

# Numerical analysis of internal complex flow in a deep - sea gas - liquid two - phase flow multistage centrifugal pump

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**Abstract.** Based on a multistage gas-liquid centrifugal pump as the research object, considering the buoyancy and surface tension of gas at the same time, the  $N-S$  equations and the standard two equations  $k-\varepsilon$  turbulent model were used for the numerical simulation of triple-stage pump of different gas content, the external characteristics and the internal flow field of the pump were analyzed. The results indicated that with the increasing of gas content, the head and efficiency of the pump were reduced, and the decrease rate increases with the increase of the gas content. Under the action of centrifugal force, the gas phase in the impeller mainly gathers near the hub. In this paper, the method of opening the return hole was used to optimize the pump, and the modification was calculated by steady numerical simulation. By analyzing the pump liquid velocity distribution, the axial force and the performance of the pump, to provide the basis for optimizing the performance of the pump.

## 1. Introduction

Pump is a widely used general machinery, and it is a very common power equipment in daily production [1-4]. The actual pumping medium is gas-liquid two-phase flow mixed fluid in the engineering practice, which brings a series of hazards to the operation of the pump, especially in the deep sea oil and gas exploration projects. At present, China's research is not very mature for high efficiency, high lift and mixed pump in actual production engineering under high pressure [5]. Therefore, the research and improvement of gas-liquid two-phase flow pump is of great significance to the promotion of related projects [6]. In this paper, the object is multi-stage gas-liquid mixing pump, ANSYS CFX16.0 software was used to simulate the three-stage full-flow steady-state numerical simulation of the pump under various operating conditions. The external characteristics and internal flow field of the pump were analyzed.

According to the main purpose of this paper, the author used the Euler - Euler model to simulate the gas - liquid mixed pump [7-8].

## 2. Model building and meshes

### 2.1. Basic parameters subsection

This paper chosen the multi-stage centrifugal gas-liquid mixed pump, which designed by the national natural science foundation of china as the main research object. In order to meet the design goals and efficiency of the demand, the pump was designed as a 25 stage centrifugal pump, each level includes



runner, diffuser and diversion cavity three main parts. In the study of numerical simulation, taking into account the constraints of the calculation conditions, the flow field of the study was simplified, and for the triple-stage pump, a full-flow numerical simulation was carried out. The relevant design parameters of the pump are shown in table 1.

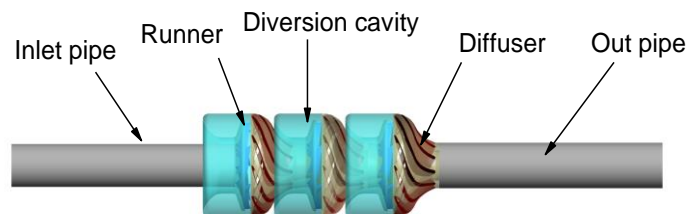
**Table 1.** Design parameters of gas - liquid mixed pump

Flow rate $Q(\text{m}^3/\text{h})$	Rotate speed $N(\text{r}/\text{min})$	Lift $(\text{m})/\text{stage}$	Pressure range $(\text{MPa})$	Efficiency $H(\%)$
26.5	3500	26	30~35	70

## 2.2. Model Description

Using Solid works and UG software, three-dimensional solid modeling of multi-stage centrifugal pump was designed and geometric model was simplified. In the process of simplification, the simplified structure did not affect the effect of numerical simulation.

The main flow components of the multi-stage gas-liquid mixed pump include the runner, the diffuser and the diversion chamber. The number of impeller blades is  $z=7$ , the number of vane leaves is  $y=10$ , the conveying mediums are gas and water. In this paper, the geometric model of the triple-stage centrifugal pump is shown in figure 1.

