Numerical investigation of centrifugal compressor stability enhancement using vaned diffuser endwall contouring technology

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Abstract
To improve the stable operating range of centrifugal compressor stage, endwall contouring technology is introduced on the hub-side wall of the vaned diffuser for NASA CC3 centrifugal compressor as a promising technique to redistribute the flow field near the endwall in this paper. The main contents of this paper are listed as follows. First, a specific parameterized endwall contouring guideline is developed based on sinusoidal function and Bézier curve, and three factors are considered, namely, peak height, peak radial position, and frequency coefficient. Then, centrifugal compressor stages with baseline diffuser and contoured endwall diffusers are numerically investigated to reveal the influence of endwall contouring on the stable operating range. Results show that endwall contouring is an effective method to achieve stability enhancement with no evident reduction in stage performance on the design point. CEW13-12 has the most successful combination of endwall contouring parameters, and stall margin is increased by up to 40% for CEW13-12. Finally, the influence mechanism of endwall contouring on the stable operating range is discussed. At the near stall condition, the contouring endwall forces the fluid within the semi vanless space to deflect toward the suction side to increase the momentum of low-energy fluid in the suction surface boundary layer and reduce the adverse pressure gradient on the suction side, which delay the development of flow separation on the vane leading edge near the shroud side to suppress the onset of rotating stall. At the near choke condition, the contouring endwall has more potential to increase the throat area, resulting in a decrease of throat blockage to improve choke margin. The results show that the endwall contouring technology is a reliable method to achieve stability enhancement of centrifugal compressor stage.

Abbreviations: B1–B8, circumferential cut profiles on contouring endwall; Baseline, original case; CEW, contouring endwall; LE, leading edge; OP, operating point; PS, pressure side; R1–R9, radial points on each cut profile; SS, suction side; TE, trialing edge.
1 INTRODUCTION

Highly loaded centrifugal compressors play a critical role in the state-of-the-art industries of metallurgy, petrochemical, natural gas transportation, refrigeration, and power, due to the advantages of simple structure, high single-stage pressure ratio, and broader stable operating range. To satisfy the high-pressure ratio and high-efficiency requirements, modern centrifugal compressors utilize high-pressure ratio stages with vaned diffusers.\(^1,2\) However, compared with centrifugal compressors with vaneless diffusers, the centrifugal compressor with vaned diffusers causes the stable operating range to decrease substantially, and the peak efficiency to move toward the surge limit. Therefore, the main challenge faced by many researchers is the way to reach a broad operating range while simultaneously meeting the high-pressure ratio requirements throughout the entire centrifugal compressor map.

The stable operating range of the centrifugal compressor is dominantly limited by aerodynamic instability, known as rotating stall or surge at a lower mass flow rate and choke at a higher mass flow rate. Numerous studies on centrifugal compressors with vaned diffusers have focused on the mechanism analysis causing aerodynamic instability. The existence of spike or modal stall precursors before surge was discovered numerically and experimentally in a centrifugal compressor by Spakovszky,\(^3\)–\(^5\) which indicated flow separation at the vane leading edge (LE) because the high incidence near the shroud side induced stall inception. Furthermore, the stability for centrifugal compressors with vaned diffusers was mainly determined by the diffuser vaneless and semi vaneless space because the stall inception occurs in this region.\(^6\)–\(^8\)

Therefore, numerous control methods\(^9\) have attempted to alter the flow field within the diffuser inlet region to enhance the operating range of centrifugal compressors with vaned diffusers near the surge condition. According to Skoch,\(^10\) the forward-tangent and reverse-tangent air injection techniques applied to the shroud surface in vaneless space of the NASA CC3 compressor could improve surge margin but decrease pressure ratio, and the improvement in stable operating range was a consequence of decreasing the average swirl angle along the spanwise direction. Similarly, Skoch\(^11\) applied air injection technology on the hub-side wall of the diffuser, but no notable improvement in surge margin was shown. Unexpectedly, the injector nozzles without air injection improved a small amount of surge margin, and the nozzles became obstructions to affect the flow field in the vaneless space. Evidence showed that the control method must act upon the flow field within the vaneless space of diffuser to improve stability. Additionally, Marsan\(^12\) proposed a multi-slot boundary-layer suction technique on the radial vaned diffuser of a centrifugal compressor, which achieved a remarkable increase in stable operating range by reducing the interaction between the hub-corner separation vortex and the impeller trailing edge vortex shedding. As noted above, active control methods require additional energy input and need to be equipped with complex configurations.

