Design and hydraulic analysis of a multiphase pump

R Yao, J W Cao, S H Ahn, B Guo, Y X Xiao and Z W Wang

State Key Laboratory of Hydroscience and Engineering & Department of Energy and Power Engineering, Tsinghua University, Beijing, 100084, China

E-mail: wzw@mail.tsinghua.edu.cn

Abstract. With the development of offshore oilfields, foreign companies and research institutes spend plenty of resources on the research of subsea multiphase pump. The helico-axial multiphase pump is generally characterized by large flow and high head unit, and has the advantages of good anti-sand performance, simple structure and small volume. This paper introduces the design of the two-stage compression unit model of a helico-axial multiphase pump by using CFD method. A numerical simulation of full three-dimensional flow field is carried out to study the blade number and the wrap angle influence on external characteristics of the pump. The optimum design combination is selected as the research object and the influence of different inlet gas volume fraction on the internal and external characteristics of the pump is discussed. The design and hydraulic analysis of the multiphase pump can provide a basis for its use in the offshore oil-gas gathering and transportation system.

1. Introduction

In recent years, the development of offshore oil and gas fields is in the ascendant. The products from offshore oil fields are multiphase mixtures containing oil, gas, water and various impurities. Its gas-liquid ratio exceeds the normal operating range of ordinary pumps or compressors [1]. There are two types of offshore oil and gas gathering and transportation: oil and gas separate transportation and multiphase mixing transportation [2]. Compared with oil and gas separate transportation, multiphase mixing transportation has the following advantages: multiphase mixing transportation can reduce about 30% in the investment of capital construction, reduce wellhead back pressure and increase oil and gas production in offshore oilfields compared with oil and gas separate transportation [3]. It is reported that the wellhead back pressure can be reduced by 50% and the oil and gas production can be increased by about 15% [4]. Multiphase mixing transportation does not need to separate oil and gas, which effectively reduces the consumption of light oil.

In the 1980s, Institut Francais Du Petrole(IFP), Statoil ASA and Total cooperated to develop the “Poseidon” project. In 1987, the industrial prototype pump P300 was developed and tested with good results in Sidi Ei Ytayem oilfield in Tunisia. Subsequently, P301 and P302 were developed and put into use. After the helico-axial multiphase pump was developed by the “Poseidon” project, Shell and Framo developed SMUBS (Shell Multiphase Underwater Booster Station), which was put into production in Norway’s Draugen Oilfield in 1994 [1]. Framo also developed the ELSMUBS system of the electric-driven in 1996, which was applied in Lufeng Oilfield of 330m deep in South China Sea in 1998 and several oilfields in Equatorial Guinea in 2002 [1].
Helico-axial multiphase pump has the advantages of simple structure, stable operation, and large flow. But it has the disadvantage of small gas-liquid ratio range [5]. Its diffuser has the functions of diffusing and shearing, which can break the air mass at the outlet of the impeller and make gas and liquid mix again. The impeller and diffuser force the medium to move along the axis to avoid phase separation, which is different from the centrifugal pump. Therefore, the helico-axial pump can transport medium with higher inlet gas volume fraction (IGVF). In addition, there is a gap between the blades, which allows small particles of solid impurities to pass through, so that the sand resistance is better. Helico-axial multiphase pump is usually equipped with high speed and variable frequency motor, which has fast speed and transport medium and large flow, and provide medium and low head [6]. Helico-axial multiphase Pump has been studied theoretically, numerically and experimentally.

Based on the theoretical analysis of the energy balance and phase separation process in the rotodynamic pump, Li et al. proposed that avoiding phase separation is the key factor to improve the gas-liquid ratio and pump hydraulic efficiency, and designed helico-axial multiphase pump prototype [7,8]. Ban et al. established a model to predict the performance of helico-axial multiphase pump, which takes into account the compressibility of gas and the change of flow area. Miao et al. proposed a method to obtain the head-flow curve of helico-axial multiphase pump through theoretical analysis [9].

Based on the bubble flow hypothesis and the two-fluid model, Yu et al. carried out a 3D numerical simulation of the two-phase flow field in the impeller. The simulation results show that the smaller radial size difference of the runner can avoid gas-liquid separation and there is a low-pressure vortex area at the inlet, which needs to be further improved [10]. Zhang et al. established a model to predict the performance of multiphase pumps, and used the NSGA-II (non-dominated sorting genetic algorithm-II) to optimize and the results are better through numerical simulation [11]. Zhang et al. also carried out steady and unsteady numerical simulation of the flow field in a three-stage helico-axial multiphase pump, and found that the pressure field and two-phase distribution of helico-axial multiphase pump were changed due to the difference of centrifugal force [12].

