Numerical analysis of the double suction centrifugal pump with different tongue shape

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Abstract. The paper deals with the influence of the tongue shape on the flow characteristics at the near-tongue region of the volute in the double suction centrifugal pump with long-tongue, middle-tongue and short-tongue respectively. For this study, the computational fluid domains including suction chamber, impeller and volute was simulated by means of the commercial CFD software that solved the Navier-Stokes equations for three-dimensional steady flow. The results show that the tongue shape has a significant impact on the pump head and efficiency in a range of 0.4Qd-1.4Qd flow rate. Under small flow conditions, the head and efficiency of the short-tongue model are higher than that of the middle-tongue model and long-tongue model. Under large flow conditions, the head and efficiency curves of the three models are relatively close. In addition, the tongue shape has a great influence on the pressure and the streamline distribution in the near-tongue region. Therefore, effectively changing the tongue shape can improve the internal flow pattern of the double suction centrifugal pump and reduce the pressure pulsations. The tongue shape also has a great influence on the turbulent kinetic energy distribution. By changing the tongue shape the numerical value of the turbulent kinetic energy can be effectively reduced, thus the generation of the turbulent vortex and the energy loss of the pump will be reduced. It is shown that the tongue shape has a great influence on the performance of the double suction centrifugal pump. This study has certain reference value to improve the pressure pulsation of the double suction centrifugal pump.

1. Introduction
Pressure pulsation in single-suction centrifugal pump has become a major research topic in the last decades. In order to resolve this problem, many scholars have done a lot of research work. A number of investigators have considered that geometry modifications have great influence on the pressure pulsation of the single-suction pumps, and the characteristics of the fluid-dynamic interaction are very dependent on the geometries tongue [1]. Studies have shown that a proper gap between the volute tongue and the outlet of the impeller can reduce the noise and vibration, and the efficiency increased slightly in the single-suction pump. In addition, it is easy to cause noise, vibration and cavitation at the volute tongue due to the fluid flow obstruction with a too small gap, and there is backflow in the near tongue regions causing energy lost and efficiency decreased with a too large gap. Zhu Lei et al. [2] presented a study on the flow characteristics in a single suction centrifugal pump with long-tongue, middle-tongue and short-tongue respectively. The author pointed out that the short tongue model can
raise the pump head and widen high efficiency zone compared with the long and middle tongue model. Guo Pengcheng [3] carried out a numerical simulation of a single stage and single suction centrifugal pumps. The author divided the tongue into 3 types (long-tongue, middle-tongue and short-tongue) according to the different stagger angle of the volute tongue, and it is pointed out that the change of the area ratio of the three types is well-distributed, and indicated that the performance curve of the centrifugal pump is mainly caused by the shape of tongues. However, there are not so much effort has been devoted to similar studies in double-suction centrifugal pumps.

Double-suction centrifugal pumps are widely used in various fields since the flow rate of the pump is twice as much as the single suction pump with the same impeller diameter, and the axial force is theoretically balanced [4]. However, due to the high energies involved in these pumps, it is tend to suffer more from pressure pulsations than single-suction pumps. In addition, the pressure fluctuations will cause a lot of vibration and noise during the operation of the pump. Many studies [5, 6] have shown that the pressure fluctuation caused by the interference between rotating impeller and stationary parts is an important factor affecting the stability of centrifugal pumps.

In this paper, we put forward that a proper change of tongue shape can improve the performance of double suction centrifugal pump. In order to verify this idea, a three-dimensional numerical simulation of a double suction centrifugal pump was carried out.

2. Pump geometry
The geometry model is built by UG software, shown in figure 1.

![Suction chamber](image.png)

**Figure 1. CFD computational domain of the pump**

Table 1 provides a list of the main characteristics of the pump. The main structural parameters of the spiral case are the base circle, which is cut at the head of the tongue, represented by \(D_3\) [7].

| Parameters                        | Value  |
|-----------------------------------|--------|
| Impeller inlet diameter, \(D_1\) (mm) | 403.7  |
| Impeller outlet diameter, \(D_2\) (mm) | 640    |
| Impeller blade number, \(Z\)      | 6      |
| Impeller outlet width, \(b_2\) (mm) | 143.94 |
| Design flow rate, \(Q_d\) (m³/h)   | 4000   |
| Head, \(H\) (m)                   | 40     |
| Rated speed, \(n\) (r/min)        | 980    |
| Specific speed, \(n_s\)           | 168    |

In this paper, we divided the volute tongue into 3 types (long-tongue, middle-tongue and short-tongue) according to the size of the base circle diameter, and kept the other parameters exactly the same. The diameter of base circle is 858mm, 685mm and 660mm respectively. The shape of the volute tongue is shown in figure2.
3. Numerical model
The governing equations of the steady flow of incompressible fluid are continuity equations and Navier-Stokes equations [8], shown as the equation (1) and equation (2).

