Effects of impeller blades with bow trailing edge on centrifugal compressor performance

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ABSTRACT
The performance of centrifugal compressors can be improved by reducing the reverse flow intensity of their impellers’ outlet. This study used numerical simulations to investigate the effects of applying bow trailing edge to the impeller blades on the performance of a centrifugal compressor with a wedge diffuser. A peak efficiency 0.87% higher than that of the baseline centrifugal compressor is gained by using positive bow trailing edge on impeller blades. Flow in both the impeller and the diffuser is improved. The results of numerical simulations show that blades with positive bow trailing edges strengthen secondary flow near the blade suction surface in the radial part of the impeller, weaken reverse flow due to the “jet-wake” flow structure, reduce blockage, and render the flow field of the impeller outlet more uniform in spanwise direction. Furthermore, the impeller with positive bow trailing edge blades reduces the separation in the diffuser by improving the distribution of the flow angle at the impeller outlet. This study provides a valuable reference for applying impeller blades with bow trailing edge to advanced centrifugal compressors design in future research.

INTRODUCTION
Centrifugal compressors are widely used in small and medium-sized aero turbine engines because of their high pressure ratio in a single stage and reliability (Krain, 2005). Due to the viscosity and curvature of the blade passage, flow in the impeller has been described as a “jet-wake” structure (Eckardt, 1976). The pressure ratio has increased with advances in the design of centrifugal compressors, and has given rise to the challenge of diffusing flow with a high Mach number at the impeller outlet. In open-type impeller, on the one hand, the convergence of boundary layer is an important source of wake, and on the other hand, the leakage flow further significantly increases the region of the wake (Hong, 2012). The existence of the wake not only reduces the efficiency of the impeller, but also limits the performance of the diffuser locating downstream (Ziegler, 2013). Therefore, it is important to improve the flow in the blade passage of the impeller.

Inspired by axial compressors, designers have introduced lean to centrifugal compressors. Howard et al. (1994) concluded that an appropriate lean can reduce leakage flow by decreasing the load of blade tip. Chary et al. (2017) got an approximate conclusion by using numerical simulations and experiments. They both concluded that the negative lean blades suppress the regions of wake flow and improves the performance of the impeller either in terms of higher pressure or a greater stall margin. Oh et al. (2011) found that a slightly positive lean at the impeller outlet is better than the original design. Arunachalam et al. (2008) obtained similar results through numerical simulations. He et al. (2016) found that the
lean influences flow fields by affecting the shock structure, spanwise pressure gradient, axial-to-radial flow separation, and secondary flow structure. He also concluded that the positive lean has potential to improve compressor performance. Although different studies have reached different conclusions about lean, bow (which can be considered as an improved lean that has different lean angles at different spans) was introduced to impeller design. Hiradate et al. (2015, 2017) experimentally and numerically investigated the effects of blades with bow trailing edge on a fully shrouded impeller, and found that the efficiency improved by positive bow is due to the blade force rendering the distribution of the radial velocity uniform at the impeller outlet. However, just as the author said, the static pressure distribution at cross section in blade passage is not enough to support the explanation. Tsukamoto et al. (2015, 2019) studied the effects of the bow trailing edge on an open-type impeller, and the efficiency improvement of the impeller and vaneless diffuser was attributed to the blade force brought by the positive bow to reduce the tip leakage flow at the impeller outlet. Mosdzien et al. (2018) and Wittrock et al. (2018) improved the performance of two different centrifugal compressors by using bow impeller blades generating from numerical optimize method. These studies have reached the conclusion that bow trailing edge has the potential to improve the performance of centrifugal compressors, more analysis focuses on the analysis of the flow structure of the impeller outlet, while ignoring the development process of the internal flow of the impeller.

It can be concluded from the above that using the lean and further bow trailing edge blades in an impeller is still an immature method to control flow in its blade passage at current. It mainly manifests as the lack of understanding of the mechanism of the flow development in the blade passage has led to different results in different studies. Also, most work has focused on impeller only or the vaneless diffusers, and few studies have considered when a wedge diffuser is located downstream the impeller. Therefore, this study investigated a transonic centrifugal compressor with a wedge diffuser. By comparing the performance and flow fields of the baseline compressor and ones with bow trailing edge impeller blades, the mechanism of flow is analyzed in detail.