Zhu et al. carried out indoor and field experiments on the fourth-generation helico-axial multiphase pump, then obtained performance curves and analyzed the pressurization at different speeds [13]. The flow visualization method of high-speed imaging can reveal the local flow mechanism leading to the performance degradation and instability of the pump. Serena et al. carried out an experimental analysis of the flow in the multiphase pump by means of high-speed imaging [14].

2. Numerical model

2.1. Model

2.1.1. Design objects. The main components of helico-axial multiphase pump are suction pipe, compression unit, pump shaft, discharge pipe, bearing and shaft sealing instrument. Impeller and diffuser (compression unit) are the important components of multiphase pumps. Their hydraulic performance directly affects the performance of the pump. In this paper, the two-stage compression unit of helico-axial multiphase pump is designed, which includes two-stage impeller, two-stage diffuser, front and rear extension section. The front and rear extension section are located at the inlet of the first stage impeller and the outlet of the second stage diffuser respectively, which can create uniform inflow and outflow conditions, thus ensuring the accuracy of calculation. Its first compression unit is shown in figure 1.

2.1.2. Design parameters. Blade number and wrap angle are selected as variable design parameters, and the impellers with different structures are designed to improve the overall hydraulic characteristics of the helico-axial multiphase pump.

In the design process of helico-axial multiphase pump, the number of diffuser and impeller blades should be prime as far as possible, and the pressure fluctuation can be reduced. The number of diffuser is 7 and the blade number of impeller in this design combination is 3, 5 and 7. The impeller and diffuser
are dynamic and static respectively, and they are prime numbers, so periodic boundary conditions can’t be adopted in the numerical simulation. In order to conduct more comprehensive analyses of the interaction between different parts, 360° full channel is selected in the design and calculation process.

The angle between the lines which are connected by the inlet and outlet with the center of the circle is called the blade wrap angle, and its magnitude directly affects the length of the runner. According to experience, the blade wrap angle of impeller is 185°, 200°, 215°, 230°, 245° respectively.

![Figure 1. First stage compression unit.](image1)

2.2. Model of the compression unit

Three-dimensional model of the single-stage impeller and diffuser with different blade numbers when the blade wrap angle is 185° are shown in figure 2. The two-stage compression unit of helico-axial multiphase pump when the blade number is 3 and the blade wrap angle is 215° are shown in figure 3.

![Figure 2. Model of single-stage impeller and diffuser.](image2)

![Figure 3. Two-stage compression unit.](image3)
2.3. Meshing

2.3.1. Mesh generation. There are many components in the compression unit of the helico-axial multiphase pump and the performance of computer is limited. Therefore, the front extension section, impeller, diffuser and rear extension section are separately meshed into structured grids. Figure 4 is the mesh generation in fluid domain when the blade number is 3 and the blade wrap angle is 215°.

![Figure 4. Mesh generation in fluid domain.](image)

The boundary layer theory points out that when the fluid flows around the solid wall at high Reynolds number, the influence of viscous force on the boundary layer can’t be neglected, and there is a considerable velocity gradient along the normal direction of the wall at the boundary layer. In order to get more accurate results, the fluid domain near the blade wall adopt mesh refinement, as shown in figure 5(a).

![Figure 5. Mesh refinement.](image)

In addition to the blade wall, mesh refinement is also needed in some locations where the flow conditions are complex. In the leading edge and trailing edge of the blade, the curvature of these locations is larger, which causes more complex flow conditions. The inlet and outlet of the impeller are affected by the way of combination, so there will be unstable flow. Gas blocking is easy to occur at the inlet of impeller under the high GVF [10]. Mesh refinement is also adopted in these areas, as shown as figure 5(b).

![Figure 6. Mesh generation of interface between the second stage diffuser and rear extension section.](image)

Front extension section, rear extension section, impeller and diffuser are meshed independently, which results in discontinuity of mesh number at the interface of these regions. The accuracy of calculation can be increased by controlling the number of mesh nodes on both sides of the interface. figure 6 is the mesh generation of interface between the second stage diffuser and rear extension section.
2.3.2. Grid independence verification. In order to exclude the influence of grid number on the calculation results, the numerical grid dependency test is conducted. By observing the hydraulic efficiency of helico-axial multiphase pump (impeller blade number is 3, blade wrap angle is 215°) under single-phase operation condition (outlet flow is 100m³/h, speed is 4500rpm), the number of grids is selected appropriately.

