Effect of lean mode of blade trailing edge on hydraulic performance for double-suction centrifugal pump

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Abstract. For three-dimensional inverse design of water pump, both blade-loading distribution and stacking condition of high-pressure side are important factors affecting the quality of design. For example, inclined trailing edge of blade can suppress the pressure fluctuation and reduce the sound pressure level and sound power of noise for single-suction centrifugal pump. And the lean mode of blade trailing edge can influence hydraulic performance of pump-turbine in pump mode. However, there is no clear view on the effect of lean mode of blade trailing edge on hydraulic performance for double-suction centrifugal pump. Therefore, three different linear lean modes, namely positive lean (+20°), zero lean (0°) and negative lean (-20°), are adopted to simulate the transient flow field respectively for a middle specific speed double-suction centrifugal pump. Concretely, results show that different lean modes of blade trailing edge have little effect on the external characteristics of the pump, and all three schemes can meet the requirements of basic performance. And in operating range of $0.6Q_0~1.2Q_0$, positive blade lean can suppress the jet-wake structure better. In the vicinity of rated condition, both positive blade lean and negative blade lean can suppress the pressure fluctuation to a certain degree. But with the changes of flow rate, the effect of positive blade lean on pressure fluctuation is unstable. In addition, when pumping sediment-laden water, both positive blade lean and zero blade lean are not conducive to improving the anti-erosion performance of the blade, and especially for the positive blade lean. In summary, in order to improve the integrated performance of low and middle specific speed double-suction centrifugal pump, under the condition of guaranteeing good blade-loading distributions firstly, negative lean mode of blade trailing edge is suggested.

1. Introduction

For three-dimensional inverse design of fluid machinery, both blade-loading distribution and stacking condition of high-pressure side are significant factors affecting the quality of design[1]. Specifically, stacking condition of high-pressure side is the blade lean mode of high-pressure edge. According to previous researches, for many working machines such as fans, pumps and compressors, lean trailing edge of blade has a certain influence on its performance. For instance, Nikkhoo et al.[2] have conducted experimental studies to prove that blade lean can obviously change the vortex pattern of flow channel for squirrel cage fan and improve its efficiency. Denton et al.[3] and Gabriele et al.[4] have tried numerical simulation to prove that blade lean can reduce the endwall losses, postpone
corner stalling and improve cascade characteristics for compressor. Yang et al.[5] and Zhu et al.[6] have compared the flow structure to prove that lean mode of blade trailing edge can remarkably influence the efficiency and stability of pump-turbine. Moreover, jet-wake structure can be effectively suppressed for axial-flow pump with inclined guide vane[7]. Secondary flow near the pressure surface can be markedly suppressed by the blade lean for centrifugal pump with semi-open impeller[8]. And inclined trailing edge of blade can suppress the pressure fluctuation and reduce the sound pressure level and sound power of single-suction centrifugal pump, which is useful for the vibration and noise reduction design[9-11].

In engineering applications, double-suction centrifugal pumps are widely used in the situations of long distance water diversion projects and high lift irrigation. However, there is no clear view on the effect of lean mode of blade trailing edge on hydraulic performance for double-suction centrifugal pump. Therefore, as an important factor of double-suction centrifugal impeller design[12], it is necessary to be analyzed for this issue. Concretely, in this paper, three different linear lean modes are adopted to simulate the transient flow field respectively for a middle specific speed double-suction centrifugal pump with staggered and closed impellers, and the hydraulic performance has been compared in terms of external characteristics, flow pattern, pressure fluctuation and erosion performance, which aims to reveal the quantitative effects that are conducive to improving the design quality of double-suction centrifugal pump.

2. Lean mode of blade trailing edge

In accordance with the related definition of inverse design method, positive blade lean is the scheme that the intersection point of hub and blade precedes that of shroud and blade in the rotational direction, while negative blade lean is the scheme that the intersection point of shroud and blade precedes that of hub and blade in the rotational direction. Concretely, meridian-surface streamlines are adjusted on the basis of zero blade lean (prototype impeller) to form the mode of positive lean and negative lean, while the lean angles are +20° and -20° respectively. Moreover, in order to provide a brief method for description and analysis, three modes, namely positive lean, negative lean and zero lean, are defined as X-shaped scheme, Π-shaped scheme and Λ-shaped scheme in turn, which is in the light of the intersection relationship between shroud streamline and hub streamline in the blade wood pattern. It is evident from Figure 1 that X-shaped blade is back-swept (local return effect) and Π-shaped blade is forward-swept (local diffusion effect) in the rotational direction.

