Numerical Study on Axial Spiral Turbine for Down-hole Generator

Xiaodong Zhang, Huiping Lu, Xuehu Liu, Yan Gong, Wenwu Yang

Abstract—**In order to study the hydraulic performance of spiral turbine for down-hole generator, three dimensional flow field simulation model of turbine was established. According to the actual working condition of down-hole generator, theoretical calculation method of axial spiral turbine was given based on design theory of spiral turbine with constant pitch and uniform thickness. Furthermore, the performance of spiral blade was studied from the three main aspects of the stator and rotor's spiral angle , rotation angle and blade number. By comparing simulation results with theoretical values, it verified the validity of the simulation model and the correctness of the theoretical calculation. Meanwhile, the influence of blade number, spiral angle and rotation angle on output torque, pressure drop and efficiency of the turbine were investigated. The results show that the influence degrees on torque value among the rotor's and stator's blade number, spiral angle, rotation angle are different. The blade number of stator and rotor is best for prime number, and it is relatively appropriate that the number of rotor blades should be more than the stator's. For the blade number between stator and rotor is different, the corresponding values of optimal rotation angle and spiral angle are different. Spiral angle and rotation angle should not be too big or small. And the rotor's spiral angle should be less than stator's. Compared with the rotation angle, spiral angle plays a greater role in turbine hydraulic performance. In practice, the appropriate rotation angle should be selected to be matched with the corresponding spiral angle according to the design requirements.**

Index Terms—**Down-hole generator, Spiral turbine, Spiral angle, Rotation angle, Blade number, Numerical study**

1. Introduction

Along with the continuous development of the directional well, cluster well and horizontal well technology, the development of logging and drilling technology is becoming more and more widely[1-2]. Power consumption of down-hole intelligent tool increases constantly. At present, there are mainly two common ways of power supply. One is battery power, while the other is a kind of under-well turbine generator to produce electricity. The way of conventional battery power supply is easily affected by environmental factors such as high temperature and high pressure. The battery needs to be changed constantly, and it can not meet the need of the underground power supply for a long time. Turbine generator utilizes drilling fluid to make electricity secular and persistent, and it can adapt to the environment of high temperature and high pressure in down-hole. Turbine generator has more advantages in the development prospect of down-hole power supply.

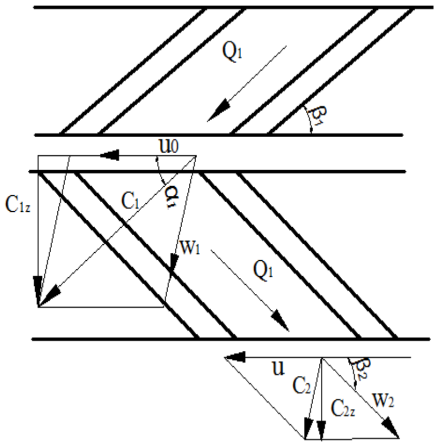
Therefore, it is of great significance to conduct the thorough research on turbine, which is the energy conversion part in down-hole generator. At present, due to the limitation of the installation size, down-hole turbine generator often utilizes spiral blade for convenience of structure design and processing. Many foreign scholars and research institutions discussed design methods and relevant test methods of turbine[3-6]. But domestic research on turbine generator starts late. X.Y. Zhang et al. [7] studied the helical turbine, and compared the performance characteristics of these three spiral turbines which were the uniform pitch and variable pitch, the width-variable blade. The results provided a strong basis for turbine spiral structure improvement. Z.Q.Yi et al. [8] researched on the turbine straight blades of small diameter under-well turbine generator computational using the fluid dynamics analysis, got the stator and rotor's distribution of internal flow field. X.Y. Zhang et al. [9-10] studied turbine of equivalent thickness and constant spiral angel, and simple theoretical calculation formula was given. However, the theoretical calculation method was only applicable to single model of rotor blades. X. Yang et al. [11-13] modified shape of turbine blades according to flow field characteristics, but blade overlap coefficient was determined by experience. The effect of rotation angle on the performance of the blade was not taken into concrete consideration. The way of optimizing was relatively simple, and the effect of improvement was slightly less perfect. Therefore, on the basis of these studies, this paper aimed at analyzing a kind of stator and rotor of generator set of turbine, its theoretical calculation method was given, and the influence of blade number, spiral angle, rotation angle of stator and rotor on hydraulic performance of turbine were analyzed. The optimum ranges of main structural parameters of the rotor and stator were discussed. The results of the study will provide valuable insight for improving the hydraulic efficiency of traditional spiral turbine.

