Effect of Different Rotational Speed on the Cavitation of Deep-Sea Mining Pump under Multiphase Flow

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Abstract. According to the phenomenon of cavitation in the deep-sea mining pump and the instability of the pump’s speed, the pump’s cavitation characteristic under different speeds is studied. Based on the theory of multiphase flow and cavitation nucleus, the Euler model and Schnerr-Sauer cavitation model are used to simulate the pump’s steady state of cavitation under different rotational speed. The pressure distribution and gas volume distribution are analyzed as well as the pump’s external characteristics, and it is verified by experimental. The results show that the low pressure and cavitation appears on the inlet of the impeller blade at the pump’s first stage. Along the streamline direction, the pressure increases gradually and the gas disappears. With the rotational speed increases from 960r/min to 2000r/min, the area of the low pressure expands rapidly as well as the cavitation, the efficiency of the pump and the NPSHa drop to 30% and 1.05m respectively, the H adds to 90.89m. The calculation formula of the pump installation height is derived as (21). When the speed is the designed 1450r/min, the pump has the best installation position about 108m under the sea level.

1. Introduction
Cavitation is a phenomenon of mass exchange between the liquid and gas when the pressure reduced, and its occurrence often affects a series of problems such as the normal flow of the fluid in the pump, the change of vibration, noise and external characteristic parameters, even the entire system would stop working[1]. Now there are many researches about the cavitation in the single-phase flow conditions for different pumps[2-3], but the cavitation in the multiphase flow conditions is in the stage of experimental. Chang Jin-shi got the result that the critical cavitation pressure and the initial cavitation pressure show a linear increase trend with the increase of sediment concentration by experiment[4]. Bostjan obtained the variation of the hydrofoil cavitation development with the sediment concentration through the water hole experiment[5]. Madadnia’s research show that the destructive effect of cavitation was higher under sandy flow[6]. The experimental study is costly and it has a poor reusability. So the numerical simulation of cavitation under multi-phase flow is of great significance.

As a key part of the deep-sea mining hydraulic lifting equipment, the deep-sea mining pump directly affect the efficiency of the equipment[7]. At present, the research mainly focuses on the structural design of the pump and the flow field analysis of solid-liquid two-phase flow, while there is almost no research on the cavitation characteristics. Yoon C H[8], Ding H[9] did the simulation and performance tests about the designed centrifugal pump, it show that solid-liquid slurry pump is difficult to meet the requirements of hydraulic pipeline lifting system. Based on the theory of vertical pipeline hydraulic transportation, Yang Fang-qiong put forward the theoretical formula of the slurry pump’s design parameters, and he analyzed the influence of the geometric parameters on the pump performance by
using the numerical simulation method[10-11]. The size of nodules which transported by deep-sea mining pump are much larger than sand, and when the cavitation occurs, there is a complex three-phase flow. In most cases, the actual speed of the pump often deviates from the designed. Therefore, Combined the two issues, the cavitation characteristics of the pump under different rotating speeds is explored through simulation, and the pump’s best speed is determined, then the installation height of the pump is researched. It has a great significance for the improvement of pump’s structural and the operation in the actual project.

2. Numerical Algorithm

2.1. Multiphase Flow

The flow pattern of the solid-liquid flow in the slurry pump belongs to the multi-phase flow. The simulation of the Eulerian phase flow model can be used to get the real flow characteristics of the discrete phase and the continuous flow. The Euler model assumes that the discrete solid phase is the quasi-fluid. The flow is still described by the conservation equation in macro continuum theory. The liquid phase continuity equation of the solid-liquid phase turbulence under the rotating coordinate is:

\[ \frac{\partial \rho_l}{\partial t} + \frac{\partial (\rho_l v_i)}{\partial x_j} = 0 \] (1)

among them, \( \rho_l \) is the liquid density; \( v_i \) is the liquid velocity vector; \( i,j \) is the direction of coordinates.

Continuity equation of solid phase is:

\[ \frac{\partial \rho_s}{\partial t} + \frac{\partial (\rho_s v_s)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \frac{v_s \partial \rho_s}{\sigma_s \partial x_j} \right) \] (2)

where the \( \rho_s \) is the density of solid phase; \( v_s \) is the solid velocity vector.