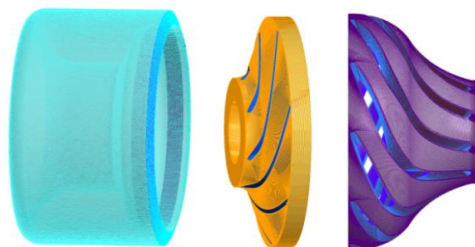


**Figure 1.** Sketch of flow-through components of pump

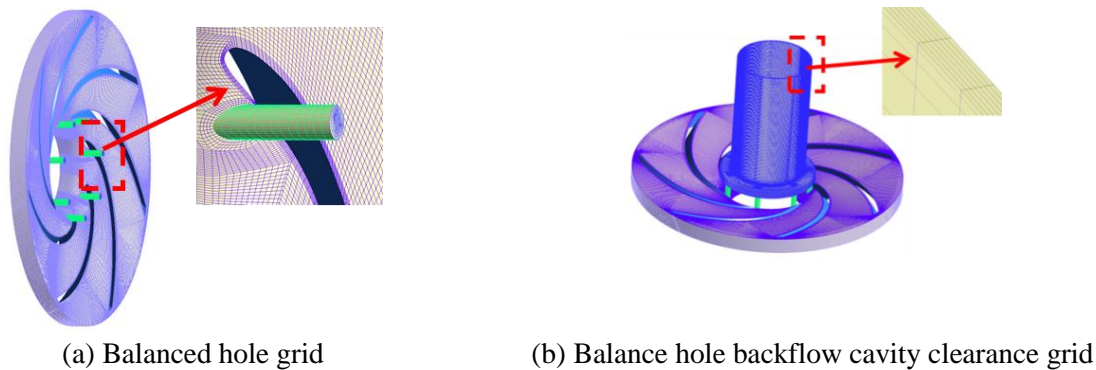
## 2.3. Meshes division

The ICEM software was used to divide the mixed pump, and the fluid domain was separated by the hybrid grid. The runner and the diffuser are the main overcurrent components of the pump, which also were the focus of the study on the computational fluid domain of the pump. The dimension of the grid is controlled at the connecting surface of the diversion chamber and the runner and the diffuser to realize the docking of each part of the grid. Then the unstructured grid is divided into the two sides of the source grid. Through the test of mesh independence, finally determine the size and number of grids. The grid distribution results are shown in figure 2.

The axial force problem of multistage pump has been the key problem of multi-stage pump selection and design [9]. In order to reduce the axial force of the pump, the method of setting the return hole in the rear plate of the runner was studied, and the effect of the method on the performance and axial force of the pump is analyse in this paper. The grid of the balance hole is shown in figure 3.



**Figure 2.** Grids of main computational domains



**Figure 3.** The local grid after the modification

### 3. Numerical calculation method and boundary condition

The numerical calculation of the multi - stage centrifugal pump was carried out in ANSYS CFX 16.0 software.

#### 3.1. Basic assumptions

This paper calculated the external characteristics and internal flow characteristics of the pump in the case of both water and gas-liquid two-phase media. The following basic assumptions were used in the calculation [10]:

The inlet of the fluid domain is mixed with gas and water two-phase uniform. The main phase of water and the second phase for the gas.

Considering the compressibility of the gas phase, the gas phase and the liquid phase are all continuous phases under different gas volume fraction (GVF) conditions.

As the medium flowed through the device for a short time and the heat dissipation was small, it is assumed that there was no heat exchange between the system and the outside. The heat exchange between the gas and liquid phases was ignored.

#### 3.2. Numerical simulation

Boundary conditions setting: The inlet of the fluid domain is a stationary part, with a total pressure inlet condition,  $p=3$  atm, the reference pressure is 1 atm. The export used the mass flow boundary condition, and calculates the domain wall with the non-slip mesh function. In the steady-state calculation process, the "Frozen rotor" is used for the interface of the dynamic and static domain, the calculation was set to the high-order solution precision, the convergence residual RMS was set to  $10^{-5}$ , the physical time step was used, the Physical Timescale was set to  $1 / \omega = 0.003s$ , the definition of the rotor part of the rotation field, the speed of 3500 r/min.

The two-phase flow model selects the mixed model (Mixture), the phase transfer unit is  $0.5mm_2$  and is calculated by velocity-pressure coupling. Considering the buoyancy of the gas, the direction is the same as the flow direction of the fluid, along the positive direction of the Z axis, the gravity is  $9.81m/s^2$ . Finally, the surface tension coefficient was set at  $0.073$  N/m to reflect the effect of surface tension on fluid flow. The standard  $\kappa$ - $\varepsilon$  turbulence model was selected in the CFX numerical simulation software.

