Effects of Compound Sweep and Lean on the Aerodynamic Performances of Transonic Centrifugal Impellers

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Abstract. The blade lean and sweep has a significant effect on the centrifugal compressors performance. This paper modifies the blade leading edges of two transonic centrifugal impellers with two compound sweep and lean (CSL) methods: the blade tip forward sweep with a negative lean (SFB), the blade tip backward sweep with a positive lean (SBB), and the CSL effects are numerically studied. For the unshrouded impeller, the SFB improves the highest isentropic efficiency and total pressure ratio. The SBB decreases the total pressure ratio and almost has no effect on the isentropic efficiency. The SFB expands the stall margin. Conversely, the SBB decreases the stall margin. The CSL effects on the shrouded impeller are less obvious compared with the unshrouded impeller. By analyzing the flow field, it is found that, the CSL has important effects on the shock, the accumulation of low momentum fluid and the blade loading near the blade leading edge for unshrouded impeller and shrouded impeller. Since the CSL effects on the stall margin of the unshrouded impeller are associated with the blade tip loading and tip clearance leakage flow, the CSL does not influence the stall margin of the shrouded impeller without a tip clearance.

1. Introduction
To meet the needs of reducing the energy consumption and decreasing the compressor size, the single-stage pressure ratio and efficiency of centrifugal compressors need to be improved without decreasing the stall margin. There will be shocks near the centrifugal impeller inlet if the pressure ratio is high enough. In addition, because of the low blade aspect ratio of centrifugal impellers, the boundary layer along the solid wall will have larger space ratio. Therefore, the flow field of transonic centrifugal impellers will be more complex, including the boundary layer separation, tip leakage vortex, and the interaction between shock and boundary layer, etc. These phenomena not only increase sharply the impeller losses but also decrease the stall margin. This paper is aimed to studying the compound lean and sweep effects of blade leading edge on the performances of the unshrouded and shrouded transonic centrifugal impellers, and then proposes an effective compound sweep and lean method of improving the centrifugal impeller performance.

Changing some local features of centrifugal impellers while keeping the basic geometric and performance parameters such as design mass flow rate and design rotation speed unchanged is an effective, fast and convenient way to improve the impeller performance, and it has been used and studied widely. For example, in order to extend the stall margin, different casing treatments were applied to the impellers[1]. Swain and Engeda[2] used flow trimming or axial trimming to modify an existing impeller to meet a new flow rate or pressure ratio. Jung et al.[3] studied the effects of recessed blade tips on the performance and flow field in a centrifugal compressor by comparing two different recessed blade geometries with the flat tip blade. They found recessed blade tip was an effective
method of reducing the tip leakage flow. It could improve the total-to-total pressure ratio and isentropic efficiency over the whole operating range.