Momentum equation of liquid phase is:

\[ \frac{\partial (\rho_l u_i)}{\partial t} + \frac{\partial (\rho_l u_i u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \mu_l \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] - \frac{\partial \rho_l}{\partial x_i} + 2 \rho_l w r u_k + \frac{\rho_l}{\tau_{rs}} (u_i - u_k) \] (3)

where \( w, r, \mu_e, u, P \) and \( \tau_{rs} \) denote the angular velocity, the radius, the equivalent viscous coefficient, the liquid dynamic viscosity, the equivalent pressure of centrifugal force and the particle relaxation time, respectively.

The momentum equation of solid phase is:

\[ \frac{\partial (\rho_s u_s)}{\partial t} + \frac{\partial (\rho_s u_s u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \mu_s \left( \frac{\partial u_s}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \frac{\partial}{\partial x_j} \left[ \frac{v_s}{\sigma_s} \left( \frac{\partial \rho_s}{\partial x_j} u_i + \frac{\partial \rho_s}{\partial x_j} u_j \right) \right] - 2 \rho_s \epsilon_{ijkl} w_{i1} u_k + \frac{1}{2} \frac{\partial}{\partial x_i} \left( \rho_s w^2 r^2 \right) + \frac{\rho_s}{\tau_{rs}} (u_i - u_s) \] (4)
2.2. Cavitation Model

When cavitation occurs, mass exchange between liquid and gas. The main equations include transport equation, mass transfer equation and continuity equation of gas phase \[12\].

**Transport equation of gas phase:**

\[
\frac{\partial}{\partial t} (\alpha \rho_v) + \nabla (\alpha \rho_v \mathbf{V}_v) = R_E - R_c
\]

(5)

where $\mathbf{V}_v$, $\alpha$, $\rho_v$ denote the gas phase, the volume fraction of the gas phase and the velocity of the gas phase.

$R_E$ and $R_c$ are the mass transfer source, which is related to the growth and collapse of the steam bubble, and describes the mass transfer between the liquid phase and the gas phase, it is modeled based on Rayleigh-Plesset equations for single vapor bubble growth.

**Mass transfer equation:**

\[
R \frac{d^2 R_B}{dt^2} + \frac{3}{2} \left( \frac{dR_B}{dt} \right)^2 = \frac{P_B - P}{\rho_g} - \frac{2\delta}{\rho_g R_B} - 4 \frac{\mu}{\rho_g R_B} \frac{dR_B}{dt}
\]

(6)

where $R_B$ is the radius of bubble, $\delta$ is the surface tension coefficient of liquid, $P_B$ is the the surface pressure of bubble and $P$ is the the far-field pressure of local.

Equation (6) can be simplified as:

\[
\frac{dR_B}{dt} = \left( \frac{2(P_v - P)}{3\rho_l} \right)^{1/2}
\]

(7)

**Continuity equation:**

**Liquid phase:**

\[
\frac{\partial}{\partial t} \left[ (1-\alpha) \rho_l \right] + \nabla \left[ (1-\alpha) \rho_l \mathbf{V}_l \right] = -R
\]

(8)

**Gas phase:**

\[
\frac{\partial}{\partial t} (\alpha \rho_v) + \nabla (\alpha \rho_v \mathbf{V}_v) = R
\]

(9)

According to the equation (8) and (9), the continuity equation of mixed phase is got:

\[
\frac{\partial}{\partial t} \rho_m + \nabla (\rho_m \mathbf{V}) = 0
\]

(10)

$\rho_m$ is the density of mixed phase and it is expressed as $\rho_m = \alpha \rho_v + (1-\alpha) \rho_l$

3. Simulation Design of Cavitation

3.1. Geometric Model and Meshing

According to the transportation of big size particles in the pump, we adopt the principle of amplifying flow design to improve the ability of transporting the large particles, the structural is shown in Figure 1, in order to facilitate the installation of pipe, it was designed for the cartridge-type overall structure, the motor, cylinder pump, the inlet flange and the outlet flange were fixed on the same axis through the bolt cylinder. The lifting tube was connected with the inlet flange and the outlet flange. Multi-stage centrifugal impeller and submersible motor connected through the coupling. The main design parameters are as follows: inlet diameter of pump $D_0=240$mm, diameter of impeller inlet $D_1=244$ mm, diameter of impeller $D_2=401.7$ mm, width of impeller outlet $b_2=54$ mm , the number of impeller blades $Z_1=3$, the number of guide vanes $Z_2=4$. Design flow $Q=800$m$^3$/h (working flow 420m$^3$/h), head $H=70$m, rotating speed $n=1450$r/min, cavitation allowance $NPSH_r=6$m.
Figure 1. Mineral pump structure diagram

The 3D model of deep-sea mining pump was made by Pro/E(Figure 2), and the whole pump is composed of impeller, guide vane, inlet pipe, outlet pipe and shell.