2. The structure design and modeling of down-hole turbine

Turbine is one of the most important part of down-hole turbine generator. This study focuses on spiral turbine composed of a stator and rotor. The stator acts as a guide roller, which is fixed in front of turbine, adjusts the mud flow direction of shocking to blade surface to improve the efficiency. The rotor is the power component. Drilling fluid impacts the turbine blades to produce circumferential force which makes turbine rotate, thus promoting the generator to achieve power generation.

2.1 Design theory of down-hole turbine

The theoretical basis of this paper is based on design theory of spiral blades that have equal pitch and equal thickness. Velocity triangle of drilling fluid at the inlet and outlet of rotor is shown in Fig.1 below.



Q1—actual flow through turbine channel
w—relative velocity of flow fluid through blades
u0—circumferential velocity of liquid rotating with rotor blades
c—absolute velocity of flow fluid
α1—inlet flow angel of rotor
β—spiral angel
1— parameters of footnotes at outlet of stator and inlet of rotor
2— parameters of footnotes outlet of rotor
z—parameters of footnotes axial velocity
u—parameters of footnotes circumferential velocity

Fig.1 drilling fluid velocity triangle at the inlet and outlet of the rotor

C_1 is inlet velocity of the rotor, which is decided by axial velocity C_{1z}. α_1 is the inlet flow angle of the stator, $\alpha_1=\beta_1$, β_1 is spiral angle of stator. Once drilling fluid enters the rotor channel, it will rotate to u_0 with the rotor, and circumferential velocity u_0 can be expressed by:

$$u_0=\frac{\pi n D_0}{60} \tag{1}$$

C_{1u} and C_{2u} represent the circumferential components of C_1 and C_2 of absolute speed at the inlet and outlet of the rotor, respectively, they can be calculated by:

$$C_{1u}=\frac{C_{1z}}{\tan\beta_1}\ ,\ C_{2u}=u_0-\frac{C_{2z}}{\tan\beta_2} \tag{2}$$

Where β_1 is spiral angle of stator, u is circular velocity, β_2 is spiral angle of rotor, C_{1z} is actual axial velocity of fluid in stator, C_{2z} is actual axial velocity of fluid in rotor.

By the liquid velocity triangle at inlet and outlet of rotor, the liquid impacts blades to produce torque M ,it can be calculated by:

$$M=\rho Q(C_{1u}R_1-C_{2u}R_2) \tag{3}$$

Where ρ is the density of water, Q is flow rate, for axial flow turbine, there is $R_1=R_2=D_0/2$, D_0 is the middle diameter of turbine blades.

From Eq. (1) , (2)and(3), torque M can be expressed by:

$$M=\rho Q\frac{D_0}{2}\left[\left(\frac{C_{1z}}{\tan\beta_1}+\frac{C_{2z}}{\tan\beta_2}\right)-u_0\right] \tag{4}$$

Output power of turbine blades:

$$P=\omega M=\frac{\pi n}{30}M \tag{5}$$

Where n is rotational speed, ω is angular velocity.

Because blades have a certain thickness and number, excretion coefficient should also be considered for the influence of axial velocity, so actual axial velocity should be calculated as:

$$C_{1z}=\frac{C_z'}{\varphi_1},C_{2z}=\frac{C_z'}{\varphi_2} \tag{6}$$

Here, C_z' is theoretical axial velocity , φ_1 and φ_2 are the excretion coefficient:

$$\varphi_1=1-\frac{Z_1 m}{\pi D_0\sin\beta_1},\varphi_2=1-\frac{Z_2 m}{\pi D_0\sin\beta_2} \tag{7}$$

Where Z_1,Z_2 are blade number of stator and rotor respectively, m is blade thickness.

