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ITJ

ISSN 1812-5638

# INFORMATION TECHNOLOGY JOURNAL

ANSInet
Asian Network for Scientific Information
308 Lasani Town, Sargodha Road, Faisalabad - Pakistan

Information Technology Journal 12 (23): 7621-7625, 2013
ISSN 1812-5638 / DOI: 10.3923/itj.2013.7621.7625


# Design and Experiment of a Small Flowrate Solid-liquid Two-phase Flow Centrifugal Pump

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**Abstract:** The solid-liquid two-phase flow centrifugal pumps are widely used to deliver the solid particles and the liquid media in all kinds of departments. Based on the velocity coefficient method, a low specific-speed solid-liquid two-phase flow centrifugal pump with small flowrate is designed in this paper. The experimental results of external performance show that the head curve of the pump model displays the drop tendency with monotonous characteristics which fully manifests that for most of working points, the designed centrifugal pump model is of good operating reliability.

**Key words:** Small flowrate, centrifugal pump, velocity coefficient, solid-liquid two-phase flow, external performance

## INTRODUCTION

Solid-liquid two-phase flow centrifugal pumps are widely used to deliver solid particles and liquid media in agriculture, pharmacy industry, petrochemical industry, etc. Up to now, a lot of researchers have carried out many investigations (Zhu *et al*., 2008; Shi *et al*., 2010; Wu *et al*., 2001). Compared with the case delivering pure liquid such as water, the solid-liquid two-phase flow is more complex. The complicated two-phase flow makes the design theory and design method of solid-liquid two-phase flow centrifugal pump unmature. The shortages of lower transmission efficiency and severe abrasion are always existent in solid-liquid two-phase flow centrifugal pump. Many professor such as (Cai, 1984; Baoyuan, 1991) and (Xu, 1992; 1995; 1998) put forward new design theories and new design methods of solid-liquid two-phase flow centrifugal pump, but in the process of actual design, the empirical correction is given priority to designing the kind of pumps by far. This method is also able to be regarded as the experience statistics velocity coefficient method based on water pump. On the basis of design theory of pure water pump and abrasion data of solid-liquid two-phase flow centrifugal pump, the geometric parameters of single phase flow pump are appropriate modified to some degree. As such, the main purpose is to reduce the magnitude of absolute velocity and relative velocity, meanwhile improve the through capacity of solid-liquid mixture to reduce the wall abrasion. In this method the main consideration is wear, structure material and geometric parameters, while the efficiency is second. As is well known, most of solid-liquid two-phase flow centrifugal pumps are designed based on this method at present. In this study a small flowrate solid-liquid two-phase flow centrifugal pump is designed by means of this method and the experiment of the external performance is also carried out to analyze it's hydraulic performance.

## HYDRAULIC DESIGN

**Basic parameters:** In this study, the rated parameters of the solid-liquid two-phase centrifugal pump are as following. Flowrate is 6 $m^3\ h^{-1}$, pump head is 8 m (in the case of pure water) and rotational speed is 1450 r $min^{-1}$. Meanwhile, it is required that the pump model is of good operating reliability at all working points. The specific-speed can be calculated according to the following formula:

$$n_s = \frac{3.65n\sqrt{Q}}{H^{0.75}} \qquad (1)$$

The magnitude of the calculated specific-speed is 45 which belongs to low specific-peed small flowrate centrifugal pump.

- **Suction diameter $D_j$:** At the inlet of solid-liquid two-phase pump, the flow velocity of liquid medium should be required to be greater than the sedimentation velocity of solid particles. Thinking of this anti-avitation performance, the flow velocity at

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pump inlet should be 1.0 and 1.8 m $sec^{-1}$ when the suction diameter is less than 250 mm. In this study the designed centrifugal pump is mainly used to transmit micro solid particles in which the maximum diameter is usually less than 0.2 mm and the maximum density is usually less than 2500 kg $m^{-3}$, therefore it has a larger freedom for the selection of entrance velocity. According to our design experience, the magnitude of entrance velocity is choosed as 1 m $sec^{-1}$ for preliminary calculation. The suction diameter of the pump model can be calculated according to the following formula and then is modified by means of our design experience

$$D_j = \sqrt{\frac{4Q}{\pi V_s}} \tag{2}$$

- **Discharge diameter $D_d$:** In order to decrease pump volume and discharge diameter, the discharge diameter is usually required to be less than the suction diameter. In general, the discharge diameter can be preliminary determined according to the following formula:

$$D_d = (0.65 \sim 1.0)D_j \tag{3}$$

**Centrifugal impeller**

- **Inlet diameter of impeller $D_1$:** The equivalent diameter of impeller inlet can be calculated by means of the following formula:

$$D_0 = K_0 \cdot \sqrt[3]{\frac{Q}{n}} \tag{4}$$

where, n is the design rotational speed, r $min^{-1}$. $K_0$ is coefficient which can be choosed by design experience and the following principle. In the case of that the cavitation performance is especially considered, $K_0$ = 3.5~3.8. In the case of that the efficiency is especially considered, $K_0$ = 4.2~4.5. When considering both of them, $K_0$ = 3.8~4.2.

