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Study on casing treatment and stator matching on multistage fan

Chuangliang Wu¹, Wei Yuan², Zhe Deng¹

¹ Master Degree Candidate, School of Energy and Power Engineering, Beihang University, Beijing 100191, China
² Associate Professor, School of Energy and Power Engineering, Beihang University, Beijing 100191, China

Abstract. Casing treatments are required for expanding the stall margin of multi-stage high-load turbofans designed with high blade-tip Mach numbers and high leakage flow. In the case of a low mass flow, the casing treatment effectively reduces the blockages caused by the leakage flow and enlarges the stall margin. However, in the case of a high mass flow, the casing treatment affects the overall flow capacity of the fan, the thrust when operating at the high speeds usually required by design-point specifications. Herein, we study a two-stage high-load fan with three-dimensional numerical simulations. We use the simulation results to propose a scheme that enlarges the stall margin of multistage high-load fans without sacrificing the flow capacity when operating with a large mass flow. Furthermore, a circumferential groove casing treatment is used and adjustments are made to the upstream stator angle to match the casing treatment. The stall margin is thus increased to 16.3%, with no reduction in the maximum mass flow rate or the design thrust performance.

1. Introduction
An important benchmark for developing fourth-generation advanced turbofan engines is to improve their thrust-to-weight ratio. Recent advancements in high-load fan technologies are particularly important for achieving this goal. Currently, the three-stage fan technology is widely used in developed countries. For example, the total pressure ratio of GE’s F110 three-stage turbofan reaches 3.2, whereas that of Pratt and Whitney’s three-stage F119 turbofan reaches 4.0[1]. Higher-load fans used in next-generation engines remain in the laboratory phase of research. In 1987, the United States military initiated the Integrated High Performance Turbine Engine Technology [2] program, which achieved great conceptual strides towards the design of high-load compressor fans. In 2002, NASA’s Green Centre achieved a stage pressure ratio exceeding 3[3]. In the early 1990s, several Chinese labs performed numerical simulations and experimental studies to investigate the performance of high-load inlet-stage fans. Two experiments on single-stage fans have been successfully completed, achieving pressure ratios of 2.2–2.3[4-5].

For achieving a simple structure, good stability and high reliability, casing treatments have been widely used in aerospace engineering applications. Casing treatments follow two typical structures, with either axial grooves or circumferential grooves. Axial groove casing treatments have good stability-
enhancing effects [6-8]; however, they require some axial space in front of the rotor, restricting their use to single-stage fans. To ensure compactness of the fan, circumferential groove casing treatments are usually used on the rear stage of multistage fans. Researchers have devoted considerable attention towards optimizing the casing parameters and understanding the mechanisms behind the stability extension caused by circumferential groove treatments. There still remains a controversy associated with the optimal axial position of circumferential grooves. Bailey [9] and Moore [10] found that the optimum position of circumferential grooves is at the middle of the blade chord. Furthermore, Muller [11] and Raba and Hah [12] suggested that the leading edge of the blade is the best position. Wu Anhui [13] and Bennington [14] reported that the stability extension achieved through circumferential groove treatments decreases as the groove width increases. In addition, Bailey [9] and Bennington [14] reported that the stability extension effect increases with increasing groove depth. Several other groups have attempted to explain the effectiveness of circumferential grooves. From the stable flow point of view, Muller [11] and Wu et al. [13] suggested that the circumferential grooves delay the overflow at the leading edge of blade tips, thereby enlarging the stall margin. Zhao et al. [15] studied the unstable flow effects of the circumferential grooves on the leakage flow at blade tips. Hah [16] found that the unstable fluctuations of the circumferential suction and ejection have an important influence on the stability extension effect.

It has been difficult to overcome the lack of aerodynamic stability for high-load turbofans. Many researchers have extended the stable operating range of fans by installing circumferential slot casings with some success. Dang [17] used circumferential groove slots to reduce the stall point of a high mass-flow rate fan by 11.5%. Duane [18] installed various circumferential groove slots on a high-load fan and increased the stability margin by 7.3%. Li [19], Liu [20] and other researchers have separately studied the optimum application and the mechanism of action for circumferential groove slots installed around high-load compressor fans. However, most studies focused on improving the stall margin of the fan while ignoring the side effects of the casing treatment on the right-hand branch of fan’s performance curves. Most studies are further limited by focusing only on single-stage fan designs. In fact, casing treatments in real-world applications will reduce the mass-flow rate through the point of maximum flow while enlarging the stability margin. For high-load compressor fans, the performance design points are often near the right-hand branch of the performance curve. Casing treatments may therefore reduce the design thrust of a turbofan engine. Considering these complications, we study a two-stage high-load transonic fan design. Using numerical simulation tools, we investigate a circumferential groove casing treatment and propose an upstream stator-matching scheme that can eliminate the drawbacks of casing treatments.

