INFLUENCE OF ENDWALL ASPIRATION ON AERODYNAMIC PERFORMANCE IN COMPOUND LEAN COMPRESSOR CASCADES

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ABSTRACT
The application of compound lean cascades in conjunction with endwall aspiration configurations is discussed in this paper. A thorough study on the reciprocal effect of blade lean angle, aspirated flow fraction and aspiration slot start position is conducted by a numerical method. The calculations show that the application of blade corner fillet has a significant effect on the parameter choice of endwall aspirated lean blade design. The fillet application makes the blade suction surface curvature increased near the endwalls, and gets the endwall slots farther away from the real blade surface, thereby aggravating the spillage in the forepart of the endwall slot and leading to the deterioration of corner blockage and loss, especially in the positive lean cascades. Consequently, the optimal aspirated lean design around 0 deg lean angle is divided into a negative lean and a positive lean design by fillet application. This influence can be inhibited dominated by moving the endwall slot start position upstream. In aspirated lean cascades, blade positive leaning can enhance the radial secondary along the blade suction surface, balance the radial distribution of boundary layer by stronger radial pressure gradient to transfer a portion of corner low momentum fluid towards the midspan, resulting in a more obvious increase in blockage in the midspan. Blade negative leaning converges the low momentum fluid towards the suction surface corner by enhanced pitchwise pressure gradient, so as to facilitate the endwall aspiration, and eliminate mixing loss remarkably after the trailing edge, more significant than the decrease of profile loss in the blade passage. The optimal aspirated lean angle in main flow is around -20 deg in this adopted compressor cascade. Higher aspirated flow fraction cannot affect the spanwise and pitchwise secondary, and is limited only to suppress the corner blockage further, hardly affects the midspan blockage. The effect of aspiration start position almost can only be found in the blade passage, unafect the mixing loss after the trailing edge. Especially in negative lean cascades, moving aspiration slot start position more upstream can linearly increase the real suction area and contribute to the decrease in aspirated flow loss and suction power needed.

INTRODUCTION
Aspiration has the potential to further improve blade loading with higher flow turning and acceptable profile loss in the rear stages stators of a multi-stage axial compressor. Since the concept of active flow control techniques to control separations in compressor stators was first proposed by Loughery et al.[1]. Aspiration or bleed has long been investigated to eliminate the design limitation of compressor stages further. From the end of the 90s of the last century to the beginning of this century, Kerrebrock and Merchant et al.[2-4] from MIT carried on a systematic research on the design of aspirated compressors and the application of aspiration to control flow in highly loaded compressors. Later, a series of aspirated axial, or even counter-rotating compressors have been demonstrated by experiments to verify the feasibility of aspiration to enhance blade loading[5-7].

Limited to the structural reliability and added manufacturing cost, aspirating became more attractive as an approach to control 3D corner separation in stators. Godard et al.[8, 9] developed a 2D subsonic cascade design method, combing blade shaping and blade suction surface aspiration. Validated by calculations and experiments, a highly loaded cascade with approximately 65 deg flow turning can be realized by only 3.3% of inlet mass flow. Later, Power[10] carried out a numerical and experimental investigation on a high subsonic aspirated cascade with diffusion factor more than 0.7, enabling a novel pressure-rise mechanism for stator design with boundary layer aspiration on blade suction surface. Compound suction configurations of blade suction surface and endwall aspiration have been developed in recent years, the capacity to control corner separation and eliminate corner stall have been further enhanced[11-13], and the internal flow of aspiration plenum has also been studied initially[14].
However, application of aspiration slots to the stators may not be possible on blade suction surface due to the size of the blade. It would be easier to apply flow control to the endwalls of the blade passages in this instance, and this approach is considered in Gbadebo’s [15]. Gbadebo et al. [15] show that significantly smaller aspirated flow fractions are achievable to remove the 3D separation in the corner entirely. To date, few effort has been focused on improving aspiration performance via passive flow control techniques. Song et al. [16] conducted numerical investigations on different suction surface slot configurations in compound lean cascades. But the study rarely has conclusions about the synergism of aspiration and blade leaning. While in Liesner et al. [17], passive flow control like boundary layer fences is employed to lower aspirated flow fraction to improve cascade performance.