2. INVESTGATED COMPRESSOR AND NUMERICAL METHOD

The transonic centrifugal compressor considered here was designed by the Institute of Engineering Thermophysics of the Chinese Academy of Sciences. It is composed of an impeller and a wedge diffuser installed downstream. Its main design parameters are illustrated in Table 1.

The definition of the bow is shown in Fig. 1. The bow angle in this study is set to 20 degrees, which is identical to the lean angle of the baseline impeller outlet. In the process of applying the bow, the blade angle distribution of the hub is kept unchanged, and the distribution of blade angle at tip and middle span was inevitably changed. The bow distance has maximum at the trailing edge, and decreases upstream smoothly to ensures that the blades before the 50% streamwise position are consistent with the baseline. Both the positive bow (the direction of the bow is the same as that of the impeller rotation) and the negative bow (the direction of the bow is opposite to that of the impeller rotation) were considered in this study. Three-dimensional (3D) diagrams of the baseline compressor and that with bow trailing edge impeller blades are also illustrated in Fig. 1.

| Table 1 Main parameters of the compressor |
|-------------------|---|
| Parameter                  | Value |
| Design mass flow (kg/s)    | 1.51 |
| Design total pressure ratio| 5.9  |
| Number of impeller blades  | 12+12 |
| Tip clearance (mm)         | 0.15 |
| Relative inducer tip Mach number | 1.4 |
| Impeller outlet rotating velocity (m/s) | 551 |
| Impeller outlet blade lean angle | 20 |
| Impeller blade backward angle (°) | 26 |
| Number of diffuser blades  | 23  |
| Diffuser inlet blade angle (from tangential, °) | 18 |
| Diffuser outlet diameter (mm) | 121 |

All numerical simulations were performed using the Numeca Fine/Turbo Euranus solver. To reduce computational cost, steady-state simulations were chosen to obtain the performance of the compressors and the flow fields by solving the RANS (Reynolds-average Navier–Stokes) equations using the explicit fourth-order Runge–Kutta integration scheme. The Spalart–Allmaras turbulence model was applied to close the turbulence terms. Several acceleration methods, including the multigrid strategy, local time-stepping, and implicit residual smoothing, were applied to accelerate the convergence to the steady state. To save the time needed for the computations, only a single passage of the impeller and diffuser was chosen to mesh with a structured grid, and the rotor-stator interface was set as a mixing plane. The average value of the dimensionless distance y+ was less than five which is enough for this turbulence model to precisely capture the
characteristics of viscous flow near the wall. The standard total pressure and total temperature were specified at the inlet boundary. Average static pressure was chosen for the outlet condition. The computation began from the choke point, and the static pressure at the outlet was increased continuously until the computation could no longer converge. Although the steady-state computation could not precisely capture the stall limit, it could still reflect changes to stall limit.

Grid independence was evaluated based on the baseline compressor to achieve grid convergence. Several sets of grids were computed. When the number of nodes achieved 3 million, the choke mass flow rate and peak efficiency can be considered independent of the number of grids. Therefore, grids with 3 million nodes were chosen for this study after consideration. The reliability of the computations was validated by comparing the data on performance obtained from the numerical computations of and experiments on the baseline compressor. Fig. 2 compares the compressor performance calculated by the grids with different nodes and experiment result at the design rotational speed. It can be concluded that the numerical computation in this study was adequate for the purposes of this study.

![Fig. 1 Geometry of the compressor used in this study](image1)

(a) Definition of bow  
(b) 3D view

![Fig. 2 Comparison of the performance characteristics of the compressor between numerical calculation and experiments](image2)

(a) Total pressure ratio  
(b) Isentropic efficiency
3. RESULTS AND DISCUSSION

3.1 Effect of the bow trailing edge on the compressor performance characteristics

Fig. 3 compares the total pressure ratio and the isentropic efficiency of these three compressors. It shows that the compressor performance characteristic is significantly affected by bow trailing edge impeller blades. Compared with the baseline, the peak isentropic efficiency increases by 0.87% when the positive bow trailing edge is used, and decreases by 1.42% when the negative bow trailing edge is used. In the case with positive bow trailing edge, the total pressure ratio improves by 2.6% under the design mass flow rate. To discuss the flow mechanism, it is necessary to separately compare the performance of the impellers and the diffusers (Figs. 4 and 5). The performance of the impeller is not affected by the bow positive trailing edge. However, when the bow negative trailing edge is used, its isentropic efficiency decreases, and a higher total pressure ratio is obtained. The loss coefficient of total pressure of the diffuser increases when the negative bow trailing edge is used and decreases when positive bow tailing is used.