2. Research object and methods

2.1. Two-stage fan and casing treatment
This research focuses on a high-load two-stage fan that includes guide vanes. Its pressure ratio is above 4.5 at the design speed, and the tangential speed of the rotor blade tips exceeds 500m/s (Mach 1.46). The load coefficient of the first-stage rotor reaches 0.4, whereas that of the second-stage rotor reaches 0.35.

Figure1. Two-stage high-load fan
Figure 1 shows a three-dimensional (3D) model of the fan. The design parameters of the circumferential casing treatment are listed in Table 1, and a detailed schematic of the circumferential groove casing treatment is shown in Figure 2.

![Image of a three-dimensional model of a fan]

**Figure 2.** Circumferential groove casing treatment

| Groove number | Groove depth/mm | Groove width/mm |
|---------------|-----------------|-----------------|
| 7             | 10              | 4               |
| Overlap length/mm | Groove spacing/mm | Distance to leading edge/mm |
| 40            | 2               | 2               |

### Table 1. Geometric parameters of the circumferential groove casing treatment

2.2. **Numerical simulation method**

We used ANSYS CFX, a numerical simulation software, with a finite-volume method to solve 3D Reynolds-averaged Navier–Stokes equations (N–S equations). In addition, the SST turbulence model was used. Considering the accuracy of our results and the computational load involved, unstable simulations were performed only at the near-stall and design points using the transient blade row (TBR) method in the CFX solver and stable simulations were used for all other conditions. Because the entire computational domain (including the casing treatment) is axisymmetric and the intake conditions are uniform, all calculations were performed with a single channel.

The computational grid (Figure 3) was generated using the IGG module of the NUMECA software. The blade surface adopted an O-type grid, the upstream and downstream sections of the blade adopted an H-type grid, the top clearance of the blade adopted a butterfly grid and the casing treatment adopted an H-type grid. When verifying mesh independence, the mesh structure remains unchanged and the flow field structure remains sufficiently equivalent when the grid number is greater than 2 million. In our final model, the grid number of the solid-wall fan was approximately 2.8 million, that of the circumferential groove casing treatment was approximately 500,000 and the y+ value was less than 3.
3. Numerical results and analysis

3.1. Flow field analysis of the solid-wall fan

The fan performance curve is shown in Figure 4. The parameters in the plot are dimensionless, and the mass-flow rate, pressure ratio and efficiency are the actual values divided by the values obtained at the design points of the solid-wall fan. The stall margin of the fan is 13.6%. Most of that stall margin is the pressure margin, and there is basically no flow margin with a flow margin of only 0.55%. These preliminary results show that enlarging the stall margin of the fan is quite essential.

![Performance curves of the solid-wall fan](image)

**Figure 4.** Performance curves of the solid-wall fan

We define the stall margin in Eq. (1):

\[
SM = \left[\frac{\pi_{K1}/m_{av}}{\pi_{K0}/m_{a0}} - 1\right] \times 100\% \tag{1}
\]

The mass-flow margin is expressed in Eq. (2):

\[
MFM = \left[\frac{m_{a0}}{m_{av}} - 1\right] \times 100\% \tag{2}
\]
\( m_{as} \) and \( \pi_{Ks}^* \) are the mass-flow rate and pressure ratio, respectively, of the fan near the stall point; \( m_{as0} \) and \( \pi_{K0}^* \) are the mass-flow rate and pressure ratio, respectively, of the rotor under the designed operating conditions.

