Numerical simulation of solid-liquid two-phase flow in a centrifugal pump with different wear blades degree

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Abstract. Based on the SST liquid turbulence model, using the Euler-Lagrange method to simulate solid-liquid two-phase flow in the centrifugal pump with different degree of wear blades. When the pump was transporting silt-laden flow, numerical simulations with the Euler-Lagrange method to analyse the trajectory of the sands and the mechanism of wear. According to the simulation results, when the flow rate is less than the design flow, the efficiency decreases gradually with the increase of the wear of the runner blade. When the flow rate is larger than the designed flow rate, the efficiency will decrease first and then increase. As the degree of wear increases, the head and power are increasing. With the increase of flow rate, the volume fraction of sand grains distributed in the pressure face of the blades increases, and the distribution of sand grains in the volute outlet change from the uniform distribution of small flow rates to the concentrated distribution of the volcanic surface at large flow rates.

1. Introduction

Dredging pumps are the core equipment in dredging industry, they have been widely used in the projects such as landfill, port and harbor construction, improvement of water environment and ecological restoration and marine energy development. During the operation, there will be a flow separation, reflux and other complex flow phenomenon in the dredging pumps, resulting in the components worn and shorten their service time.

At present, the methods of multi-phase flow field in pumps include theoretical, the model test and the numerical simulation. As the multiphase flow test is difficult and its cost is high. With the development of science and technology, numerical simulation has become the main means of analyzing multiphase flow. The methods of numerical multiphase flow include Euler-Lagrange method and Euler-Euler method according to the methods of solid-phase calculation. [1] At present, the study of solid-liquid two-phase flow include the magnitude of the force between the solid-liquid phases. Li [2], Dong wenlong[3] analyzed the magnitude of each phase force at the different particle size. It suggested that the Basset force should not neglect when the particle size is more than 0.5 mm. When the particle size is larger than 2 mm, the virtual mass force should take into account. When the particle size increases to 5 mm or more should consider Magnus force and Saffman force. Wang Jiaqiong et al [4] predict the solid-liquid flow in the centrifugal pump based on the particle model. Peng et al. [5] used the Euler-
Lagrange method accurately predicts the wear law of the Francis turbine. Many scholars have focused on the trajectories of solid particles in the pump and the solid distribution characteristics in the impeller. However, the numerical simulation of solid-liquid two-phase flow in centrifugal pumps with different degrees of wear and the analysis of the pulsation characteristics and the energy performance of the centrifugal pump under different flow rates is less. So in order to analyze the change of the external characteristics of the centrifugal pump and the change of the wear position in the different wear state of the runner, the numerical simulation of the centrifugal pump with different wear blades is calculated.

2. Numerical Methods

Using Euler-Lagrange model to simulate the solid-liquid two-phase flow, and using the Lagrange method to compute the solid that considered as dispersed phase, while the liquid phase considered as a continuous medium at the Euler point. In the calculation, the velocity difference between the solid phase and the liquid phase is considered. In the numerical simulation, the liquid flow field is preferentially calculated, and then analysis the motion trajectory by calculating the force of the discrete phase model. Through the statistical analysis of the trajectories with a large number of particles, the motion profiles of the particles obtain, and the mutual coupling between the two phases is used.

The assumption in the calculation are as follows:

- The liquid phase is an incompressible continuous fluid, the solid phase is a dispersed phase, the physical properties of each phase are constant, there is a velocity difference between the solid phase and the liquid phase and the temperature remains the same;
- The solid phase is spherical particles with uniform particle size;
- Each group of particles moves from a position along a separate orbit, and the mass and velocity of each group are calculated and tracked along the orbit;
- The mass, momentum, and energy source of the particle phase acting on the fluid are evenly distributed in the unit of the continuous fluid phase in an equivalent amount.

The basic equations in this model including:

Mass conservation equation of continuous fluid phase:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j} (\rho v_j) = - \sum n_k m_k$$

(1)

The continuity equation of solid particles:

$$\frac{\partial \rho_k}{\partial t} + \frac{\partial}{\partial x_j} (\rho_k v_{ki}) = n_k m_k$$

(2)

Continuous fluid-phase momentum equation:

$$\frac{\partial}{\partial t} (\rho v_j) + \frac{\partial}{\partial x_j} (\rho v_j v_i) = - \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i} \left[ \mu \left( \frac{\partial v_j}{\partial x_j} + \frac{\partial v_i}{\partial x_i} \right) \right] + \Delta \rho g_i + \sum \rho_k (v_{ki} - v_i) / \tau_k + \rho_k v_i S_k + F_{k,i}$$