Continuity equations:
\[
\frac{\partial u_i}{\partial x_j} = 0
\]  

(1)

Steady Navier-Stokes equations in three-dimensional:
\[
\frac{\partial u_i}{\partial x_i} + u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\mu}{\rho} \frac{\partial^2 u_i}{\partial x_j \partial x_j} + f_i
\]

(2)

A sensitivity analysis of the numerical model was performed in order to impose appropriate parameters regarding grid size, time step size and turbulence model. A brief summary of this work is contained below.

3.1. Grid generation
The pump is split into three component parts for modelling. The component parts including: (a) suction chamber, (b) impeller, (c) spiral case. The grid is the basis for converting the model into the computational domain of flow field and performing CFD calculation. At present, mesh generation mainly consists of two classes: structured mesh and unstructured mesh. For this study, a structured grid is used for the runner and an unstructured grid is used for the volute and suction chamber.

Due to the size and complexity of the pump care was taken regarding the distribution of grid elements in the model. Considering the configuration of the computer and the computation time, a detailed grid independence check was conducted, with the influence of the volute on the flow in the impeller grid being factored into the check [9]. This concluded that the total mesh number of the pump was 5,922,700 (the impeller consisted of 2,942,400 elements, the suction chamber consisted of 1,591,100 elements and the spiral case with short-tongue consisted of 1,389,200 elements), which was sufficient to reliably model the flow characteristics. Care was taken to concentrate grid in the boundary layer and the cutwater region of the volute. Figure 3 provides an indication of the component of the model mesh.
3.2. Pre-processing
The flow field in the centrifugal pump is numerically calculated by means of ANSYS CFX 16.0 commercial software, and the SST turbulent model was used to solve the Reynolds averaged equations. Although a number of boundary conditions were examined, the parametric study was conducted using a mass flow at inlet and static pressure at outlet as this set of boundary conditions had been found to be more stable and converge faster than other combinations without a significant loss in accuracy. Assuming that the walls are adiabatic and satisfy no slip conditions, the near wall region is treated by standard wall functions. The impeller was set in a rotating frame of reference, the suction chamber and spiral case were set in a static frame of reference. The interfaces between rotating and stationary frames were modelled using the rotor/stator interface option; interface between components in the same frame of reference use general grid interface (GGI) option. In the iterative process, the convergence of the calculation results is judged by detecting the residual value [10], with the higher order solution precision, the convergence accuracy is 10-4.

4. Numerical simulation results and discussions
In this section, we will analyse and discuss the result of the numerical simulation. A brief summary of this work shown as follows.

4.1. Pump performance curves analysis
Kept the other parameter settings unchanging, six working conditions (0.4Qd, 0.6Qd, 0.8Qd, 1.0Qd, 1.2Qd, 1.4Qd) of three models of the double suction centrifugal pump were numerically simulated. The pump head is defined as the equation (3), the pump shaft power is defined as the equation (4) and the hydraulic efficiency is described as the equation (5).

\[ H = \frac{P_{out} - P_{in}}{\rho g} \]  
\[ P = M \omega \]  
\[ \eta = \frac{\rho g Q H}{M \omega} \]

In the formula, \( H \) is expressed as the pump head, \( P_{out} \) as outlet pressure, \( P_{in} \) as inlet pressure, \( \rho \) as fluid density, \( P \) as the pump shaft power, \( M \) as torque, \( \omega \) as angular velocity. \( \eta \) as efficiency, \( Q \) as flow rate.

The hydraulic performance of the pump under three models was predicted, as shown in figure 4. From figure 4, it can be seen that the short-tongue model shows an obviously higher hydraulic head and relatively better efficiency at small flow rate compared with the middle-tongue model and long-tongue model. In addition, the head and efficiency curves of the three models are very close in the design condition and large flow rate conditions. The pump shaft power of the long-tongue model and middle-tongue model are larger than the short-tongue model at the small flow rate and the design
conditions, with the increased of the flow rate, the pump shaft power of the short-tongue model is larger than the middle-tongue model and long-tongue model.

(a) Q-H curves at different flow rates

(b) Q-P curves at different flow rates

(c) Q- η curves at different flow rates

**Figure 4.** Hydraulic performance curves
4.2. *Pump flow field analysis*

The internal flow characteristics of the double suction centrifugal pump with different tongue was analysed further. Under 0.4Qd, 1.0Qd and 1.4Qd conditions, the velocity contour around volute tongue of the pump with different tongue on middle layer was obtained, as shown in figure 5. Under the design condition, the static pressure field, streamline and turbulent kinetic energy of the pump with different tongue on middle layer was obtained, as shown in figure 6, figure 7 and figure 8 respectively.