**Figure 5.** Mach number contour at the blade tip of the solid-wall fan

First, we analysed the blade-tip flow field of the solid-wall fan. In Figure 5, the solid line indicates the shockwave position and the dotted line represents the trajectory of the leakage flow. In transonic rotors, leakage flow will pass through the shockwave at the leading edge of the blade tip. For the first stage rotor, moving from the design point to the near-stall-point plots, the position of the channel shock wave at the blade tip moves forward; however, the shockwave remains in the channel throughout the process. The leakage flow passes through the blade-channel shockwave near the pressure side of the blade and forms a low-velocity zone, and this low-velocity zone also exists at the design point. From the design point to the near-stall point, the low-velocity zone has no tendency to increase and no extensive blockage occurs at the channel; thus, the low-velocity zone cannot be the key factor causing the fan to stall. On the other hand, the leakage-flow trajectory does not significantly move forward from the design point to the near-stall point and the overall leakage-flow structure does not change. For the second-stage rotor, the position of the channel shockwave at the blade tips significantly moves forward from the design point to the near-stall point. In addition, at the near-stall point, the shockwave moves to the leading edge of the blade. This movement of the shockwave forewarns a stall. At the same time, the trajectory of the leakage flow certainly moves forward at the near-stall point so that the axial momentum of the leakage flow drops to such an extent that the leakage flow strikes the pressure side of the adjacent blades at their leading edges, blocking the flow through the channel of the blade tips. Based on the Mach number contour plot at the blade tips, we conclude that stalls begin at the second-stage fan.

3.2. *Stability enhancement analysis of circumferential groove casing treatment*

Because stalling begins at the second stage of the solid-wall fan, we tried many different schemes of circumferential groove casing treatments. By combining our preliminary simulations with the previous research on circumferential groove casing treatments [5-10], we determined the circumferential groove parameters, as listed in Table 1. Figure 6 shows that the circumferential groove casing treatment clearly
enhances the stability. The stall margin (SM) of the fan increases from 13.6% to 16.3%, whereas the mass-flow margin (MFM) of the fan increases from 0.55% to 3.28%. The effect of stability enhancement therefore meets the demands of real-world applications.

Comparing the two casings with the same mass-flow rate, Mach contours at the blade tips of the second-stage rotor operating near the stall point are shown in Figure 7. The solid line represents the shock position, whereas the dotted line represents the trajectory of the leakage flow. In general, when the fan functions stably, the shockwave at the blade tip stays within the blade channel. As backpressure gradually increases, the shockwave gradually moves forward; however, when the shockwave moves to the leading edge of the blade, the stability of the fan considerably decreases. Figure 7 shows that the blade-channel shockwave for the solid-wall fan has been pushed to the leading edge of the rotor under near-stall conditions. After the casing treatment, the channel shock wave is clearly located rearwards and the shockwave remains within the blade channel. Therefore, the backpressure of the fan can continues to increase and the stall margin of the fan can be widened. At the same time, the low-velocity zone caused by leakage around the solid-wall fan strikes the pressure side of adjacent blades at the 30% chord, blocking the mainstream flow. Despite the leakage smoothly flows through the blade passage, the leakage flow around the grooved-wall fan does not strike the pressure side of adjacent blade.

Figure 6. Performance curves for fans with solid walls and groove casing treatments

Figure 7. Mach contour at the blade tip of the second rotor

Figure 8 shows the static pressure distribution at the blade tips of the second-stage rotor. First, as the suction-side static pressure rises, the shock stagnation point for the solid-wall fan is clearly located farther forward than that of the grooved-wall fan, which is consistent with the conclusion obtained from
the blade-tip Mach contour map. Second, the static pressure for the solid-wall fan is higher than that for the grooved-wall fan, showing that the load at the blade tips of the solid-wall fan is greater than that for the grooved-wall fan. In addition, in the range of 0%–35% blade chord length, the static pressure difference between the pressure and suction sides of the solid-wall fan is significantly greater than that for the grooved-wall fan. The static pressure difference between the pressure and suction sides is the main driving force for the blade-tip leakage flow. Therefore, in the range of 0%–35% blade chord length, the leakage flow around the solid-wall fan is more serious than the leakage flow around the grooved-wall fan.

![Static pressure distribution at the blade tips of the second-stage rotor](image)

**Figure 8.** Static pressure distribution at the blade tips of the second-stage rotor

Figure 9 shows the static pressure distribution at the blade tips of the second-stage rotor and the radial velocity distribution in the circumferential grooves at four different moments in a rotational period. The background flow field represents the static pressure contour. The radial velocity distribution in the grooves (the upward direction is positive) clearly shows that the phenomena of ‘suction’ at the pressure side and ‘spray’ at the suction side occur. Although the suction and spray effects are different at different moments, they are mainly concentrated in the first five grooves around the casing. Figures 8 and 9 demonstrate that the circumferential grooves form a channel between the pressure and suction sides of the blade tip. The pressure difference between the pressure and suction sides is reduced by the suction and spray effects of the grooves.
Figure 9. Static pressure contours at the blade tip and radial velocity distribution in grooves