(3)

The momentum equation of Solid phase particle:

$$\frac{\partial}{\partial t} (\rho_k v_{ki}) + \frac{\partial}{\partial x_j} (\rho_k v_{ki} v_{ki}) = \rho_k g_i + \rho_k (v_{ki} - v_i) / \tau_k + \rho_k v_i S_k + F_{k,Mi}$$

(4)

The motion equation of solid particles:
\[
\frac{du_p}{dt} = F_D(u - u_p) + \frac{g(\rho_p - \rho)}{\rho_p} + F
\]

In the formula, \( F_D(u - u_p) \) indicates the resistance of the unit mass particles; \( F \) expressed as the sum of other external forces received by the solid phase, which mainly includes the drag force caused by the inter-phase velocity slip, the pressure gradient force when the velocity gradient is present in the flow field and the turbulence diffusion force.

The force model will be described below:

2.1. Drag force
Drag force refers to resistance of the viscous fluid between the two phases due to the existence of speed slip, and the drag force is related to particle Reynolds number, particle concentration, continuous phase turbulence and other factors.

Schiller-Naumann’s drag coefficient \([6]\) in the numerical simulations:

\[
C_D = \left(1 + 0.15 \Re^{0.87}\right) \frac{24}{\Re_p}
\]

2.2. Pressure gradient force
The pressure gradient force refers to the additional non-uniform pressure caused by the pressure gradient of the continuous phase in the flow field where the discrete phase particles are present in the pressure gradient.

For multi-fluid models, most people think that pressure is only a function of the spatial position, and that the phases are at the same pressure at the same location. Thus, the pressure gradient force per unit volume of the mixed fluid subjected to the discrete phase particles:

\[
F_{\text{pp}} = -\alpha_p \nabla p
\]

Continuous phase will be subject to pressure gradient force, liquid pressure gradient force are as follow:

\[
F_{\text{pl}} = -\alpha_l \nabla p
\]

The negative sign indicates that the pressure gradient force is opposite to the direction of the pressure gradient. Subscript l, p, respectively, represent the liquid phase, solid phase.

2.3. Turbulence diffusion force
Turbulence diffusion force is the additional force the dispersed phase in the continuous phase velocity pulsating by the high concentration region to the low concentration region of Burns \([7]\) proposed a Favre Averaged Drag model (for example, the FAD model) for calculating the turbulence diffusivity of the dispersed particles in the multiphase mixture. In the mixed fluid of per unit volume, the turbulent diffusion force of the dispersed phase particles is:

\[
F_{\text{TD,p}} = -\frac{3}{4} C_{TD} C_{D,lp} \alpha_p \rho_l \frac{\nu_{l,l}}{\sigma_{l,l}} |u_l - u_p| \left( \frac{\nabla \alpha_l}{\alpha_l} - \frac{\nabla \alpha_p}{\alpha_p} \right) F_{\text{D,p}}
\]

In the formula, \( \sigma_{l,l} \) is the continuous phase of the turbulence Schmidt number, the empirical value of 0.9: CTD the turbulence diffusion coefficient, the empirical value is 1.0 \([7]\). Due to the lack of relevant test data, the empirical coefficients in the model take the recommended empirical values.

The wear prediction model used in this paper is Finnie model, which mainly considers the influence
of discrete particle velocity, collision angle and particle quality on wear [8]. The wear rate on the wall is predicted by the following equation (8):

$$\rho_{\text{Err}} = \sum_{p=1}^{N_{\text{parties}}} \frac{m C(d_p) f(\alpha) V^{b(v)}}{A_{\text{face}}}$$

In the formula, $\rho_{\text{Err}}$ is the wall wear rate, $m$ is the Particle mass flow rate; $C(d_p)$ is the function of particle diameter, $\alpha$ is the collision angle of the particle trajectory with the wall; $f(\alpha)$ is the function of the collision angle; $v$ is relative particle velocity; $b(v)$ is the function of relative particle velocity; $A_{\text{face}}$ is the area of the grid area on the wall.

3. Physical model and boundary conditions

The object of this study is a three-blade centrifugal dredging pump. The model consists of an inlet section, a runner and a volute, and an extension of the outlet. The main parameters of the pump are: design flow rate $Q=200 \text{ m}^3/\text{h}$, design head $H=32 \text{ m}$, design rotating speed $1450 \text{ rpm}$, the blade of outer diameter is $175\text{mm}$, the inlet diameter is $150\text{mm}$. The inlet and outlet sections are extended the inlet diameter by 3 and 4 times, respectively. I don’t consider gap between the blade and the runner during the calculation. The centrifugal pump three-dimensional modeling shown in figure 1, and the mesh are shown in figure 2.