Figure 2. Pump’s three-dimensional modeling

ICEM was used to carry out full flow grid. The quality of the mesh has a significant impact on the accuracy of the simulation results. The impeller and guide vane are distorted in the space, so the idea of "bottom-up" topology creation was used to create a topological rectangular block for a part of a single flow channel, then extend the small topology and cut the redundant blocks according to the actual model. At the same time the point, line and surface were associated. Next, the block structure similar to the entity was obtained, and the structure grid was generated by the block. Finally the number of nodes, the number of grids, the node density and the grid spacing near the wall were adjusted. O-type grids are used at the inlet pipe and outlet pipe. The full-channel grids are obtained by assembling the single-channel grids, and the grid quality is 0.6 above (Figure 3).

Figure 3. Specific requirements of maps

3.2. Physical Parameters and Boundary Conditions

Deep-sea mining pump delivers slurry containing manganese nodules. The relevant parameters of manganese nodules are based on the technical indicators in the report of the Head of the Tenth Five-year Mining and Testing System Group[13]; the physical properties of seawater are related to their geographical distribution. According to the seawater average salt concentration of 3.5%, refer to
the "Chlor-Alkali Industrial Physical and Chemical Constituents Manual" ,the seawater properties were obtained; the nature of the gas phase reference to "Engineering Thermophysical Properties Handbook." The physical properties of each phase are shown in Table 1.

Table 1. Physical parameters

| Physical properties                        | Value     |
|-------------------------------------------|-----------|
| Density of seawater (kg.m⁻³)               | 1025      |
| Viscosity of seawater (Pa.S)               | 1.673×10⁻³|
| Saturated vapor pressure of seawater (Pa)  | 806.45    |
| Density of air (kg.m⁻³)                    | 0.01927   |
| Viscosity of air (Pa.S)                    | 8.8e-06   |
| Density of Manganese nodules (kg.m⁻³)      | 2040      |
| Average particle size of Manganese nodules (mm) | 20—50   |
| Concentration of Manganese nodules (%)     | 5—10      |

Reasonable boundary conditions provide a guarantee for the convergence of the simulation. There were several types of boundary conditions in the deep-sea mining pump: import and export boundary, solid wall boundary and dynamic coupling interface. The inlet boundary took the velocity inlet, the velocity was set perpendicular to the inlet cross-section, and the value was calculated according to equation (11); The outlet boundary was set as free flow because of the outflow condition was unknown; The impeller wall was set as a rotating wall; and other boundary conditions were set as the stationary wall. Because the impeller is a rotating part, so the interface must be used between the rotating part and the static part.

\[ v = \frac{Q}{A} = \frac{4Q}{\pi D_0} \]  

where \( A \) is the cross-sectional area of inlet.

4. Simulation Results and Analysis

The cavitation and external characteristics of the pump were explored at different rotational speeds under the conditions with the working flow \( Q = 420 \text{ m}^3/\text{h} \), particle size \( d = 30 \text{ mm} \) and solid volume concentration \( C_v = 8\% \). The steady-state simulation analysis was carried out with the different rotating speed, because of the article length , there are three conditions to be analyzed, 960 t/min, 1450 t/min and 2000 t/min.

At the speed of 1450 r/min, the static pressure of the pump with the full flow was analyzed. The inlet static pressure and outlet static pressure reach -200000 Pa and 800000 Pa respectively. The pressure at the primary impeller is much lower than the pressure at the secondary impeller, it shows as the figure 4. Then the pump guide vane and impeller are selected to be analyzed, The pressure at the inlet of the first-level impeller blades is the lowest, and it is easy to cavitation. Therefore, this paper analyzes the pressure and gas phase distribution of the first impeller, also calculates the necessary \( NPSH \) and head \( H \) of the pump to explore the effect of different rotational speeds on the cavitation characteristics of the pump.