At the recent past, using blade bow, lean and sweep to inhibit the boundary layer development and adjust the shock structure has been an effective method of improving the performance of transonic compressors and extending the stable operating range. In the early twentieth century, the blade bow, lean and sweep were first introduced to the axial compressor, and a tremendous amount of published literatures about the blade bow, lean and sweep of axial compressors can be found. Hah et al.[4] introduced the forward swept and backward swept leading edge to the fan blade. The study reported that a forward swept blade could suppress the radial-wise secondary flow and the tip entropy generation, and a backward swept blade could decrease the shock loss. For both forward swept blade and backward swept blade, the effect on efficiency were not great. The study also showed that the forward swept blade could increase the stall margin. Conversely, a backward swept blade suffered a decreased stall margin. Govardhan et al.[5] investigated the forward swept and backward swept blades effect on the overall performance and flow field in the axial compressor rotor at different operating points. The results indicated that both forward swept and backward swept blades decreased the pressure ratio of the axial rotor. The forward swept blade extended the stall margin as forward sweep changed the streamlines pattern in such a way that the streamlines on the suction side were deflected towards the hub and the streamlines on the pressure side were deflected towards the casing, but an opposite phenomenon was observed in the backward swept rotor, which decreased the stall margin. Amano et al.[6] studied the effect of blade forward sweep and backward sweep on a transonic compressor performance, and drew the same conclusion as Govardhan et al.[5] that forward swept blade extended the stall margin. Conversely, a backward swept blade decreased the stall margin. In another paper, Xu et al.[7] reported neither forward swept blade nor backward swept blade had an effect on the efficiency. Because the sweeps had different effects on the shock structure and second flow, they could not decide a better sweep style. Passrucker et al.[8] and Corsini et al.[9] only studied the forward swept blade. Passrucker et al.[8] indicated increased efficiency with forward swept blade, and observed the deflecting of fluid towards the blade tip, which extended the stall margin. Corsini et al.[9] used the forward swept blade to extend the stall-free operational range in axial fan rotors by attenuating span-wise secondary flow. However, Gallimore et al.[10-11] reported different results about the swept blade. Although backward swept blade induced flow separation near the hub, the blade achieved the best overall performance which was achieved mostly by the perfect performance at the blade mid-span. The blade forward sweep led to the increase of blade loading near the leading edge, the decrease of blade loading near the trailing edge at the mid-span and the increase of the blade tip clearance loss which offset the benefit of blade forward sweep. Jang et al.[12] optimized a transonic axial compressor by introducing blade bow, lean and sweep, and indicated that blade bow was the best way of improving the rotor efficiency.

Considering the front axial portion of the centrifugal impeller is similar to axial rotor, it is possible to introduce the blade leading edge sweep or lean to the centrifugal impeller to improve the performance. However, the available literatures about blade leading edge lean or sweep of centrifugal impellers are very limited up to now. Krain and Hoffmann et al.[13] indicated higher efficiency and larger chock margin with backward sweep of the blade leading edge. Xu et al.[14] found both positive lean and negative lean of the blade leading edge could improve the centrifugal impeller efficiency. Negative lean achieved the highest efficiency, but the widest stable operating range was achieved with positive lean.

There are three problems in above studies about the effect of blade leading edge sweep or lean on the centrifugal impeller performance. Firstly, the available studies only quantitatively or qualitatively analyzed the sweep or lean effect, but the explanation for mechanism was not involved. Secondly, there was not a consistent conclusion about the sweep or lean effect. Thirdly, the sweep or lean is not defined explicitly. Whether the other parameters are constant or not is unclear when the blade leading edge sweep or lean is introduced.
To solve the above problems and understand further the mechanism of blade leading edge sweep or lean effect, this paper defines the sweep and the lean clearly, and then proposes a new definition of compound sweep and lean. Two compound sweep and lean cases are employed for both the unshrouded impeller and the shrouded impeller to determine a more effective compound sweep and lean design. The origin unshrouded impeller and the original shrouded impeller are used as the baseline cases and compared with compound sweep and lean cases. At the same time, this study compares the compound sweep and lean effects of the blade leading edge on the flow characteristics and the performances of the unshrouded impeller and the shrouded impeller.

2. Method

The original unshrouded impeller is a part of a single-stage compressor tested by the author’s team. The original shrouded impeller is achieved by moving the shroud of the unshrouded impeller, so the shrouded impeller has the same geometric parameters as the unshrouded impeller, except for the impeller outlet width (the difference is the tip clearance). The impellers and their basic parameters are presented in figure 1 and table 1, respectively.

![Figure 1. Impeller model.](image)

### Table 1. Basic impeller parameter.

| Parameter               | Model 1 | Model 2 |
|-------------------------|---------|---------|
| Rotation speed (r/min)  | 45000   | 45000   |
| Design mass flow (kg/s) | 1.3     | 1.3     |
| Tip relative Ma         | 1.6     | 1.6     |
| Tip clearance (mm)      | 0.5     | 0       |
| Blade number            | 11/11   | 11/11   |
| Outlet diameter D2 (mm) | 200     | 200     |
| Outlet width b2 (mm)    | 7.6     | 7.1     |

