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## Research on the characteristic of noise and pressure fluctuation in the oil-gas mixing transportation of screw-axial pump

To cite this article: Hui Quan *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **240** 032031

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## Research on the characteristic of noise and pressure fluctuation in the oil-gas mixing transportation of screw-axial pump

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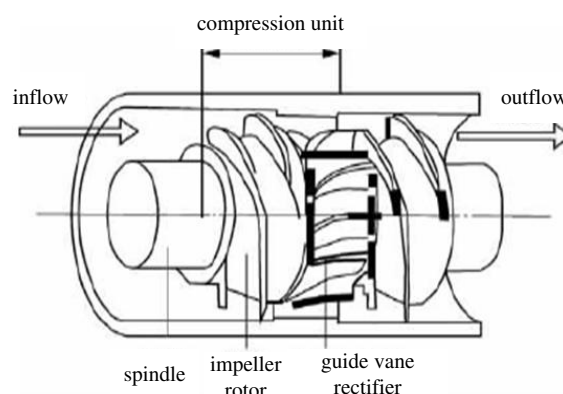
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**Abstract:** In order to study the spiral gas-liquid mixing loss and noise characteristics of the oil-gas mixing transportation of screw-axial pump, with the oil-gas mixing transportation of screw-axial pump as the research object, based on the RNG  $k-\epsilon$  psilon internal unsteady flow model to simulate the pump parameter change of gas and liquid delivery information, extract the unsteady gas-liquid transfer information parameters and pressure fluctuation signals, using acoustic software LMS Virtual Lab IBEM indirect boundary element method is used to solve the helical axial multiphase mixture pumps in dipole source caused by noise. The distribution law of pressure field, pressure pulsation and sound pressure level in the pump and outlet of the helical axial flow multiphase flow pump is analyzed. The results show that the gas-liquid mixing in the helical-axial-flow oil-gas mixing pump, from impeller inlet to impeller outlet, is aggravated, the pressure increases and the flow is turbulent. From the inlet of the guide vane to the exit of the guide vane, the bubbles gradually move from the hub to the middle of the runner, then the pressure gradually decreases, and the low voltage area leads to the vortex phenomenon; for the internal pressure fluctuation of helical axial flow type oil-gas mixing pump, the pressure gradient in the static and static interference area between the impeller and the guide blade is large, which is an important area affecting hydraulic performance. The pressure pulsation in the cascade is affected by the blade passing frequency and the mutual interference between the rotor and stator blades. The pressure pulsation maximum value is mainly corresponding to the blade frequency and its doubling frequency, and under the interaction of rotor - stator cascade, the pressure pulsation of the monitoring point near the static and static interface is superimposed, and the main frequency is the blade's triple leaf frequency. For screw axial flow noise in the oil-gas mixture pumps, screw axial multiphase flow pump hydrodynamic noise exports more than imports, sound pressure level frequency in low frequency band is evident, for impeller pressure impulses and its frequency doubling, after three times the pressure impulses, sound pressure level decline slows, the distribution of sound pressure level along with the increase of flow rate, shows the tendency of increase after decreases first. The research results are of great theoretical value and practical significance for the unsteady turbulent gas-liquid mixed transport, pressure fluctuation and noise characteristics in the oil-gas mixed transport pump.



## 1. Introduction

With the development of offshore oilfield, helical axial flow multiphase flow pump has become a research hotspot with its effective working mode and economic benefit [1]. Because of its complex structure, different types and the properties of the flowing media, there are often some Vibration and noise, Figure1 is a single-stage structure of a helical axial flow type oil/gas mixed transport pump. With the development of hydraulic machinery, it is found that in order to achieve high efficiency, wide working conditions and strong safety requirements, it is necessary to deeply understand and study the flow condition and energy loss mechanism of the impeller machinery [2]. Relative movement between rotating blade and stationary guide blade in the pump, circumferential movement of water flow in the suction chamber when deviating from the optimal working condition, local cavitation and secondary flow, et al., may lead to rapid change of fluid pressure in the pump with time, that is, the phenomenon of pressure fluctuation occurs. The pressure fluctuation law may be periodic or can be random, but the pressure fluctuation may aggravate the vibration of the unit equipment, can cause resonance when serious [3]. The local cavitation can be further developed and the flow field structure is destroyed. Therefore, it is of great theoretical value and practical significance to study the unsteady turbulent pressure fluctuation characteristics of the oil-gas mixed transport pump[4].



**Figure1 Single-stage structure of a spiral axial flow oil and gas mixed pump**

Because the helical axial flow type oil-gas mixing pump has excellent adaptability to the high gas-bearing fluid, the industrialized countries such as France, Norway, Italy, America, Germany, Britain and Japan have studied it and made it industrialized application in the field of ground and underwater oil and gas mixing [5]. In January 1984, total of France, Institute of France Petroleum (IFP) and the Norwegian National Petroleum company Statoil invested a large amount of money in the research and development of the series of spiral-axial multi-phase hybrid pumps of "Poseidon". In 1991, P300 Industrial prototype that installed in the Tunisian Onshore oil field carried out experiments [6]. In 1994, The Norwegian state oil company installed the P301 ground screw axial-flow multiphase hybrid pump on its North Sea Gullfaks A platform, marking the "Poseidon" project's long-term development plan into the industrialized stage [7]. Since then, the spiral axial multiphase mixture pumps have been used in a number of submarine pressure on projects, such as by Norkem Shell and Shell International provides technical and economic assistance, the Norwegian Framo Engineering manufacture hydraulic driven underwater booster station SMUBS ELSMUBS and electric

drive, Mobile and MEPS Framo Engineering cooperation development project, developed by institute of French oil Nomad production system, et al. At the same time, Italy's AGIP, NUOVO Pigeon and Snampro Get-ui Company, the United States Weirs Co., LTD. and ABB Group, Germany's Bornemann, the British Bhra, Norway's Framo and other companies have joined the development of helical-axial flow-type mixed pump, development and field experiments [8-11].

