Effects of Impeller Trimming on Performance in a Double-suction Centrifugal Pump

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Abstract: In order to research the relationship between impeller trimming and performance in a double-suction centrifugal pump, impellers were trimmed 11mm and 22mm respectively. Numerical methods were applied combined with the original one, and experimental validation was also carried out on original and 11mm model. The performance and difference of total pressure between volute inlet and outlet was calculated, meanwhile, distributions of velocity and turbulence eddy dissipation on volute middle section were analyzed. The results indicate that the head drops with the trimming process and discontinuity appears when trimmed 22mm. The impeller trimmed 11mm is the most efficient under part-load and design conditions, while the original model has the lowest efficiency. Efficiency in trimmed cases decreases dramatically under over-load condition, especially when trimmed 22mm. Due to rotor-stator interaction between impeller and volute receded by the enlarged clearance, the head losses inside the volute decrease with impeller trimming but increase significantly under $1.4Q_d$ when trimmed 22mm. The extremely high shock losses resulting from exaggerated impeller outlet flow angle and dissipation near volute tongue account for it. Moreover, the losses in volute diffuser channel increase due to the vortexes generated by flow separation and backflow. The losses in volute will be improved by impeller trimming within reasonable limits in a double-suction pump. This research provides theoretical foundation for impeller trimming in double-suction centrifugal pumps.

1. Introduction
Impeller trimming is a common method to make a pump suitable for various requirements without redesign. Double-suction centrifugal pumps are wildly applied in many fields, irrigation, water supply and et. al. Compared with single-suction centrifugal pumps, the internal flow inside a double-suction centrifugal pump appears more complex due to the back-to back blades [1]. Impeller trimming increases the clearance between impeller and volute to recede rotor-stator interaction, and influences pump performance characteristics in different degree [2-3]. But the outlet flow angle will increase according to the velocity triangle. In single-suction centrifugal pumps, pump performance will deteriorate rapidly, if the trimming size is out of reasonable limits [4]. Impeller trimming has positive effect on noise and cavitation in single-suction pumps [5]. Different trimming methods also influence the performance and pressure fluctuation [6-7].

The trimming processes mainly depend on similarity law and empirical equation, which work well in single-suction. traditional theory is not completely suitable for trimming a double-suction impeller due to the special structure. Liu [8] proved the applicability of turbulence models in numerical in
Double-suction centrifugal pumps. The impeller arrangement influences the internal flow and pressure fluctuation in volute [9]. Different trimming methods were proved that the performance characteristics variate with the shape of modification at trailing edge in double-suction impellers [10]. However, available researches pay little attention to the differences of energy losses inside the double-suction centrifugal pumps, which have been profoundly studied in single-suction pumps [11-15].

In this study, a double-suction centrifugal pump (Model 250GS40) from Shandong Shuanglun Co. Ltd was investigated. This pump was trimmed to meet user’s demand and found a peculiar change in efficiency. The impeller was trimmed 11mm and 22mm in sequence. Numerical methods were adopted to reveal the flow losses inside the volute by calculating the total pressure difference between volute inlet and outlet. The distribution of turbulence eddy dissipation and its relationship between velocity were also analyzed. Moreover, experimental validation was taken out to verify the accuracy of numerical simulation.

2. Numerical Simulation

2.1 Model and Calculation Grids

The research was carried out on the double-suction centrifugal pump with the specific speed of \( \text{ns}=89.5 \) and the basic parameters are given in Tab.1. As shown in Figure 1(a), computational fluid domains were composed of a suction chamber, a shrouded double-suction impeller without stagger and a volute casing. As the pump is disassembled horizontally, there is no clearance between impeller outlet and volute inlet. A discharge pipe was added to the volute outflow, not shown in the picture, to decrease the effect of backflow in this region in simulation. Impellers were trimmed for 11mm and 22mm respectively, shown in Tab.2, and impeller outlet area remained unchanged in these models. To elaborate numerical results, a cartesian coordinate system was established for analysis, exhibited in Figure 1(b), where the rotating axis was z-axis, pump symmetry plane is x-y plane, and \( \theta \) is the angle of volute spiral, beginning at \( \theta=-71^\circ \).