2.1. The definition of compound sweep and lean

In the present paper, the sweep or lean is only applied to the main blade, keeping the splitter unchanged, and the word “blade” refers to the main blade of the impeller for convenience. The shifted blades are designed by shifting the blade leading edge along the meridional chord line (axial direction), maintaining the same blade angle $\beta$ distribution along the meridional chord line at both the hub and shroud sections. Because of the three dimensional blade structure, this method introduces sweep and lean at the same time, which is the so-called “compound sweep and lean”(CSL). In order to isolate the CSL effect of the blade leading edge, it is need to keep the aft portion of the CSL blade the same as that of the origin one. The inlet blade angle is unchanged eliminating the incidence effect. The blade wrap angle $\theta$ is defined by equation (1):

$$\theta = \theta_0 + \int_{m_2}^{m_1} \frac{\tan \beta}{r} dm$$

(1)
Where $m$ is the meridian line length, $m_T$ is the trailing edge position, $r$ is the local radius and $\theta_T$ is the wrap angle at the trailing edge. The integration is started from the trailing edge. If the blade angle distribution and the wrap angle $\theta_T$ are constant, the warp angle at the leading edge will be changed as the leading edge moves along the meridional chord line.

The paper conducts a comparative study between the unshrouded and shrouded impellers, and designs two kinds of different CSL blades (blade tip forward sweep with positive lean (SFB) and blade tip backward sweep with positive lean (SBB)) and an original blade without sweep or lean (UOB) for each impeller as shown in figure 2. In the figure, the red radial lines are the UOB blade leading edges, and the red leaned lines are the shifted blade leading edges. The two kinds of CSL blades share the lean angle. The blade models are presented in figure 3. The CSL only changes the blade profile near the blade leading edge, the aft portion of the blade is unchanged.

2.2. Numerical Method

In the present paper, high-resolution steady CFD simulation is conducted with the commercial CFD software ANSYS-CFX. As shown in figure 4, the single passage computational model consists of the impeller, the intake passage upstream and the vaneless diffuser downstream. The multi-block structured meshes are used for the impeller and intake passage, which are formed using the Automatic Topology and Meshing (ATM) feature in ANSYS TURBOGRID, and the meshes in the near wall regions are refined. The vaneless diffuser meshes are formed in ANSYS ICEM. The impeller meshes are shown in figure 5. By the grid independence check, the computations are carried out with about 1193792 notes for the unshrouded impeller. For the shrouded impeller, the number of notes is about 1000000.

It has been verified that the SST turbulence model presents the best agreement with the experimental data in the author’s previous study[15], and the SST turbulence model has been validated and used by many other studies[3,16]. Therefore, the $k-\omega$ shear stress transport (SST) turbulence model and automatic wall function are chosen. The unshrouded impeller studied in this paper is a part of a single-stage compressor consisting of impeller, diffuser, and volute, which has been tested by the author’s team. The numerical results were compared with the experimental ones in the previous study[17], showing good quantitative and qualitative agreements over most operating range.

At the intake passage inlet, the total pressure (99638 Pa) and total temperature (288.15 K) are assumed, and the flow is axial. Mass flow rate per passage is specified at the diffuser outlet, and static pressure is given at the diffuser outlet at near-choke point. The face between two adjacent main blades is periodic. The hub, shroud and blade surfaces are assumed as smooth, no slip and adiabatic walls. The computations are ended until residuals fall below 1.0e-05. Because only taking the residuals as the convergence criterion may be misleading in some cases, another convergence criterion is that the error in mass flow rates between the inlet and outlet of the computational domain is less than 0.1%.
3. Results

3.1. CSL effects on the performances of the unshrouded and the shrouded transonic impellers

Figure 6 presents the CSL effect on the unshrouded impeller performance. The CSL has significant effects on the isentropic efficiency, total pressure ratio and stable operating range. Compared with the UOB case, the SFB improves the highest impeller efficiency by 0.18%. The SBB almost has no effect on the efficiency.