The noise of the fluid mechanical system can be divided into the radiated noise and the fluid dynamic noise of the mechanical structure vibration according to the excitation generated [12]. Chu, Dong, Langthjem, Parrondo, et al. discussed the influence of the volute geometry on the flow-induced noise from the mechanical structure [13]. Wu Renrong, Geng Shaojuan, Huang Junxiong, Tan Minggao, Liu Houlin et al. studied the relationship between geometrical parameters of impeller and blade form and fluid-induced noise from the aspect of mechanical structure [14-15]. Wang, KATO, JIANG, Si Qiaorui, et al discussed the correlation between flow and structural coupling and noise radiation [16]. Wang, Argarin, Mathey, siling, Yang Yanli, Wu Wenhua, et al. studied the effects of hydrodynamic convection noise [17-18]. The research results show that the interaction between dynamic and static leaves is the main cause of local pressure fluctuation and far-field noise at low Mach number.

In order to analyze the influence of gas-liquid mixed transmission and pressure pulsation on the flow-induced noise in a helical-axial multiphase flow pump, this paper studies the internal pressure pulsation characteristics of an axial-flow helical multiphase flow hybrid pump, which is based on the unsteady numerical calculation of RNG  $\varepsilon$  turbulence model[19], and the relationship between pressure fluctuation characteristic and flow induced noise in cascade is further analyzed by finite acoustic boundary element method in order to provide reference for the stable operation and flow induced noise control of the helical axial flow multiphase flow pump unit [20].

## 2. Calculation model establishment and numerical method

The main design parameters of helical axial flow multiphase flow pump are as follows: design Flow  $Q_d = 100\text{m}^3/\text{h}$ , design head  $H=80\text{m}$ , rotational speed  $N=4500\text{r/min}$ , the impeller and guide vane hub are tapered, and the geometrical parameters of the model are obtained by hydraulic design, such as table 1 is shown.

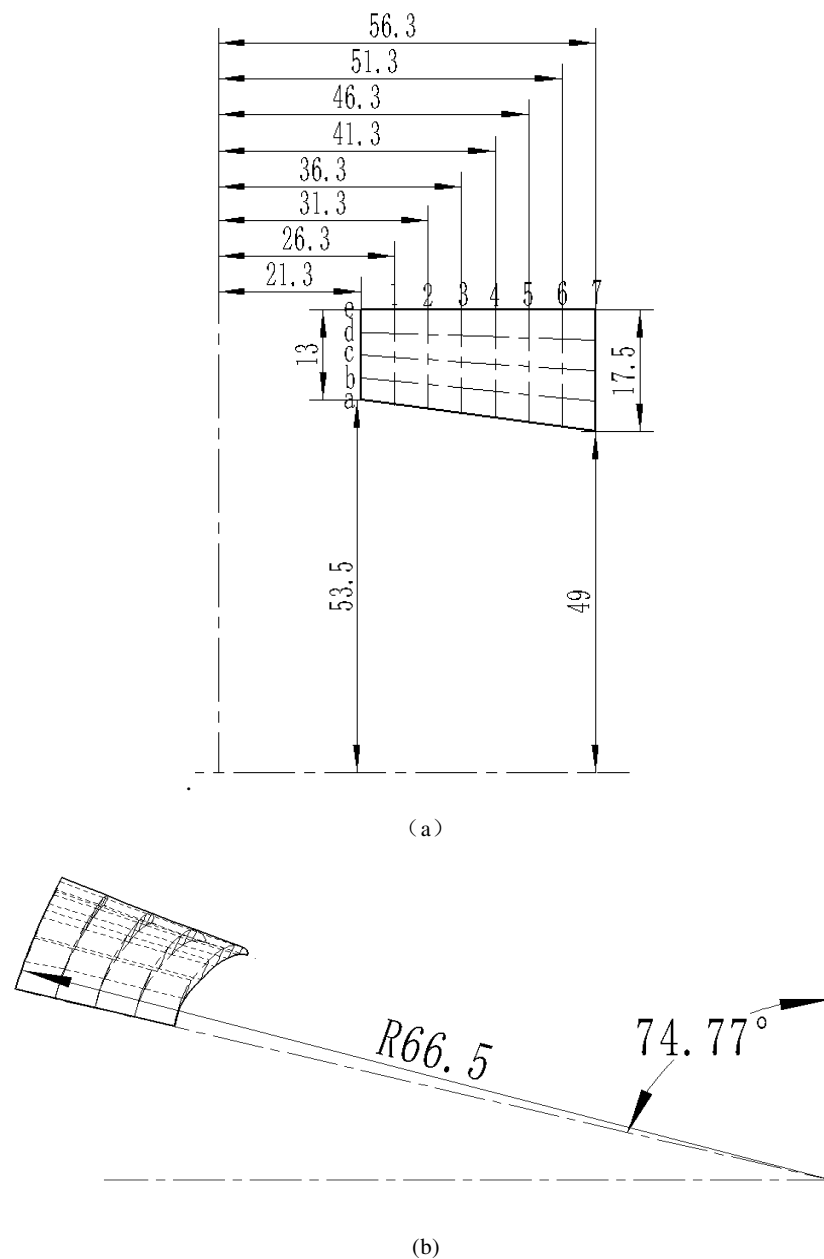
**Table 1 model pump main flow parts geometrical parameters**

	parameters	numerical
impeller	hub inlet diameter /mm	100
	hub outlet diameter /mm	110
	flange outer diameter /mm	142
	leaf number/Z	4
	the cascade solidity/ $\delta$	2
guide vane	hub inlet diameter /mm	110
	hub outlet diameter /mm	100
	flange outer diameter /mm	142
	leaf number /Z	13
suction chamber	inlet diameter /mm	100
discharge chamber	base diameter/mm	110

baffle tongue blade angle /°	40
Eighth section to the exit vertical height H/mm	120