After the modified optimization of the runner, the numerical simulation setting was slightly different. The runner and the balance hole rotate together, so it was set as the rotation field. The return cavity part was connected with the rotating balance holes, therefore, it was necessary to set the rotating wall, with the wheel speed consistent [11].

### 4. Results analysis

#### 4.1. Analysis of external characteristics

The calculation of the head and efficiency of the multiphase pump is different from that of the single-phase pump. Because of the compressibility of the gas phase, the Euler formula of the lift head is no longer applicable. Through learning from previous studies on multiphase pump characteristics definition, eventually without considering the gas energy change and the whole system with no external heat exchange under the condition of constant temperature [12], this paper simplify the pump head and efficiency calculation formula. The results are as follows:

Mixing pump head calculation:

$$H = (1 - \alpha)H_l + \alpha H_g \quad (1)$$

The expressions of liquid phase and gas head are as follows:

$$H_l = \left[ \frac{P_2 - P_1}{\rho_l} + \frac{1}{2}(\overline{v_c^2} - \overline{v_j^2})_l + g(z_c - z_j)_l \right] / g \quad (2)$$

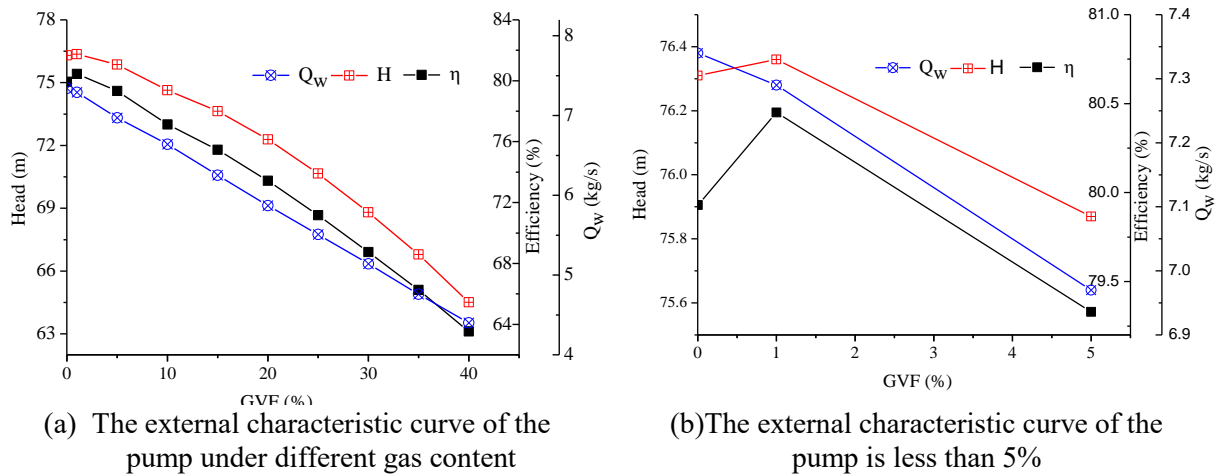
$$H_g = \left[ R_n T \ln \frac{P_2}{P_1} + \frac{1}{2}(\overline{v_c^2} - \overline{v_j^2})_g + g(z_c - z_j)_g \right] / g \quad (3)$$

Where:  $\alpha$  is the gas mass fraction,  $\alpha = m_g / (m_l + m_g)$ ;  $R_n$  represents the molar constant of the gas of the unit mass,  $R_n = R / M$ ,  $M$  represents the molar mass of the gas, and  $T$  represents the calculated temperature is 298.15K. Formula for calculating efficiency of multiphase pump:

$$\eta = \frac{P_{out}}{P_{in}} \quad (4)$$

Where: pump input power  $P_{in} = Torq \cdot R_o$ , pump output power:  $P_{out} = \rho_{mix} g Q H$ .

The hydraulic performance of the pump was predicted under several different gas content conditions, as shown in figure 4.