According to figure 6(b), the SFB improves the total pressure ratio in the whole stable operating range. The SBB reduces the total pressure ratio. The stall margin and choke margin are presented in table 2. According to table 2, the SFB extends the stall margin. Conversely, the SBB reduces the stall margin. The CSL has little effect on the choke margin.

| Case      | UOB | SFB | SBB |
|-----------|-----|-----|-----|
| Stall margin | 11.5% | 15.4% | 7.7% |
| Choke margin | 12.4% | 13% | 13.4% |

In order to clarify the mechanism of CSL effects on the performances of different impellers, this paper studies the CSL effect on the shrouded impeller performance by comparing with the unshrouded impeller. The CSL effect on the overall performance of the shrouded impeller is presented in figure 7. It is because the shrouded impeller is achieved by moving the shroud of the unshrouded impeller that the shrouded impeller has smaller mass flow rate at choke point. Obviously, the CSL has great effect on the choke margin of the shrouded impeller. The CSL has almost no effects on the efficiency and total pressure ratio of the shrouded impeller at low mass flow rate conditions. The two CSL cases have
the same stall margin as the UOB case for the shrouded impeller, which is different from the unshrouded impeller.

According to the above analysis, the CSL influences the stall margin of the unshrouded impeller, but not for the shrouded impeller. We can predict that the CSL effect on stall margin may be associated with the tip clearance leakage vortex.

![Figure 7. Comparison of overall performances for the shrouded impeller.](image)

3.2. CSL effect on the flow field of the unshrouded transonic impeller

3.2.1. CSL effect on shock in the impeller inducer. Figure 8 presents the CSL effect on the shock strength at the highest efficiency operating point. Relative Mach number distributions are plotted on the blade-to-blade surfaces at 50% span and 95% span. As shown in figure 8, at 95% span, the SFB attenuates the shock strength as well as the shock loss compared with the UOB case. Conversely, the SBB enhances the shock strength, and increases the shock loss. At 50% span, the shock strength is weaker, and the CSL has less effect on the shock strength. From the above analysis, we can obtain that the CSL mostly affects the shocks near the blade tip. Because the shocks near the blade tip have significant effects on the loss and the stall margin of impellers, we can predict that the CSL may be an effective way of improving the centrifugal impeller performance.

![Figure 8. Mach number distribution on the blade-to-blade surfaces.](image)

3.2.2. CSL effect on the accumulation of low momentum fluid near the blade tip. The CSL influences the moving of low momentum fluid by three aspects: sweep, lean and shock. Near the blade leading edge, the total radial equilibrium equation is known as:

$$\frac{\partial P}{\partial r} = \frac{\rho \omega^2}{r}$$  \hspace{1cm} (2)

Where \(P\) is the static pressure, \(r\) is the local radius and \(\rho\) is the gas density. The right term is the centrifugal force, deciding the radial pressure gradient.

Figure 9 shows the sweep effect. For the UOB case, the pressure contours are nearly radial near the blade leading edge and perpendicular to the incoming flow. According to figure 10, the low
momentum fluid inside the boundary-layer moves radially outward to the blade tip due to the centrifugal force. For the SFB case, because the blade is forward swept, the pressure contours near the blade leading edge are tilted upstream, so there will be an additional radial force $\Delta P$, which is opposite to the centrifugal force. The equation (2) will become:

$$\frac{\partial P}{\partial r} = \rho \frac{V^2}{r} - \Delta P$$

(3)

The $\Delta P$ offsets a part of the centrifugal force, preventing the radial migration of low momentum fluid towards the blade tip. Otherwise, because the pressure contours are inclined upstream, the low momentum fluid will be deflected downstream as shown in figure 10. Hence, the accumulation of low momentum fluid near the blade leading edge tip is reduced. For the SBB case, the $\Delta P$ is in the same direction as the centrifugal force, promoting the radial migration of low momentum fluid, and the low momentum fluid will be deflected upstream as shown in figure 10. Hence, the accumulation of low momentum fluid near the blade leading edge tip is increased.

![Figure 9](image1.png)

Figure 9. The pressure contour distribution on suction side at the highest efficiency point.