Theoretical axial velocity can be expressed by:

$$C_z'=\frac{Q}{\pi D_0 b} \tag{8}$$

Where b is radial height of blade

By (3) ~ (5),they can be converted to:

$$\left(\frac{60P}{\rho n Q^2}+\frac{n\pi^2 D_0^{\ 2}}{60Q}\right)\left(1-\frac{Z_2 m}{\pi D_0\sin\beta_2}\right)\left(1-\frac{Z_1 m}{\pi D_0\sin\beta_1}\right)*$$
$$\tan\beta_1\tan\beta_2 b=\left(1-\frac{Z_2 m}{\pi D_0\sin\beta_2}\right)\tan\beta_2+$$
$$\left(1-\frac{Z_1 m}{\pi D_0\sin\beta_1}\right)\tan\beta_1 \tag{9}$$

According to Eq.(9), if flow rate Q, output power P, rotational speed n are given according to design requirements, the relation between the spiral angle of β_1 and β_2 can be calculated with a known number of blades(Z_1,Z_2)and blade thickness (m). Because the theoretical calculation is suitable for the ideal model, namely, the rotor and stator should be completely symmetrical structure. Therefore, $\beta_1=\beta_2$, $Z_1=Z_2$.

Initial design parameters: outer diameter of blade D= 128 mm, inner diameter of blade d=98 mm, radial height of blade b=15mm, medium diameter D0=113mm, blade thickness m= 5mm, blade number $Z_1=Z_2=7$;

Design requirements: the output power P⩾1000W, flow rate Q=27 -35L/s, rotational speed n=1500-2500r/min. In this paper the initial flow rate is 27L/s, the rotation speed is 1500 r/min. Taking the known parameters into Eq.(9), it can be obtained: $\beta_1=\beta_2$ =44.71°.

Introducing another indirect calculation parameters of blade structure, θ wrap angel at blade section or rotation angle of blade, it can be expressed by:

$$\theta=\varepsilon\frac{2\pi}{Z} \tag{10}$$

Where ε is overlap coefficient, Z is blade number.

Axial height h of blade：

$$h=\frac{\theta D_0}{2}\tan\beta \tag{11}$$

Rotation angle and spiral angle are known, assuming primary value of θ is 90°, θ_1 and θ_2 present rotation angel of stator and rotor, respectively. The axial height can be identified according to Eq.(11). Hence, 3D model can be presented based on known basic parameters of turbine.

3 Numerical simulation of flow field

3.1 Numerical simulation model

In the process of modeling, in order to reduce the difference between the fluid flow and the actual flow at the inlet and outlet for an approximate steady solution, specifically, it often extends upward for the inlet boundary of stator, downward extension for outlet boundary of rotor. The three-dimension model of the turbine passage was established by UG soft, and then it was imported into ICEM-CFD software to mesh. After modified and improved, the three-dimensional grid of the turbine model was shown in Fig.2.

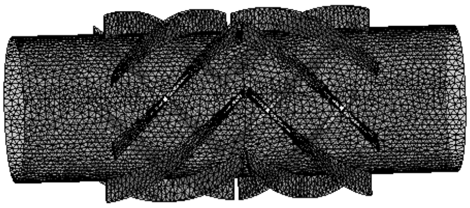


Fig.2 grid computing model

3.2 Governing equations and boundary conditions

CFD calculation equation is Navier Stokes equation. The flow field is calculated based on the Navier-Stokes equations as the governing equations for the accurate description of the actual flow:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u_i)}{\partial x_i}=0 \tag{12}$$

$$\rho\left[\frac{\partial u_i}{\partial t}+u_j\frac{\partial u_i}{\partial x_j}\right]=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_i}\left[u_j\frac{\partial u_i}{\partial x_j}\right]+\frac{\partial R_{ij}}{\partial x_j} \tag{13}$$

Where R_{ij} is the reynolds stress tensor, μ_t is turbulence viscosity coefficient, K is the turbulent kinetic energy, S_{ij} is deformation rate tensor.