**Figure 4.** Performance curve of multiphase pump with different GVF

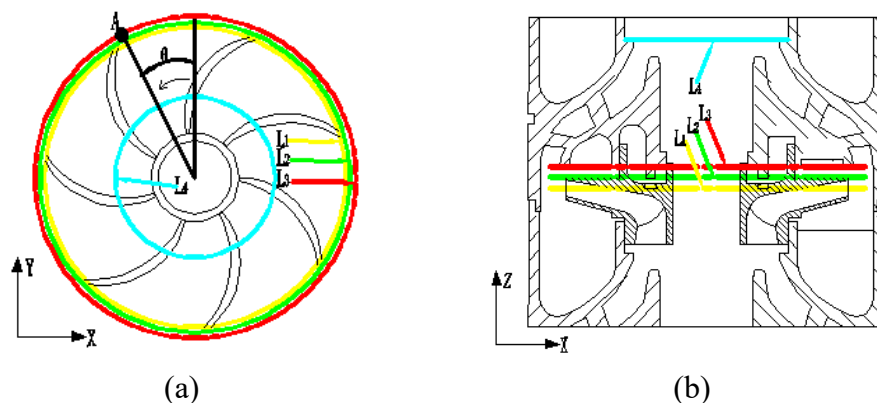
Figure 4 (b) can be seen 0-5% gas content of the head and the efficiency of the specific circumstances, and there is a maximum in the 1% gas rate conditions. This paper calculates the gas content of 1% as the initial condition of the mixed flow pump, comparing with other working conditions. In figure4 (a), it can be seen that as the inlet gas rate increases, the lift curve of the pump is significantly reduced. Corresponding to the minimum gas content of 1% of the case, the head reaches a maximum of 76.36m. When the gas content of 40%, the head down to 64.51m, a total reduction of 11.8m. This is mainly because the existence of gas made the mixed liquid in the pump flow state changes, especially when the gas content increases, the gas phase and liquid phase mutual impact, resulting in two-phase head were produced to varying degrees of reduction, and ultimately have a great impact on the head of the pump. The efficiency curve is similar to the trend of the head curve.

With the increase of the gas content, the efficiency of the pump is declining and the descending rate is increasing. When the gas content is 1%, the efficiency of the pump is up to 80.45%. When the gas content is 40%, the efficiency of the pump is reduced to 63.53%. It is easy to see through the mass flow curve that the mass flow rate is decreasing as the gas content increases, when the volume of the inlet pump is constant.

#### 4.2. Internal flow analysis of mixed pump

Under the condition of design flow, the internal flow problem of the mixed pump in the cases with mixed pump inlet gas volume fraction are 1%, 10%, 20%, 30% and 40% respectively were explored, and the simulation results were analysed.