![Figure 10](image2.png)

Figure 10. Sweep effect on the low momentum fluid.

Figure 11 shows the effects of lean. For the UOB case, the blade is radial near the leading edge. As given in figure 11, the blade force $F$ that is applied to the fluid is perpendicular to the blade without the radial force component. For the SFB case, the negative lean leads to a radial blade force $F_r$ on the
suction side, which is opposite to the centrifugal force. The radial force $F_r$ further prevents the radial migration of low momentum fluid. Hence, the equation (3) becomes:

$$\frac{\partial P}{\partial r} = \frac{\alpha^2}{r} - \Delta P - F_r$$

(4)

For the SBB case, the positive lean introduces a radial force $F_r$ on the suction side, which is from hub to shroud. The radial force $F_r$ further promotes the radial migration of the low momentum fluid.

Figure 11. Lean effect on the low momentum fluid.

Figure 12 shows the effects of shock. For the UOB case, the shock is nearly radial and perpendicular to the incoming flow. The velocity component $V_n$ that is perpendicular to the shock will be decreased across the shock, but the tangential component $V_t$ is unchanged. As presented in figure 13, when the fluid passes through the shock, the velocity is reduced, but its direction is unchanged. For the SFB case, the shock is inclined upward, as a result of blade tip forward sweep. When the fluid passes through the shock, the variation of velocity is shown in figure 13. The fluid is deflected towards the hub, which prevents the radial migration of the low momentum fluid. In the cases of SBB, the shock is inclined downward as a result of blade tip backward sweep. When the fluid passes through the shock, the variation of velocity is shown in figure 13. The fluid is deflected towards the shroud, which promotes the radial migration of the low momentum fluid.

Figure 12. Mach number distribution near the suction side.
Figure 13. Variation of velocities across the shock.

From the above discussions, it can be concluded that the SFB deflects the low momentum fluid towards the hub, reducing the accumulation of low momentum fluid near the blade tip. Whereas the SBB deflects low momentum fluid towards the shroud, increasing the accumulation of low momentum fluid near the blade tip. Figure 14 presents the limited streamlines distribution on the blade suction side at two different operating points.

As shown in figure 14(a), at the highest efficiency operating point, for the UOB case, the flow separation is very obvious from the blade root to 70% span. For the SFB case, the flow separation occurs ranging from the blade root to 30% span. The second flow deflects downstream at the 30% span, and the low momentum fluid reaches the blade tip at about 45% streamline position. For the SBB case, the second flow deflects downstream at 80% span position, and reaches the blade tip quickly at about 20% streamline.

As shown in figure 14(b), at near-stall operating point, the flow separation area continues to increase along both spanwise and streamline direction. For the UOB case, the second flow is deflected downstream at about 85% span, and then reaches the blade tip very quickly. For the SFB case, the second flow is deflected downstream at 50% span. The accumulation of low momentum fluid at the blade tip is further from the leading edge than that of the UOB case. For the SBB case, the second flow reaches the blade tip near the leading edge.

Therefore, we can draw a conclusion that the SFB reduces the accumulation of low momentum fluid near the blade tip, and makes it further from the blade leading edge. The SBB causes the accumulation of low momentum fluid closer to the leading edge, makes it closer to the blade leading edge.

Figure 14. Limited streamlines distribution on the suction side.
3.3 CSL effect on the flow field of the shrouded impeller
The CSL effects on the shock and limited streamlines distribution on blade suction side of the shrouded impeller at the highest efficiency point are shown in figure 15 and figure 16, respectively. The SFB attenuates the shock strength, and the accumulation of low momentum fluid at the blade tip is further from the blade leading edge. However, the SBB has the opposite effects. From the above discussion, it can be concluded that the CSL has the same effect on the flow field of shrouded impeller as that of the unshrouded impeller.