ICEM CFD was applied to generate structure meshes for all components, shown in Figure 2, and refined grids were utilized near solid walls to obtain accurate data. The influence of elements number on the original model is shown in Figure 3, based on pump head and hydraulic efficiency under nominal condition. Obviously, the head and hydraulic efficiency are almost constant when cells number reaches \( 4.96 \times 106 \). Finally, the grids with approximately five million elements for each scheme were adopted for numerical simulation.

| Table 1. Main parameters of model pump | Value |
|---------------------------------------|-------|
| Design flowrate, \( Qd \) (m3/h)      | 500   |
| Rotation speed, \( n \) (r/min)       | 1480  |
| Design head, \( Hd \) (m)             | 40    |
| Pump inlet diameter, \( Ds \) (mm)    | 250   |
| Pump outlet diameter, \( Dd \) (mm)   | 200   |
| Impeller inlet diameter, \( D1 \) (mm)| 192   |
| Original impeller outlet diameter, \( D2 \) (mm) | 365 |
| Original impeller outlet width, \( b2 \) (mm) | 46  |
| Blades number, \( Z \)                | 6     |
| Volute inlet diameter, \( D3 \) (mm)  | 365   |
| Volute inlet width, \( b3 \) (mm)    | 100   |
Table 2. Impeller trimming schemes

| Scheme  | $\Delta D = D_2 - D_2'$ (mm) | $D_2'$ (mm) |
|---------|-----------------------------|-------------|
| Original| 0                           | 365         |
| Scheme 1| 11                          | 354         |
| Scheme 2| 22                          | 343         |

Figure 1. Flow field and coordinate system

(a) Computational domains  
(b) Cartesian coordinate of flow domain

Figure 2. Structure grids

(a) Suction  
(b) Impeller  
(c) Volute

Figure 3. Grid sensitivity based on head and hydraulic efficiency
2.2 Computational Settings

Commercial software ANSYS CFX 18.0 was employed to analyze the external characteristic and inner flow under steady condition, and SST $k-\omega$ turbulence model was adopted to solve RANS equation. Settings of boundary conditions are given in Tab.3. Moreover, inlet total pressure was assigned 1 atm, and wall roughness was 0.025mm. The interfaces of suction to impeller and impeller to volute were set as Frozen Rotor, while static interfaces were default. For each case, the convergence criterion was 10^{-4}, and the maximum iteration number was 500. All performance parameters were represented by the arithmetic average value of the last 50 steps to reduce error.

Table 3. Boundary conditions

| Type            | Location            | Option            |
|-----------------|---------------------|-------------------|
| Inlet, Suction inlet | Total Pressure      |                   |
| Outlet          | Discharge pipe outlet| Mass Flow Rate    |
| Wall            | Entire physical walls| No Slip Wall      |

Numerical simulations were proceeded under various operating conditions ranging from $0.4Qd$ to $1.4Qd$. As the volumetric and mechanical losses were not considered in simulation, the numerical results of pump efficiency($\eta$), volumetric efficiency($\eta_V$) and mechanical efficiency($\eta_m$) are estimated by equation (1)-(3)[17], where $\eta_h$ is hydraulic efficiency.

\[
\eta = \eta_h\eta_v\eta_m
\]
\[
\eta_v = \frac{1}{1 + 0.68n_r^{-2/3}} 
\]
\[
\eta_m = 1 - 0.07\left(\frac{n_r}{100}\right)^{-2/6}
\]

To evaluate the power losses inside the volute, the total pressure difference between volute inlet and outlet was calculated by equation (4).

\[
\Delta H_{volute} = \frac{P_{total-volute-in} - P_{total-volute-out}}{\rho g}
\]

2.3 Experimental Setups

The reasonability of numerical simulation is verified by comparing experimental and numerical external characteristics. In the present study, the experiment was conducted on the open test rig in Shandong Shuanglun Co. Ltd, and Figure 5 shows the schematic diagram of the test rig and test pump. In the experiment, water was indrawn from the tank via the test pump and then return to the tank. An electromagnetic flowmeter was used here to measure the flowrate. Uncertainty of all instruments were less than 0.5% and experiment procedures were in accordance with GB/T 3612-2016 (equal to ISO9906:2012). The experiments were operated on the original model and Scheme 1. The electric power was measured and converted into shaft power by the test system itself.
3. Results and Discussion

