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Non-overload design of low specific speed submersible pump

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ABSTRACT

Based on the design of low specific speed submersible pump, the influence of impeller geometrical parameter on shaft power of pump is analyzed in this paper. Under the condition of the maximum value of shaft power being no more than the input power of matching electromotor, the method of reducing the maximum value of pump power is researched further in order to reduce the power of the matching electromotor. Then the design correlation formula of the geometrical parameters of impeller is presented. The value of shaft power was predicted by the software FLUENT. The results show that the ratio of maximum power of pump to the power at designing condition is less than 1.2 when the correlation formula presented is used. The experimentation of prototype shows that the maximum value of shaft power of pump reduces greatly whilst every parameter meets the rated requirements. The results demonstrated the guiding role of the discriminated formula in practice, so the desired aim was achieved.

INTRODUCTION

Low specific speed submersible pumps are often used to extract groundwater, and have been widely applied in rural areas 、 mines 、 water supply company 、 geothermal development and other areas. In particular, as most people are seeking more energy, low specific speed submersible pump in the coal mine industry demand is also growing. However, the actual operating performance of this pump is very unstable causing relatively short life of the pump, one of the important reasons is that pump is prone to overload and will lead to burning of motor.

Low specific speed submersible pump's shaft power curve rises with the flow increases. The lower specific speed, the shaft power curve upward trend is more obvious[1~2]. In the process of coal mine drainage, a kind of pump is used in variety of occasions. Especially in emergency reccue, the pump of high rated head is used to run in lower head condition, that cause the shaft power increases rapidly with head reduce and flow decline.

Currently in China, motor power matching conditions of low specific speed submersible pump are generally rated shaft power of 1. 2～1. 3 times the bep power. However, when the head is low, the maximum shaft power is likely to exceed the rated operating conditions of 1.5 times. The nature of high-head low specific speed submersible pumps used in low head conditions will lead to burning of motor overload. Therefore, the reseach of no overload for low specific speed submersible pump has great significance.

NON-OVERLOAD DESIGN OF LOW SPECIFIC SPEED SUBMERSIBLE PUMP

It was found that the design of the pump have significant influence to the non-overload performance of pump after experimental study. The study of low specifilc speed submersible pump which non-overload performance means to make the characteristic curve of pump shaft power more flat and the maximum of shaft power lower (less than the rated shaft power 1.2 times) by new impeller design technology. It will not give overload problems and reduce the matching of motor power in all pump heads range, and so could save energy.

When the impeller of low specifilc speed submersible pump is designed, the geometrical parameters affecting the maximum of shaft power. Should be considered looking at the impeller pump basic equations and principle of two-phase flow.

Design based on the fomula

The interaction of impeller design parameters and shaft power of clean water centrifugal pump were investigated by many people. The fomula of maximum shaft power of clean water centrifugal pump is derived by Yuan[1]:

$$P_{\max} = \frac{1}{4\eta_m} \rho \sigma^2 u_2{}^3 \pi D_2 b_2 \psi_2 \tan \beta_2 \qquad (1)$$

$$Q_{\max} = \frac{1}{2} h_0 \tan \beta_2 \eta_v \pi D_2 b_2 \psi_2 u_2 \qquad (2)$$

$$\sigma = \frac{u_2 - \triangle v_{u2}}{u_2} = 1 - \frac{\triangle v_{u2}}{u_2} = 1 - \frac{\pi u_2 \sin \beta_2}{z u_2} = 1 - \frac{\pi}{z} \sin \beta_2 \qquad (3)$$

$$\psi_2 = 1 - \frac{z s_{u2}}{D_2 \pi} \qquad (4)$$

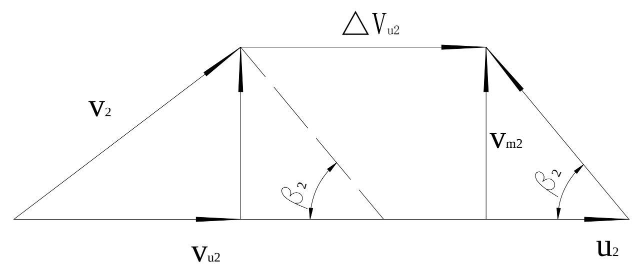
It is found that the maximum shaft power of low specific speed submersible pump is generally greater than $P_{\max}$ of clean water centrifugal pump, but not more than $0.275 \rho \sigma^2 u_2{}^3 \pi D_2 b_2 \psi_2 \tan \beta_2 / \eta_m$. Therefore, to meet the maximum shaft power of low specific speed submersible pump is not greater than rated shaft power of 1.2 times. We set the following fomular:

$$0.275 \rho \sigma^2 u_2{}^3 \pi D_2 b_2 \psi_2 \tan \beta_2 / \eta_m \le 1.2 \rho g Q_e H_e / \eta_e \qquad (5)$$

where ρ is the pumping medium density（kg/m^3）; σ the slip coefficient; u_2 the circumferential speed of impeller outlet（m/s）; D_2 the impeller diameter（m）; b_2 the outlet width of impeller（m）; ψ_2 the blade blockage coefficient of impeller outlet; β_2 the blade outlet angle（°）; z the blade number of impeller; n the speed（r/min）; η_m the mechanical efficiency of pump; g the acceleration of gravity（m/s^2）; Q_e the pump flow at rated condition（m^3/s）; H_e the pump at rated condition of single stage head（m）; η_e the pump efficiency at rated condition, s_u the circumferential blade thickness of calculation points.

Head fomular requirements on parameters

The choice of geometrical parameters of impeller which must meet the rated flow of head, is obtained from the following basic equation of pump:



$$\frac{gH}{\eta_h} = u_2 (\sigma u_2 - \frac{v_{m2}}{\tan \beta_2}) - u_1 v_{u1} \qquad (6)$$

where H the single stage head of pump（m）; η_h the pump hydraulic efficiency; v_{m2} the axial velocity of impeller outlet（m/s）; u_1 the circumferential speed of impeller inlet (m/s); v_{u1} the axial velocity of impeller inlet (m/s).

We achieve that full-head non overload performance of pump and maximum pump shaft power can be controlled in less than 1.2 times of rated shaft power, as long as impeller structural parameters meet the equations（5）、(6）.

Influence of solid particles onparameter choice

Low specific speed submersible pump used in coal mines is different from ordinary clean water pump, because the impeller design must take into account solid paticles affecting the flow of liquid. Known by two-phase flow theory: at impeller inlet, the velocity of solid particles is smaller than the flow rate of water, so solid particles have a ”relative plug” for water. On the contrary at impeller outlet, the velocity of solid particles is greater than the water flow and solid particles have a “relative suction” for water. [3~4]

Therefore, for non-overload design, blade inlet incidence angle $\Delta\beta_1$ should be appropriately increased and blade outlet angle β_2 should be properly decrease. In order to meet the capacity of sewage through flow, the blade outlet width b_2 need to be increased.

EXAMPLE OF DESIGN AND TEST ANALYSIS

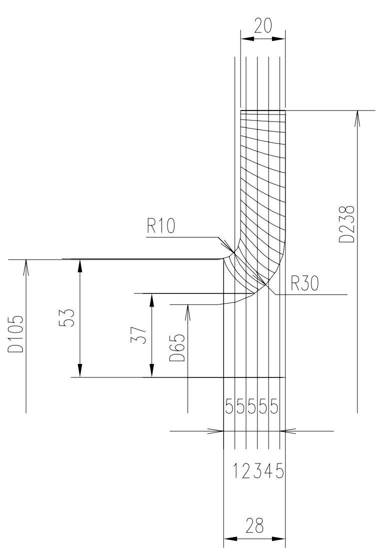
According to the design points of BQS80-180/3-90 low specific speed submersible pump, impeller geometrical parameters are shown below: diameter of impeller inlet D_j =0.105m, blade outlet diameter D_2 =0.238m, blade outlet width b_2 =0.02m， blade outlet angle β_2 =9° ， blade number z =4， blade outlet thickness s_2 =0.005m， estimated mechanical efficiency of pump η_m =70%， left end of geometrical formula (3) is

$$0.275\rho\sigma^2 u_2{}^3 \pi D_2 b_2 \psi_2 \tan\beta_2 / \eta_m = 29484\text{W}$$。

The parameters of rated conditions are: flow: Q_e =80/3600=0.022m^3/s, single stage head: H_e =60m, efficiency of pump: η_e =52.1%[5], the right end of correlation formula is $1.2\rho g Q_e H_e / \eta_e$ =30114W。

Obviously, left of the formula is less than right of it, that is the maximum shaft power is less than the rated shaft power 1.2 times.The parameters into basic equation of pump, give the single stage head H =61.03m. where v_{u1} is the circumferential velocity at impeller blade inlet generated from guide vane outlet.

Hydraulic design of impeller and 3D model after removing the front cover, as shown in Fig..1 and Fig.2.