![Fig. 3 Comparison of the performance characteristics of the compressors](image)

**Fig. 3** Comparison of the performance characteristics of the compressors

(a) Total pressure ratio
(b) Isentropic efficiency

The stall limit is also affected by the bow trailing edge impeller blades. It has been proved that, at the design speed stall is triggered at the inlet of the diffuser in previous study. The negative bow trailing edge doesn’t appear to affect the stall limit of this compressor. However, the stall limit of the compressor with positive bow trailing edge significantly decreases by 21%. The reason of stall limit change will be discussed below.

3.2 Flow mechanism of the bow trailing edge affecting the impeller flow field

Even though the diffusers exhibit the most significant change in performance, the impellers provide the inlet conditions for them. It is thus important to analyze changes in impellers flow field first. The chock mass flow rate of the compressor nearly remains constant under different bow tailing edge types because the throat is located at the inlet of the inducer. Therefore, the design mass flow rate is chosen to analyze the mechanism of changes in the impellers flow field.
The reverse flow at impeller outlet is consisted of the boundary layer and the tip leakage flow. This part of the low-energy fluid cannot resist the adverse static pressure gradient due to the increase in the radius of the impeller, and forms a large-scale flow separation that further leads to the reverse flow. Fig. 6 compares the distribution of $V_\text{r}$ at the outlet of the impeller, and the proportion of the reverse region is also shown. It shows that the range of reverse flow is clearly changed by the bow trailing edge of the impeller blades. With positive bow trailing edge, this region of reverse flow invading the impeller is remarkably reduced. On the contrary, the impeller efficiency is significantly reduced due to reverse flow increasing. To further analyze reasons for this change, it necessary to discuss the development of secondary flow in the blade passage. The secondary velocity vectors are defined by their components perpendicular to the streamwise oriented mesh lines as the mesh is body-fitted (Kang et al, 2001). When the flow starts to change from the axial direction to the radial direction, the strength of the secondary flow in the impeller is significantly enhanced. Rather to the outlet, the change of secondary flow here more should be discussed. Fig. 7 compares the intense of secondary flow and the secondary flow structure at 70% section of meridional length under different type of bow trailing edge. The secondary flow in each blade passage can be summarized into two blade surface vortices and one casing surface leakage vortex. Since the loading of the main blade is greater than that of the splitter blade, the surface vortices on the suction and pressure surfaces of the main blade and the leakage vortex on the main blade suction side is stronger, which is also changed the most obviously. The suction surface vortices move upper in spanwise direction under the positive bow trailing edge, and lower in spanwise direction under negative bow trailing edge (as a marked in Fig.7). The movement of the pressure surface vortices is opposite to the suction surface ones. The strength of the main blade suction surface vortex is significantly enhanced under the positive bow trailing edge, and the strength of tip leakage vortex on the main blade suction side has the same trend of change under suction surface vortex induction (as b marked in Fig.7). The fluid that constitutes the main blade suction surface vortex has the maximum relative velocity in the cross section. Also, the boundary on the pressure surface is much thicker than the one on suction surface. Therefore, it can be considered that the two blade surface vortices play opposite roles in the formation of the wake. The pressure surface vortex continuously transports low-energy fluid to the casing surface, and part of it becomes the leakage flow. This fluid is the main source of the wake. On the contrary, the suction surface vortex continues to convey high-speed fluid to the casing surface. That means suction surface vortex continues to convey kinetic energy to boundary layer of the casing, further suppresses the separation. In the case of using positive bow trailing edge, the enhancement of the suction surface vortices and the weakening of the pressure surface vortices reduce the size of the recirculation zone. Especially, because the strength of the suction surface vortex of the main blade is the largest and the change is also the largest, the reduction of the wake area on the suction surface side of the main blade is the most obvious.