3.1 Analysis of Performance

Figure 6 presents the comparison between experimental and computational results. Evidently, calculation performance curves are accord with experimental curves well for both models when flowrate ranges from 0.6\(Q_d\) to 1.4\(Q_d\). The largest deviation of head is 3.26% in original model under 1.2\(Q_d\) condition, and 3.49% in Scheme 1 under 0.6\(Q_d\) condition. Meanwhile, the maximal difference in efficiency appears under 1.4\(Q_d\) condition, 3.37% and 6.51% individually. Therefore, the numerical results are accurate and applicable for detailed analysis.

As illustrated in Figure 6(a), the pump head drops with the decreasing of \(D_2\). The head curves are regular when \(\Delta D\) is less than 11mm, while discontinuity appears under part-load condition when the impeller is trimmed for 22mm. It is possible that there will be a hump if the trimming size continues to increase. However, efficiency variates in an extremely different trend, shown in Figure 6(b). Scheme 1 has the highest efficiency under part-load and design conditions, but it drops dramatically under high flowrate conditions. The same situation appears in Scheme 2, this model turns to be the worst one under 1.4\(Q_d\) condition. The efficiency of original model is the lowest until the flowrate reaches \(Q_d\), but there is no huge gap compared with others. Nevertheless, the efficiency gets a lower descent rate under over-load condition if the impeller has not been trimmed, which means this model is more efficient when over-load and has a wider high efficiency region than others.

Figure 7 shows the head losses in volute. The \(\Delta H\) decreases with the increasing of trimming size when the flowrate is lower than 1.4\(Q_d\). Apparently, a bigger clearance will promise less energy losses.
within this range, as the rotor-stator interaction between impeller and volute recedes, and all models achieve the minimum loss at the design point. In original case and Scheme 1, the head losses change mildly under over-load condition. On the contrary, the ΔH between 1.2Qd and 1.4Qd increases significantly in Scheme 2, resulting in more power wastes than other two pumps. Moreover, the remarkable rise of losses may account for the lowest efficiency and head. The trimming process decrease the rotor-stator interaction between impeller and volute and increases impeller outlet flow angle, so the flow is more uniform when enter the volute under part-load condition, accounting for the higher efficiency and low losses under part-load and design conditions. But the sharp efficiency decline under over-load condition needs further investigation.

3.2 Analysis of inner Flow and Loss Distributions

To determine where high energy losses occur and the relation with flow pattern, contours of turbulence eddy dissipation, which is already available in the post processing, and velocity distribution under different operating condition inside the volute studied.

Figure 7 depicts the turbulence eddy dissipation and velocity distribution on the middle section (z=0) of volute under 0.6Qd condition. The overall turbulence eddy dissipation falls with the increase of ΔD. High turbulence eddy dissipation can be observed near volute inlet and the highest value appears in the region close to the tongue in each case. This situation may contribute to the lower head loss in volute under part-load conditions in trimmed cases. The highest value of turbulence kinetic decreases with trimming size slightly. Comparing Figure 7(a) and (b), high turbulence eddy dissipation and high velocity appear in the same place, and the proportion changes in the similar trend. Additionally, the turbulence eddy dissipation, along with velocity, decreases progressively from volute tongue to the outlet in volute diffuser. Backflow can be found at outlet in original model, and this phenomenon disappears in Scheme 1 and 2, making flow pattern more uniform in Scheme 1 and 2. The impeller outlet flow angle increases due to the decreasing circular velocity decreases and invariable meridional velocity. This angle becomes more reasonable in trimmed impellers that fluid enters the volute more smoothly.
Compared with those under 0.6\( Q_d \) condition, distinct difference can be found in Figure 8(a) in three schemes under design condition. High energy losses region near the tongue shrinks in present graphs. Turbulence eddy dissipation in Scheme 2 are much lower than other two schemes and more uniform, while the highest in original model, which is the reason that Scheme 2 has the lowest head losses and the original efficiency is the lowest under design condition. Combine the velocity distribution shown in Figure 8(b), backflows are found at outlet in original model and Scheme 1, but the backflow does not waste too much energy in the two cases due to the low velocity.