![Figure 4. Full-flow static pressure cloud and impeller vane static pressure cloud](image-url)
4.1. Stress Analysis
It's plotted in figure 5 with the data which was extracted from the first level impeller blade suction surface and the pressure surface of the middle line hydrostatic pressure. It can be seen from the figure that the fluid pressure on the suction side of the impeller is lower than that on the pressure side. Under the three speed the pressure difference is all about 150kpa at the entrance, but it is different at the end of the blade, with the speed increasing the pressure difference increases from 12kpa to 347kpa. The mixed fluid pressure on the pressure side of the impeller blade increases greatly with the increase of the rotational speed. The increase of the fluid pressure on the pressure surface increases from 120kpa at the speed of 960r/min to 460.4kpa at 2000r/min. It can be seen that the greater of the speed, the more effects of the exchange from the kinetic energy to the pressure energy through impeller. When the rotational speed is 960r/min, the pressure of the suction surface of the blade increases with the relative length, and both are higher than 300kpa. From the point of pressure, there is no cavitation under this rotational speed. With the speed increasing to 1450r/min, The minimum pressure of suction surface is -248kpa, it means that the cavitation should occur. At the speed 2000r/min, minimum pressure of the suction is basically the same with 1450r/min, but the trend of pressure rising is slower along the streamline, it means that the blade’s suction side has a large area in low pressure, and the distribution of the vacuoles is more extensive.

![Figure 5](image)

**Figure 5.** Pressure distribution curves on both sides of blades at different rotational speeds

4.2. Analysis of Gas Volume Fraction
Figure 6 shows the vapor volume fraction of the first stage impeller at different rotational speeds. It can be seen from the figure that cavitation occurs mainly at the inlet of the impeller’s back. When the rotational speed is 960 r/min, there is no cavitation occurs, and it's the same as the prediction during the pressure analysis. The kinetic energy is low when the pump at a low speed, and it is difficult to form a low pressure zone which provide the condition for the occurrence of cavitation. During the process of rotating speed from 1450r/min to 2000r/min, bubbles appeared at the inlet of the blade’s suction side, and the bubble distribution gradually disappeared along the streamline. When the rotating speed reaches 2000r/min, a large area of bubbles appeared which takes the 3/4 area of the all, and the gas phase volume fraction is mainly about 0.9. The main reasons are as follows: the fluid transported by the pump is a solid-liquid phase flow, and the manganese nodules are not infiltrated and have greater tensile stress than the rupture of the seawater. Under the effect of virtual mass force and high rotating speed, the fluid gets high kinetic energy, and it leads the local pressure dropping down that promotes the occurrence of cavitation; While at a high speed, the particles have a greater inertia. And the velocity difference between liquid and solid phases will be formed because of the different density, it makes the decrease of pressure near the particles. Therefore, the greater rotational speed, the bigger of the velocity difference between the two phases, the more significant the cavitation is.
4.3. Analysis of Pump Characteristics

1) Lift and efficiency

Head can be calculated by the pump inlet and outlet pressure, the formula is as follows:

\[ H = \frac{P_{\text{out}} - P_{\text{in}}}{\rho g} + \Delta z \]  

(12)

\( \Delta z \) is the vertical distance of the import and export.

The torque \( M \) of the pump’s impeller can be obtained in the simulation. And according to the equation (13), the pump’s hydraulic efficiency can be calculated.

\[ \eta = \frac{QH \rho g}{M \omega} \times 100\% \]  

(13)

2) \( NPSHa \)

The \( NPSHa \) is pump’s device cavitation margin, and it’s also called the effective cavitation margin which represents the surplus energy against the pump’s cavitation. The greater value of \( NPSHa \), less prone to cavitation. And it can be calculated by the following formula:

\[ NPSHa = \frac{P_{\text{in}} - P_{\text{c}}}{\rho g} \]  

(14)