![Grid independence verification graph](image)

Figure 7. Hydraulic efficiency with total grid number under single-phase operation condition.

The variation of hydraulic efficiency with total grid number under single-phase operation condition is shown in figure 7. It can be seen that with the increase of the total grid number, the hydraulic efficiency decreases gradually. When the total grid number is greater than 1.87 million, the variation of the hydraulic efficiency is less than 0.1%. At this time, the influence of the total grid number on hydraulic efficiency can be neglected. Therefore, the total grid number is 1.87 million (same as the 2nd point). The number of regional grids corresponding to the 2nd point is shown in table 1.

Table 1. Number of nodes and grids in each part of 1.87 million grids.

| Domain            | The front extension | Impeller | Diffuser | The rear extension | Total     |
|-------------------|---------------------|----------|----------|--------------------|-----------|
| Number of nodes   | 177480              | 840672   | 847028   | 241700             | 2106880   |
| Number of elements| 162400              | 741198   | 743568   | 221200             | 1868366   |

3. CFD methods

The blade number and wrap angle are used as variable design parameters and the steady 3D flow field of helico-axial multiphase pump under different combinations are simulated. The most suitable combination of blade number and blade wrap angle is selected under the best efficiency and head. Under the optimum design combination, the effects of different IGVFs on the head and hydraulic efficiency are analyzed. In addition, the velocity, pressure and two-phase distribution are analyzed. The calculated parameters are shown in table 2 and the setting of IGVF is shown in table 3.

Assuming that the gas flow pattern is bubbly flow, the average diameter of the bubble is 0.3mm, and the surface tension coefficient is 0.073N/m.

The force between two phases only considers the dragging force \( M_k^D \). Schiller Naumann drag force model is used to calculate the drag force, as shown in formula (1) and formula (2).

\[
M_k^D = \frac{3}{8} \frac{C_D}{r_g} \rho_l \left| u_l - u_g \right| \left| u_l - u_g \right|
\]

(1)
wherein, \( r_g \) is the gas radius, \( \text{Re} \) is the relative Reynolds number of liquid phase and gas phase.

Two-phase flow has been extensively studied and developed, which makes it possible to solve 3D two-phase flow problems. The establishment of two-phase flow model is the premise of numerical simulation. In the two-fluid model, each phase of fluid is treated as a continuous medium filled with fluid field. The flow of single-phase fluid and the interaction between two phases are considered, and the effects of velocity slip and interphase coupling on the fluid are also considered. This model is suitable for high flow rates or homogeneous mixing of two-phase fluid and the results are easy to compare with the experimental results. Therefore, it is chosen as two-phase flow model for numerical simulation. The SST model is chosen as the turbulence model for numerical simulation [15]. It takes into account the transfer effect of shear stress in the turbulence, which can predict the starting point of flow separation and determine the size of separation zone.

### Table 2. Setting of calculation parameters.

| Material          | Angular velocity | Inlet pressure | Outlet flow | Wall              | Turbulence model | Two-phase flow model |
|-------------------|------------------|----------------|-------------|-------------------|------------------|----------------------|
| Water/ Air at 25°C| 4500rpm          | 1atm           | 100m³/h     | No slip wall boundary | SST             | Two fluid model      |

### Table 3. Setting of IGVF.

| Conditions | IGVF (%) |
|------------|----------|
|            | 5        |
|            | 10       |
|            | 15       |
|            | 20       |
|            | 25       |
|            | 30       |

### 4. Results

#### 4.1. The optimal combination

4.1.1. External characteristics. The external characteristic parameters of the pump include head, hydraulic efficiency, power and NPSH (Net Positive Suction Head). These parameters are the external manifestation of movement of fluid in the pump. The head and hydraulic efficiency of helico-axial multiphase pump are selected as the external characteristic parameters. Numerical simulation of 15 combinations are carried out. The head and hydraulic efficiency of each combination under single-phase operation conditions are compared.

The difference between the outlet of the rear extension section and the inlet of the front extension section is the pressure-increase \( \Delta p \).

\[
\Delta p = p_{out6} - p_{in1}
\]

Wherein, \( \Delta p \) is the pressure-increase, \( p_{in1} \) is the inlet pressure, \( p_{out6} \) is the outlet pressure.