The alteration of pressure gradient on the blade suction surface especially near the endwalls makes the design of compound lean and endwall aspiration interact with each other. And the reciprocity and synergism between these flow control techniques are the focus of this paper.

FLOW CONFIGURATION

A highly loaded parallel-annulus cascade with a camber angle of 60 deg is utilized as the baseline of this study. Previous experimental results showed that the peak diffusion factor at the hub is over 0.60 and the De Haller number is equal to 0.74. The geometry and inlet aerodynamic conditions of the baseline cascade are shown in Table 1.

![Figure 1. Stacking line of compound lean cascades.](image1)

The endwall aspiration slots adopt in this study start from three different streamwise locations to 98% chord length. The three slot start positions (ssp) are 1) the position just downstream of the peak Mach number, 2) the separation point on the blade suction surface in the straight baseline cascade, and 3) 20% chord length downstream of the separation point. The slots are configured adjoined to the corner fillet and parallel to the blade suction surface.

![Figure 2. Main passage clustering and endwall aspiration slot grid, ssp=0.46.](image2)

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NUMERICAL METHODS

The computational grids were generated by AutoGrid™ and IGG™ from NUMECA™, as shown in Figure 2. An O4H block topology was adopted for the main flow passage, including an O-grid block around the blade. To simulate the real flow in the compressor cascades, corner fillets were involved in the computational domains, and the radii of them were set to be 5% of the blade chord length. The aspiration slots were on both the shroud and hub endwalls, and two meridional control lines were placed to axially cluster the grids to locate the axial position of the aspiration slots. Axial inlet and exit grid planes were placed about one axial chord length upstream of the leading edge and two axial chord lengths downstream of the trailing edge respectively.

| Table 1. Compressor cascade geometry and inlet aerodynamic parameters. |
|--------------------------|--------------|
| **Blade geometry**       |              |
| Profile                  | NACA65-24A10 |
| Chord                    | 100mm        |
| Solidity                 | 1.25         |
| Aspect ratio             | 1.0          |
| Blade camber angle       | 60 deg       |
| **Inlet aerodynamic parameters** |            |
| Reynolds number          | 5.27×10^5    |
| Mach number              | 0.23         |

To exclude the interference of blade sweep, the lean blade investigated here is defined as the leaning normal to the airfoil section chord line. The lean stacking line mentioned in this paper can be defined as two parts, as shown in Figure 1, two NURBS-line segments at both ends, similar to the Ref. [18]. The spanwise height of the joinpoint of NURBS-lines is defined as the lean length \( h_b \). The lean angle \( \beta_b \) is defined as being the angle made between the endwall normal direction and the NURBS-line segments at both ends. As shown in Figure 1, the suction surface forms an obtuse angle with the endwall while the lean angle is positive.

A first cell wall spacing of 1×10^4 meters was applied to all blocks to maintain the average values of \( y^+ \) below 1. With such mesh topology employed, a grid sensitivity study was performed to quantify the numerical uncertainties of the flow phenomena in the investigated annulus cascade. Almost no
The modification of the results was observed when further refined the mesh employed in this paper. The numerical simulations were performed with the finite volume technique based flow solver CFX™ from ANSYS™. The high-resolution scheme embedded in CFX was employed for the advection scheme and turbulence modeling. The k-ε turbulence model provided in CFX/Solver was used to deal with the large-scale separation in high-turning compressor cascade, referring to Ref. [19, 20].