To avoid the disadvantages of active methods, passive control methods have been proposed to enhance the stable operating range of centrifugal compressor stages. As noted by Galloway et al.,\(^13\) the application of porous throat diffuser with a common side cavity improved compressor stable range by up to 26.8% in a high-pressure ratio centrifugal compressor. This study proved that the porous throat diffuser equalized the throat pressure to create a more homogeneous circumferential pressure distribution, which was beneficial to improve the compressor stability. Then, in a high-pressure ratio turbocharger compressor stage, a second study by Galloway et al.\(^14\) used a vaned diffuser recirculation technique on either hub or shroud endwalls to recirculate air from the latter part of the diffuser passage to the diffuser inlet. The results showed that 58% of the stable operating range was gained over the baseline case by the recirculation configuration on the diffuser shroud side due to the increase of diffuser LE incidence angle near the shroud side. The other passive methods raise the stable operating range by optimizing the blade geometry or the passage geometry. Gunadal and Govardhan\(^15\) obtained 19.3% of stall margin enhancement by using 6° leaned vanes in the low-soldiety vaned diffuser of a centrifugal compressor stage. The diffuser with leaned vanes accelerated the flow in the diffuser inlet region and reduced the losses of incidence angle, which weakened flow separation and delayed the onset of rotating stall. Recently, nonaxisymmetric endwall contouring technology proved its effectiveness in turbines\(^16,17\) and axil compressors\(^18–20\) to suppress the secondary flow and flow.

**KEYWORDS**
centrifugal compressor, endwall contouring, flow separation stability enhancement, vaned diffuser
separation. Zhou et al.\textsuperscript{21} performed a numerical optimization on non-axisymmetric contouring endwall for a vaned diffuser hub-side wall of a centrifugal compressor, which showed the total pressure loss of the diffuser decreased under design condition, but the stable operating range of compressor stage decreased as well. Hermann et al.\textsuperscript{22} utilized a hub-side endwall contouring technology on the vaned diffuser inlet region of an open impeller centrifugal compressor stage. As a result, the stable operating range of the compressor stage was expanded by 8%.

Past studies clearly indicated a relationship between controlling the flow separation in the diffuser inlet region and enhancing the stable operating map of centrifugal compressors. Moreover, the above existing literature show that the endwall contouring technology can improve stage performance or enhance the stable operating range of the centrifugal compressor stage. No specific endwall contouring method exists, and a comprehensive understanding on the control mechanism is not yet available. In the present paper, the first objective is to develop an effective, specific parameterization endwall design method to produce different endwall concave and convex patterns. The second objective is to reflect the effect of the contoured diffusers on the stable operating range of the centrifugal compressor stage numerically. And the third objective is to provide an insight into the flow control mechanism on stability enhancement.

2 | CASE DESCRIPTIONS

The NASA CC3 high-speed centrifugal compressor designed by Allison Engine Company is used for the present paper. Diagrams of NASA CC3 compressor with wedge diffuser are depicted in Figure 1. The left panel of Figure 1 shows the model diagram. The right panel of Figure 1 displays the meridional flow path and the aerodynamic parameter measurement station, and station B represents the measurement position of flow parameters at the outlet of the wedge diffuser. Table 1 shows the specific geometric and aerodynamic design parameters of the NASA CC3 compressor reported by McKain and Holbrook.\textsuperscript{24}

3 | NUMERICAL METHODS AND VALIDATION

3.1 | CFD method

The NASA CC3 centrifugal compressor stage with wedge diffuser is investigated numerically using ANSYS CFX software. For the sake of saving the computation cost, single-passage numerical simulations are performed on the assumption of periodicity. A rotating impeller passage (a main blade and a splitter blade) and a stationary diffuser passage make up the computational