The head \( H \), also known as the head of the pump, refers to the energy obtained by the flow of fluid per unit weight through the pump.

\[
H = \frac{\Delta p}{\rho g}
\]

Wherein, \( \rho \) is the density of water, \( g \) is the acceleration.
Hydraulic efficiency $\eta$ is the ratio of pump effective power to shaft power, and its numerical value can reflect the efficiency of pump energy conversion.

$$\eta = \frac{P_e}{P} \times 100\% = \frac{\rho g H Q}{2\pi n T} \times \frac{30 \Delta p Q}{\pi n T}$$

(6)

Wherein, $\eta$ is the hydraulic efficiency, $P_e$ is the effective power, $P$ is the shaft power, $Q$ is the flow of the pump, 100 m$^3$/h, $n$ is the angular velocity, 4500rpm, $T$ is the torque of rotating parts to axis.

**Figure 8.** Hydraulic efficiency and head of pump with different impeller design combinations.

It can be seen from the figure 8 that when the blade wrap angle is 215° and blade number is 3, the head is 74.4m and the hydraulic efficiency is 56.7%. The head and hydraulic efficiency are the maximum of all combinations. The head and efficiency of 5-245, 7-215, 7-230 and 7-245 are all negative, which indicates that the fluid in the pump has reversed flow, and there is no positive pressure. The structure of these combinations is unreasonable.

4.1.2. Influence of impeller blade number on external characteristics. Figure 9 shows the variation of head and hydraulic efficiency with different blade numbers (3, 5, 7) when the blade wrap angle is fixed as 185°, 200°, 215°, 230° and 245° respectively.

**Figure 9.** Variation of head and hydraulic efficiency of impeller with different blade number.
It can be seen that when the blade wrap angle is fixed, there will be an optimal blade number. When the blade wrap angle is smaller, the hydraulic efficiency and head will increase with the increase of the blade number; but when the blade number continues to increase, the runner will be narrow, the flow rates and the friction loss increases, thus the hydraulic efficiency and head gradually decreases. In some cases, the flow situation will be worse and even the reverse flow phenomenon occurs. When the blade wrap angle is larger, the hydraulic efficiency and head will decrease with the increase of the blade number.

4.1.3. Influence of impeller blade wrap angle on external characteristics. Figure 10 shows the variation of head and hydraulic efficiency with different blade wrap angle (185°, 200°, 215°, 230° and 245°) when the blade number is fixed as 3, 5 and 7 respectively. It can be seen that when the blade number is fixed, there will be an optimal blade wrap angle. When the blade wrap angle is smaller, the hydraulic efficiency and head will increase; but when the blade wrap angle continues to increase, the flow distance will be longer and the friction loss increase, so, the hydraulic efficiency and head gradually decreases. When the blade number is larger, the hydraulic efficiency and head will decrease with the increase of the blade wrap angle.

![Figure 10. Variation of head and efficiency of impeller with different blade wrap angles.](image)

4.2. Influence of IGVF on external characteristics

The results of the pressure-increase and hydraulic efficiency with different IGVF are presented, as shown in figure 11. It can be seen that the pressure-increase is 0.73 MPa and the hydraulic efficiency is 56.7% under pure liquid phase. With the increase of the IGVF, the pressure-increase and hydraulic efficiency decrease gradually, and the rate of decline is slower and slower. When the IGVF is 30%, the pressure-increase has been reduced to 0.27 MPa, and the hydraulic efficiency has been reduced to 35.4%. This is because the gas content increases gradually with the increase of the IGVF. The density of liquid is larger than gas, the work done to liquid is greater than that to gas when the flow rate and angular velocity of the helico-axial multiphase pump are constant. In addition, as the IGVF increases, the flow inside the pump becomes more and more disordered, and the working capacity of the pump will be worse, which will also lead to the decrease of hydraulic efficiency and the pressure-increase.
Figure 11. The pressure-increase and hydraulic efficiency with different IGVFs.

4.3. Influence of IGVF on inner flow field

4.3.1. Pressure field analysis. Figure 12 shows the average static pressure distribution in the whole field on the axis of the two-stage compression unit with different IGVF. It can be seen that in the radial direction of the impeller, the static pressure increases gradually from hub to shroud due to centrifugal force at the same IGVF; under different IGVFs, the static pressure at the outlet decreases with the increase of IGVF. Meanwhile the pressure-increase over the pump decreases, and the working capacity of the impeller decreases with the increase of IGVF. It also shows that the static pressure of the diffuser is larger than the outlet of impeller in each stage, because the diffuser can convert kinetic energy into pressure energy.
Figure 12. Average static pressure distribution in the full field of the axial surface at different IGVFs.