In this study the rotational speed of the pump model is relatively low, therefore the cavitation performance can be easily meet. As such, the pump model emphasizes the efficiency aim, namely that $K_0$ = 4.2~4.5. According to design experience, the final hub diameter is designed as 22 mm ($d_h$). Therefore the inlet diameter of impeller can be preliminary determined by means of the following formula:

$$D_1 = \sqrt{D_0^2 + d_h^2} \tag{5}$$

- **Impeller diameter $D_2$:** In the case of that the rotational speed is less than 3000 r $min^{-1}$ and the specific-speed is greater than 30, the following formula is widely used to preliminary calculate the impeller diameter of low specific-speed centrifugal pump. The formula is based on velocity coefficient method

$$D_2 = K_{D2} \cdot \frac{\sqrt{2gH}}{n} = 19.2\left(\frac{n_s}{100}\right)^{1/6} \frac{\sqrt{2gH}}{n} \tag{6}$$

It was necessary to state that the formula is in view of the pure water pump. Considering the non-hump design requirements, the calculated results would be corrected by the appropriate changes according to design experience.

- **Width of impeller inlet $b_1$:** As is well known, the geometric parameters at impeller inlet play prominent role in cavitation performance. The width of impeller inlet can be preliminary determined by the following formula:

$$b_1 = (0.1175 \ln n_s - 0.3727)D_2 \tag{7}$$

- **Width of impeller outlet $b_2$:** From the viewpoint of anti-wear, the velocity at impeller outlet is as small as possible, but this will reduce the pump head. In order to meet the pump head and wear, the width at impeller outlet should be larger which would make sure that the solid particles in liquid phase can smoothly pass. On the one hand, the larger width can reduce the relative velocity at impeller outlet and decrease wall wear. On the other hand, it weakens the jet-wake effect at impeller outlet and makes the velocity distribution more uniform. At the same time, the width at impeller outlet can not be increased too much, otherwise it will cause the hump phenomenon which can result in that the operation is not stable. The width of impeller outlet can be preliminary determined according to the following formula:

$$b_2 = 2.413\left(\frac{n_s}{100}\right)^{0.977} \frac{\sqrt{2gH}}{n} \tag{8}$$

- **Blade angle ($\beta_1$, $\beta_2$):** As is well known, the blade outlet angle has an important effect on pump performance. According to our design experience, the larger blade inlet angle $\beta_1$ and the smaller blade outlet angle $\beta_2$ can efficiently weaken the wall wear. In the case of that the solid particles is relatively larger, the blade outlet angle can be choosed as the larger value, namely $\beta_2$ = 20 and 30°. Otherwise $\beta_2$ = 15 and 25°. In order to weaken jet-wake flow structure, the blade outlet angle should be smaller for solid-liquid two-phase flow centrifugal pump. As such, the relatively higher pump head and operating stability can be obtained in the case of less blade number. At the same time, reducing blade outlet angle $\beta_2$ can make the flow velocity in volute decrease, this will reduces hydraulic loss in volute and weakens wall wear

- **Blade number Z:** Less blade number can reduce crowding phenomenon at impeller inlet and outlet, meanwhile it is also favor of overcoming the hump phenomenon. So in solid-liquid two-phase flow centrifugal pump, blade number is usually 4~7
- **Wrapping angle of blade ø:** Too large wrapping angle leads to longer passageway which will increase hydraulic loss. Otherwise smaller wrapping angle shortens the efficient length of passageway. In order to ensure that the pump has no overload characteristic within all operating points, the wrapping angle of blade should be choosed as larger value. According to experience, ø = 120 and 220°. Larger wrapping angle of blade and smaller blade angle usually can weaken the wall wear in the solid-liquid two-phase flow centrifugal pump. At the same time, we also notice that too large wrapping angle easily causes stalling, therefore the wrapping angle should be consider overall

**Volute chamber:** Spiral volute is widely choosed in the process of pump design because of meeting the law of conservation of moment, therefore it has the advantage of good hydraulic performance and wider high efficiency. In this study, the volute section is semi-circle section which is helpful for conveying liquid containing individual larger solid particles.

- **Basic diameter $D_3$:** In order to ensure that solid particles can smoothly pass, the tongue clearance in solid-liquid two-phase flow centrifugal pump is usually much larger than the value in common centrifugal pump, but this will result in that the pump efficiency decreases. So on the premise of delivering successfully solid particles, the tongue clearance should be as small as possible. In general, the basic diameter of volute can be preliminary determined by the following formula

$$D3 = D2+2d0+4\sim4\sim10 \text{ mm} \quad (9)$$

where, $d_0$ denotes the maximum diameter of solid particles.

- **Volute width $b_3$:** The inlet width of volute can be preliminary calculated by the following formula

$$b_3 = b_2 + 2s + 2 \sim 6 \text{ mm} \quad (10)$$

where, s is the thickness of hub and shroud of impeller.