| Parameter                           | Value  |
|-------------------------------------|--------|
| Design mass flow (kg/s)             | 4.54   |
| Design rotating speed (r/min)       | 21,789 |
| Total pressure inlet (Pa)           | 101,325|
| Total temperature (K)               | 288    |
| Design total pressure ratio         | 4      |
| Impeller inlet tip diameter (mm)    | 210    |
| Impeller exit diameter (mm)         | 431    |
| Impeller back sweep angle (°)       | 50     |
| Numbers of impeller blades          | 15 full + 15 splitter |
| Diffuser divergence angle (°)       | 7.791  |
| Diffuser exit diameter (mm)         | 714    |
| Number of diffuser vanes            | 24     |
| Diffuser vane height (mm)           | 17     |
domain, as shown in Figure 2. The inlet section is extended upstream of the impeller in the axial direction to obtain the stead, uniform axial inflow field, and the outlet section is properly extended downstream of the diffuser in the radial direction to weaken the effect of the outlet boundary condition on the outflow field. AUTOGRID is utilized to generate the structured computational grid. The specific computational mesh is shown in Figure 3. Obtaining the accurate solution into the boundary layer is crucial, especially in the case of compressible flows. Thus, finer grids near the blade surfaces and the solid walls are necessary to ensure that the $y^+$ of the first-layer wall mesh is less than 2. At the same time, the mesh around the impeller–diffuser interface is refined to capture the flow field near the rotor–stator interface as accurately as possible.

3D compressible steady RANS equations are solved using the commercial element-based finite volume CFX code. The SST (shear-stress transport) turbulence model is adopted to ensure highly accurate predictions of flow separation under adverse pressure gradients in the impeller and diffuser channels. High-resolution discrete scheme is applied on the advection term to improve calculation accuracy. The total pressure inlet boundary condition is defined as the axial inflow, constant total pressure (101,325 Pa), and total temperature (288 K) at the inlet of the impeller. At the near choke condition, such that the outlet boundary condition uses the average static pressure at the high flow rate. At the near surge condition, the performance curve is relatively gentle, and pressure ratio is insensitive to mass flow rate. The mass flow rate boundary condition is assigned at the diffuser outlet at the low flow rate. The solid walls in the computation domain are assumed to be no-slip, adiabatic walls. The periodic boundary condition is applied on the circumferential boundary. The impeller–diffuser interface employs the stage interface model to transfer flow information between the rotating row and the stationary row. The convergence criteria used for steady simulation are that the RMS residuals fall below $10^{-4}$, and the fluctuations in total-to-total adiabatic efficiency and inlet mass flow rate are less than 0.05% and 0.01 kg/s, respectively. In this paper, with decreasing mass flow rate, stall is deemed to be achieved whenever the numerical simulations diverge, and the last operating point before stall is regarded as the near stall operating condition.

### 3.2 Numerical method validation

To verify the accuracy of the numerical method, first, a grid-independent analysis is performed on an identical geometrical model to assess the relationship between mesh size and simulation precision. Five different mesh resolutions are adopted, namely, very coarse (0.7 million), coarse (1.17 million), moderate (2.33 million), fine (3.05 million), and very fine (4.2 million). The variation tendency of the average total pressure loss coefficient of the wedge diffuser with the mesh size is shown in Figure 4. The total pressure loss coefficient of the wedge diffuser is calculated as follows:

![Diagram of computational domain.](image1)

![Diagram of computational mesh.](image2)

![Validation of grid independence.](image3)
\[ C_l = \frac{P_{14} - P_4}{P_3 - P_3} \]  

(1)

where \( P_{13}, P_{14}, P_3 \) are the total pressure at the diffuser inlet, the total pressure at the diffuser outlet, and the static pressure at the diffuser inlet, respectively. The total pressure loss coefficient of the diffuser drops sharply as the total grid number is less than 2.33 million, whereas the loss coefficient slightly changes when the grid number is higher than 2.33 million. In addition, to save on computation cost, the grid number of 2.33 million is finally selected as the computation mesh size.

Then, to ensure the validation of the turbulence model on the numerical simulation, the \( \kappa \varepsilon \) and \( \kappa \omega \) SST turbulence models are used for simulation based on the moderate mesh selected in the grid-independent analysis. Table 2 compares the total pressure ratio and the adiabatic efficiency of centrifugal compressor stages for the two turbulence models with the experimental results on the design point. The results indicate that the \( \kappa \varepsilon \) turbulence model overestimates pressure ratio and efficiency, whereas the \( \kappa \omega \) SST turbulence model obtains extremely slight differences in stage performance. This result may be caused by the diverse applications of different turbulence models. The \( \kappa \varepsilon \) turbulence model is suitable for compressible/incompressible flow problems without separation and is not accurate for the simulation of nonslip wall, adverse pressure gradient, and strong curvature flow. However, the \( \kappa \omega \) SST model breaks through the above limitations. Thus, the \( \kappa \omega \) SST turbulence model is more suitable for solving the internal flow field of the centrifugal compressor.