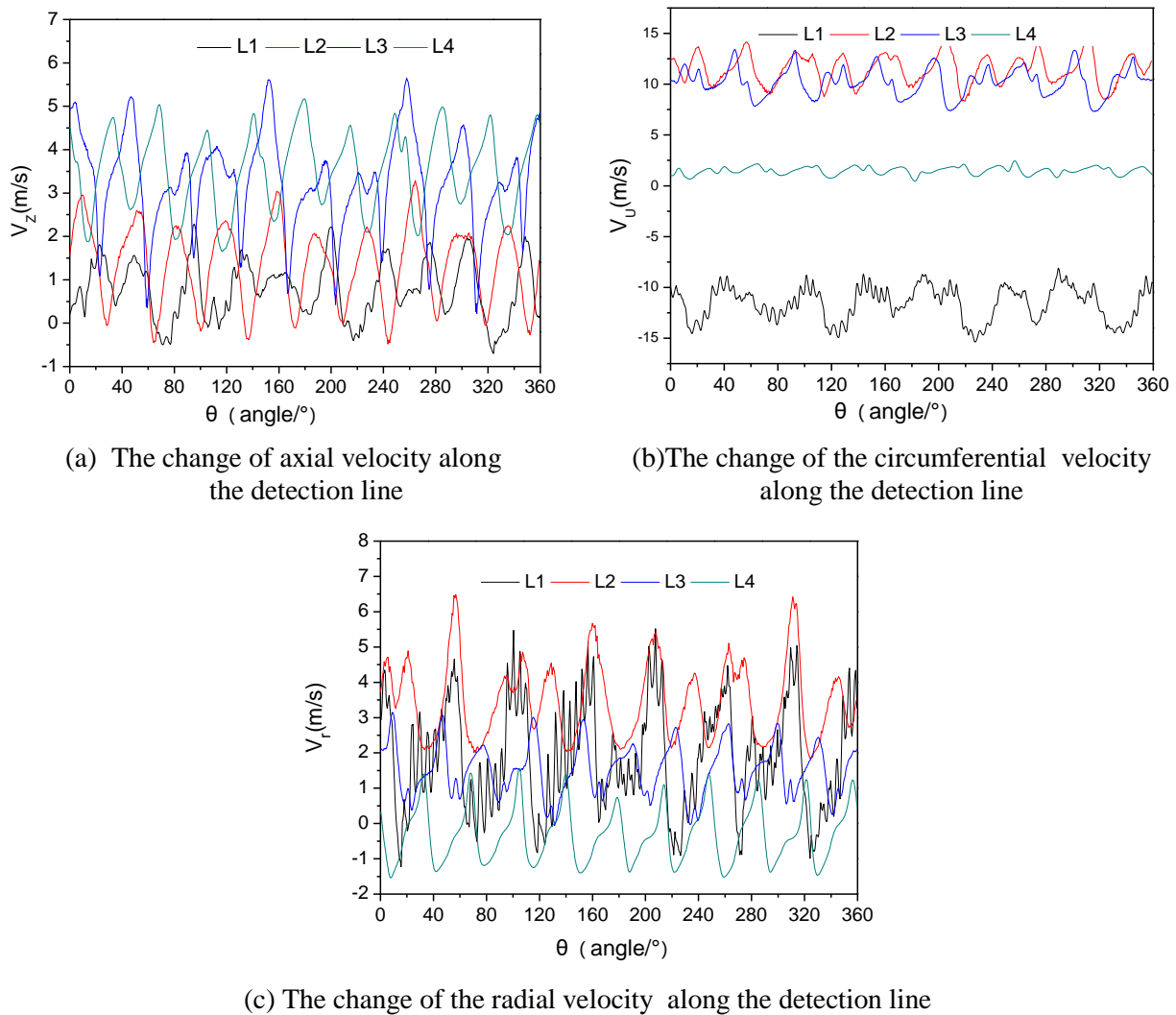
**4.2.1. Analysis of liquid flow in static and dynamic coupling region.** In order to analyze and study the flow state and the change of the velocity component of the liquid phase fluid in the static and dynamic coupling surfaces of the runner and the diffuser, detection lines L1, L2, L3 and L4 are selected respectively in the static and dynamic combined region. The angle is rotated counterclockwise from the beginning of point A in figure 5 (a), the range of  $\theta$  is  $0^\circ$ - $360^\circ$ , used to detect changes in liquid flow. figure 5 (b) shows the different positions of the four test lines on the Z axis, and L1 is the center of the outlet of the pump runner to detect the velocity of the liquid phase after passing through the runner. L2 for the mixing pump runner and diffuser transition intermediate position, used to detect the region of liquid fluid flow changes in the intermediate process. L3 for the mixing pump diffuser inlet center position, used to check the pump into the diffuser liquid speed components of the specific circumstances. L4 is the location of the outlet of the pump diffuser, used to compare the speed of the liquid phase at the inlet and the exit speed of the diffuser.



**Figure 5.** The position of the test line in the pump

Figure 6 shows the changes in the axial, circumferential and radial velocity components of the liquid phase velocity of the pump on the first stage of the mixing pump at 20% gas rate. Other cases are similar and the analysis results are as follows:

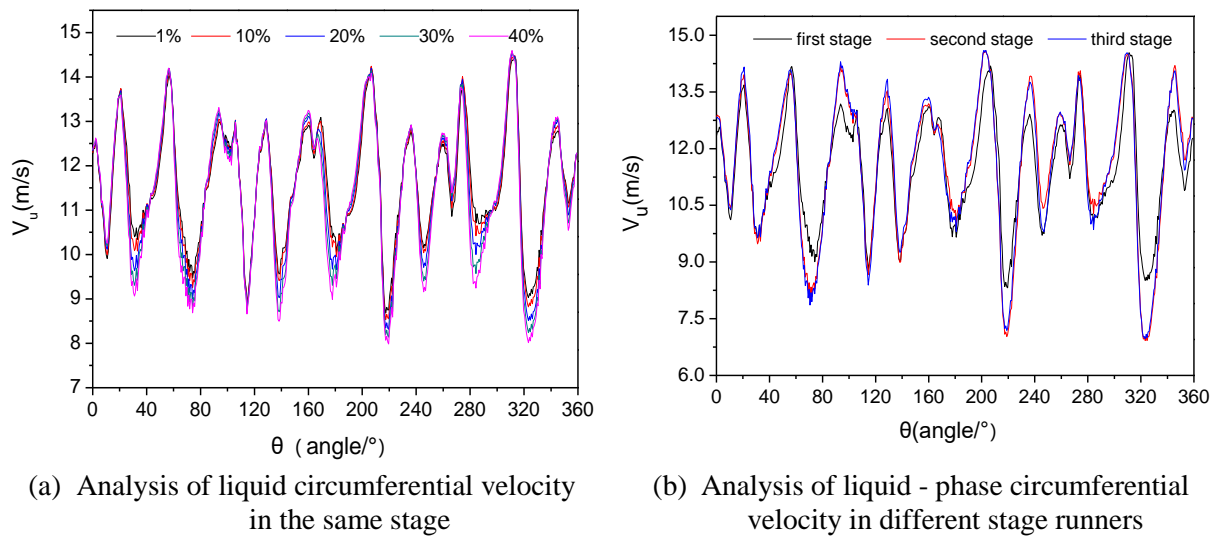
By comparing and analyzing the three velocity components, it can be seen that the axial velocity component on the runner exit detection line L1 is the smallest, and the liquid phase is mainly radial and tangential velocity under the centrifugal force of the pump runner, the circumferential velocity direction of the runner exit is opposite to the tangential velocity of the diffuser inlet. This is related to the design of the guide vane in the diffuser, and the opposite circumferential velocity can better convert the kinetic energy of the circumferential velocity of runner into axial kinetic energy, to achieve the purpose of raising the liquid head. Compare the size relationship of the three velocity components on the L4 detection line of the diffuser exit, the circumferential velocity is the smallest and relatively stable. The radial velocity component changes along the detection line with 10 major peaks, which is consistent with the number of guide vanes of the diffuser, and the variation range is between -1.5m/s and 1.5m/s. That is, the guide vane has a certain influence on the radial flow.



**Figure 6.** The change of the velocity components of the liquid phase along the detection line

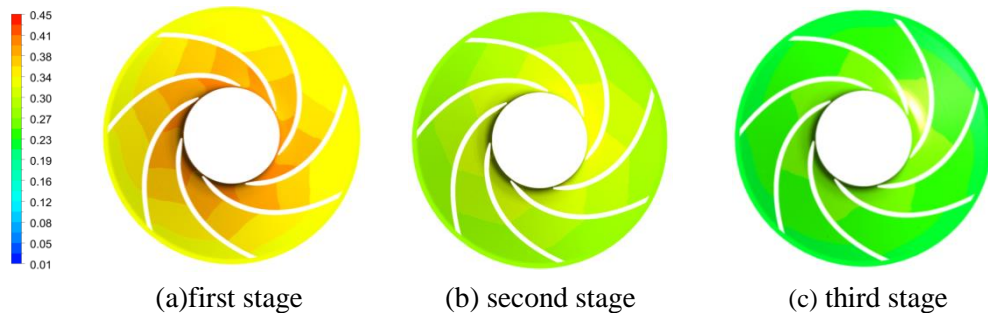
Figure 7 (a) in different gas rate conditions, the circumferential velocity component changes in primary runner L2 detection line. It can be seen from the figure that the change of circumferential velocity component on the detection line is similar to that under different gas-bearing conditions. The change is affected by the common influence of the two structures of the runner and the diffuser, there are 10 major peaks (related to the guide vane setting position) and some small variations. The peak position of the set is the corresponding position of the flow channel, and the trough is mainly the corresponding position of the guide vane. In the vicinity of the guide vane inlet, the higher the gas content is, the higher the circumferential velocity of the liquid phase is, the stronger the instability. Figure 7 (b) shows the liquid circumferential velocity distribution of the mixing pump on the detection line L2 at a gas content of 20%. It can be seen from the figure that the variation of the circumferential component of the liquid velocity at all levels of runner is very similar in the dynamic and static regions. This is because under the same gas content, the internal structure of the hybrid pump plays a major role in the velocity distribution of the liquid phase. The design of the structure of the impeller and the diffuser at all levels of the hybrid pump is consistent. Therefore, the distribution of circumferential velocity components is similar.