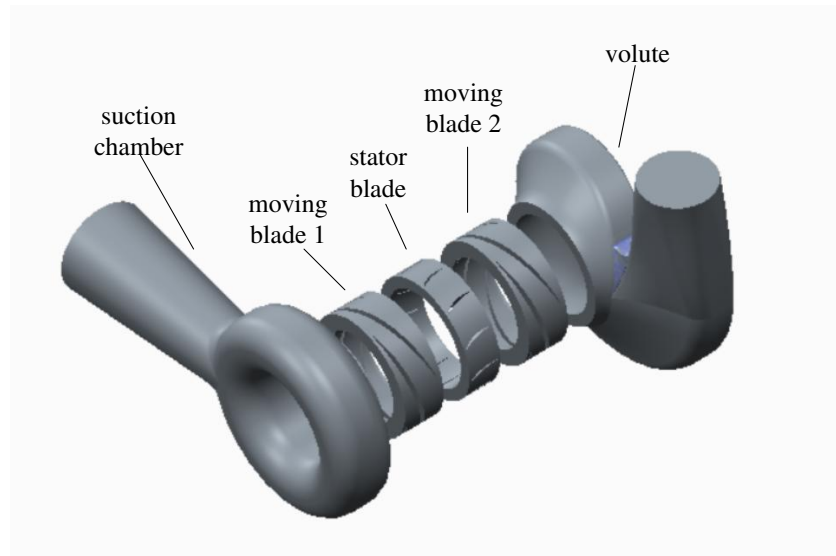
### 2.1. Flow field calculation

According to the geometrical parameters in table 1, the hydraulic design drawing of the blade is shown in Figure2.



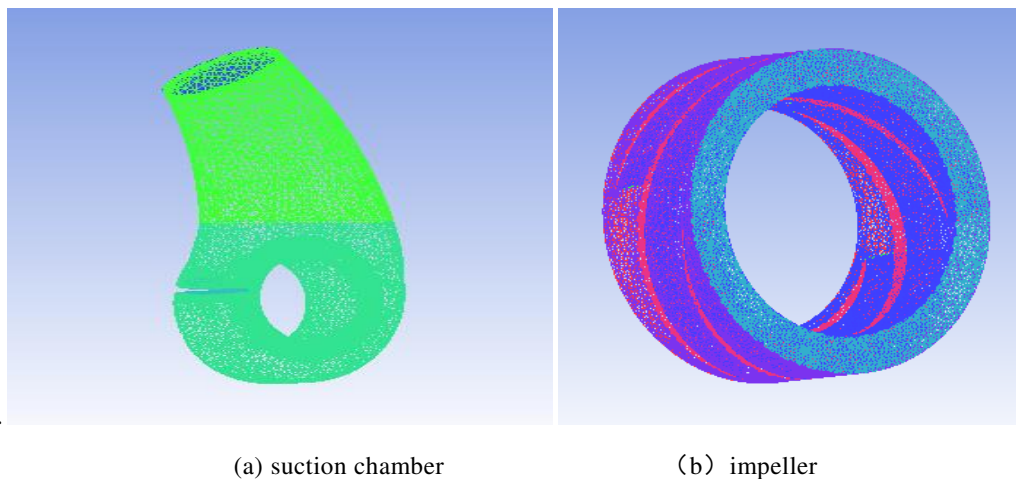
**Figure2 Blade design process**

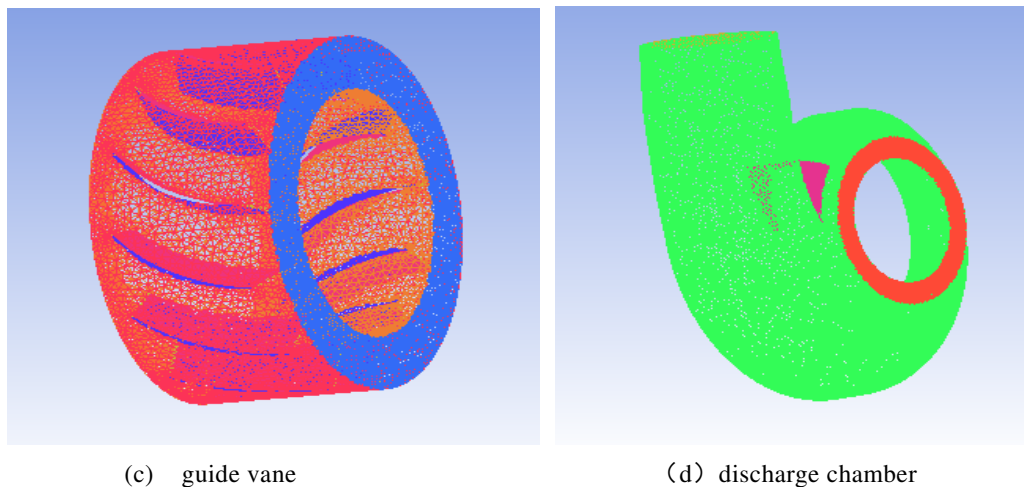
Figure 3 is the calculation domain of a helical axial flow multiphase flow pump. The calculation domain includes suction segment, two compressed water bodies and volute water bodies.