S_{ij} is defined as below:

$$S_{ij}=\frac{1}{2}\left|\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right| \tag{14}$$

R_{ij} is defined as below:

$$R_{ij}=-\rho\overline{u'_j u'_i}=2u_t S_{ij}-\frac{2}{3}\mu_t\frac{\partial u_k}{\partial x_k}\delta_{ij}-\frac{2}{3}\rho k\delta_{ij} \tag{15}$$

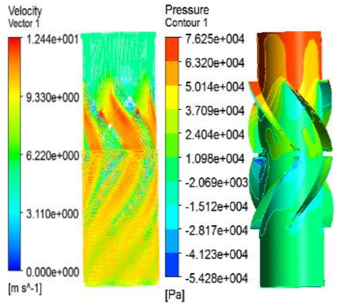
Set the boundary conditions as follows: he type of interface was set as frozen rotor. And for all of the CFD simulations of this paper, it was achieved by specifying a normal speed at the inlet and a total pressure at the outlet. Normal speed at the inlet of the stator was set to 5.1m/s, and the pressure at the outlet was set to 1 atm. No slip walls were allowed for the rest boundary of main walls.

4 Effects of structural parameters on turbine performance

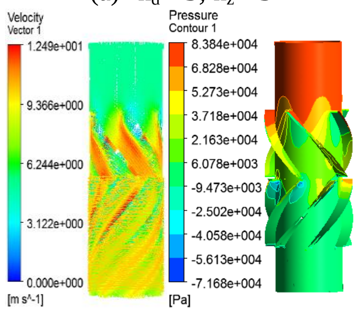
Turbine as a key part of generator, its structure parameters of blades affect the hydraulic performance of turbine. Rotation angle was ignored in the existing study of hydraulic performance of turbine. Then the following main three aspects: blade number, spiral angle and rotation angle were studied.

4.1 Analysis on different number of blades

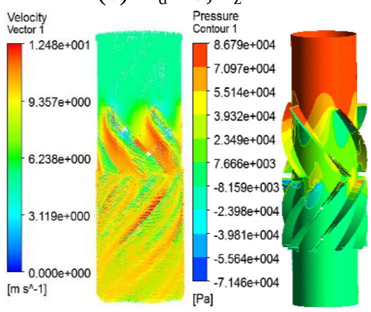
The even number of blades can easily cause resonance, so the whole blades will be forced badly. Therefore, an odd number of blades was investigated in this paper. Taking initial parameters $\theta_1=\theta_2$ =90°, $\beta_1=\beta_2$ =44.71° for example to research on turbine flow field in the corresponding situation, when blade number of rotor n_z = 7, n_d takes 5, 7, 9 and n_d = 5, n_z takes 5, 7, 9 . Concrete analysis results are as follows：



(a) n_d =5, n_z =5



(b) n_d =5, n_z =7



(c) n_d =5, n_z =9

Fig.3 turbine flow field in different number of stator blades

Fig.3 shows turbine flow field of different blade number of rotor. As blade number increases, turbine pressure and fluid velocity increase gradually. Because cascade flow space of turbine decreases, fluid at outlet of the guide roller has larger contact area with the turbine blade surface so as to improve the efficiency of absorption in unit fluid momentum and transformation with the increasing of blade number.

Table 1 n_z=7flow field data in different stator blade

Turbine parameters	Data			
Blade number/slice	5	7	9	13
Pressure drop/(Mpa)	0.071	0.065	0.073	0.089
Torque /(N·m)	6.120	4.979	5.475	7.708
Shaft power/w	960.84	781.7	859.58	1210.16
Efficiency	0.502	0.446	0.437	0.504

Table 2 n_d=5 flow field data in different number of the rotor blade

Turbine parameters	Data			
Blade number/slice	5	7	9	13
Pressure drop/(Mpa)	0.063	0.071	0.074	0.083
Torque /(N·m)	5.411	6.120	6.394	7.251
Shaft power/w	849.53	960.84	1003.8	1138.3
Efficiency	0.5	0.502	0.503	0.508

The Table 1 shows, When $n_z < n_d$, as the number of stator blades increases, shaft power increases gradually. There will be more difficulties in manufacture if the blades are beyond the quantity. And flow channel space is too small to easily form circumfluence and secondary flow phenomena. When $n_z > n_d$, the less blade number of stator is, the greater power output of turbine is. But too few blades are not conducive to improve the utilization rate of fluid energy. From the Table 2, as the number of rotor blades increases, output power of shaft increases gradually, and efficiency increases slowly. When $n_z = n_d$=7or 5, torque simulation values are respectively 4.979 and 5.411 N·m. The torque values calculated by Eq.(4) were respectively 4.637 and 5.021 N·m. Simulation results show excellent agreement with the theoretical calculation, which proves the correctness of the theoretical calculation. When numbers of stator and rotor blade are equal, the output power and efficiency of the turbine are less than the asymmetric turbine's when n_d=5,n_z=7. By considering shortcomings of too many blades and synthesizing data of table 1 and Table 2, it can be concluded that it is more appropriate when the blade number of stator is 5 and blade number of rotor is 7. It's illustrated that turbine performance is better when structure of stator and rotor are designed for the asymmetric structure.