According to the formulas, the \( H \) and efficiency as well as \( NPSHa \) were calculated under different rotational speeds, and the external characteristic curve of pump was obtained like figure 7. As can be seen from the figure, at the rotational speed of 960 r/min, the pump’s cavitation margin value is up to 55.08m, with sufficient ability to resist the occurrence of cavitation, the efficiency has the highest value of 57.29% too, but the head is only 25.04m; when the rotational speed increases to 2000 r/min, the pressure drops quickly, and the cavitation phenomenon is serious with the \( NPSHa \) drops to 2.35m rapidly, the loss of pump's energy causes the lower efficiency with a drop of 27% comparing to the low speed. But at a high speed, the fluid has great kinetic energy under the action of the impeller, and it changes into high pressure energy, so the \( H \) reaches 90.89m. When the speed is the designed, the \( H \) of the pump is 76.64m, which meets the requirement of 70m, the pump cavitation margin value is 5.52m, slightly smaller than the designed, so cavitation will occur, and the pump efficiency drops to 32%. In a summary, the greater speed of the pump, the \( H \) will increase sharply, but the cavitation margin decreases rapidly as well as the efficiency.
In order to avoid the cavitation of deep-sea mining pump, the pump's installation position needs to be discussed. Deep-sea mining pump works below the sea level, and with the increase of depth the pressure increases rapidly, and the lower pump's position, the less prone to cavitation. However, this is not conducive to the pump's installation and maintenance, it also has high requirements on the pump's material and sealing performance. Therefore, at the premise to meet the delivery system requirements, the higher of the pump's installation, the better. According to the equation (15) of land pump’s installation position, it deduced the calculation formula of the installation height of the deep-sea mining pump, and the installation height of the pump under different rotational speeds was discussed [14].

\[ H_s = \frac{P_s}{\rho g} - \frac{P_c}{\rho g} - [NPSH] - \Delta h \]  

(15)

where \( H_s \) and \( P_c \) denote the pump installation height and the pressure of pump inlet flange. 

\([NPSH]\) is permissible cavitation allowance, which is the cavitation allowance for determining the service conditions (such as installation height) of the pump, and it should be greater than the effective cavitation allowance to ensure that the pump will not cavitate during operation. In general \([NPSH] = NPSHa + k \), \( k \) takes (0.3-0.5) m.

\( \Delta h \) represents the hydraulic loss from suction inlet to pump inlet, and it is related with the pressure loss \( \Delta P \) as:  

\[ \Delta h = \frac{\Delta P}{\rho g \mu} \]

The size of solid particles transported by deep-sea mining pump is about 30mm on average, so Englemann’s theory could be used to predict the pressure losses of the system [15]. When the seawater and ore in the pipeline mixed well, the pressure loss in the pipeline which caused by the friction is determined by the following formula:

\[ dP = [(1 - C_v) \rho_\lambda \frac{u_i^2}{2D_i} + C_v \rho_\lambda \frac{u_s^2}{2D_i}]dX \]  

(16)

where \( dX \), \( D_i \), \( C_v \), \( u_i \) and \( u_s \) denote the pipe differential length, the pipe diameter, the ore concentration, the seawater flow rate and the ore particle flow rate.

\( \lambda, \lambda_s \) are the drag-line pipe frictional resistance coefficient to seawater and ore particles.

According to the formula above, the loss pressure can be obtained from \( X = 0 \) to \( X (z) \):
\[ \Delta P[X(z)] = \int_0^{X(z)} [(1 - C_\nu) \rho_i \lambda_i \frac{u_i^2}{2D_i} + C_\nu \rho_i \lambda_\nu \frac{v_x^2}{2D_\nu}] dX \]  

(17)

In accordance with the Englemann empirical formula, the frictional resistance coefficient has the following relationship:

\[ \frac{\lambda_\nu}{\lambda_i} = 48.9 \left( \frac{d_s}{D} \right)^{2.1} \left( \frac{u_i}{gD_i} \right)^{-1.6} \left( \frac{m_i}{m_\nu} \right)^{0.7} \left( \frac{\rho_\nu}{\rho_i} \right)^{2.0} \]  

(18)

\( m_i \) and \( m_\nu \) are the weight flow rate of ore particles and seawater respectively, they can be calculated by \( m_i = Q_i \rho_i \) and \( m_\nu = Q_\nu \rho_\nu \).

According to the Nicholas formula:

\[ \lambda_i = \frac{1}{[1.74 + 2 \log(D_i / 2\Delta)]^2} \]  

(19)

\( \Delta \) is the roughness of pipe wall, and the value is 0.01 in the deep-sea mining system.