**Figure3 Computational domain of a spiral axial flow multiphase flow mixing pump**

By using ICEM CFD software to mesh the computational domain, considering the complexity of the model structure of the oil-gas helical axial flow multiphase flow pump [21], the wall surface of impeller blades and vane blades is encrypted by using the highly adaptive unstructured tetrahedral meshes [22]. The computed domain grid partition results are shown in Figure tow. The grid of each part of screw axial flow oil and gas mixture pump is shown in figure 4.





**Figure 4 Grid partitioning**

According to the calculation, when the grid number is 661292, the head calculation is the most accurate, and the calculation time is reasonable, so select the grid number to calculate.

The steady and unsteady flow field is calculated using CFX. When calculating, it is assumed that the working medium is water and incompressible fluid, and considering the heat capacity of the water. The heat generated by viscous dissipation can be neglected and the energy equation is not considered in the calculation. In order to achieve the transmission of interface data, the rotor interface is set as frozen rotor interface during steady calculation [23]. In unsteady calculation, the dynamic interface is set as the transient rotor interface. Combined with the structural characteristics of the oil-gas helical axial flow multiphase flow pump, the RNG  $\epsilon$  turbulence model can accurately simulate the complex flow phenomena inside. The time step for setting the unsteady calculation is  $1.1111 \times 10^{-4}$  s, that is, the impeller is calculated once per rotation of  $3^\circ$ . In the multi-coordinate system, the leaf rotation field is calculated in the rotating coordinate system, and the suction chamber, the guide vane and the volute are calculated in the stationary coordinate system. The import condition is 1 standard atmospheric pressure. The outlet condition is the mass flow outlet, and the wall uses the non-slip boundary condition. The near wall area uses the standard law of wall [24-25]. The residual error  $10^{-5}$  is one of the criteria for judging the degree of convergence.

In the calculation, the monitoring points are set up to collect the pressure fluctuation information in the flow field, and the monitoring points are arranged in the middle of the rotary surface and the spacer of the spiral axial flow multiphase flow pump, as well as the P1~P14 point of the monitoring point is set up near the inlet of the impeller.

## 2.2. Sound field calculation

Because the sound field and the flow field are unified in nature, the control equations are also Navier-Stoke equations. It is possible to solve the pressure fluctuation at the same time through the solution of N-S equation convection field, but it is confronted with the difficulties of high-precision turbulence mode, high-precision time-space format and large computational quantity[26-27]. In this paper, the computational fluid dynamics and computational acoustics are combined to calculate the method, which is mainly based on Lighthill acoustic analogy theory. The complex flow process is replaced by an equivalent sound source, then the acoustic wave equation is solved to calculate the acoustic propagation. The basic control equation is

$$\frac{1}{c_0^2} \frac{\partial^2 p'}{\partial t^2} - \frac{\partial^2 p'}{\partial x_i^2} = \frac{\partial^2}{\partial x_i \partial x_j} (\rho v_i v_j - \sigma_{ij}) - \frac{\partial f_i}{\partial x_i} + \frac{\partial^2}{\partial t^2} \left( \frac{p'}{c_0^2} - \rho' \right) \quad (1)$$

Where  $p$  is the fluid pressure;  $i, j$  is an arbitrary exponent;  $\sigma_{ij}$  is the viscous stress;  $C_0$  is the speed of sound;  $x_i, x_j$  is the differential of the spatial function in the  $i$  and  $j$  directions;  $v_i, v_j$  is the velocity vector at  $i, j$  the differentiation in the direction;  $f_i$  is the differentiation of the physical force coordinate function in the  $i$  direction.

The right-end source of the equation contains 3 components, are four-polar, dipole and monopole 3 kinds of sound sources. Among them, the quadrupole sound source is closely related to non-linear flow; the dipole sound source is caused by surface pulsation excitation; the monopole source is generated by volume change of fluid medium and the internal field noise source is mainly dipole sound source without cavitation in the pump.

In this paper, the indirect boundary element method of LMS Virtual Lab is used to calculate the internal noise of the helical axial flow multiphase flow pump. The unsteady pressure pulsation excitation of the inner surface of the pump is defined as the acoustic boundary condition after the fast Fourier transform, and the inlet and outlet of the pump define the full sound absorption property. In the calculation, the vibration of the pump casing is neglected, that is, the pump housing is assumed to be total reflection wall surface.

### 3. Characteristics analysis

The head efficiency of the oil-gas mixture pump under different working conditions is calculated, and the characteristic curve of the pump is obtained as shown in figure 5.