Experimentally measured profiles of stagnation pressure, stagnation temperature from experimental inlet sections and a simple radial equilibrium static pressure condition were adopted to define the aerodynamic conditions of the simulations. Axial and pitchwise velocity components were assigned to produce a desired inlet flow angle. The aspirated flow fraction was specified as a fraction of the cascade inlet mass flow, and a specified aspirated flow fraction was obtained by adjusting the static pressure of the aspiration slot exit.

Compared to the experimental results in Figure 3, numerical separation lines and other flow field details were approximately in good agreement with the measurement, giving confidence in the use of numerical method adopted in this paper to draw conclusions about details of the flow.

RESULTS AND DISCUSSION

Several metrics are adopted to judge whether the application of blade leaning and aspiration is useful to control corner separation and improve the aerodynamic performance of compressor cascades.

Stagnation Pressure Loss Coefficient. The stagnation pressure loss coefficient of aspirated cascade is designed as the stagnation pressure difference between the inlet and both main flow path and aspiration slot exit divided by the inlet dynamic pressure head of the main flow path. The stagnation pressure loss coefficient is given as

\[
\omega_p = \frac{m_2 \times (p_i' - p_s') + m_s \times (p_i' - p_s')}{m_1 \times (p_i' - p_i)}
\]  

(1)

To isolate the effect of aspiration from the internal flow of the slot, the relevant parameters of aspiration is extracted at the slot inlet on the endwall surface.

Figure 4 shows the variation of stagnation pressure loss coefficient with blade lean angle and aspirated flow fraction at different aspiration slot start positions. Response surface methodology is used to describe the sample space and obtain the optimization strategy for the design parameters of an aspirated lean cascade. As shown in Figure 4, three different aspiration slot start positions are investigated in this paper. The aspiration slot in Figure 4.left starts just downstream of the peak suction on the blade, the aspiration slot in Figure 4.mid originates from the saddle point in the suction corner, and the last slot is entirely configured in the separation region. There exist two optimization designs while the aspiration slot starts downstream of separation start point, but only one negative lean optimal design can be found in the Figure 4.left.
The optimal aspirated positive lean angle is always around the optimal unaspirated blade lean angle (about 25 deg positive lean angle). And the optimal negative lean design changes with the slot start position. With the aspiration slot start position moves downstream, the magnitude of the optimal negative lean angle gets bigger and less aspirated flow is needed to minimize the stagnation pressure loss. Meanwhile, the unfavorable effect of slot spillage is suppressed, since the aspiration slot is completely in the separation region.

Figure 4. Response surface of lean angle $\beta_b$ and aspirated flow fraction $\kappa$, $i=0$ deg. Left: six-order RSM, $R^2=0.9931$, ssp=0.26. Middle: seven-order RSM, $R^2=0.9868$, ssp=0.46. Right: eight-order RSM, $R^2=0.9993$, ssp=0.66.

The appearance of two optimal aspirated lean designs at the same aspiration position is due to the application of fillet in the blade corner. There is only one optimal aspirated lean design in the cascades without fillet while the aspiration slot is arranged at the same position, as shown in Figure 5, and the optimal aspirated lean angle is around 0 deg.

The introduction of blade corner fillet has a noticeable impact on both the main flow and aspirated flow loss. The optimal aspirated negative lean angle around -20 deg in the main flow path is almost unaffected, but the stagnation pressure loss in the positive lean side of Figure 5.mid is raised with different blade lean angles. At the same time, the aspirated flow loss contour shape alters significantly in the positive lean side of Figure 5.right, and the most obvious sign is the migration of the minimal aspirated flow loss point.

Figure 5. Variation of stagnation pressure loss with lean angle $\beta_b$ and aspirated flow fraction $\kappa$, $i=0$ deg, ssp=0.46.
However, as shown in Figure 4, all the alteration of the main flow and aspirated flow by applying corner fillet will be inhibited dominated by moving the endwall slot start position upstream.