The secondary flow change must result from the static pressure changed by introducing bow trailing edge into the impeller. Fig.9 compares the distribution of static pressure at this section. Take the main blade as example, the low pressure region on the suction surface tip side is significantly increased, and the high pressure area on the pressure surface tip side is enhanced under the positive bow trailing edge. The normal direction of the isoline of static pressure can be considered as the direction of the pressure gradient projection on this section (as black arrow show in Fig.9). It can be found that the angle of the pressure gradient near suction surface and the spanwise direction increases, on the contrary, the angle of the pressure gradient near pressure surface and the spanwise direction decreases under the positive bow trailing edge. Also, the component of pressure gradient in spanwise direction increases on suction surface, and decreases on the pressure surface. Due to the change in the magnitude and direction of the pressure gradient, the suction surface vortex increases and moves up, and the pressure surface vortex weakens and moves down. The trend of static pressure change under negative bow trailing edge is just opposite. Changes in the static pressure gradient occurs due to additional the inertial force caused by the curvature of migration on the blade surface in cross section when using bow trailing edge. The migration concentrates on the upper span of the blade. The inertial force is shown as red arrow in Fig.9. The pressure gradient changes its direction and magnitude to balance the inertial force. Also, the spanwise direction changed by bow trailing edge further aggravate the change of angle between pressure direction and spanwise direction. It is worth to mention that, the pressure difference between the suction and pressure surface of the blade tip increases under the positive bow trailing edge. This may cause a larger flow rate of tip leakage flow and lead to no improvement in efficiency of the impeller when the outlet reverse flow region is reduced.

### 3.3 Flow mechanism of the trailing edge bow affecting the diffuser flow field

Changes in the conditions of the diffuser inlet due to the impellers leads to changes in the performance of the diffusers. Entropy generation downstream of the impeller occurs in both the vanelss and the vaned region. In the former, entropy generation is reduced due to the weakening of the boundary layer separation on the casing in case with positive bow trailing edge. This not only reduces loss due to the separation itself, but also reduces loss due to a lower the gradient of shear stresses in the axial direction by improving the uniformity in the spanwise direction. This part of the loss is non-negligible in the diffusers, and a greater difference is noted in the vaned region. Fig. 10 compares the circumferential average flow angle (defined as the angle between the direction of flow direction and tangential direction) at the inlet of the diffuser at the design mass flow of these cases (the negative flow angle means reverse flow). It shows that changes in the absolute
flow angle on the casing and the hub sides have opposite tendencies. The flow angle on the casing side decreases in the case with positive bow trailing edge and increased in case with negative bow trailing edge. Meanwhile, the flow angle on the side of the hub is increases in case with positive bow trailing edge and decreases in case with negative bow trailing edge. Fig.11 compares the Vr and ab absolute Vt distribution at the inlet of the diffuser under design condition. As the blade angle at the impeller outlet doesn’t change in these cases, it can be considered that the distribution of Vt is almost the same. In other words, the change in the flow angle occurs mainly because of changes in Vr. With positive bow trailing edge, the impeller suppresses the reverse flow on the side of the casing further improves flow capacity on the side of the casing. Under the same mass flow rate, more flow through the side of the casing leads to a more uniform velocity distribution in spanwise direction. This leads to a reduction in Vr and the flow angle on the hub side.