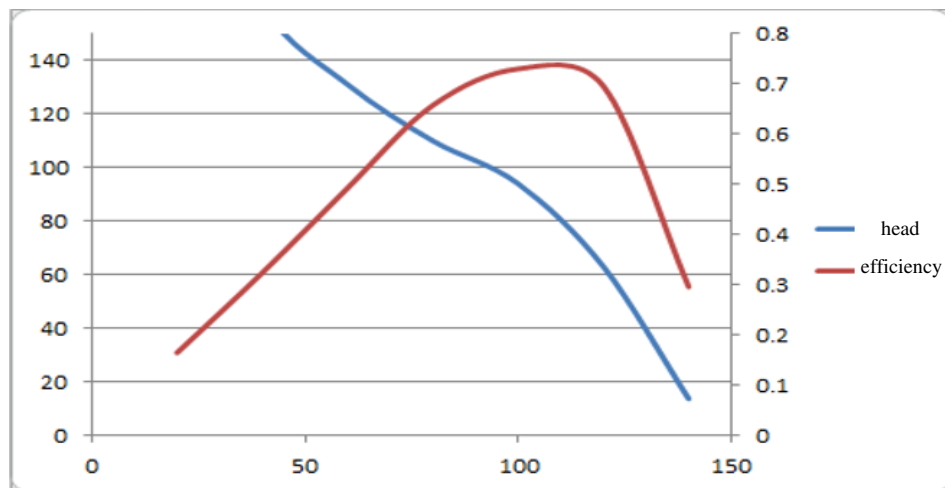


Figure 5 Characteristic performance

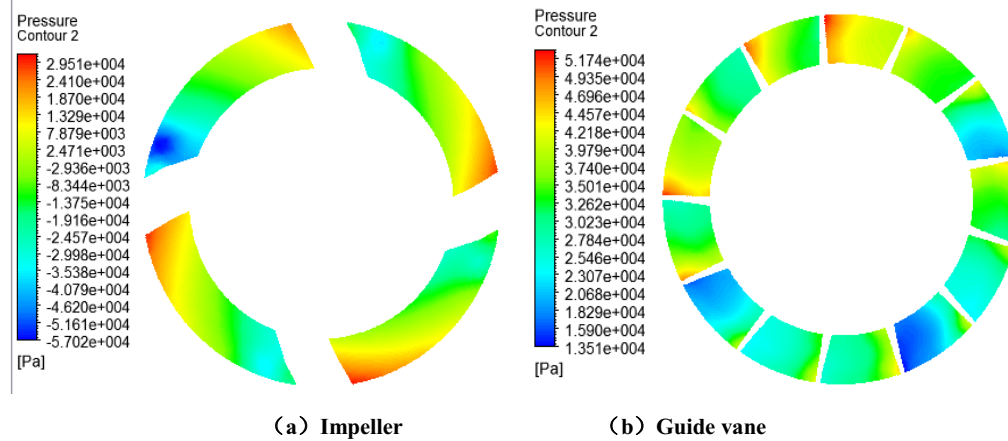
According to the analysis in Figure 5, the rated head of the oil-gas mixture pump is 80 m. It can be seen from this design that the head under rated working condition is 100m, which increases by 20m. Except for the low experimental efficiency at large flow rate and small flow rate, the other working conditions don't change much, indicating that the hydraulic design has improved compared with the previous design.

### 4. Analysis of internal flow field



#### 4.1. Pressure field analysis

Figure 6 is a helical axial flow multiphase flow pump under different operating conditions of the static pressure distribution and according to the change monitoring point layout.

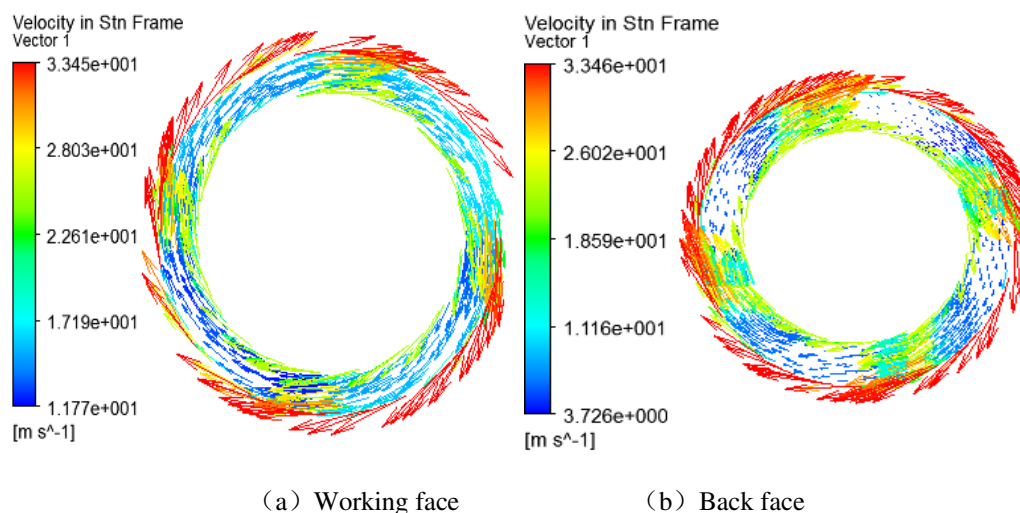


**Figure6 Stress nephogram**

According to the analysis in Figure 6, the pressure decreases gradually from the working face to the back of the impeller on the same section. From the inlet of the impeller to the outlet of the impeller, the high-pressure area of the working face gradually transitions to the back and focuses on the wheel rim, and forms a low-pressure area at the back near the hub; For the guide vane, the pressure gradually increases from the suction surface of the guide vane to the pressure surface, and the high-pressure area focuses on the pressure surface near the wheel rim.

#### 4.2. Velocity field analysis

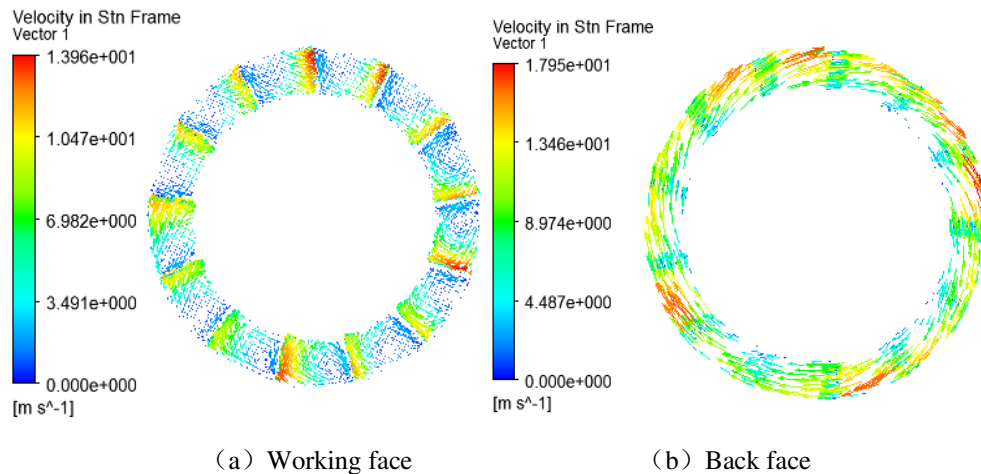
The relative velocity distribution on the impeller show in figure 7.