Figure 13 shows the static pressure distribution in 0.5 span rotary surface of the impellers and diffusers. Wherein, the inlet of the first stage impeller is on the down side and the outlet of the second stage diffuser is on the up side. It can be seen that the static pressure in the impeller increases gradually at the same IGVF and the static pressure on the pressure surface of the impeller is larger than that on the suction surface. There is a low-pressure distribution area at the suction surface near the inlet of impeller. There is a small high-pressure distribution area near the inlet of impeller because of impact action by the fluid. In the diffuser, the static pressure increases steadily. Under different IGVFs, the outlet pressure of impeller and the pressure-increase both decrease with the increase of IGVF.

4.3.2. Velocity field analysis. The absolute velocity distributions of gas and liquid in 0.5 span rotary surface at different IGVFs are obtained, as shown in figure 14 and figure 15. It is found that the absolute velocities of the gas phase and the liquid phase has little difference at the same IGVF, but it indicating that there is velocity slip between the gas phase and the liquid phase.

The impellers do work on the fluid by rotating, so the kinetic energy of the fluid increases, and the velocity from the inlet to the outlet of the impeller increases gradually. Fluid velocity at the inlet increases because of impact action by the fluid. There is a high-speed distribution area near the suction surface at the outlet of impeller, and the velocity in this area increases with the increase of IGVF.
diffuser transforms the kinetic energy of the fluid into the pressure energy, so the velocity of the fluid in the diffuser decreases gradually.

Figure 14. Gas absolute velocity distribution in 0.5 span rotary Surface.

Figure 15. Liquid absolute velocity distribution in 0.5 span rotary surface.

Figure 16 shows the gas relative velocity distribution in 0.5 span rotary surface when the IGVF is 0%. Figure 17 and figure 18 show the relative velocity distribution of gas and liquid in 0.5 span rotary surface with different IGVFs respectively. It is found that the flow of gas and liquid is relatively uniform, and the flow direction is basically parallel to the wall. However, vortexes appear near the suction surface at the trailing edge of the blade and the flow is unstable with the increase of IGVF, which may be caused by flow separation. Under the rectifying action of diffusers, gas and fluid mixes again and a large vortex appears, which leads to the increase of flow loss.

Figure 16. Relative velocity distribution in 0.5 span rotary surface (IGVF=0%).
4.3.3. Two-phase distribution analysis. The gas volume fraction distributions in different span rotary surfaces with different IGVFs are shown in figure 19~figure 24. At the same IGVF, GVF of suction surface is higher than that of pressure surface because of pressure distribution. The gas-liquid separation occurs near the suction surface at the outlet of impellers, and the GVF increases. Gas-liquid separation also occurs at the inlet of impellers because of impact action by fluid. The GVF from hub to shroud of impeller decreases gradually. Under the rectifying action of diffuser, gas and liquid mixes again and a large vortex appears. Under different IGVFs, the GVF in the runner increases and the proportion of high GVF area increases with the increase of IGVF.
Figure 20. Gas volume fraction distribution in different span rotary surfaces (IGVF=10%).

(a) 0.05 span (near the hub)  (b) 0.5 span  (c) 0.95 span (near the shroud)

Figure 21. Gas volume fraction distribution in different span rotary surfaces (IGVF=15%).

(a) 0.05 span (near the hub)  (b) 0.5 span  (c) 0.95 span (near the shroud)

Figure 22. Gas volume fraction distribution in different span rotary surfaces (IGVF=20%).

(a) 0.05 span (near the hub)  (b) 0.5 span  (c) 0.95 span (near the shroud)

Figure 23. Gas volume fraction distribution in different span rotary surfaces (IGVF=25%).

(a) 0.05 span (near the hub)  (b) 0.5 span  (c) 0.95 span (near the shroud)

Figure 24. Gas volume fraction distribution in different span rotary surfaces (IGVF=30%).

13
5. Conclusions

A model of two-stage compression unit in the helico-axial multiphase pump is established. Numerical simulations of 3D full flow field are carried out. The effects of blade number and blade wrap angle on the external characteristics of the pump are discussed. The optimal design combination is selected, and the effects of different IGVFs on the internal and external characteristics of the pump under the combination are discussed. The work and results are as follows:

1) Fifteen different combinations of impellers are obtained by changing the blade number and the blade wrap angle on the basis of the first compression unit of the helico-axial multiphase pump, and 3D modeling is carried out separately.