**Blockage.** To evaluate the three-dimensional flow field in the main flow path. Khalid’s[21, 22] definition for endwall blockage is introduced to quantify the blockage region found in the endwall region of compressors. The concept of boundary layer displacement thickness is introduced to define the endwall blockage, which can be written as

\[ A_{blo} = \int_{A} \left[ 1 - \frac{\rho v_m}{\rho_{\beta} v_{\beta}} \right] dA \]  

(2)

The quantity \( v_m \) is the velocity component in the main flow direction, \( v_{\beta} \) is the mass averaged velocity in the core region, and the integral is taken over the defect region. Specifically, the edge criterion is based on \( \nabla (\rho v_m) \), as the cascade exit flow is designed to turn to the axial conditions in this study, the velocity component perpendicular to the axial direction can be used to calculate the blockage area in a transverse section downstream from the trailing edge. Referring to Khalid’s[21, 22], as shown in Figure 6, the criterion of the boundary layer edge is taken as

\[ \nabla (\rho v_m) / (\rho v_{1,m} / c) = 2 \]  

(3)

The application of blade corner fillet directly induces the variation of corner blockage in compressor cascades. Referring to Taylor’s[23], the effect of blade leaning can be seen most significantly in the central 60% mass flow which is bounded by two stream surfaces at the 20% and 80% spanwise locations, as shown in Figure 6.left. And each 20% span near the endwalls can be seen as the corner and endwall region, the corner blockage adopt in this paper is defined as the integration of these regions. Figure 7.upper-left two contours present the variation of corner blockage in aspirated lean cascades with and without fillet application. To distinguish the effect of the fillet application on corner blockage and main flow loss associated, Figure 7.upper-right and lower-left present the corner blockage and main flow loss increased by fillet application. The main flow stagnation pressure loss is obviously raised by the higher corner blockage, especially as the aspirated flow fraction is comparatively small or the blade lean angle is positive.

![Figure 6. Definition of blockage.](image)

![Figure 7. Variation of aerodynamic performance with lean angle \( \beta_b \) and aspirated flow fraction \( \kappa \) with the application of fillet, \( i=0 \deg, ssp=0.46 \).](image)

While the aspirated flow fraction is comparatively small, the main flow loss and the aspirated flow loss is increased
remarkably, due to the deterioration of the unfavorable effect of spillage in the forepart of the endwall slot. As shown in Figure 7.lower-right, more spillage increases the mixing loss in the main flow path with the application of corner fillet.

As can be seen in Figure 8, the application of blade corner fillet makes the curvature of blade suction surface increase near the endwalls. Meanwhile, the endwall slots adjoined to the blade suction surface get farther away from the real blade surface. The synergy between blade leaning and corner fillet makes the corner blockage increase as can be seen in Figure 8. Therefore, the synergistic effect forms a visible region of loss deterioration as shown in Figure 7.lower-left from -20 deg lean angle with small aspirated flow fraction to +40 deg lean angle with more aspirated flow.

Some typical aspirated lean designs in Figure 4 are chosen to comparatively study the design parameters such as the blade lean angle and aspirated flow fraction. The BSL case in Figure 9 is the unaspirated straight cascade calculated as the baseline experiment. The two cases in the middle of Figure 9 represent the positive and negative lean cascade with comparatively small aspirated flow fraction. And the main flow optimal aspirated lean design is shown in Figure 9.right.

In unaspirated cascades, the positive lean blade balances the radial distribution of boundary layer by stronger radial pressure gradient to transfer a portion of corner low momentum fluid towards the midspan of the suction surface, the decrease in loss is due to the decrease of blockage in corners. Meanwhile, the negative lean blade converges the low momentum fluid towards the suction surface corner, the increase in loss of unaspirated negative lean blade is also due to the increase of blockage in corners.

