Design of an Axial Turbine Analysis

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Abstract

This paper make use of ANSYS to establish a occupy space on the edge of internal pipeline of high performance hydraulic turbine model, from the theoretical analysis to enhance the flow, speed on the turbine efficiency have positive role, and based on the theory of fluid dynamics (CFD), using CFX software to establish 3D model, simulation and analysis of the influence of turbine structure on the output power; simulation results show that: turbine blade number can affect non axial turbine generator output power, and in turbine blade number for 9 hydraulic characteristics to the best state. The above conclusions provide the basis for further improving the output power of the special generator and the study of the performance of the non axial turbine structure.

Keywords

Oil pipeline; non shaft turbine; turbo generator; flow field.

1. Introduction

In remote, no light, no wind and the long-term in the harsh environment, such as the rainy season, the area, the long distance oil pipeline valve chamber of low cost instrumentation for power supply has become a major problem, due to the influence of the external condition. Do not use the such as wind and solar power to provide long-term stability of power, and only rely on people regularly to the valve chamber operation to replace the power supply system. In this model the free turbine shaft power device, the key components of turbine as the generator energy transformation, with small size, high power, anti abrasion and high pressure characteristics. So it is necessary to the turbine leaf form, modeling and manufacturing method are studied, so as to enable the generator system structure more simple, longer life, power generation efficiency higher. Research in the field of blade of wind power equipment, can design and three-dimensional simulation analysis and optimization of genetic algorithm for small wind turbine blades based on using fluent 3-D simulation get favourable parameters, using genetic algorithm optimization of the parameters of the wind turbine to achieve wind energy conversion maximization; at the same time, in the field of underground drilling turbine power generator blade, logging while drilling (LWD) with turbine impeller hydraulic performance analysis, using CFD software modeling and simulation analysis, calculation of the hydraulic characteristics of the blade, prediction of the output characteristic of the impeller. In this paper, a method of using SoildWorks software to establish a three-dimensional model of the non axis of twist surface, thickness, lift angle and blade number is variable. The method of computational fluid dynamics is adopted, and theANSYS software CFX module simulation. Finally through to different leaf blade root, leaves leaves top static pressure difference and the output power are compared and analyzed, so as to obtain the energy conversion rate high, the blade stress, good hydraulic performance of turbine type.

2. Turbine structure design and modeling

2.1 Introduction to the working principle of the non axial turbine

Oil pipeline of the poor environment, high pressure, with strong corrosive and contains a large number of sludge, in view of the requirements of the industrial design, the design of physical turbine shaft as shown in Figure 1. The pipe turbine is used as an axial flow impeller, and the blade is used for absorbing the kinetic energy of the flow of oil to be converted into mechanical energy, which drives the turbine to rotate and generate electricity

Figure 1 non axial turbine in kind

2.2 non axial turbine design theory

The design of the long distance transmission pipeline is mainly including the parameters such as the number of blades, the thickness of the blade, and the lifting angle. The effective power output of the turbine can be calculated by the formula (1), (2) and (3):

$$T_N = \frac{30P_u}{n\pi} \tag{1}$$

$$P_p = \frac{P_u}{\eta_l\eta_e} \tag{2}$$

$$M_P = \frac{P_u}{n} \tag{3}$$

Seek to obtain:

$$p_u = \frac{1}{30} M_P \pi n \eta_l \eta_e \tag{4}$$

T_N— Turbine theory output torque, N•m;

P_u—Pipeline motor theory output power, W;

n—Turbine stable speed, r/min;

η_1—Electromagnetic conversion efficiency, 0.8;

η_e—Mechanical efficiency of generator, 0.8;

When the blade is of equal thickness and equal pitch angle, the axial velocity V, the thickness of Z, and the number of M of the turbine blade are:

$$v = 1 - \frac{zm}{\pi \bar{D} \sin\gamma} \tag{5}$$

$\bar{D}$– Blade mean diameter, mm;

B–Blade helix angle, °;

The flow rate of the flow through the cascade is calculated:

$$q = (1 - \frac{\bar{d}}{D})Q \tag{6}$$

D—Inner diameter of turbine, mm;

$\bar{d}$—Average diameter of the top of the leaf, mm;

Axial velocity C_Z "and the actual axial velocity of oil due to known axial velocity C_Z"

$$c_z^{'} = \frac{1}{\pi \bar{D} b} q \tag{7}$$

$$c_z = \frac{c_z^{'}}{v} \tag{8}$$

The circumferential velocity of the outlet fluid can be obtainedu_z:

$$u_2 = \frac{\pi n \bar{D}}{60} \tag{9}$$

The output torque of the turbine isM_P:

$$M_p = \rho q \bar{D}(\frac{c_z}{\tan\gamma} - \frac{\pi n \bar{D}}{60}) \quad (10)$$