Figure 9(a) illustrates turbulence eddy dissipation distribution under 1.4\( Q_d \) condition. It is obviously that the region with high turbulence eddy dissipation near tongue has the highest value under conditions presented for each scheme, and this region spreads with impeller trimming, especially in Scheme 2. Although high dissipation region is small in original case, the value is much higher at the beginning of volute spiral, marked on the graph. This may explain why the efficiency in trimmed cases drops dramatically under over-load conditions.
The dissipation in diffuser may be completely different compared with the previous. Turbulence eddy dissipation in the downstream region of the tongue is the highest in all three schemes under presented conditions as the turbulence eddy dissipation is the lowest in previous cases. Its distribution is nonuniform and changes slightly with impeller trimming. The relatively higher head loss of Scheme 2 under 1.4Qd condition may account for the large area of turbulence eddy dissipation. When refer to the Figure 9(b), there exists severe backflow and vortexes in high turbulence eddy dissipation regions. Impeller trimming reduce the backflow at outlet, shown in Scheme 1, but recurs in Scheme 2, even worse than original. Serious backflows and vortexes occur in the upstream and downstream region of tongue and volute spiral, detailed graphs shown in Figure 10 with the clearance between impeller and volute. The backflows in volute spiral and tongue upstream region are mainly caused by rotor-stator interaction between impeller and volute, and the interaction is more powerful in original case owing to the tiny clearance. In Scheme 1, the extended clearance weakens the interaction. In Scheme 2, due to the increased impeller outflow angle, the impeller force fluid to impact directly on tongue, and increases shock losses near tongue despite of the enlarged clearance. There still exists vortexes in downstream region of the tongue. In original case, the backflow at outlet blocks the passage partly on the side of tongue, thus, the blocked flow in the tongue downstream region transforms into vortex. The vortex and backflow result in high energy losses in the diffuser channel. In Scheme 1, the backflow and vortex almost disappear, therefore the flow in diffuser channel is uniform compared with others. In Scheme 2, as the plugging is relieved relatively, the vortexes are generated by flow separation.
4. Conclusions
In this article, numerical methods are applied to analyze the performance and flow losses in the double-suction centrifugal pump with three different impellers, the original one and trimmed 11mm and 22mm respectively. Experiments are also conducted to verify the accuracy of numerical results. Comparing turbulence eddy dissipation and velocity distribution under three typical conditions in three schemes, impeller trimming has noticeable influence on performance and energy losses. The following conclusions have been obtained:

(1) The pump head drops when the trimming size increases, and discontinuity appears in Scheme 2,
which means there will be a hump if the trimming size is out of the reasonable range.

(2) Efficiency is improved under part-load and design conditions, and Scheme 1 is the most efficient one. However, efficiency drops dramatically in trimmed cases under over-load condition, thus, the original impeller has a wider high efficiency range. Impeller trimming is no suitable for double-suction pumps that need to operate under over-load condition frequently.

(3) The head losses and turbulence eddy dissipation in volute decrease with trimming size. Exceptionally, head loss increases significantly under 1.4Qd condition in Scheme 2. The extremely high turbulence eddy dissipation near the tongue and vortexes in volute diffuser may explain this phenomenon. The trimmed impeller decreases the rotor-stator interaction by enlarging the clearance, but also changes the impeller outlet flow angle and the shock losses near tongue increases under over-load condition.

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Nomenclature

\[ b \] Width [mm]  \[ D \] Diameter [mm]

\[ g \] Gravitational acceleration [m/s^2]  \[ H \] Head [m]

\[ n \] Rotation speed [r/min]  \[ n_s \] Specific speed,  \[ 3.65n\sqrt{Q/H^{3/4}} \]

\[ p \] Pressure [Pa]  \[ Q \] Flowrate [m^3/h]

\[ Z \] Blades number  \[ \eta \] Efficiency

\[ \rho \] Fluid density [kg/m^3]  \[ \Delta D \] Impeller trimming scale

\[ \Delta H \] Head loss [m]

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