**Figure 7.** The liquid velocity analysis on the L2 test line

**4.2.2. Gas distribution in the cross section of runner.** Figure 8 shows the gas volume fraction of the 0.5 span of runner at all stages of the gas pump with a gas content of 40%. It can be seen from the figure8 that the gas is mainly distributed at the position of the wheel near the hub of the runner, the accumulation of gas often causes the blockage of the runner channel, which is easy to cause the unstable development of the fluid state in the flow channel and reduce the performance of the pump. The gas volume fraction of the first runner is the largest, and the final volume is the smallest, it is mainly because the gas phase is the compressible gas. With the increase in the series, the basin pressure continues to increase, the gas was constantly compressed, so the gas volume fraction gradually reduced.



**Figure 8.** The GVF distribution of the 0.5 span

## 5. Analysis of calculation result after modification

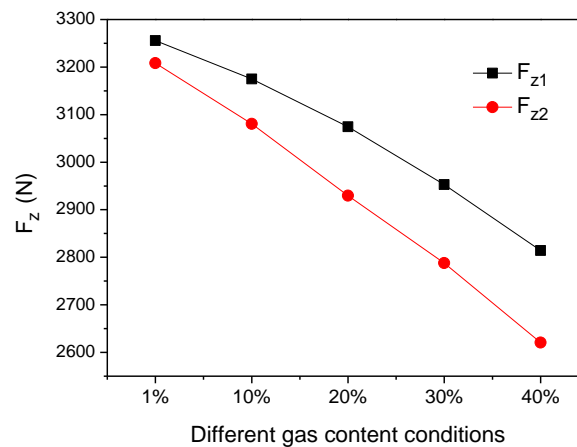
### 5.1. Analysis of axial force of mixing pump runner with different gas content after modification

For the mixed pump, the axial force is the vector sum of the force in the direction of the Z axis, the positive direction of the Z is positive, and the opposite direction is negative. The axial force of the rotor of the pump should be composed of the following parts: the axial force of the outer surface of the impeller front cover is  $F1$ , the axial force of the outer surface of the impeller rear cover is  $F2$ , the dynamic reaction of the fluid in the runner at each level is  $F3$ , the axial force of the inner surface of the impeller front cover is  $F4$ , the axial force of the inner surface of the impeller rear cover is  $F5$ , the resultant force of gravity along the Z axis is denoted by  $F6$  [13]. The internal force of the inner and outer surfaces of the front and rear of the pump were regarded as the internal force of the system, and

its resultant force is often recorded as 0. In this section, the change of the axial force is mainly studied and the influence of gravity is neglected. Finally, the axial force of the rotor of the mixed pump can be expressed as the axial force:

$$F = F_1 + F_2 + F_3 \quad (5)$$

Figure 9 is the change of the axial force of the pump before and after the modification.  $F_{z1}$  indicates the axial force of the pump before modification,  $F_{z2}$  indicates the change in the axial force of the pump after the modification. It can be seen that the axial force on the pump increases as the gas content increases. Because the interference of gas-liquid two-phase flow increases, and the energy obtained by the fluid is reduced. So the dynamic reaction force of the front and rear cover plates and the runner blades of the mixed flow pump will be reduced. It can be clearly seen from the figure that the axial force of the pump rotor is lower than that before the adjustment. The difference in the total axial force increases with the increase of the gas content. Obviously, when the maximum difference is 40% gas content, the axial force before and after the modification is reduced by 193.65N.



**Figure 9.** The change of axial force of runner before and after modification under different gas content conditions

### 5.2. Changes in the performance of mixed pump before and after the modification

The purpose of opening the balance hole is to reduce the axial force of the multi-stage pump, but because of the high pressure fluid back to the relatively low pressure wheel hub, the runner must form a certain disturbance, affecting the stability of the fluid within the runner. It often brings the shortcomings of reducing the efficiency of the pump. Therefore, the opening of the balance hole and the search for the best conditions for opening the balance hole is the key to perfect the balance hole to reduce the axial force.