Moreover, to evaluate the accuracy of the above numerical model, the numerical simulation results of NASA CC3 compressor stage are compared with experimental data. Figure 5 provides the numerical and experimental comparison of stage adiabatic efficiency and stage total pressure ratio with respect to nondimensional mass flow rate at design rotating speed. Given that the calculational choking mass flow rate is higher than that in the experiment, the mass flow rate acquired by the calculational and experimental results is nondimensionalized by their own choking mass flow rate, as shown in Figure 5A,B. These two figures show that the variation trends of stage performance obtained by numerical simulation show a good agreement with the experimental results. At the design mass flow rate, the computational adiabatic efficiency is 0.9% higher than the test data, and the difference in total pressure ratio is approximately 1.76%. Numerical prediction overestimates the adiabatic efficiency and total pressure ratio at the lower mass flow rate. The maximum differences in adiabatic efficiency and total pressure ratio are approximately 4.54% and 4.8%, respectively. Taken together, these results suggest a reasonable error margin between numerical prediction and experimental measurement.

| Turbulence Model | Pressure Ratio | Efficiency (%) | Pressure Ratio, % Difference | Efficiency, % Difference |
|------------------|----------------|----------------|-----------------------------|-------------------------|
| \( \kappa \varepsilon \) | 4.22           | 86.55          | 6.30                        | 4.03                    |
| \( \kappa \omega \) SST | 4.04           | 83.95          | 1.76                        | 0.90                    |
| Experiment      | 3.97           | 83.20          | —                           | —                       |

**FIGURE 5** Stage performance comparison between calculation and experiment. (A) Adiabatic efficiency and (B) total pressure ratio.
because the working medium, the geometry model and the turbulence model used in the simulation are not exactly the same as the actual working process. Consequently, the numerical simulation method is reliable for predicting the internal flow characteristics of the NASA CC3 centrifugal compressor with a wedge diffuser.

4 | DESIGN METHOD OF ENDWALL CONTOURING

Endwall contouring technology is a passive control method that constructs a profile with concave and convex curvatures to change in local flow condition. The control mechanism is based on the principle, proposed first by Rose,25 where the concave profile on the suction side produces decelerating flow that increases local static pressure, whereas the convex profile on the pressure side leads to accelerating flow that decreases local static pressure. Flow losses can be effectively reduced by controlling the gradient of static pressure.

In the present paper, a parameterization endwall contouring method is proposed to design different forms of hub-side walls. The full circular diffuser hub-side wall is divided into 24 flow passages by the camber lines of vanes. Single-flow passage is selected to introduce the design method of endwall contouring, and other flow passages are obtained by periodic array in circumferential direction. First, to achieve flexible adjustment of endwall profile, the flat diffuser endwall needs to be discretized into a parametric model. As shown in Figure 6, the contouring region is defined by eight curves along the circumferential direction, called as B1–B8, and B1 and B8 are based on two adjacent camber lines (Camber1 and Camber2 in Figure 6) while other curves are uniformly arrayed in a circumferential direction between the two camber lines. Then, nine controlled points uniformly distribute on each cutting curve, named as R1–R9, and the contouring region consists of 72 control points. To ensure the smooth connection between the contouring region and the noncontouring region, 30 points at the border are fixed (red points in Figure 6), and 42 other points located in the inner region are free (blue points in Figure 6). Different types of contouring endwall can be produced by adjusting the free points.

Based on the above discretized endwall, the contouring endwall is designed by the superposition of two functions in the streamwise and pitchwise directions. The controlling function in the streamwise direction determines the radial position and the height of the concave and convex shapes. Each cutting curve is defined by a Bézier curve, which is a function of the streamline position. Bézier curve \( A(r) \) is defined as follows:

\[
A(r) = \sum_{i=0}^{n} P_i B_{i,n}(t), \quad 0 \leq t \leq 1
\]

\[
B_{i,n}(t) = \frac{n!}{k!(n-k)!} t^k (1-t)^{n-k}, \quad t \in [0, 1] \quad (2)
\]

where \( P_i \) is the control point of the Bézier curve. Figure 7 displays the schematic of the cutting curve along the streamline direction. By adjusting the control points, peak position \( r_{\text{peak}} \) along the streamline direction and peak height \( A_{\text{peak}} \) of the concave and convex change. \( r_{\text{peak}} \) can vary from R3 to R7 streamwise position, and \( A_{\text{peak}} \) represents the amplitude of the concave and convex profile, which is the percentage of the diffuser vane height.