![Mach number distribution at 95% span.](image1)

![Limited streamlines distribution on the suction side.](image2)

4. Discussion
Figure 17 shows the impeller efficiency distribution along spanwise at 30% streamline. For both the unshrouded impeller and the shrouded impeller, the efficiencies of the SFB case is higher than that of the UOB case at above 80% span, which is in accordance with the above analysis that the SFB case decreases the blade tip shock strength, and the accumulation of low momentum fluid is at 50% streamline. The low momentum fluid migrates towards the hub, a large area of low Mach number region occurs ranging from 0% to 30% span as presented in figure 12. Therefore, the efficiencies of the SFB case is lower than that of the UOB case from 0% to 30% span. Conversely, the efficiencies of the SBB case is lower than that of the UOB case at above 80% span as a result of stronger shock and more low momentum fluid at the blade tip. From the above discussion, the CSL has important effects on the local loss and changes the efficiency distribution along the span.

![Impeller efficiency distribution along the span.](image3)
For the shrouded impeller, the CSL has the same effect on the accumulation of low momentum fluid as that of the unshrouded impeller. However, the CSL has no effect on the stall margin of the shrouded impeller. Therefore, the prediction that the CSL effect on the stall margin of the unshrouded impeller is associated with the accumulation of low momentum fluid near the blade leading edge tip is not true. Here we focus on the blade tip clearance which is the difference between the unshrouded impeller and the shrouded impeller. For the unshrouded impeller, the tip leakage flow (TLF) from the blade leading edge to 10% streamline interacts with the incoming flow, resulting in the tip clearance leakage vortex (TLV). The TLV separates itself from the suction side, moving downstream and towards the hub. This is an important flow structure in the impeller passages, and it varies with the operating points.

As the flow mass rate reduces, the blade tip loading near the blade leading edge becomes larger, and the TLV trajectory becomes more tangential. At the stall operating point, the TLV trajectory is parallel to the circumferential direction near the leading edge of main blade. The phenomena represent the stall of the centrifugal impeller. From the above analysis, the blade leading edge loading decides the distribution of the TLV, influencing the stall margin. Figure 18 shows the blade loading distributions along the streamline direction. The SFB case obtains smaller blade tip loading than the UOB case from 0% to 15% streamline, but the SBB case has larger blade tip loading. The TLV trajectory is more tangential for the SBB case as a result of its higher blade tip loading, so it has smaller stall margin. Conversely, the SFB case has larger stall margins. So we can get that the CSL effect on stall margin is achieved by influencing the blade leading edge loading and the TLV.

For the unshrouded impeller, the SFB reduces the blade tip loading, resulting in less tangential TLV trajectory and larger stall margin. Conversely, the SBB reduces the stall margin. Because the shrouded impeller has no blade tip clearance, the CSL has no effect on the stall margin of the shrouded impeller. It is verified that the CSL effect on the stall margin is achieved by influencing the blade leading edge loading and the blade tip clearance vortex for the unshrouded impeller.

For the unshrouded transonic impeller, the SFB is the most effective method of improving the efficiency, total pressure ratio and stall margin.

5. Conclusions
For both the unshrouded and shrouded impellers, the CSL influences the local losses by controlling the shock strength and the accumulation of low momentum fluid at the blade tip. The SFB improves the highest efficiency. The SBB has no effect on the highest efficiency. The SFB case improves the total pressure ratio. On the contrary, the SBB case decreases the total pressure ratio.

The CSL influences the moving of low momentum fluid by three aspects: sweep, lean and shock. The SFB reduces the accumulation of low momentum fluid near the blade tip. Whereas the SBB increases the accumulation of low momentum fluid near the blade tip.

For the unshrouded impeller, the SFB reduces the blade tip loading, resulting in less tangential TLV trajectory and larger stall margin. Conversely, the SBB reduces the stall margin. Because the shrouded impeller has no blade tip clearance, the CSL has no effect on the stall margin of the shrouded impeller. It is verified that the CSL effect on the stall margin is achieved by influencing the blade leading edge loading and the blade tip clearance vortex for the unshrouded impeller.

For the unshrouded transonic impeller, the SFB is the most effective method of improving the efficiency, total pressure ratio and stall margin.
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