The middle warehouse is the first place to store ore in the deep-sea mining transportation system, and then upgrade, the middle warehouse position is under the sea surface about \( L \), the pump inlet position can be approximated as the middle warehouse position. Then the distance from the suction inlet to the pump installation is \( L - H_i \). The pressure loss at this distance is:

\[ \Delta P = (L - H_i) [(1 - C_\nu) \rho_i \lambda_i \frac{u_i^2}{2D_i} + C_\nu \rho_i \lambda_\nu \frac{v_x^2}{2D_\nu}] \]  

(20)

The deep-sea mining pump installation height can be introduced according to the above formula:

\[ H_i = \left( \frac{P_s}{\rho_m g} - \frac{P_i}{\rho_m g} - [NPSH] - L \times [(1 - C_\nu) \rho_i \lambda_i \frac{v_i^2}{2D_i} + C_\nu \rho_i \lambda_\nu \frac{v_x^2}{2D_\nu}] \right) \]  

\[ \times \left( 1 - \frac{1}{\rho_m g} \left( [(1 - C_\nu) \rho_i \lambda_i \frac{v_i^2}{2D_i} + C_\nu \rho_i \lambda_\nu \frac{v_x^2}{2D_\nu}] \right)^{-1} \right)^{-1} \]  

(21)

According to the formula(21) and the related data obtained from the simulation, the installation altitude of the deep-sea mining pump is calculated at different rotational speeds respectively, and Table 2 is obtained. It can be seen from the table that when the speed is 1450r/min, the installation height of the pump is closest to the sea level about 108m; at low speed conditions, the pump can provide cavitation surplus energy, but the low kinetic energy results a low pressure energy, so the pump needs to be installed deeper to make up for the pressure difference; in high speed conditions, the pump cavitation is serious, and it needs to be installed in the depth to reduce the cavitation damage.

**Table 2.** Pump installation height

| Speed(r/min) | Depth of sea level(m) |
|--------------|-----------------------|
| 960          | 112                   |
| 1450         | 108                   |
| 2000         | 119                   |

5. Experiment

Using the simulation experiment system of deep sea mining built by Central South University and Changsha Institute of Mining and Metallurgy, the cavitation performance of the pump was tested. The experimental system consists of five subsystems: a lifting subsystem, a water supply regulator subsystem, an artificial calibration subsystem, a feeding subsystem, a control and measurement subsystem. In the experiment, due to the lack of high-speed cameras and some other equipment, the observation and analysis of the vacuole was not carried out. The E-magE(Electromagnetic Flowmeter), 3151SG(Pressure Transmitters) and SINAMICS6150(Torque speed sensor) were chosen
to measure the inlet gauge pressure, outlet gauge pressure, torque, shaft power and other parameters. Then according the equations to calculate pump’s external characteristics. (figure 8).

![Laboratory equipment](image)

(a). deep-sea mining pump  (b). riser and return pipe  (c). control cabinet

**Figure 8.** Laboratory equipment

During the experiment, with the increase of speed the vibration and noise of the pump are more serious. And according to the calculate, pump head increases significantly with the increase of speed, though the cavitation margin decreases significantly, from 960 r/min to 1450 r/min the $H$ increases 46.6m, the $NPSHa$ and efficiency decreases to 3.87m and 38%. When the rotating speed increased to 2000 r/min, the $H$, $NPSHa$ and efficiency are 82.3m, 0.93m and 28% respectively. By comparing the simulation with the measurement results (figure 9), we can know that the two results are mostly similar, and the law is basically the same. There are also some different because of the hydraulic loss in the real transportation. It verifies the accuracy of the simulation through the experiment.

![Data comparison](image)

**Figure 9.** Data comparison between simulation and experiment

6. Conclusion

According to the research, the main conclusions are as below:

1. Cavitation will affect the flow field in the deep-sea mining pump, which should be considered when analyzing the pump. The squeezing between the blades makes the kinetic energy convert to pressure energy gradually, and the pressure of the first-stage impeller blades is lower than the second-stage impeller blades’. The lowest pressure zone is at the inlet of the first-stage impeller blades where is the area cavitation occurs.

2. With the increase of pump speed, the impeller has a great area of low pressure at the first stage and the cavitation becomes serious gradually, it creates a serious impact on the pump’s blades at high speed. In general the $NPSHa$ and pump efficiency has a negative correlation with speed, $H$ has a positive correlation with speed.
3. The formula of the installation height of the deep-sea mining pump is deduced, and the pump has the best installation height when the revolving speed is 1450r/min. Under this speed the $H$ and $NPSHa$ are good satisfied with the work requirements, that indicates the rationality of the pump speed design. In this study, numerical simulation and simulation experiments are used to study the cavitation characteristics of deep-sea mining pump, and it makes some progress. In the next step other parameters’ effects to the pump’s cavitation characteristics will be explored, and it will be further tested in the sea test.

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