The function in the pitchwise direction produces the concave profile on the suction side and the convex profile on the pressure side. The pitchwise function is defined as a sinusoid starting from the camber line on the suction side (B1 line) to the adjacent camber line on the pressure side (B8 line). Sinusoid \( f(\theta) \) is a function of circumferential position, defined as follows:

\[
f(\theta) = \sin \left[ \frac{\lambda \pi}{8} \left( \theta - \frac{\theta_8 + \theta_0}{2} \right) \right], \quad \theta_0 \leq \theta \leq \theta_8,
\]

where \( \theta_0 \) and \( \theta_8 \) stand for circumferential positions of the starting and ending line, respectively, and \( \lambda \) is the
frequency coefficient. Different frequency coefficients produce varied forms of the concave and convex shapes along the circumferential direction, as shown in Figure 8. The center of the concave and convex shape is far from the suction side or the pressure side when the frequency coefficients vary from 1 to 3.

Then, the contouring endwall is defined as follows:

\[
\Delta z(r, \theta) = A(r)f(\theta),
\]

where \( r, \theta, \) and \( z \) stand for the coordinates along the radial direction, the circumferential direction, and the axial direction, respectively, and \( \Delta z \) means the axial coordinate offset of the control points located in the contouring endwall. Once function \( A(r) \) and function \( f(\theta) \) are determined, the 3D contouring endwall is produced based on the axial coordinate offset. The schematics from Figures 6–8 enlarge the contouring region than the actual contouring endwall to demonstrate the principle clearly. Figure 9 provides an example of the height distribution of a contouring diffuser endwall at the hub side. The contours show that the endwall treatment is located upstream of the diffuser, and the hub-side wall has a convex structure on the pressure side and a concave structure on the suction side.

After performing a preliminary calculation, the contouring region is finally limited to the range from 5% to 30% relative streamline, which is extended to the vaneless space of the diffuser. Moreover, two different frequency coefficients, five different peak positions, and five different peak heights are considered to reveal the endwall contouring effect and control mechanism. The baseline model is the original diffuser configuration. The contoured models follow the naming conventions, where CEW stands for the contouring endwall, the first number is the frequency coefficient, the second number means the peak radial location on the cutting curves, and the last number is the percentage of peak height to the diffuser vane height. For example, CEW15-12 represents the contouring diffuser endwall with the frequency coefficient 1, the peak position at the R5 radial position, and a peak of 12% diffuser vane height.

5 | RESULTS AND DISCUSSION

With the objective to assess the effect of endwall contouring on the performance characteristics of NASA CC3 centrifugal compressor with vaned diffuser, CFX simulation is performed on the calculation models at 100% rotating speed. Then, the effect of endwall contouring and flow control mechanism is discussed in detail as follows.

5.1 | Comparison of performance characteristics

The global effect of endwall contouring is presented in the form of stage performance characteristic for the centrifugal compressor stage with baseline and contoured diffusers in Figure 10. To quantify the effect of the
endwall contouring on the compressor stability, the evaluation criteria of compressor stability extension and efficiency losses on design point are defined by formulae (5)–(8):

$$SR = m_c - m_s,$$

$$\Delta SR = \frac{SR_{CEW} - SR_{Baseline}}{SR_{Baseline}} \times 100\%,$$

$$\Delta SM = \frac{m_{s, Baseline} - m_{s, CEW}}{SR_{Baseline}} \times 100\%,$$

$$\Delta CM = \frac{m_{c, CEW} - m_{c, Baseline}}{SR_{Baseline}} \times 100\%,$$

where $SR$ is the stable operating range of centrifugal compressor stage; $\Delta SR$, $\Delta SM$, $\Delta CM$ are the percentages of the stable operating range extension, the stall margin extension, and the choke margin extension, respectively; $m$, $\pi$ are the mass flow and pressure ratio, respectively; subscripts $s$, $c$, $CEW$, $Baseline$ denote the stall point, choke point, the contouring cases and the baseline case, respectively.