4.2 Analysis on different rotation angle

The stator acts as the role of the guide roller, improving the utilization of energy by changing the inlet flow angle. Improper selection of overlap coefficient will have adverse effects on the hydraulic efficiency of blade, and it associates with the selection of rotation angle.

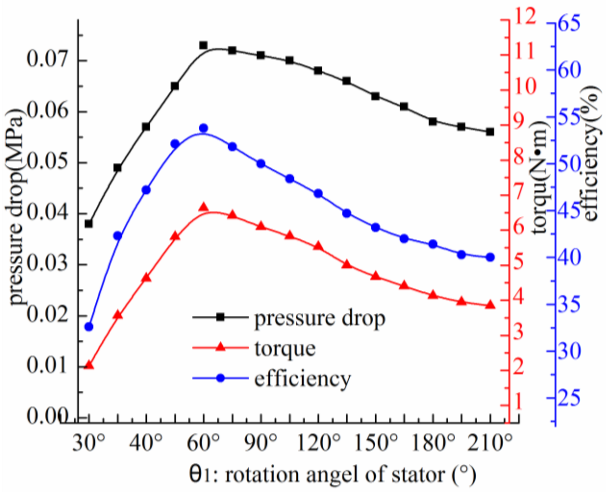


Fig.4 impacts of θ_1 on hydraulic performance of turbine

Hence, it is very necessary to analyze how rotation angle of stator blade impacts on hydraulic performance of turbine. Taking 5 as the stator blade number, 7 as the rotor blade number, and 44.71° as the rotor spiral angle. By changing rotation angle of stator to analyze turbine flow field, specific results of the analysis as shown in Fig.4.

It can be seen from the Fig.4, as rotation angle of stator blade increases gradually, the pressure drop, efficiency ,output torque of turbine increase firstly and then decrease. When stator's rotating angle is 60 °, at this time, it achieves the maximum torque values and efficiency. So in this example of turbine, in order to obtain the maximum shaft power, rotation angle of stator should be about 60 °.

Changing the rotation angle of rotor blade will affect overlap coefficient of the rotor blade. If overlap coefficient is too small, utilization of fluid energy will be reduced. Fig.5 is about the relationship between torque, pressure drop ,efficiency and rotation angle of rotor blades. When rotation angle is less than 45 °, torque , pressure drop and efficiency will all increase as the rotation angle increases. And their changing trends are almost similar. When rotation angle is more than 45 °, torque, pressure drop, and efficiency linearly decrease with the increase of rotation angle. Seen from the Fig.5, when torque comes to a maximum, pressure drop and efficiency are also the maximum. Therefore，he best rotation angle of rotor blade is about 45 ° in this example. Too big or too small angel will lead to smaller values of the torque. Hydraulic efficiency of blades was reduced. Comparing Fig.4 with Fig.5, it can be concluded that when blade number of stator and rotor is different, the best rotation angle of corresponding value is also different.