At the same flow point and under near-stall conditions with a solid-wall casing, the blade-tip leakage flow around the second-stage rotor is shown in Figure 10. The leakage-flow structure is divided into three types. Type-A leakage is normal leakage flow due to the existence of tip clearance. The Type-A leakage flow smoothly passes around the blades with the mainstream; this flow commonly occurs throughout the compressor process. Type-B leakage is a kind of leakage flow starting from the leading edge of the blade and flowing to the pressure surface of the adjacent blade before leaving the blade channel with the adjacent blade pressure surface. This leakage flow does not directly form the tip blockage but reduces tip-flow area so that the tip deflects the mainstream flow in the circumferential direction, causing axial momentum to decrease, at which point the tip-flow capacity considerably decreases. Further development of this cascade can lead to a stall. Type-C leakage is a cross-channel leakage flow that occurs at the leading edge of the blade. This leakage flow will pile up at the leading edge of the blade, blocking fluid passage around the leading edge, which causes the compressor to quickly enter the stall process. For a solid-wall casing, Type-B and Type-C leakages may obviously cause stalling. With a casing treatment, the blockage of the blade tips disappears only for the normal Type-A leakage. Therefore, circumferential slot casings improve the tip-leakage flow and enhance blade-tip flow capacity.

Figure 10. Leakage flow at the blade tips of the second rotor

3.3. Influence of the casing treatment on the performance of the fan at the design point (right branch)

We next compare the performance of the fan with a solid wall and that with the casing treatment (Figure 7). While widening the stability margin, the casing treatment has a negative impact on the right-branch performance of fan. For this fan, the design point is at the large-flow point of the right branch and the casing treatment reduces the design-point flow by 0.2%. The decrease of the flow at the design point reduces the engine cruise thrust, which is considerably detrimental to engine performance. Our analysis of the design-point flow field shows that the key factor reducing the fan flow is the reduction in the blockage at the tip of the second-stage rotor, as shown in the low-speed zone of Figure 11.
The blockage factor $B_m$ is a measure of flow clogging, and our research refers to the definition reported in Khalsa et al. [21]:

$$A_m = \iint (1 - \frac{\rho \omega}{\rho_e \omega_e})\,dA$$

(3)

$$B_m = \frac{A_h}{A_{exit}}$$

(4)

Where $\rho$ and $\omega$ are the local density and local axial velocity, respectively, of the blockage area; $\rho_e$ and $\omega_e$ are the density and axial velocity, respectively, at the boundary of the blockage area. $A_{exit}$ is the area of the measuring section, $A_h$ is the blocking area and $B_m$ is the blockage factor. Determining the clogging boundary is the key to analyzing the clogging. Sunder [22] provided a criterion based on the flow velocity gradient:

$$C_s = |\nabla_r (\rho v_m)| + |\nabla_\theta (\rho v_m)| \geq \delta.$$  

(5)

To ensure that the threshold for judging the boundary is uniform along the direction of the flow, we revise the judgment criterion with reference to the dimensionless treatment LiuBaojie has made to Eq. (5):

$$C_m = |\nabla_r (\frac{\rho v_m}{\rho v_{avg}})| + |\nabla_\theta (\frac{\rho v_m}{\rho v_{avg}})| \geq \delta$$

(6)

Where $v_{avg}$ is the average flow velocity for each section.

**Figure 11.** Blockage region with the second rotor at the design point

Figure 11 shows the flow-direction blocking distribution of the second-stage rotor with a solid-wall fan and a groove casing treatment fan at the design point. The red zone in the figure represents a blocking area that shows a clear boundary with the mainstream flow. It is evident from the figure that the clogging of the blade tip is not serious at the design point for the solid-wall fan and that the blockage area is
exactly the same as the trajectory of the leakage flow. This result clearly indicates that leakage flow caused the blockage. For the casing treatment fan, the blockage under design-point operation is more serious and the position of the blockage is different from that for the solid-wall fan. In this case, the blockage originates from the leading edge of the blade and expands downstream. In contrast to the radial velocity distribution in the slot shown in Figure 10, the blockage of the blade tip in the casing treatment fan operating at the design point is related to the radial ejection of the circumferential grooves near the suction surface. The ejected fluid blends with the mainstream flow at the blade tip, which hinders the mainstream flow and forms a low-velocity vortex, the area of which dramatically expands at the reverse pressure gradient.

