and pointed out the objective function of optimal design in combination with the working characteristics of the axial flow pump and turbine motor.
4) Finally, the optimization design process is established, referring to the system optimization and optimiza-tion design method.

At the beginning of the design, a complete optimization model was not established in the details, but only a macro optimization design scheme was provided. In the next step, this part should be specified in the detailed de-sign study, so as to verify the correctness of the scheme.

$Hi\infty$	Head of axial flow pump	[m]
U_2	Circumferential velocity of pump outlet	[m/s]
Q, Q_i	Pump flow	[m^3/s]
F, F_p, F_2	Shroud diameter of impeller	[m^2]
M	Torque on pump shaft	[N.m]
P	Liquid density	[Kg/m^3]
Ω	Rotating speed	[rad/s]
w_∞	Inlet speed of pump	[m/s]
H	Efficiency	[%]
M_i	Turbine produces torque	[N.m]
B_1, β_1	Turbine rotor inlet angle	[°]
B_2, β_2	Turbine rotor outlet angle	[°]
R_0	Average turbine radius	[m]
R_1	Pump inlet radius	[m]
R_2	Pump outlet radius	[m]
F_T	Flow area of turbine cascade	[m^2]
Q_{in}	Lift pump injection flow	[m^3/s]
Np	Axial flow pump power	[W]
H_i	Head of pump	[m]
P_i	Power of turbine	[W]
V_{u1}	Circumferential component of pump runner inlet velocity	[m/s]
V_{u2}	Circumferential component of pump runner outlet velocity	[m/s]
C_1	Inlet angle of pump guide vane	[°]
C_2	Outlet angle of pump guide vane	[°]
D_1	Inlet angle of pump runner	[°]
D_2	Outlet angle of pump runner	[°]
P_{x1}	Pump axial load	[N]
C_{x1}	Axial component of pump flow velocity	[m/s]
A_1、α_1	Outlet angle of turbine stator	[°]
A_2、α_2	Inlet angle of turbine stator	[°]
n	Rotating speed	[rpm]
C_z	Turbine liquid axial velocity	[m/s]
T	Axial load of turbine	[N]
ΔP_s、ΔP_r	Pressure drop in turbine stator and rotor	[Pa]
u	Turbine circumferential speed	[m/s]
Q_{ou}	Upward discharge of the lifting pump	[m^3]
N_t	Turbine power	[W]

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