![Figure 1](image1.png)
**Figure 1.** Centrifugal pump three-dimensional fluid domain

![Figure 2](image2.png)
**Figure 2.** Rotor and volute mesh

Through the grid-independent inspection, the final grids are as follows: the import section of about 250000, the export section of about 750000, volute grid number of about 100 million, and the wheel part of about 1.2 million. The three-dimensional shapes of the blades with different degrees of wear shown in the following figure 3, the blade of non-worn is orange, the blade of $1.72\%$ worn rate is blue and the blade of $5.56\%$ worn rate is grey.
The settings for the two-phase flow calculation parameters and the boundary conditions shown in the following table:

**Table 1. The boundary conditions of the two-phase flow**

| inlet               | Fluid velocity, solid mass flow, velocity |
|---------------------|--------------------------------------------|
| outlet              | Static pressure                            |
| Boundary conditions | No slip boundary condition                  |
| interface           | Frozen rotor                                |
| Turbulent model     | SST k-ω                                     |
| Particle size       | 0.5mm                                      |
| Particle density    | 2300kg/m³                                   |
| Solid phase mass concentration | 10%                                      |

The recovery coefficient of the vertical collision between the solid phases in the calculation is set to 0.8 and the recovery factor for the parallel collision is set to 1.0. Because the relative position of the impeller and the partition has a great influence on the flow characteristics and hydraulic performance of the solid-liquid two-phase flow, the relative position of the three blades is calculated for each flow condition, and the calculated parameters are average of three relative position with the relative positions of the vertebral tongue.

4. Analysis of numerical simulation results

4.1. **The change of the external characteristics at the different degree of wear blades**

The simulation of solid-liquid two-phase based on the accurate numerical simulation results of flow field. The simulation is steady. Based on the numerical simulation of the flow field, the solids are injected from the inlet. Because of the different flow field distribution of the solid-liquid two-phase flow in the impeller with different wear degree under the same flow rate, the Qd, 0.5Qd and 1.3Qd flow rates are studied in three different wear states, which are unworn, 1.72% wear rate and 5.56% wear rate. The measure of the degree of wear is the ratio of the mass of the worn blades to the total mass of the blades. After analyzing the numerical simulation results, the external characteristics of the centrifugal pump are as follows:
We can found out that when the flow rate isn’t more than the design flow rate, the efficiency gradually reduces, the head and power increase with the increase of the blade worn; When the flow is greater than the design flow rate, with the increase of the blade worn, head and power increase, efficiency reduce first and then increased. This is because when the blades are light worn, the flow field within the wheel deteriorated, the efficiency is reduced; However, when the wear occurs further, the increase in the degree of wear is equivalent to an increase in the flow area, to some extent equivalent to reduce the flow rate. According to the performance curve of the centrifugal pump, when the flow rate reduced, the head increases. When the flow rate is less than the design flow, the centrifugal pump efficiency reduces when the flow rate decreases; when the flow rate is greater than the design flow, the centrifugal pump efficiency increase when the flow rate decreases but is still larger than the design flow.

4.2. The characteristics of Sand movement
Based on the simulation of water, solid particles add. Because of the similarity of particle motion at the same flow rate, the flow field and the trajectories of sand particles in non-wear state at 0.5\( Q_d \), \( Q_d \) and 1.3\( Q_d \) were calculated. The figure 6 below shows the flow characteristic for solids and liquids.
Observe the centrifugal pump solid-liquid trajectory in different flow rate can found out that there are more swirl in impeller at small flow rates. With the increase of flow rate, the internal flow of the impeller gradually improved, the whirl reduce, the coincidence of the solid particles and the trajectory of the liquid was gradually increased, the followability of the solid particles was improved, and the wear of the impeller blades reduce. After leaving the runner, the particles due to the role of centrifugal force, a large number of particles tend to volute surface.

4.3. The characteristics of blades wear

Particle movement is different with the different flow rates, which leads to different particle trajectories. The figure below shows the distribution of the sand at the same wear rate (5.56% wear rate) but different flow rates, the cross-sectional position shown as the middle section of the impeller at 50% span.
Figure 7. Particle concentration distribution at different flow rates

After analysis of the particle concentration distribution we can find out that when the flow is less than the design flow, the particles are concentrated in one of the runner passages, and there are swirls inside the runner. A large number of sand are distributed in the hub and the runner flow path, the spherical surface of the volute is low. Due to the presence of axial vortices, the distribution of sand in a passage is uniform. In the design flow, the sand flow within the runner is roughly equal, the sand concentration gradient is large, and the distribution within the runner is periodic. As the sand has a centrifugal force, volcanic surface sand particles are more distributed. When the flow rate is larger than the designed flow rate, the sand distribution in the runner is roughly equal, but the sand concentration gradient is small, and the sand distribution in the runner is also periodic, volcanic surface sand particles are more distributed. The uniformity of the distribution of sand grains do not develop due to the sands grains to the pressure face of the blades. With the increase of flow rate, the volume fraction of sand grains distributed in the pressure face of the blades increases, and the distribution of sand grains in the volute outlet change from the uniform distribution of small flow rates to the concentrated distribution of the volcanic surface at large flow rates.