- **Volute tongue angle $\phi_0$:** The magnitude of volute tongue angle should ensure the smooth link between spiral part and diffuser part of volute. Moreover, it also can reduce the radial size
- **Volute shape:** In order to facilitate manufacture, Archimedes spiral is choosed to design the voute shape in this study

$$\rho = 82.5 + \frac{15\theta}{360} \quad (11)$$

whereñis the radius of spiral volute, θ denotes the angle starting from the volute tongue.

- **Throat area A:** In low specific-speed centrifugal pump, the flow velocity at volute throat is relatively high, therefore the larger throat area is usually choosed to reduce flow velocity and hydraulic loss. In this study, the velocity at throat should be

$$u_{th} = (0.62 - 0.0043n_s)\sqrt{2gH} \quad (12)$$

According to:

$$A = \frac{Q}{u_{th}} \quad (13)$$

The final throat area will be determined through the modification.

- **Diffuser:** In order to reduce hydraulic loss, the diffuse angle of diffuser is about 8°~12°. In this pump the angle is 8°

**Design results:** In this study the designed centrifugal pump can be seen in Fig. 1.

The main geometric parameters of the designed small flowrate low specific-speed solid-liquid two-phase flow centrifugal pump are shown in Table 1.

Table 1: Main geometric parameters of pump model

| Suction diameter $D_j$ $mm^{-1}$ | Discharge diameter $D_d$ $mm^{-1}$ | Impeller inlet diameter $D_1$ $mm^{-1}$ | Impeller diameter $D_2$ $mm^{-1}$ | Impeller inlet width $b_1$ $mm^{-1}$ | Impeller outlet width $b_2$ $mm^{-1}$ | Blade inlet angle $\beta_1$/(°) |
|---|---|---|---|---|---|---|
| 50 | 40 | 48 | 160 | 20 | 10 | 25 |
| Blade outlet angle $\beta_2$/(°) | No. of blades Z | Blade wrapping angle $\phi$/(°) | Volute basic diameter $D_3$ $mm^{-1}$ | Volute inlet diameter $b_3$ $mm^{-1}$ | Volute angle $\phi_0$/(°) | Throat area A $mm^{-2}$ |
| 25 | 5 | 137 | 165 | 15 | 13 | 176 |

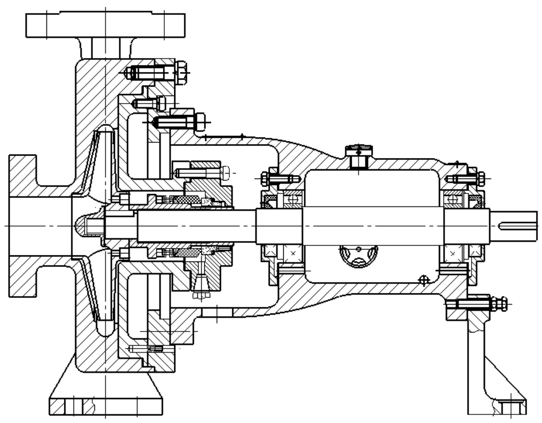

Fig. 1: Sectional view of centrifugal pump model

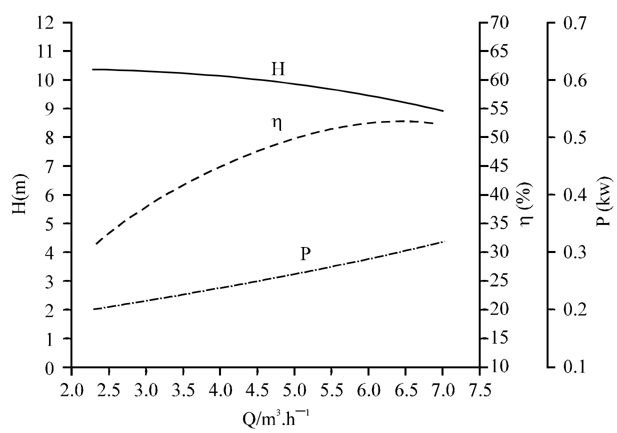


Fig. 2: Experimental results of external performance

## EXPERIMENT OF EXTERNAL PERFORMANCE

The experiment of the external performance is carried out in the case of pure water (Zhang *et al*., 2013). The experimental results are shown in Fig. 2.

It can be seen from Fig. 2 that the head curve of the centrifugal pump shows the drop tendency with monotonous characteristics in the range of operating conditions which fully manifests that although the designed centrifugal pump is a small flowrate low specific-speed pump, the pump model is of good operating reliability. At the rated operating point, the pump head and efficiency are respectively 9.42 m and 53%, it reaches the design aim.

## CONCLUSIONS

The design method and design process of solid-liquid two-phase flow centrifugal pump are stated in detail in this study. Based on the velocity coefficient method of experience statistics, a small flowrate solid-liquid two-phase flow centrifugal pump is designed. The experimental results of the external performance show that the designed centrifugal pump has excellent operating reliability and has completely reached the design requirements.

## ACKNOWLEDGMENTS

The authors thank the support under Zhejiang Provincial Natural Science Foundation of China (No. LY12E06002), Zhejiang Provincial Education Department (No. pd2013448) and Academic Foundation of Quzhou University (No. XNZQN201102).

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