The output effective power of the output shaft turbine generator () can be produced from the relation type (1) and (11):

$$p_u = \frac{1}{30}\pi n \eta_l \eta_e \rho Q \bar{D}[\frac{Q\cos\gamma}{b(\pi\bar{D}\sin\gamma - zm)}(1-\frac{d}{D}) - \frac{\pi n \bar{D}}{60}] \bullet (1-\frac{d}{D}) \quad (11)$$

According to the formula (11), the value at the given parameters of the leaf, helix angle gamma, electromagnetic conversion efficiency ETA L and generator mechanical efficiency ETA e parameters can be calculated axial turbine generator output theory effective power.

2.3 The establishment of a non axial turbine model and the parameterization of blades

In the establishment of a non axial turbine model, the more the number of blades, the smaller the flow channel space, the greater the degree of overlap. In the optimization process of the analysis of the turbine flow field, we should focus on the optimization of the turbine model to improve the efficiency of the analysis, so as to establish the model of the data in Table 1 and the corresponding flow passage model.

Table 1 Parameters of the main design of the turbine model

Feature item	Turbine	Guide
Blade number m / A	(5-11)	7
Blade thickness /mm	4	4
Leaf diameter /mm	70	70
Inlet flow channel length /mm	-	70
Outlet flow channel length /mm	70	-
Elevating angle / ^{O}	45	
Leaf tip diameter /mm	55	55
The mean diameter of the blade is /mm	63	62.5
Blade shaft length /mm	65	70
Ye Pingjun high /mm	15	15

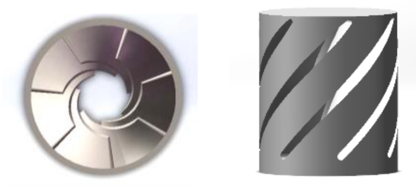


Fig. 3 7 blade turbine and corresponding full channel model

In turbine blade design, because blades inside the turbine, processing more difficult and according to the blade root and pipe wall stress situation, imports and the original import of the blade at the same, exit design approximately 5 / 4 of the original export, dispersed impact of the fluid on the blade, so as shown in Figure 4 using arc and comparison method to gamma to design the blade profile, leaves the maximum thickness as the thickness of the blade and model blade edge at the tip of the sleek design, favorable weakened in the inlet ofblade stress concentration; this design leaves increased leaf life span and maximum ceiling fluid kinetic energy.

Fig. 4 the composition of the turbine blade profile

3. Optimization and Simulation of the turbine flow field

For distorted surface of turbine blade is more complex, with the number of turbine blade increased, the cascade passage becomes narrow and more, so the use of unstructured grid based on finite volume method Tetra/Mixed. At the same time in these parts of the turbine blade inlet and blade suction surface near the basin suitable when increasing grid density, as shown in Figure 5, the total number of grid 455677, number of nodes for 653335. By using this method, it is not only able to make the generation of geometric patch and shell mesh shorter, but also to the numerical simulation of the impeller machine.

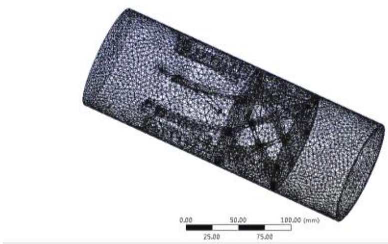


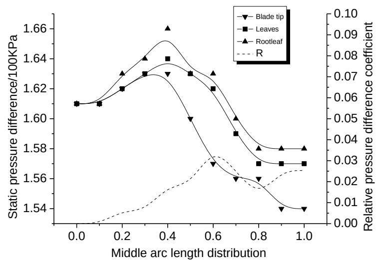
Fig. 5 grid model of turbine flow field

The turbine blade region as a regional rotation, and the velocity of fluid pipeline parallel, fluid design for water, in the static temperature 20 water dynamic viscosity coefficient for mu = 1.007 x10-6m2/s, which water density p =1000kg/m3, inlet velocity v1=5m/s, outlet pressure p2=0.3MPa, the rotational speed of the turbine design for n=150r/min, the wall surface of the wall slip, the flow to the Z axis as center of rotation.