**Figure7 Relative velocity distribution on impeller**

The analysis of the relative velocity distribution on the impeller in figure 7 shows that the radial component of the relative velocity gradually increases from the working face to the

back face. The maximum relative velocity within the two sections is almost unchanged, but the minimum value is constantly decreasing. Near the outlet of the impeller, the velocity gradient increases and the flow becomes turbulent.



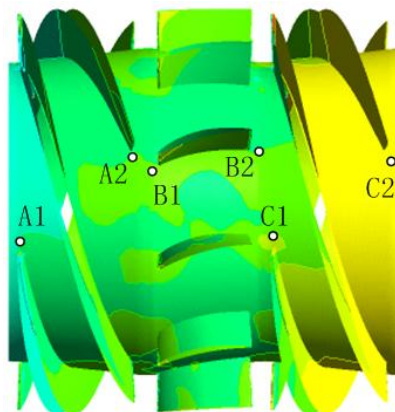
**Figure 8 The relative velocity distribution on the guide vane**

The analysis of the relative velocity distribution on the guide blade in Figure 8 shows that, from the pressure surface to the suction surface, the relative velocity gradually decreases, and with the blade inlet to outlet, the high-speed area gradually concentrates on the pressure surface of the blade. Near the exit of the guide vane, there is obvious vortex phenomenon in the flow passage.

## 5. Analysis of pressure pulsation frequency domain

### 5.1. The selection of monitoring point

The head impeller-guide vane-secondary impeller of helical axial flow multiphase flow pump is a compression work unit, which is the object of the work unit and the monitoring point is arranged [29-31]. Among them, monitoring points A1 and A2 are distributed at the inlet and outlet of the first impeller of the helical axial-flow multiphase pump, while monitoring points B1 and B2 are arranged at the inlet and outlet of the first impeller. Monitoring points A1 and A2 are distributed at the inlet and outlet of the second impeller of the helical axial-flow multiphase pump. As shown in Figure 9.



**Figure 9 Layout of the monitoring point**

### 5.2. Pressure pulsation extraction scheme

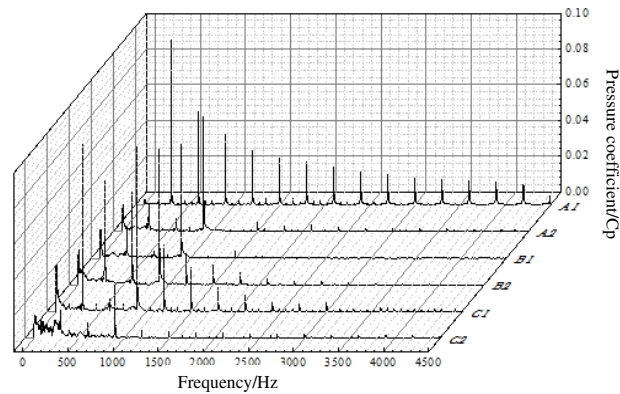
By unsteady numerical calculation, the time-domain pressure information of the monitoring point under different flow rate is obtained[32-35]. The pressure coefficient  $C_p$  is introduced to measure the amplitude of the pressure fluctuation, and the expression of  $C_p$  is

$$C_p = \frac{P_i - P_{ave}}{\frac{1}{2} \rho u_2^2} \quad (2)$$

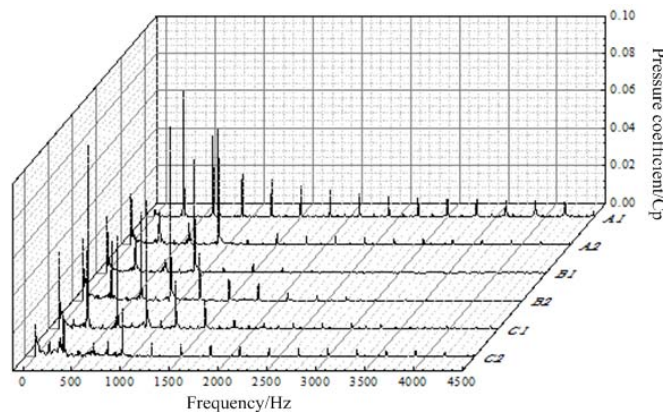
In the formula,  $P_i$  is the static pressure value of the monitoring point at a certain moment, Pa;  $P_{ave}$  for a rotational period of static pressure mean value, Pa;  $\rho$  is the working medium density,  $\text{Kg/m}^3$ ;  $u_2$  for the impeller exit circumferential speed, m/s.

### 5.3. Pressure pulsation frequency domain analysis

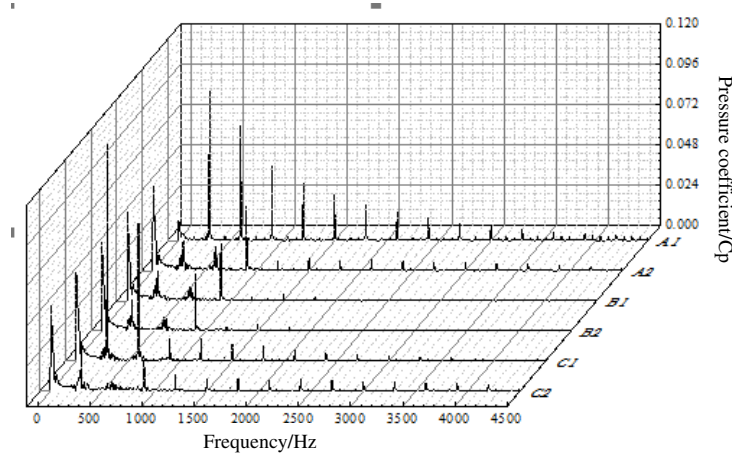
Under the condition of pressure fluctuation under different flow rate, the peak of the amplitude of pressure fluctuation mainly appears in the low frequency band. With the increase of flow rate, the amplitude of pressure fluctuation increases first and then decreases the trend. This is because, as the fluid velocity increases, the pressure pulsation amplitude increases due to the rapid flow of water on the impeller and the guide vane and other flow components, and then the pressure pulsation amplitude is attenuated under the action of guide vane rectification.