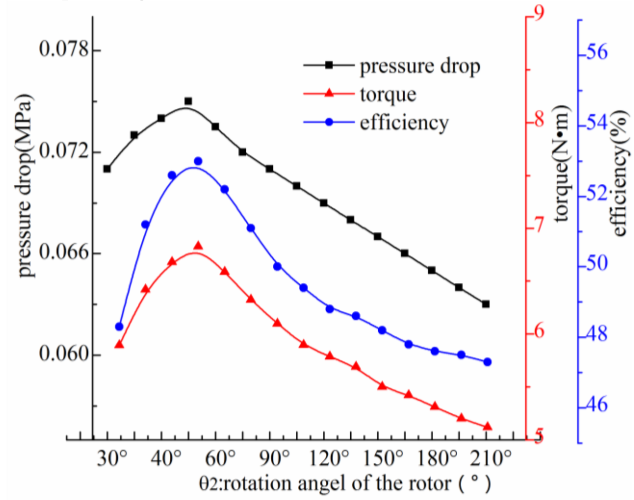


Fig.5 impacts of θ_2 on hydraulic performance of turbine

4.3 Analysis on the different spiral angle

From the Fig.6, the values of torque and pressure drop decrease with increases of stator's spiral angle, the efficiency firstly increases and then decreases. While stator's spiral angle is smaller, hydraulic efficiency of turbine is lower. When spiral angle of stator ranges from 37° to 42 °, the hydraulic efficiency of turbine is much bigger. Thus, stator's spiral angle should not be too small. Comparing Fig.4 with Fig.6, to improve the equal size of torque value, rotation angle needs changing at least 30 °. But 1 °of spiral angle changing has the same effect on torque value. Compared with rotation angle, spiral angle has greater influence on torque and pressure drop, efficiency. Selecting appropriate rotation angle can improve the value of torque, thus improving the hydraulic efficiency of turbine and increasing the generator power. Priority should be given to appropriate spiral angle in practical engineering application and then determining the best rotation angle to match the corresponding spiral angle value.

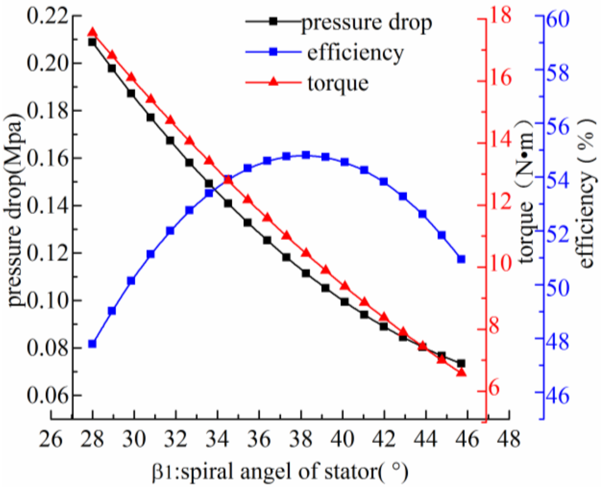


Fig.6 impacts of β_1 on hydraulic performance of turbine

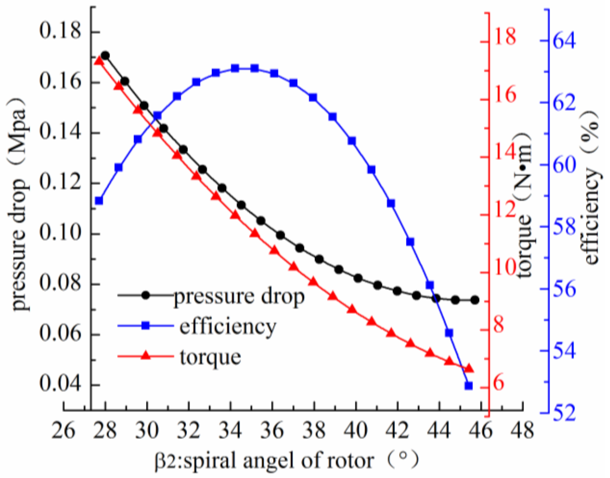


Fig.7 impacts of β_2 on hydraulic performance of turbine

Fig.7 shows relationship between torque, pressure drop, efficiency and spiral angle of rotor blades. Torque and pressure drop decrease gradually with the increase of spiral angle. And efficiency increases to a maximum with spiral angle increasing to 36°and then sharply reduces. Small torque value will lead to low hydraulic efficiency of turbine, thus it cannot reach the expected design purpose. In this example, efficiency is higher when spiral angle ranges from 31°to 38°. From Fig.6 and 7, turbine efficiency is low when stator's spiral angle is less than 32°, but for spiral angle of rotor , turbine efficiency is low while spiral angel is more than 44°. Comprehensively, spiral angle of stator should be slightly larger values than spiral angle of rotor. Moreover, best scopes of spiral angle between stator and rotor are different when they have different blade number.