![Image](image_url)

**Figure 1.** Lean mode of blade trailing edge.

3. Numerical methods

3.1. Turbulence model and near-wall region model

Flow pattern in centrifugal pump belongs to three-dimensional incompressible turbulent flow, and its numerical simulation often adopts RANS method which has high computational efficiency[13]. The mainstream region in centrifugal pump is fully developed turbulent flow with high Reynolds number.
Therefore, the SST k-ω model, which is recommended by many scholars, is used in this study. And it not only can simulate the strong swirling flow, but also improves the sensitivity to the adverse pressure gradient. And compared with the k-ε model, the simulation results of SST k-ω model are more accurate and stable because it can reduce the simulation difficulty of near-wall region[13-14]. The transport equation of turbulent kinetic energy $k$ and dissipation rate $\omega$ of the model is shown in Equation (1).

$$
\frac{\partial \rho k}{\partial t} + \rho u_j \frac{\partial k}{\partial x_j} = \frac{P_k}{Re} - \frac{\beta_p k \omega}{M_a} \frac{Re}{M_a} + \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] M_a
$$

$$
\frac{\partial \rho \omega}{\partial t} + \rho u_j \frac{\partial \omega}{\partial x_j} = \frac{P_\omega}{Re} - \frac{\beta_p \rho \omega^2}{M_a} \frac{Re}{M_a} + \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial \omega}{\partial x_j} \right] + 2(1-F_i) \frac{\rho}{\sigma_\omega \omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \frac{M_a}{Re}
$$

The physical parameters of above equation are as follows: $u_j$ - velocity, $Re$ - Reynolds number, $x_j$ - direction vector, $P_k, P_\omega$ - pressure, $\mu, \mu_t$ - viscous coefficient, $F_i$ - weighted function, $M_a, \beta_p, \alpha_t, \sigma_k, \sigma_\omega$ - closed constant.

As for the simulation of near-wall region, SST k-ω model adopts the k-ω model in the near-wall region while uses the k-ε model in the mainstream region by means of the mixture function. Thus, for the k-ω model, it is of great importance to control reasonable $y^+$, which guarantees enough computational nodes in near-wall region[13].

### 3.2. Meshing and Y-plus assurance

There are many calculation domains of the centrifugal pump whose geometrical shape is complex, so the hybrid grids with hexahedral core are adopted, which have better adaptability to the boundary and can combine the advantages of hexahedral structured grids and tetrahedral unstructured grids. The boundary layer grids of blades are taken into account, and the complex geometric region is locally refined. Additionally, in order to provide reliable results for subsequent research, the grid-independent verification is carried out for four meshing schemes, and the grid number ratio is about 2.2. Besides, the rated head of double-suction pump is used as a monitor value for grid-independent verification, and the corresponding result is given by Figure 2. Obviously, the difference between results of G3 and G4 is only 0.053%. In order to weigh a balance between calculation accuracy and calculation cost, the number of grids for the whole calculation domain is about 2,826,000 finally.

Moreover, reasonable $y^+$ must be considered to adapt to the flow characteristics of near-wall region. However, the values of $y^+$ can only be determined after the calculation is completed, the meshing and calculation are carried out iteratively, which can guarantee that the final values are within a reasonable range. For instance, the $y^+$ distribution of blade surface is given by Figure 3.
3.3. Solver control
Steady simulation is carried out firstly, and then transient simulation is carried out on the basis of the steady results. The time step of transient simulation is \(3.356 \times 10^4\)s, namely the impeller rotates 3° in each time step. Second-order upwind scheme is used for the convective terms, while second-order backward Euler scheme is used for the transient terms. As for the boundary conditions, mass flow inlet and static pressure outlet are adopted, and the surface roughness is set according to the design drawing. In terms of the calculation method, the whole implicit coupling technique is adopted in the study. Moreover, the convergence residual standard is \(1.0 \times 10^{-5}\), while monitoring the head and shaft power of the pumps.