As shown in figure 5, under 0.4Qd and 1.0Qd conditions, long-tongue model and middle-tongue model has an obvious jet zone in the tongue gap. Under the three operating conditions, stagnation point is produced near the volute head of the long-tongue model and middle-tongue model, and the stagnation point does not change with the change of the tongue type, but varies with the change of the working point.
Figure 6. Pressure contour around volute tongue under design condition

As shown in figure 6, there is a local high pressure zone around the tongue head of the middle-tongue model and long-tongue model, which is similar to that of single stage and single suction centrifugal pump [11]. This is due to the water thrown from the rotating impeller lashed against the volute tongue head, and resulting in the stagnation point. The short-tongue model has no lash on the volute tongue head, so there is no stagnation point. What’s more, the stagnation point of the long-tongue model which is compared with that of the middle-tongue has a tendency of outward diffusion. This is due to the lash of the long-tongue model larger than that of the middle-tongue model.

![Figure 6](image)

(a) short-tongue model  (b) middle-tongue model  (c) long-tongue model

Figure 7. Streamline distribution around volute tongue under design condition

As shown in figure 7, the middle-tongue model and long-tongue model are very smooth in the near-tongue region of the volute, but the short-tongue model has an obvious backflow in the near-tongue region.

![Figure 7](image)

(a) short-tongue model  (b) middle-tongue model  (c) long-tongue model

Figure 8. Turbulent kinetic energy contour in the volute under design condition

As shown in figure 8, the short-tongue model has larger turbulent kinetic energy at near tongue regions than that of the middle-tongue model and long-tongue model. It can be concluded that short tongue is easier to produce vortex at near tongue regions.

5. Conclusions

This paper studies the effects of the tongue shape on the performance of double suction centrifugal pump. The major conclusions of this study are summarized as follows:

- The tongue shape can change the head and efficiency of the double suction centrifugal pump, especially in the small flow rate conditions, the short-tongue model show an obviously higher hydraulic head and relatively better efficiency. From this we could conclude that a reasonable design of the tongue shape can improve the head and efficiency of the double suction centrifugal pump.
The tongue shape can change the velocity distribution and pressure distributions, streamline distribution and turbulent kinetic energy distribution of double suction centrifugal pump. Thus it can be concluded that a reasonable design of the tongue shape can improve the internal flow field of the double suction centrifugal pump.

The influence of the tongue shape on the performance of single stage and single suction centrifugal pump is similar to that of the single stage and double suction centrifugal pump in some respects. We have only studied a small part of the static and interference of the double suction centrifugal pump. To further study the problem of the pressure pulsation of double suction centrifugal pump, the unsteady numerical simulation of the model is needed and experiments are needed to verify the accuracy of the numerical simulations.

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References
[1] Guelich JF, Bolleter U. 1992. Pressure pulsations in centrifugal pumps (ASME J Vib Acoust 1992; 114:272-9)
[2] Zhu Lei, Yuan Shouqi, Yuan Jianping, et al. 2011. Numerical simulation for rotor-stator interaction of centrifugal pump with different tongues [J] (Transactions of the CSAE, 2011, 27 (10): 50-55 (in Chinese))
[3] Guo Pengcheng. 2009. Numerical investigation and performance prediction on 3d complex viscous flows in hydromachinery [D] (Xi’an: Xi’an University of Technology (In Chinese with English abstract))
[4] Wang, F. J. 2005. Pump and Pumping Station (China Agricultural Press, Beijing, China)
[5] Fernandez O J M, Gonzalez J, Arguelles D K M, et al. 2011. Decomposition of deterministic unsteadiness in a centrifugal turbomachine nonlinear interactions between the impeller flow and volute for a double suction pump [J] (ASME Journal of Fluids Engineering, 2011, 133 (1):011103. 1－011103.10.)
[6] Yao Z F, Wang F J, Qu L X, et al. 2011. Experimental investigation of time-frequency characteristics of pressure fluctuations in a double-suction centrifugal pump [J] (ASME Journal of Fluids Engineering, 2011, 133(10):101303.1－101303.10)
[7] Guan X F. 1995. Handbook of modern pump technology [M]( Astronautics Press, Beijing, China)
[8] Wang F J. 2004. Computational fluid dynamics analysis-principle and application of CFD software (Beijing: Tsinghua University press)
[9] FERZIGRE J H, PERIC M. 1996. Computational methods for fluid dynamics [J] Physics today, 1996, 509 (3): 80-84.
[10] Wang X Y, Wang C X, Li Y B, et al.2009. Numerical analysis of flow characteristics in centrifugal pump cavity [J] Journal of agricultural machinery, 40(4): 86-90.
[11] Yuan S Q, Heng Y G, Hong F, et al 2013 Influence of two types of cut-water shapes on simulated performance of centrifugal pump [J] Journal of Drainage and Irrigation Machinery Engineering, 2013,31(7):553-557,604 (in Chinese).