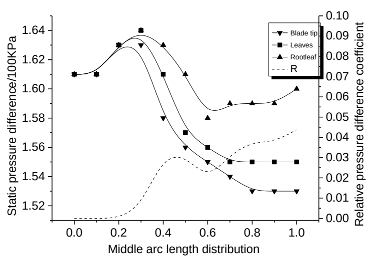
Solving incompressible turbulent flow field calculation of the segregated solver, the RNG turbulent model and the Reynolds equation combined as solving turbulent control equations, discrete interpolation method with second order upwind schemes for high precision, at the same time, the standard wall function to determine the flow near the wall.

3.1 turbine blade number m analysis

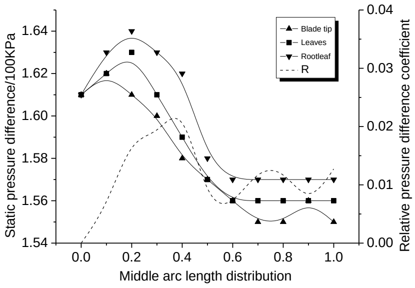
Even blades are very easy to cause resonance, so the model of the following research are odd shape, with the lifting angle of 45 degrees and the thickness of Z for 4mm simulation. Of leaf number corresponding to the flow field in turbine, the income to the pressure surface relative rotary suction surface static pressure difference P and tip Leaves Roots and leaves of relative pressure coefficient r, the pressure difference coefficient r, with leaves arc line changes the distribution is as follows:



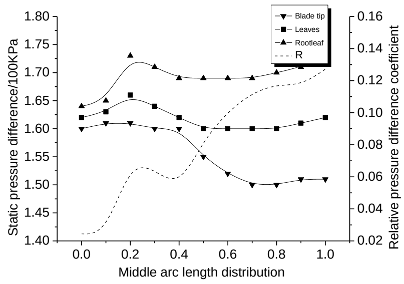
m =5



m =7



m =9



m =11

Fig. 6 5 relative rotation pressure difference and relative pressure difference coefficient of -11 blade number

From Figure 6 shows that, when the blade number when, the turbine blade by the shock, the blade leading edge and tip by the greater pressure; with the increase of the number of blades, the static pressure of the blade tip and blade root difference and relative pressure difference increases, and leaves the high-pressure area decreases gradually and moving towards the root and leaf blade leading edge relative pressure difference coefficient C maximum decreases, after leaf number more than 9 times, the relative pressure difference coefficient C increased overall, leaves in high and low pressure area very obvious in the.

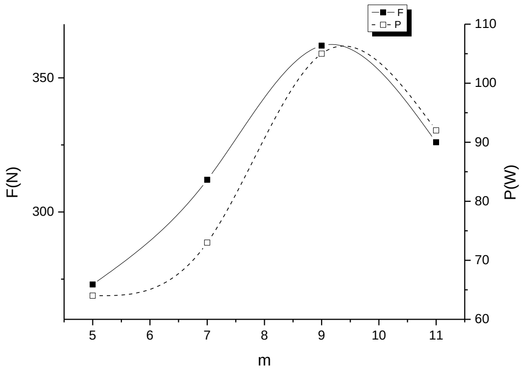


Figure 7 5-11 form for turbine blade force and power

At the same time, it can be seen from Figure 7, the axial force of the turbine blade and the corresponding output power increases first and then decreases. The reason is that: with the increase of blade number m, fluid cascade basin decreased, but the leaf resistance increases, no shaft turbine with rotating fluid, the amount of fluid through the centre of the turbine increases, near the center of the turbine blade tip by the impact of increases, the pressure surface of the blade increases the absorption unit volume flow kinetic energy and efficiency of blade the torque is increased; when the number of leaves reached 9 of the time, due to the amount of fluid through the centre of the turbine cascade fluid through too much, too little, turbine axial force and the output power decreases, while the static blade root difference compared to smaller, relying on the center of the turbine export flow is quick, blade outlet the relative tip by the impact of large, static pressure difference, the greater the damage to the blades. Moreover, too many leaves increase the difficulty and cost of the process. Therefore can choose 9 leaves as the number of turbine blade, the following research take 9 leaves.

4. Conclusion

By means of simulating the flow field of a non axial turbine model, the number of blades, the thickness of the blades and the lifting angle of the axial turbine can all affect the hydraulic performance of the axial turbine. According to the needs of industry, regulating shaft turbine blade parameters, dispersion of rotation of the static high pressure or low pressure, improve the blade stress uniformity of, to satisfy the need to maximize the power, without the protection of the working life of the turbine shaft. The number of 9 hydraulic characteristics in the non axial turbine blade is better than the best.

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