The endwall aspiration slots utilized in this paper can only remove a part of the low momentum fluid close to the slots. Different lean designs will affect the suction capacity of the endwall slots in aspirated lean cascades, as shown in Figure 9.mid. With a positive lean angle, aspiration can only remove the remainder of low momentum fluid redistributed by the stronger pressure gradient in corners. And in negative lean cascades, aspiration can remove more low momentum fluid gathered from midspan by blade negative leaning. With higher aspirated flow fraction, aspiration in negative lean cascades is limited only to weaken the corner blockage, not the midspan blockage further.
**Secondary Kinetic energy.** The local velocity component perpendicular to the main flow direction is used to define both the blockage and SKE (secondary kinetic energy) in this paper. To assess the development of secondary in different cascades, the local SKE is divided by the blade inlet dynamic pressure head.

\[
SKE = \left( \rho v_{sec}^2 / 2 \right) / Q_{in}
\]

(4)

As shown in Figure 10, aspiration can effectively control the secondary in compressor cascades. To verify the radial migration of the low momentum fluid in lean cascades, positive lean makes the blade radial secondary enhanced, and negative lean makes the blade pitchwise secondary enhanced near the endwalls. Higher aspirated flow fraction can only make the secondary enhanced near the slots region, unaffected the spanwise and pitchwise secondary.

![Figure 10. Contours of SKE in exit blockage region.](image)

To quantify the blade leaning effect on midspan and corners, integral blockage and stagnation pressure loss coefficient are taken at three different spanwise regions as shown in Figure 6. As given in Figure 11, less aspirated flow in positive lean cascades can make the corner blockage and loss at the same level as in the negative lean cascades with much higher aspirated flow fraction. But the increase in blockage in the midspan of positive lean cascades is the key issue of the increase in loss. More aspirated flow in negative lean cascade can further suppress the corner blockage and the loss associated but unaffected the midspan.

![Figure 11. Variation of blockage and exit loss component](image)

Figure 12 shows the axial development of stagnation pressure loss of typical aspirated lean cascades of different
blade lean angles and different aspiration start positions. Before the aspiration start position at 0.46 of the axial chord length from the leading edge, as shown in Figure 12.left, there is a little decrease in loss of the positive lean cascade, and a little increase in loss in the negative lean cascades. After the aspiration start position, less low momentum accumulation in the corners results in a decrease in loss, especially in the negative lean cascade. And the decrease in mixing loss after the trailing edge is remarkable in the aspirated negative lean cascade, more significant than the decrease of profile loss in the blade passage. The effect of aspiration start position can be seen in Figure 12.right. The difference almost can only be found in the blade passage, unaffected the mixing loss after the trailing edge.

Static pressure coefficient. Static pressure coefficient is introduced to quantify the suction power required to remove the high entropy fluid. It is defined as the static pressure difference between the main flow inlet and aspirated flow inlet divided by the inlet dynamic head of the main flow path.

\[
c_p = \frac{p_1 - p_a}{p_1 - p_t}
\]  

The magnitude of the static pressure coefficient represents the suction power required for carrying the aspirated flow from the main flow into the aspiration slot. Smaller static pressure coefficient indicates less suction work consumed.

A comparison of the characteristics in typical aspirated lean cascades can be found in Table 2. The different mechanism makes the positive aspirated cascade has less stagnation pressure loss in the aspirated flow, and less suction power consumed. In the negative lean cascades, the variation of aspiration start position makes a clear influence on the aspirated flow loss and suction work consumed. The earlier the aspiration starts in the axial direction, the longer the slot is. This appearance will effectively increase the real suction area and contribute to the decrease in aspirated flow loss and suction power needed.

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<th>(\kappa)</th>
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<th>(c_p) (Hub)</th>
<th>(\omega_{af}) (Shroud)</th>
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CONCLUSIONS
In this paper, we have numerically studied the effect of endwall aspiration on lean cascades. With the introduction of aspiration, the penalties to the cascades performance which result from handling the aspirated flow must be included in the cascade performance. The balance between the decrease in main flow loss and the penalties of the aspirated flow finally forms an optimal aspirated lean design. More detailed observations and conclusions about this research are enumerated below.