Figure 12 shows a plot of the blockage factor vs. fan flow. For the solid-wall fan, the blockage at the design point and its right branch is relatively small and stable, but increases rapidly near the stall point. The casing treatment reduces the growth rate of blockage in the small-flow area, indicating that the casing treatment’s relief effect on the leakage-flow blockage in the small-flow area is much larger than the blocking effect on the mainstream flow due to the slot’s ejection. This balance of factors delays the fan stall. However, at the design point and right branch, the result is just the opposite, indicating that the blockage for the grooved-wall fan is larger than that for the solid-wall fan and that the flow at the design point decreases.

![Figure 12](image)

**Figure 12.** A plot of the blockage factor vs. normalised mass flow

Figure 13 shows the axial velocity distribution around the second-stage rotor’s entrance at the design point. The axial velocity of the casing treatment fan is obviously less than that of the solid-wall fan above 70% blade height. This result indicates that the blade-tip blockage caused by the casing treatment weakens the circulation of the second rotor above 70% blade height.

![Figure 13](image)

**Figure 13.** Wz velocity at the entrance of the second rotor
3.4. Improving the flow at the design point (right branch) by adjusting the stator

Figure 14. Partial angle adjustment scheme of the first-stage stator

For the two-stage fans, the opening angle of the upstream stator has a considerable impact on the performance of the fans. A local adjustment scheme for the angles of the first-stage stator is shown in Figure 14. The S70 scheme adjusts the first-stage stator above 70% blade height with a maximum adjustment angle of 3°, whereas the S40 scheme adjusts the first-stage stator above 40% blade height with a maximum adjustment angle of 5°.

Figure 15 shows the performance curves for the solid-wall fan, along with grooved-wall fans operating with the scheme of not adjusting the stator (S0) and schemes S70 and S40 for stator adjustment. The stability and flow margins for the S0 scheme are 16.3% and 3.31%, respectively. The stability and flow margins for the S70 scheme are 16.5% and 3.04%, respectively. The stability and flow margins for the S40 scheme are 17% and 2.856%, respectively. With increasing stator adjustment range and angle, the fan-flow margin decreases; however, the stability margin increases. Although stator adjustment negatively affects the flow margin, it will increase the pressure ratio for the stall point, which increases the compression ratio margin, thereby increasing the overall stability margin.

From a comparison of the design-point flow with the upstream stator adjustment, as shown in Figure 15, it can be observed that the S70 stator adjustment scheme increases the design-point flow; however, its overall flow is still less than that for the original solid-wall fan. Furthermore, the S40 stator adjustment scheme increases the design-point flow to the level of the original solid-wall fan. Considering the effect of stator adjustment on the stability margin of the fan and the design-point flow, the S40 stator adjustment scheme is an optimal scheme for the stable operation of the fan.

Figure 15. Performance curves for partial angle adjustment of the first-stage stator with casing treatment

For the condition that the flow of the fan is same as the design-point flow, the inlet axial velocity at the second-stage rotor is distributed along the radial direction with a circumferential groove casing, as
shown in Figure 16. The figure shows that the local adjustment of the first-stage stator does enhance the flow capacity of the upper part of the second-stage rotor. Comparing this result with Figure 14, we conclude that the local adjustment of the upstream stator can lessen the adverse impact of the casing treatment on the design-point flow.

Figure 16. Axial velocity distribution before the second rotor

4. Conclusion

1. To extend the stable operating range of a multistage fan or compressor, the key stage initiating a stall must be determined first and an appropriate casing treatment should be applied around the key stall stage.

2. A channel is formed at both sides of the blade by a circumferential groove casing, and through suction on the pressure surface and spray on the suction surface, the static pressure difference between the pressure and suction surfaces is reduced, which improves the tip leakage flow and effectively inhibits the blade-tip blockage in the small-flow area, thereby increasing the stability margin of the fan.

3. Circumferential groove casings can improve the stability; however, they adversely affect the performance of the fan at the design point. Circumferential grooves cause serious blockage at the blade-tip area when operating at the design point and along the right branch of the performance curve. This blockage weakens the flow capacity of the upper part of the rotor so that total flow at the design point is reduced and the engine thrust is adversely affected.

4. Partial adjustment of the upper part of the upstream stator can enhance the flow capacity of the upper part of the downstream rotor and compensate for the casing treatment’s adverse effect on the design-point flow. The combined application of the casing treatment and a locally adjusted upstream stator can extend the stability of an engine while preserving the overall performance.

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