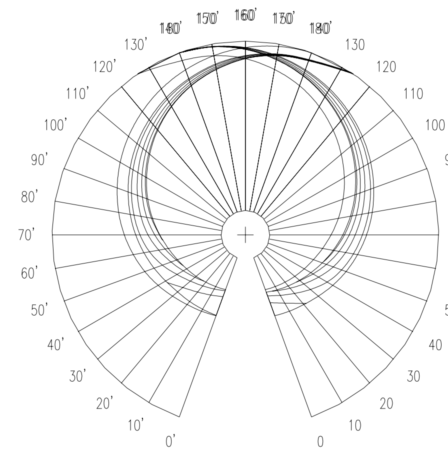


Fig.1 Impeller hydraulic picture

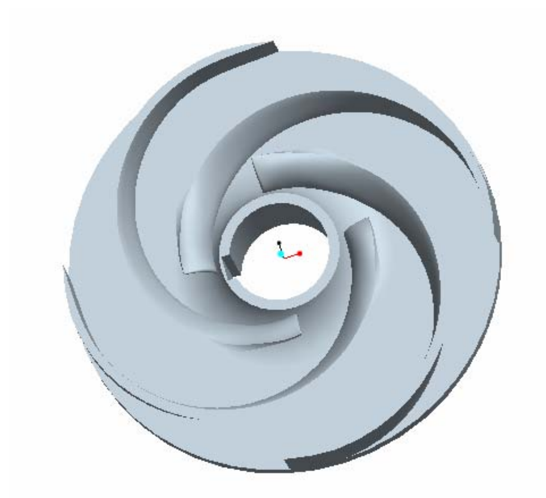


Fig.2 3D model of impeller

The main characteristics of impeller hydraulic design:

1）The blade outlet angle is smaller than for the average pump, which is the key way to achieve full head non-overload, but this will increase the difficulty of manufacturing. Here the choice of a large blade outlet width b_2 =0.02m and reduction at the number of blades z to 4 to improve the efficiency of pump;

2）The number of impeller blades is less than conventional design, which is to enable the plug does not occur because of the sewage sediment. It is approriated to reduce the number of blades, which is beneficial in improving the efficiency of pump;

3）There is the way to meet the requirements of sewage sediment without plugging, so the impeller outlet width is larger than conventional design;

4）In order to make up the lower head which is caused by a decreased outlet angle, it is needed a larger impeller exit diameter than normal. The larger exit diameter is affective to rise the lift, and increase the usage scope of low specific speed submersible pump, but not beneficial for efficiency;

5）In order to have more flat power curve, the blade wrap angle value is 180° which is than normal, and it is a significant measure to achieved non-overload power curve.

NUMERICAL SIMULATION RESULTS AND ANALYSIS

The 3D models of impeller and guide vane as shown in Fig.3. On the single stage impeller and guide vane coupled to CFD numerical simulation [7], obtained the shaft power curve of single stage impeller, as shown in Fig.4. Numerical simulation show that: when theflow is 110m^3/h, the shaft power of single stage impeller can produce maximum value, $P_{2\max}$=24.5kW. The maxium shaft power of pump (3 stage) is 73.5kW, less than the matching motor power 90KW. As the flow further increases, the shaft power curve has downward trend and can be reached full head non-overload. At rated flow, the shaft power of single stage P_{2e} is 22.6kW, $P_{2\max}/P_{2e}$ =24.5kW /22.6kW ＝1.08<1.2， achieved the purpose of reducing maximum shaft power.

The impeller relative velocity vector is shown in Fig.5. The turbulent kinetic energy distribution of impeller and guide vane is shown in Fig.6.

As shown in Fig.5, the velocity field inside the impeller is uniform and smooth, without flow diffusion and significant recirculation zone. In Fig.6, the turbulent kinetic energy distribution in impeller and guide vane is uniform. There is no larger turbulent fluctuation and the loss for turbulent flow is smaller. The above points can explain that the impeller design is in line with requirement.