4. Results and discussions

4.1. External characteristics of pumps
The CAU300GS35-typed double-suction centrifugal pump model for numerical calculation is shown in Figure 4. And the rated parameters of the pump are as follows: head \(H_r=35.5\)m, flow rate \(Q_r=800\)m³/h, power \(P_r=86\)kW, specific speed \(n_r=125\), rotational speed \(n_r=1490\)rpm, impeller outlet diameter \(D_2=352\)mm, blade number \(Z=6\), circumferential thickness of blade outlet \(\delta_r=15\)mm. Specifically, according to effective operating range of the pump, numerical simulation at seven operating points (0.6\(Q_r\sim 1.2Q_r\)) for each scheme is conducted to obtain the head-flow curve and efficiency-flow curve. In addition, the volumetric efficiency and mechanical efficiency are estimated by referring to the relevant papers[15]. Besides, in order to verify the reliability of the numerical simulation results, prototype pump experiment for external characteristics has been conducted, and the results of efficiency are shown in Figure 5.

In operating range of 0.6\(Q_r\sim 1.2Q_r\), the relative error of efficiency between the numerical simulation and the experiment is about 1.22%. It shows that the simulation results are reasonable and reliable, which can be used to perform detail analysis. Moreover, there is little difference between the results of X-shaped scheme and A-shaped scheme. And both head and efficiency of \(\Pi\)-shaped scheme decrease slightly during the low flow rate conditions. Therefore, it proves that different lean modes of blade trailing edge have little effect on the external characteristics of double-suction centrifugal pump, and all three schemes can meet the requirements of basic performance.

![3D model of the pump](image1)

**Figure 4.** 3D model of the pump.

![Simulation results of external characteristics](image2)

**Figure 5.** Simulation results of external characteristics.

4.2. Flow pattern of impeller outlet
Considering the main difference of each scheme lies in the impeller outlet area, outlet cylindrical surface of the right impeller viewed from inlet is chosen as a monitor surface, and the flow pattern affected by lean modes is analyzed by means of velocity uniformity and rotation intensity in this surface. Concretely, velocity uniformity \(V_u\) is expressed as:
\[ V_u = \left[ 1 - \frac{1}{C} \sqrt{\frac{\sum (C_i - \bar{C})^2}{n}} \right] \times 100\% \] (2)

where \( n \) is the number of grid node, \( C_i \) is absolute velocity of each node and \( \bar{C} \) is average value of above absolute velocity. Besides, rotation intensity is given by:

\[ \Omega_s = \frac{(\lambda_{ci}^{\text{max}} - \lambda_{ci}^{\text{min}})}{64 \cdot n \cdot Q^2} \sum \lambda_{ci}^2 \] (3)

\[ \lambda_{ci} = \frac{\sqrt{3}}{2} \left( \sqrt{\sqrt{\Delta} - \frac{1}{2} R + \sqrt{\Delta} + \frac{1}{2} R} \right) \] (4)

where \( Q \) is flow rate of operating condition, \( \lambda_{ci} \) is a vortex identification criterion called swirling strength, \( \Delta \) is eigenvalue equation discriminant of \( \nabla V \) and \( R \) is the third invariant of \( \nabla V \). Both \( \Delta \) and \( R \) can be calculated by referring to the relevant papers[16].

**Figure 6.** Velocity uniformity of impeller outlet.  
**Figure 7.** Rotation intensity of impeller outlet.

One of the most important problems for the flow at impeller exit is the jet-wake structure, and the mixing of the jet-wake flow is to be a source of significant disturbance and loss generation. As definite evaluation indexes, dimensionless \( V_u \) and \( \Omega_s \) can reveal this phenomenon vividly. Concretely, the velocity uniformity curve for each scheme under variable flow rates is shown in Figure 6. \( V_u \) of impeller outlet meets the maximum value in the vicinity of rated condition, while it decreases remarkably for off-design conditions. In the whole range of operating conditions, it maintains highest for velocity uniformity of \( X \)-shaped scheme, and it keeps lowest for velocity uniformity of \( \Pi \)-shaped scheme. The maximum difference between them is up to 4.46\%, which proves that positive blade lean with local return effect can suppress the jet-wake structure to a certain degree.