1) Fillet. The appearance of two optimal aspirated lean designs at the same aspiration position is due to the application of fillet in the blade corner, and there is only one optimal aspirated lean design exist which is around 0 deg lean angle in cascades without fillet. The optimal aspirated lean angle around -20 deg in the main flow path is almost unaffected. But the application of fillet makes the blade suction surface curvature increased near the endwalls. Meanwhile, the endwall slots adjoined to the blade suction surface get farther away from the real blade surface. Therefore, the synergistic effect between blade leaning and corner fillet forms a visible region of blockage and loss deterioration, especially in the positive lean cascades. While the aspirated flow fraction is comparatively small, the main flow loss and the aspirated flow loss is increased remarkably, due to the deterioration of the unfavorable effect of spillage in the forepart of the endwall slot. However, all the alteration of the main flow and aspirated flow by applying corner fillet will be inhibited dominated by moving the endwall slot start position upstream.

2) Lean angle and aspirated flow fraction. The blade positive lean can enhance the radial secondary along the blade suction surface, balance the radial distribution of
boundary layer by stronger radial pressure gradient to transfer a portion of corner low momentum fluid towards the midspan. And endwall aspiration can only remove the remainder of low momentum fluid in corners. The increase in blockage in the midspan is the key issue of the increase in loss. Negative lean makes the blade pitchwise secondary enhanced near the endwalls and converges the low momentum fluid towards the suction surface corner. Aspiration can remove more low momentum fluid gathered from midspan. Before the aspiration start position, the variation of loss in lean cascades is negligible. But after the aspiration start position, less low momentum accumulation in the corners results in a decrease in loss, especially in the negative lean cascades. And the decrease in mixing loss after the trailing edge is remarkable in the aspirated negative lean cascades, more significant than the decrease of profile loss in the blade passage. Higher aspirated flow fraction can only make the secondary enhanced near the slots region, unaffect the spanwise and pitchwise secondary. With more aspirated flow, aspiration in negative lean cascades is limited only to suppress the corner blockage and the loss associated further but unaffect the midspan.

3) **Aspiration start position.** The effect of the aspiration start position variation can only be found in the blade passage, hardly affect the mixing loss after the trailing edge. In the negative lean cascades, the variation of aspiration start position makes a clear influence on the aspirated flow loss and suction work consumed. The earlier the aspiration starts in the axial direction, the longer the slot is. This appearance will effectively increase the real suction area and contribute to the decrease in aspirated flow loss and suction power needed.

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**NOMENCLATURE**

- \( \alpha \) = flow angle
- \( \beta \) = metal angle
- \( \delta \) = boundary layer thickness
- \( \delta^* \) = displacement boundary layer thickness
- \( \kappa \) = aspiration flow fraction
- \( \omega \) = loss coefficient
- \( \rho \) = density
- \( c \) = chord length
- \( c_p \) = static pressure coefficient
- \( i \) = incidence
- \( p^* \) = stagnation pressure
- \( p \) = static pressure
- \( r \) = radius
- \( s \) = distance perpendicular to the surface
- \( ssp \) = slot start position
- \( v \) = velocity
- \( Q \) = dynamic pressure

**SUBSCRIPTS**

- \( 1 \) = inlet
- \( 2 \) = exit
- \( af \) = aspirated flow
- \( b \) = lean geometry
- \( blo \) = blockage
- \( d \) = defect region
- \( fs \) = free stream
- \( m \) = main flow direction
- \( p \) = with penalties from handing aspirated flow
- \( sec \) = secondary
- \( se \) = aspiration slot exit
- \( si \) = aspiration slot inlet

**Abbreviations**

- BSL = baseline
- CFD = computational fluid dynamics
- EXP = experiment
- LE, TE = leading and trailing edge
- RSM = response surface model
- SS, PS = suction and pressure surface
- SKE = secondary kinetic energy

**REFERENCES**