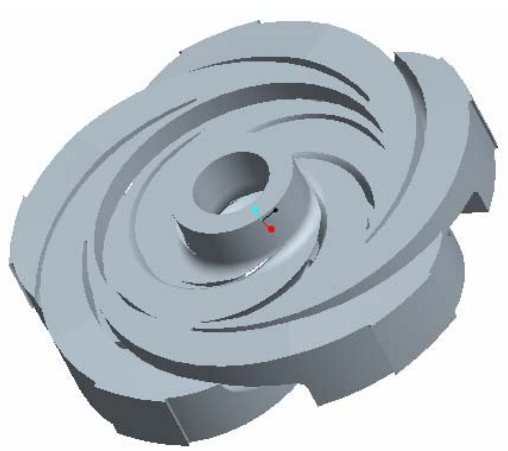


Fig.3 3D model of impeller and diffuser

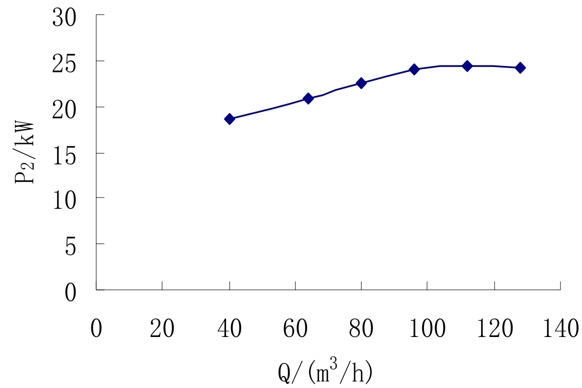


Fig.4 P$_2$—Q curve

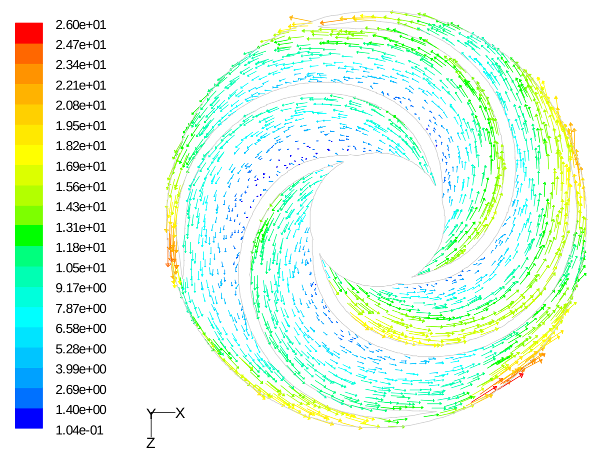


Fig.5 Relative velocity magnitude

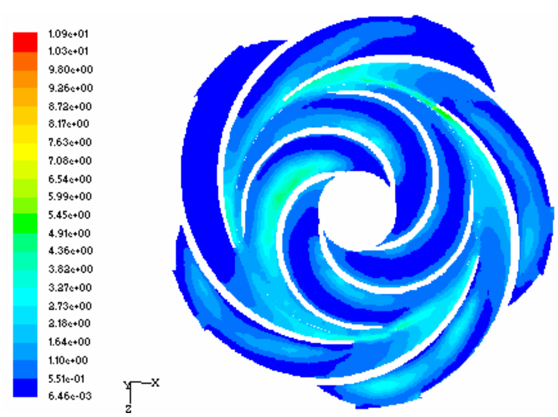


Fig.6 Turbulent kinetic energy distribution

Test results and analysis

The test results of BQS80-180/3-90 low specific speed submersible pump which three stages are shown in Table.1, in which speed is 2950r/min.

From the measurement data, the maximumshaft power of pump is 76.9KW，lower than the motor rated power 90KW, and so the requirement for non-overload power is achieved.

At rated flow Q_e =80m^3/h, the shaft power is about 74.0kW, the ratio of maxium shaft power and designed shaft power is 76.9/74.0=1.04, lower than design target requirement at 1.2 times the bep power. The results of numerical simulation using FLUENT are basically consistent with the exprimental results.

Tab.1 Testing results of performance of pump

Flow/（m^3/h）	Head/ m	Shaft power/ kW	Input power of motor/kW	Efficiency of pump/%
0.00	236.8	42.9	48.1	0.00
32.8	227.3	54.2	60.3	37.4
54.6	217.7	63.9	71.2	50.6
60.4	216.1	65.3	72.6	54.4
69.2	207.0	69.7	77.6	55.9
74.8	200.4	72.3	80.4	56.5
81.7	193.6	74.0	82.3	58.2
85.9	188.1	74.6	83.0	58.9
90.2	181.1	76.9	85.5	57.8
96.1	157.9	76.6	85.4	53.9

CONCLUSIONS

1）The effect of impeller geometrical parameters on shaft power of low specific speed submersible pump, as expressed is the formula for non-overload impeller design:

$$0.275\rho\sigma^2 u_2^3 \pi D_2 b_2 \psi_2 \tan\beta_2 / \eta_m \leq$$

$1.2\rho g Q_e H_e / \eta_e$, was clarified. According to the formula, the design objective to get maximum shaft power of low specific speed submersible pump less than the rated shaft power 1.2 times, is possible.

2）With the decrease of maximum shaft power, motor power can be appropriatly reduced. It is possible to change the previous way of using high power motor to achieve non-overload, which persuits to save energy.

ACKNOWLEDGMENTS

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