Moreover, the rotation intensity curve for each scheme under variable flow rates is shown in Figure 7. As the flow rate increases, \( \Omega_s \) of impeller outlet decreases gradually. In the whole range of operating conditions, it maintains highest for rotation intensity of \( \Pi \)-shaped scheme, and it keeps lowest for rotation intensity of \( X \)-shaped scheme. The maximum difference between them is up to 31.85\%, which proves that negative blade lean with local diffusion effect cannot suppress the trailing vortex well. Additionally, it also has a bad influence on the downstream flow and lead to more hydraulic loss in the volute casing. For example, the values of hydraulic loss for \( X \)-shaped scheme, \( \Lambda \)-shaped scheme and \( \Pi \)-shaped scheme are 3.77m, 3.85m and 3.90m respectively, and it should be the reason why head decreases slightly during the low flow rate conditions for \( \Pi \)-shaped scheme.

4.3. Pressure fluctuation characteristics
Pressure fluctuation characteristic is an important indicator of the stability of pump, while the representative fluctuation characteristic lies in the top point of volute casing. Specifically, the tongue of volute casing is straight, and its initial angle is about 33°. In this paper, the amplitude of pressure fluctuation is expressed as the pressure coefficient \( C_p \), which is defined as:

\[
C_p = \frac{p - \bar{p}}{0.5 \rho u_2^2}
\]

where \( p \) is transient pressure, \( u_2 \) is circumferential velocity of impeller outlet, \( \rho \) is density, \( \bar{p} \) is average pressure in statistical time. Moreover, due to the fact that rotational speed of the study object is 1490rpm and the number of blades is 6, rotation frequency and blade frequency of the double-suction centrifugal pump are 24.83Hz and 149Hz respectively.

**Figure 8.** Pressure fluctuation frequency characteristics under different operating conditions: (a) 0.6\( Q_0 \); (b) 1.0\( Q_0 \); (c) 1.2\( Q_0 \).

Apparent pressure fluctuation frequency characteristics under different conditions are shown in Figure 8. The main frequency for each scheme is 297.38Hz at rated condition, which is twice blade frequency and in agreement with 299.10Hz obtained by experiment, while the values of amplitude for X-shaped scheme, \( \Lambda \)-shaped scheme and \( \Pi \)-shaped scheme are 0.00463, 0.00496 and 0.00449 respectively. Besides, the main frequency is also twice blade frequency at 1.2\( Q_0 \) condition, while the values of amplitude for X-shaped scheme, \( \Lambda \)-shaped scheme and \( \Pi \)-shaped scheme are 0.00381, 0.00454 and 0.00450 respectively. However, the main frequency for each scheme is no longer twice blade frequency at 0.6\( Q_0 \) condition, while low frequency fluctuation with wideband frequency occurs
obviously. And the maximum values of amplitude for X-shaped scheme, A-shaped scheme and Π-shaped scheme are 0.01061, 0.00747 and 0.00823 respectively.

Therefore, in the vicinity of rated condition, inclined blade trailing edge can suppress the pressure fluctuation to a certain degree, and especially for the negative lean mode. Nevertheless, with the change of flow rate, the influence of positive lean blade on pressure fluctuation is very unstable. Additionally, owing to the severe changes of flow pattern, pressure fluctuation characteristics are very complex during low flow rate condition. Hence simple blade lean has no significant suppression effect on pressure fluctuation, which may need supplementing with appropriate blade-loading distribution.

4.4. Erosion characteristics of impeller

Double-suction centrifugal pumps are widely used in the Yellow River irrigation district because of their many advantages, so it is necessary to pay attention to the influence of blade lean modes on erosion characteristics of impeller. Specifically, for numerical simulation of solid-liquid two-phase flow in the double-suction centrifugal pump, sand particles whose density is 2650 kg/m$^3$ are assumed to be fully rounded, while the median diameter is chosen as 25 μm. The inlet sand volume friction is set to 0.3774%, which is in accordance with the Yellow River pumping station. Two-phase flow model is taken as Eulerian model, and interphase drag model is chosen as Wen-Yu model[18]. Besides, erosion model is taken as McLaury model[19], which can be given by:

$$ERD = \alpha_s \rho_s C (BH)^{-0.59} F_s V_s^{1.5} F(\theta)$$  \hspace{1cm} (6)

$$F(\theta) = 5.40 \theta - 10.110^2 + 10.93 \theta^3 - 6.33 \theta^4 + 1.42 \theta^5$$ \hspace{1cm} (7)

where ERD is erosion ratio density, $\alpha_s$ is sand volume friction, $\rho_s$ is sand density, BH is the Brinell hardness of the wall material, $F_s$ is particle shape coefficient, $V_s$ is sand impact speed, $\theta$ is impact angle in radians, while $n=2.41$ and $C=2.17 \times 10^{-7}$ are empirical constants.

Concretely, average values of erosion ratio density for impeller are shown in Table 1. Obviously, erosion ratio of impeller is relatively higher under low flow rate conditions, while eccentric erosion generally lies in the left impeller viewed from inlet. Moreover, it maintains relatively highest for erosion ratio density of X-shaped scheme, and it keeps relatively lowest for erosion ratio density of Π-shaped scheme. The maximum difference between them is up to 5.72%, which proves that both positive blade lean and zero blade lean are not conducive to improving the anti-erosion performance of the blade, and especially for X-shaped scheme. Therefore, in order to improve the anti-erosion performance of impeller, negative blade lean with other necessary measures is more appropriate.

| Erosion ratio ($\times 10^{-5}$ kg m$^{-2}$ s$^{-1}$) | X-shaped scheme | A-shaped scheme | Π-shaped scheme |
|-----------------------------------------------|-----------------|-----------------|-----------------|
|                                              | R-Impeller      | L-Impeller      | R-Impeller      | L-Impeller      | R-Impeller      | L-Impeller      |
| 0.6$Q_o$                                     | 2.0707          | 2.3547          | 2.3438          | 2.1429          | 1.9647          | 2.2794          |
| 1.2$Q_o$                                     | 1.4091          | 1.4927          | 1.3670          | 1.4057          | 1.3760          | 1.4120          |

On the whole, different blade lean modes have a certain influence on the hydraulic performance of double-suction centrifugal pump. However, it is not an immeasurably vast difference. Especially in the vicinity of rated condition, all schemes have almost the same external characteristics. And it should result from the other important factor, namely blade-loading distribution, which can determine the work ability of impeller directly[12]. Specifically, blade-loading distribution of blade middle streamline (0.5 spanwise) at rated operating condition is shown in Figure 9. There is almost no difference among the three lean modes obviously. In this case, the influence of simple blade lean should be quite limited. Therefore, in order to improve the integrated performance of low and middle
specific speed double-suction centrifugal pump, under the condition of guaranteeing good blade-loading distributions firstly, negative lean mode of blade trailing edge is finally suggested.

Figure 9. Blade-loading of blade middle streamline at rated operating condition.

5. Conclusions
Three different linear lean modes of blade trailing edge are adopted to simulate the transient flow field respectively for a middle specific speed double-suction centrifugal pump, while hydraulic characteristics have been analyzed in detail. The following conclusions are achieved.
(1) Different lean modes of blade trailing edge have little effect on the external characteristics of the pump in operating range of 0.6Q0–1.2Q0, and all three schemes can meet the requirements of basic performance.
(2) Positive blade lean with local return effect can suppress jet-wake structure to a certain degree, while negative blade lean with local diffusion effect cannot suppress blade trailing vortex well.
(3) In the vicinity of rated condition, inclined blade trailing edge can suppress the pressure fluctuation to a certain degree, and especially for the negative lean mode. But with the change of flow rate, the influence of positive lean blade on pressure fluctuation is very unstable.
(4) Both positive blade lean and zero blade lean are not conducive to improving the anti-erosion performance of the impeller, and especially for positive blade lean scheme. Hence negative blade lean with other necessary measures is more appropriate.
(5) In summary, the influence of simple blade lean on hydraulic performance is quite limited for low and middle specific speed double-suction centrifugal pump whose blade is relatively narrow and long. In order to improve the integrated performance, under the condition of guaranteeing good blade-loading distributions firstly, negative lean mode of blade trailing edge is finally suggested.

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