2) The flow fields of fifteen two-stage compression units are taken as the calculation objects under steady flow of single phase. The results show that there will be an optimal blade wrap angle (blade number) when the blade number (blade wrap angle) is fixed. In a certain range, the larger the blade wrap angle (blade number), the higher the energy transfer efficiency. However, when the blade wrap angle (blade number) increases again, the flow distance increases (runner narrows, flow rates increases), the friction loss increases. When blade number is 3 and blade wrap angle is 215°, the head and hydraulic efficiency reach the maximum.

3) The steady 3D numerical simulation of helico-axial multiphase pump with the optimal design combination is carried out under different IGVFs. The variation of internal and external characteristics with IGVFs are analyzed. The results show that the pressure-increase and hydraulic efficiency both decrease with the increase of the IGVF. The analysis of pressure field shows that with the increase of IGVF, pressure-increase over pump decreases; the static pressure of pressure surface is larger than that of suction surface. The analysis of velocity field shows that there is velocity slip between gas phase and liquid phase; the diffuser transforms the kinetic energy into pressure energy, which results in the decrease of the velocity of the fluid; the flow of the fluid is more uniform in the impeller, but there is a large vortex and the flow is unstable in the diffuser due to the mixing of gas and liquid. The analysis of two-phase distribution shows that the GVF of suction surface is higher than pressure surface; the GVF from hub to shroud of impeller decreases gradually; with the increase of IGVF, the GVF and the proportion of high GVF area increases.

6. References

[1] Li S Cao F and Xing Z 2011 Research and Application of Aubsea Multiphase Pump Fluid Machinery vol 03 p 40-44+51.
[2] Cheng L and Chen T 1999 Performance and Design Characteristics of Multiphase Flow Pump. Chemical Equipment Technology vol 04 p 42-45. (in Chinese)
[3] Ribeiro O.J.S. R.M.T.Camargo and C.A.S.Paulo 1996 The Impact of Subsea Boosting on Deepwater Field Development. Offshore Technology Conference p 14.
[4] Oxley K.C. and G.J. Shoup 1994 A Multiphase Pump Application in a Low-Pressure Oilfield Fluid-Gathering System in West Texas University of Tulsa Centennial Petroleum Engineering Symposium p 3.
[5] Wang R Wang L Zheng P 2017 Research Status of Oil-Gas Multiphase Pumps Chemical Enterprise Management vol 06 p 5.
[6] Li Q and Xu D 2000 Development and Research of Oil-Gas Multiphase Pump China Offshore Oil and Gas Engineering vol 01 p 47-51+56-5+28. (in Chinese)
[7] Li Q Xu D and Li Z 2004 Optimize of the Helico-Axial Multiphase Pump Prototype and Its Experimental Studies on Performances Journal of Engineering Thermophysics vol 06 p 962-964.
[8] Li Q Xue D and Zhu H 2005 Research on Hydraulic Design Concept of an Helico-Axial Multiphase Pump and Its Experimental Studies on Performances Journal of Engineering Thermophysics vol 01 p 84-87.
[9] Miao C Li Z and Li J 2007 Theoretical analysis on head-flow rate curves of multiphase pump Acta Petrolei Sinica vol 03 p 145-148.
[10] Yu Z Cao S and Wang G 2007 CFD Analysis of the Air-Water Bubbly Flow in a Multiphase Rotodynamic Pump Impeller *Journal of Engineering Thermophysics* vol 01 p 46-48.

[11] Zhang J Zhu H and Yang C 2011 Multi-objective shape optimization of helico-axial multiphase pump impeller based on NSGA-II and ANN *Energy Conversion and Management* vol 52 p 538-546.

[12] Zhang J Cai S and Zhu H 2014 Numerical Investigation of Compessible Flow in a Three-stage Helico-axial Multiphase Pump *Transactions of the Chinese Society for Agricultural Machinery* vol 09 p 89-95.

[13] Zhu H Li Q and Chen L 2007 The Experimental Studies of the Helico-Axial Multiphase Pump *Journal of Engineering Thermophysics* vol 04 p 601-603.

[14] Serena A and L.E. Bakken 2016 *Flow Visualization of Unsteady and Transient Phenomena in a Mixed-Flow Multiphase Pump* (49729) V02DT44A012.

[15] Wilcox D C. 2006 *Turbulence modeling for CFD.*