Figure 10 shows the changes in the performance curve of the pump before and after opening the balance hole, the foot sign "1" indicates the head and efficiency of the pump before the change, and the footer "2" indicates the lift and efficiency of the pump after modification. It can be seen that in the case of low gas content, the way of opening the balance hole for the pump head and efficiency of the impact is not important. But with the increase in gas content, the way to open the balance hole will result in a substantial reduction in the efficiency of the pump head, that is to say, the reduction of the axial force of the sacrificial mixing pump efficiency under the condition of high gas content is no longer applicable.

In order to further find the best applicable conditions, define two percentages  $\zeta$  and  $\psi$ , which are defined as follows:

$$\zeta = \frac{F_{z1} - F_{z2}}{F_{z1}} \times 100\% \quad (6)$$

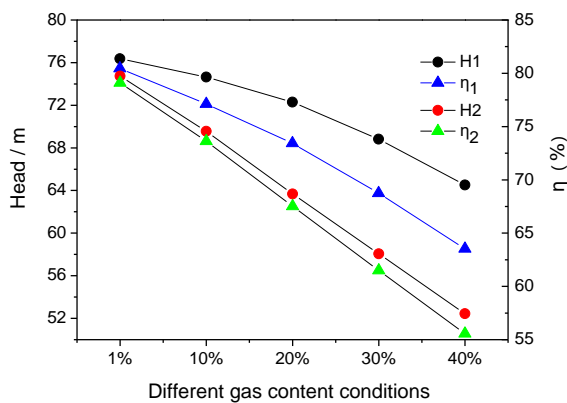


Where:  $F_{z1}$ 、 $F_{z2}$  show before and after the opening of the balance hole respectively, the pump rotor axial force of the joint force.

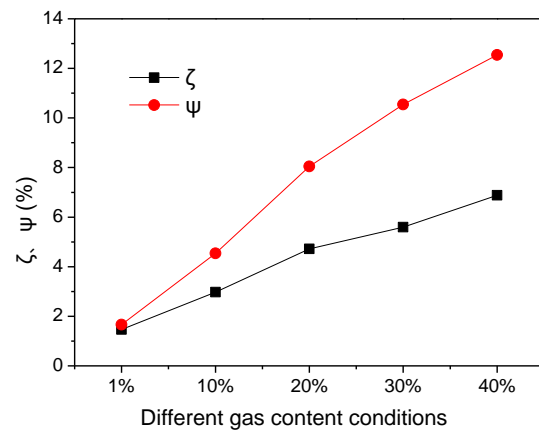
$$\psi = \frac{\eta_2 - \eta_1}{\eta_1} \times 100\% \quad (7)$$

Where:  $\eta_1$ 、 $\eta_2$  show the opening of the balance hole before and after respectively, the transmission pump efficiency.

Figure 11 shows the change in the two change coefficients as the gas content increases, it can be seen that in the case where the gas content is less than 20%, the reduction in the performance of the pump and the decrease in the axial force are still in the range of engineering acceptable. However, when the gas content is greater than 20%, the rate of change of the efficiency of the pump is increased and the decrease rate of the axial force decreases. In this case, the use of the balance hole to reduce the axial force of the way to become undesirable should be used before the comprehensive consideration.



**Figure 10.** Performance curve of pump before and after modification with different conditions



**Figure 11.** Variation coefficient of axial force and efficiency of pump before and after modification

## 6. Conclusion

The gas phase is mainly distributed in the vicinity of the wheel hub, and the volume fraction of the gas is gradually reduced due to the increasing pressure.

The flow at the static and dynamic junction of the mixing pump is mainly affected by the guide vanes in the diffuser. The results of the liquid velocity components show a certain fluctuating situation in the circumferential direction. After the diffuser, the radial and tangential kinetic energy transformed into available axial kinetic energy.

Through the opening of the balance hole can reduce the multi-stage pump axial force. When the gas content is less than 20%, the efficiency loss of the pump is relatively small, when the gas content is more than 20%, the method may no longer be applied due to a significant reduction in the performance of the pump.

In this paper, the numerical simulation of the pump is an Euler two-fluid method. This method regards the continuous phase and the discrete phase as statistically continuous. Due to the existence of the average time and space, the discrete phase is distributed over the control volume, and the real discrete phase flow image cannot be obtained [14-16]. It is necessary to find a more accurate calculation method in the actual gas-liquid two-phase flow for further study.

## Acknowledgements

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