(a)  $0.8 Q_d$



(b)  $Q_d$

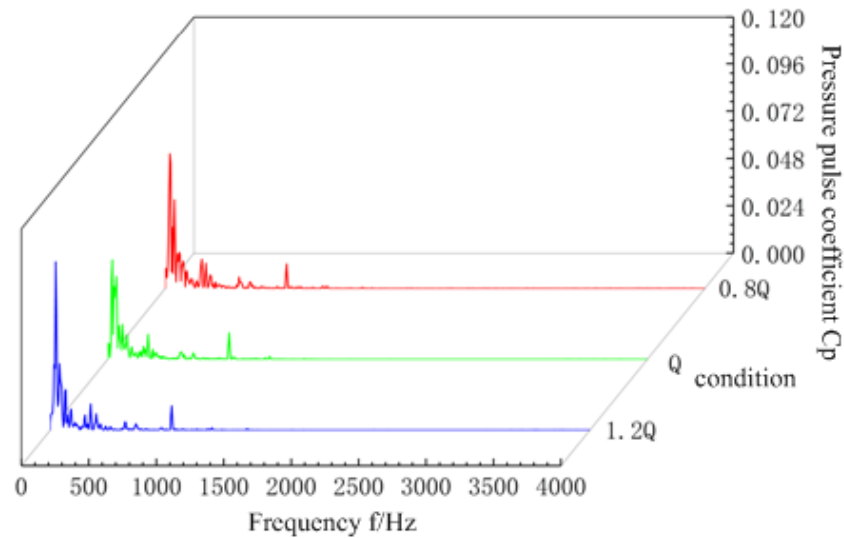
(c)  $1.2 Q_d$ **Figure10 Frequency domain diagram of internal pressure fluctuation monitoring point**

Compared with the pressure fluctuation of non-peer monitoring points, the frequency domain distribution of A1 and C1 is the same, and the pressure pulsation frequency is 300Hz, which is 4 times the axis frequency. However, second frequency is different. The first stage frequency is 600Hz and second stage frequency is 900Hz. The frequency domain distribution is the same after triple blade frequency. The pressure pulsation Peak is consistent, and the first stage pulse amplitude is larger. A2 and C2 have the same frequency domain distribution, and the main and secondary peaks of pressure pulsation are the same, namely 900Hz and 300Hz. The pulsation amplitude of the first stage in the low frequency band is larger, while that of the second stage in the high frequency band is larger. When the fluid enters the helical axial flow multiphase flow pump, because of the influence of the annular suction chamber on its flow, the inlet flow state of the first stage impeller is unstable and the flow velocity is fast, as well as the exit is affected by the number of vanes in the process of the impeller and the guide vane, which makes the static pressure gradient change obviously and the pressure fluctuation amplitude.

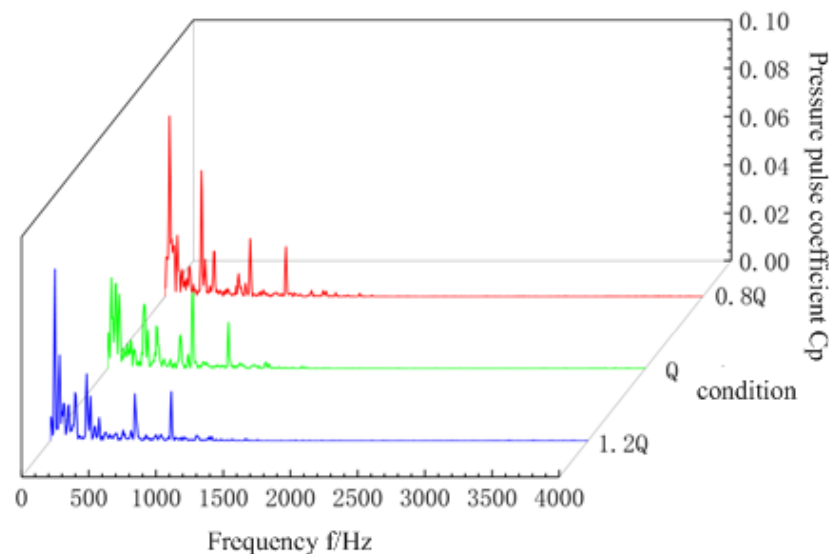
As can be seen from the pressure pulsation frequency domain diagram of the monitoring points at the inlet and outlet of the guide vane, the pressure pulsation frequency domain distribution of B1 and B2 is the same, while the frequency domain with larger pulsation amplitude is in the low frequency band. The pulsation amplitude after 1500Hz shows a significant downward trend until it is reduced to 0. The main peak of pressure pulsation at the monitoring point is 900Hz, which is 12 times the shaft frequency. The wave peaks also appear at 600Hz and 300Hz. The monitoring point of the guide Vane exit is larger than that of the B1 of the B2 at the inlet, which indicates that the pressure fluctuation in the guide vane is affected by the blade of the lower impeller. Because the guide vane is placed between the first and second impeller, and under the combined action of the number of guide vane blade and the number of impeller blade, the amplitude of pressure pulsation in guide vane at the frequency of guide vane is not obvious.