Due to the different trajectories of particles, and ultimately lead to different parts of the surface wear characteristics. Through the numerical simulation of the sand flow, the wear rate density of the surface can obtain which can qualitatively analyze of the degree of blade wear. The following figure is the blade wear distribution at the same wear degree (5.56% wear rate), different flow rate.
By analyzing the wear characteristics of the runner at different flows, it can be found that the blades' wear is concentrated on the blades' pressure and suction side at lower flow rate. As the flow increases, the wear of the blade pressure side gradually decreases, but the degree of suction side decreases larger; when the flow is greater than the design flow, the wear of blade suction side close to zero, but the blade pressure side wear is larger than the design flow. This is because when the flow rate is less than the design flow, the liquid and solid entering the runner dashed on the suction side of the blade. When the flow rate is greater than the designed flow rate, the fluid and solid impact on the pressure side of the blade. Due to the limited blade, the axial vortex exists in the runner, which makes the blade wear of the suction side more serious at small flow rate, and the blade wear of the pressure side more serious under the large flow rate.

In this paper, the effects of three flow rates, three kinds of wear degree, three relative positions of blade and volute calculated out, and the wear rate density on the runner obtained. The total wear rate on the runner is determined by the integral wear rate density on the overcurrent surface, the unit is kg/s,

The expression is as follows: $E_r = \int \rho \, ds$. The relative value of the wear rate equal to the wear rate compared with the maximum value obtained. Figure showed the relative value of the wear rate with different flow rate at 1.72% of blade wear. Figure showed the relative value of the wear rate with the effects of three flow rates, three kinds of wear degree.
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Figure 9. the relative value of the wear rate with different flow rate at 1.72% of blade wear

Figure 10. the relative value of the wear rate with the effects of three flow rates, three kinds of wear degree

At the same degree of wear, as the flow increases, the relative value of the wheel wear decreases first and then increases. This is because when the flow is less than the design flow, there is a whirlpool inside the runner, the return is more serious and the efficiency is low. As the flow rate increases, the amount of reflow from the scroll tongue decreases, but when the flow is greater than the design flow, will also cause the pressure side wear increases, resulting in increased wear relative value.

We can also found that the relative value of wear increases with the increase of wear degree when the flow rate is not more than the design flow. When the flow rate is larger than the designed flow rate, the relative value of the wear of the runner increases first and then decreases, this is because in the process of increasing the degree of wear, the efficiency of the pump first reduced and then increase.

5. Conclusion
In this paper, the solid-liquid two-phase flow is simulated by three flow conditions, three kinds of wear degree and three relative positions of blade and volute tongue. Using the SST liquid-phase turbulence model and the Euler- Lagrange method to simulate the centrifugal pump with different wear blades. The change of the external characteristics of the centrifugal pump and the distribution of the internal particles in the runner and the relative value of the wear blade under different flow rates obtain by numerical simulation. The main conclusions are as follows:

When the flow rate is less than the design flow, the efficiency decreases gradually with the increase of the wear of the runner blade. When the flow rate is larger than the designed flow rate, the efficiency will decrease first and then increase. As the degree of wear increases, the head and power are increasing.
When the flow rate is less than the designed flow, the wear of the suction side is more serious than the pressure side, when the flow is greater than the design flow the blade pressure side wear is more serious. After leaving the runner, the particles due to the role of centrifugal force, a large number of particles tend to volute surface, resulting in increased wear on the volute surface. With the increase of flow rate, the volume fraction of sand grains distributed in the pressure face of the blades increases, and the distribution of sand grains in the volute outlet change from the uniform distribution of small flow rates to the concentrated distribution of the volcanic surface at large flow rates.

The blade wear is smallest at design flow, followed at the greater than the design flow, the wheel wear the largest at less than the design flow; When the flow is greater than the design flow, with the increase in wear, the relative value of wear increased first and then reduced; When the flow rate is not more than the design flow, the relative value of the wear increases as the wear degree increases.

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