5 Conclusion

(1) On the basis of the design theory of constant pitch and uniform thickness of spiral turbine, theoretical analysis and calculation of a axial spiral turbine with stator and rotor were presented. By comparing theoretical values with CFD simulation results, the theoretical formula was verified to be practical.

(2) By carrying the numerical simulation on the structure of stator and rotor with different number of blades, it can be found that the number of stator blades should be less than the rotor's. Moreover, both of their blades shouldn't be overmuch. In this example, the number of stator blades should be five, and rotor should be seven. That is to say, the design with asymmetric structure of turbine is better.

(3) By carrying the numerical simulation on the structure of stator and rotor with different rotation angle, we find the selection of rotation angle will affect the torque and the drop pressure. And the value of the best rotation angle is different between stator and rotor when they have different number of blades. So the overlap coefficient should be selected reasonably in the actual design. And rotation angel cannot be ignored. The way of CFD method provides a reference basis for the determination of overlap coefficient, as well as to determine the design of axis height of blade.

(4) By analyzing the rotation angle and spiral angle of stator and rotor, it can be concluded that spiral angle has a greater influence on the torque and the drop pressure than rotation angle. In this example, the appropriate scope of the stator's spiral angle is 37° to 42 ° and the rotation angle should take about 60 °. Rotor's spiral angle is suitable for 31 ° to 38 °and the best rotation angle is about 45 °. It further illustrates that design of spiral turbine of down-hole generator should be asymmetric structure.

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References

[1] H.L. Qu, Z.M. Wang, J.Y. Xiao, The summary of the research on downhole turbine generators[J], Modern Manufacturing Technology and Equipment, 2010, 196(3): 6-7.
[2] Y. Shen, Y.N. Su, L. Li, Design of down-hole turbine alternator for measurement while drilling and its performance analysis[J], Acta Petrolei Sinica, 2008, 29(6): 6-7.
[3] Hoschek J, Müller R, Turbine blade design by lofted B-spline surfaces[J], Journal of Computational and Applied Mathematics, 2000,119(1):235-248.
[4] Bernard M, Maryse P, Anne M,et al, Improving efficiency of a 195 MW francis turbine using numerical simulation tools [C], 20th IAHR Symposium,Charlotte,North Carolina,U. S. A. ,2000.
[5] William H B, Edward C F, Henry M. Fluid-driven alternator having an internal impeller: US, 6607030[P], 2003-08-19.
[6] R uprecht A, Unsteady flow analysis in hydraulic turbmachinery[C], 20th IAHR Symposium, Charlotte, North Carolina, U. S. A. 2000.
[7] Zhang X.Y. Zhang, J. Feng, Hydraulic behavior research of three types of under-well helical turbine[J], Mechanical Design and Research, 2010,37(11):11-13.
[8] Yi Z.Q. Yi, Research on turbine blade of downhole generator used in small-diameter well based on simulation, Mechanical Research &Application, 2013,126(4):72-74.
[9] X.Y. Zhang, J. Feng, H.B. Luo,et al., Design of Axial-flow impeller based on CFD analyse[J], Oil Field Equipment, 2007,36(2):22-25.
[10] X.Y. Zhang, J. Feng, Hydraulic behavior research of under-well variable-pitch turbine[J], Mechanical Design and Research, 2009,36(2):26-28.
[11] X. Yang, L. Wang, Y. Liu, Numerical analysis of hydraulic performance of turbine motor[J], Modern Machinery, 2011,37(1):43-45.
[12] X. Yang, Y. Liu, Optimazition of tuebine motor structure[J], Mechanical Engineering and Manufacturing and Automation, 2010,163(6):181-182.
[13] X. Yang, Y. Liu, Research performance of the downhole turbine-generator[J], Mechanical Design and Research, 2010,37(10):39-41.

Xiaodong Zhang, Male, born in 1959. Professor and doctoral tutor in Southwest petroleum university, got a master's degree in Beijing university of science and technology of mining machinery professional in 1995. Main research direction for oil drilling equipment, new technology, modern design theory and method of teaching and oil drilling tools, especially the development of the downhole drilling tools and downhole tools.

Huiping Lu, a graduate student, research direction: downhole drilling tools, School of Mechatronic Engineering, Southwest Petroleum University, Chengdu 610500, China

Xuehu Liu, Southwest Branch Company, Sinopec Corporation, Sichuan, China