#### 5.4. Analysis of blade pressure pulsation frequency domain and noise under different working conditions

The acoustic pressure frequency distribution of dipole source at the inlet and outlet of the model pump under different operating conditions of the helical axial flow multiphase flow gas-oil pump is obtained by the acoustics calculation, as shown in Figure 11.



(a)



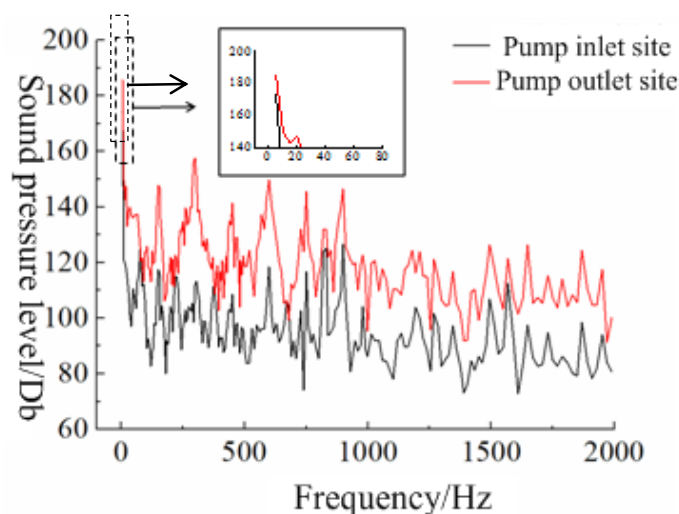
(b)

**Figure11 Guide vane frequency domain diagram**

It can be seen from the Figure11, it can be seen that the main frequency of pressure pulsation near the inlet of the first and second order guide vane under different working conditions is at the axial frequency from (a) and (b). In the figure (a), the pressure pulsation near the inlet of the guide vane of the first grade under different working conditions in the figure shows that the pressure pulsation amplitude decreases successively with the increase of frequency. When the fluid flows through the impeller inlet to the outlet of the impeller, the pressure pulsation tends to strengthen first and then weaken. The helical suction chamber can

improve the flow conditions and guarantee the uniform velocity field at the impeller inlet, so the pressure pulsation amplitude near the inlet is small; the vane has a rectifying effect, and the pressure pulsation near the impeller outlet is smaller than the middle portion.

The variation of pressure pulsation amplitude is different under three different working conditions: large, small flow rate and designed flow rate. But generally speaking, the pressure pulsation amplitude is the minimum under design conditions, the maximum under low flow conditions and the intermediate between the two under high flow conditions.



**Figure 12 Corresponding curves of sound pressure level frequency**

In conclusion, the comparison of the frequency response of the sound pressure level at the inlet and outlet of the pump under different conditions shows that the noise frequency of the sound field is mainly concentrated in the low frequency band, the waveform is disordered after 900Hz, and the sound pressure level is obviously reduced. The flow rate has little effect on sound pressure level, vane frequency, twice vane frequency and three vane frequency are its characteristic value. At the inlet and outlet points, the sound pressure level frequency response curve waveform is approximately the same, but the sound pressure level value differs greatly. Compare the frequency response curve of sound pressure level and the pressure pulsation frequency domain of cascade, it is found that the pressure pulsation has a certain effect on the inner noise of the pump, and the pressure fluctuation characteristic can be used to characterize the radiation characteristics of the sound pressure level of the pump infield noise.

## 6. Conclusion

Here are conclusions based on analyses above:

(1) The gas-liquid two-phase separation from impeller inlet to impeller outlet is aggravated, the pressure increases and the flow is disordered. From the inlet of the guide vane to the exit of the guide vane, the bubbles gradually move from the hub to the middle of the runner. The pressure gradually decreases and a low-pressure area occurs, leading to vortex phenomenon.

(2) Helical axial flow multiphase flow pump in the low frequency of 1500Hz, the monitoring point pressure fluctuation amplitude increases with the increase of the flow rate, in the middle and high frequency section of more than 1500Hz, the pressure pulsation amplitude of the monitoring point decreases as the flow rate increases, because the increase of flow velocity makes the impact action of water flow on the impeller and guide vane strengthened, The amplitude of pressure fluctuation is gradually attenuated by the effect of the guide vane rectification.

(3) The pressure fluctuation in the cascade of the helical axial flow multiphase flow pump is affected by the frequency of the blades and the interference between the static and dynamic cascades. The static pressure of both the front and back of the impeller blade shows obvious discrete spectrum characteristic, and the maximal value mainly corresponds to the blade frequency and its frequency multiplication. Because of the mutual interference between the rotating impeller and the static guide blade, the pressure pulsation at the monitoring point at the outlet of the first stage impeller and the inlet of the second stage impeller is superimposed, and the main frequency of pressure pulsation is shown as triple blade frequency.

(4) The hydrodynamic noise of screw - axial multiphase pump is larger at the outlet than at the inlet, noise radiation has obvious dipole characteristics, the sound pressure level distribution with the increase of flow, showing the trend of first increase and decrease, and the pressure fluctuation characteristics of the same trend, pressure pulsation is associated with the internal noise of the helical axial flow multiphase pump, Therefore, the pressure fluctuation characteristic can provide some reference value in the analysis of the interior noise of the helical axial flow multiphase flow pump.

### Acknowledgements

This work was partially supported by the China Postdoctoral Science Foundation (2018M633651XB), the National Natural Science Foundation of China (NSFC) (51609113, 51579125), the Natural Science Foundation of Gansu (2017GS10829) and The Hongliu Outstanding Young Talents Funding Scheme of Lanzhou University of Technology.

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