It is the improvement of the flow angle of the diffuser inlet that brings the improvement in diffuser blade passage. Fig.12 and Fig.13 respectively compare the limiting streamlines distribution of the suction surface and pressure surface of the diffuser under design point. In the baseline, the positive incidence angle at the casing side and the negative incidence angle at the hub side respectively leads to the separation on suction and pressure surface of diffuser blade (as marked in Fig.12 and Fig.13). When the impeller with positive bow trailing edge reduces the incidence angle of the diffuser both at the casing and hub side, the separation on the two surface is alleviated remarkably. By contrast, in case with negative bow trailing edge, the separation is aggravated. When the separations on the suction and the pressure surfaces are controlled, the diffuser has a lower total pressure loss coefficient under a positive bow trailing edge.
Except for the efficiency, another point needed to concern is the operating range significantly reduced by the positive bow trailing edge. To discuss the reasons for this, the flow field in the diffusers at the mass flow rate which corresponds to the near stall point of the case with positive bow trailing edge is analyzed. Fig. 14 compares the circumferential average flow angle at the diffuser inlet at this flow rate. The trend of changes in the flow angle with the bow trailing edge of the impeller blade is consistent with that at the design point, which means the incidence angle is improved by positive bow trailing edge. It can be considered that the stall is not induced by the separation of the blade surface due to the incidence angle. Fig. 15 compares the spanwise distribution of the magnitude of absolute velocity at diffuser inlet. Due to the positive bow trailing edge reduces the reverse flow at impeller outlet, the flow capacity on the casing side is increased, and the magnitude of absolute velocity on the hub side is reduced. The cross sections of the diffusers are shown in Fig. 16 to analyze the flow development under the near stall flow rate of the positive bow trailing edge impeller, and the proportion of
blockage region (defined as absolute Mach number under 0.2) is calculated at every section. Under the flow angle of diffuser inlet, the diffuser blade passage is dominated by passage vortex, and the intense is increases as the flow rate decreasing (the direction of the vortex is shown in Fig.16). Under the effect of this passage vortex, the boundary layer of the diffuser blade pressure surface moves to the hub side, and migrates to the middle of the blade with the boundary layer of hub together. Further, a low-velocity region that blocks the flow is formed on the surface of the hub. Fig.16 shows that, the region of blockage increases significantly in the case of using the positive bow trailing edge when the separation at diffuser blade pressure surface is suppressed. This confirms that, the generation of low-velocity region is not caused by flow separation. It can be considered that the lower velocity causes the boundary layer between the casing and the pressure surface of the blade to accumulate more easily under the effect of the reverse pressure gradient. Therefore, the stall under steady state calculation triggered by low-velocity flow on hub occurs under a greater mass flow rate in the case of using positive bow trailing edge.

Fig. 14 Spanwise distribution of the absolute flow angle at near stall mass flow rate

Fig. 15 Spanwise distribution of the magnitude of absolute velocity at near stall mass flow rate

Fig. 16 The flow development in the diffuser under the near stall flow rate
4. CONCLUSIONS

This study investigated the effects of the impeller blades with bow trailing edge on the performance of a transonic centrifugal compressor in terms of its flow mechanism. The aim is to provide a valuable reference for applying bow trailing edge to advanced centrifugal compressor design. The main conclusions are as follows:

(1) In a centrifugal compressor with a wedge diffuser, the use of the bow trailing edge on the impeller blades can significantly change the performance characteristics of the compressor. The peak efficiency increases by 0.87% and the total pressure ratio at the design point increases by 0.3% under the positive bow trailing edge. On the contrary, the negative bow trailing edge significantly reduces the total pressure ratio and efficiency of the compressor. Also, the operating range of the compressor is significantly affected by the use of positive bow trailing edge, where the stall limit decreases by 23%.

(2) The use of the bow trailing edge influences secondary flow inside the impeller, especially in the part when flow direction changing from axial to radial. The positive bow trailing edge enhances the suction surface vortices and suppresses the pressure surface vortices by changing the static pressure distribution in the blade passage. Further, the reverse flow region is reduced by positive bow trailing edge.

(3) The positive bow trailing edge improves the uniformity of the velocity distribution in spanwise direction of the impeller outlet. This reduces the viscous loss in vaneless region. Velocity redistribution in spanwise direction improves the incidence angle of the diffuser blade and further reduces the region of separation on suction and pressure surface. Therefore, the total pressure loss coefficient can be reduced by using positive bow trailing edge.

(4) Also, using the positive bow trailing edge has a negative impact on the stall limit. Because the positive bow trailing edge reduces the velocity at the hub side of the diffuser inlet by reducing the reverse flow on the casing side of the impeller outlet, the stall caused by the accumulation of the boundary layer on the hub and the pressure surface occurred at a greater mass flow rate.

NOMENCLATURE

\( y^+ \)  Nondimensional wall distance of first node  
\( \text{PBTE} \)  Positive bow trailing edge  
\( \text{NBTE} \)  Negative bow trailing edge  
MB  Main blade  
SB  Splitter blade  
\( \text{Ma}_{ab} \)  Absolute Mach number  
\( V_r \)  Velocity in radial direction  
\( V_t \)  Absolute velocity in tangential direction  
\( U_2 \)  Impeller outlet velocity  
\( W_{sec} \)  Magnitude of secondary velocity  
\( P_{s2} \)  Average static pressure at impeller outlet

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