In Figure 10, OP1 and OP2 represent the near surge condition and the near choke condition at 100% rotating speed, respectively. The stall margins of all contoured cases increase toward the lower mass flow rate, and the stall margin increases at most by up to 40% for CEW13-12. The choke margins of CEW15-12 and CEW25-12 increase toward the higher mass flow rate, whereas the choke margin of CEW23-12 slightly decreases. The maximum increase in choke margin is 9.3% for CEW25-12. Consequently, evident range extensions for all contouring cases are achieved at 100% speed due to the endwall contouring technology. Near the stall and design condition, nearly no reductions in the adiabatic efficiency and total pressure ratio are observed. Near the choke condition, the adiabatic efficiency and total pressure ratio for other cases are improved clearly except CEW23-12.

To evaluate the influence of three endwall contouring parameters on the compressor stability extension, the cases with different frequency coefficients, peak positions, and peak heights are performed. The evaluation results are summarized in Figures 11 and 12. Figure 11 presents the variation of stable operating range with frequency coefficient and peak position when peak height is 12% diffuser vane height.

Figure 11A shows a slight rise first and then a gradual drop in the stable operating range extension of the centrifugal compressor as the peak position moves downstream. Even the original stable operating range decreases when the peak position locates in R7. Combined with Figure 11B, the stall margin extension has a steep fall with the peak location moving downstream, whereas the choke margin extension rises at beginning and then goes down. The stability enhancement technology for centrifugal compressor aims to gain the maximized stall margin. Therefore, the endwall contouring technology will make a better gain in stable operating range if the concave and convex peaks in the radial direction are placed at R3.

With increasing frequency coefficient, the center of the concave surface is far from the suction surface, and the center of the convex surface is far from the pressure surface. According to Figure 11, an apparent difference in the extension of surge margin is caused by different frequency coefficients when the concave and convex
peaks are located in the upstream of the endwall contouring region. The cases with frequency coefficient 1 have more excellent performance in stability extension than those with a frequency coefficient 2.

According to the above analyses, the influence of peak height on stable operating range is discussed by controlling frequency coefficient 1 and peak position R3, as shown in Figure 12. With increasing peak height, the stable operating range has a sharp rise until up to 12% diffuser vane height. When peak height exceeds 12%, the stable operating range does not increase and even slightly declines. With the decrease in mass flow rate, the impeller passage is blocked by the stall cells induced by the impeller rotating stall. Therefore, the improvement of stable operating range by the diffuser endwall contouring is limited. Moreover, the change in choke margin is minimal with the variations of peak height, but the peak height makes a large difference in stall margin.

Figure 13 shows the variations of stage adiabatic efficiency and total pressure ratio for baseline and contouring endwalls with different peak heights at the near stall condition (OP1). When peak height is below 12%, the stage performances for the cases with contouring endwalls are improved slightly than Baseline. Once peak height is beyond 12% diffuser vane height, the endwall contouring technology continuously decreases the stage performances with the increase of peak height. Thus, the optimal peak height is 12% diffuser vane height for the present paper.

Overall, peak height, peak radial position and frequency coefficient have a different effect on the performance characteristics of centrifugal compressor.
stages with vaned diffusers. And the CEW13-12 has an optimal improvement in centrifugal compressor stability, which can achieve 40% stall margin extension. Therefore, the forthcoming sections will discuss the details of the flow field and reveal control mechanisms of endwall contouring based on Baseline, CEW15-12, and CEW13-12 cases at 100% speed.

5.2 Effect of endwall contouring on the flow field of vaned diffuser

With the purpose of analyzing the stability enhancement mechanisms of endwall contouring at the vaned diffuser hub-side, specific flow field analyses among Baseline, CEW15-12 and CEW13-12 at 100% speed are investigated in this section. The details of flow field are investigated at the operating points near surge condition OP1 and near choke condition OP2 to clarify the stability enhancement mechanism.

5.2.1 Near surge condition OP1

To validate the widening of surge margin owing to the effect of contouring endwall on the vaned diffuser, the spanwise distribution of circumferentially averaged absolute velocity and absolute flow angle (from radial direction) at impeller exit is shown in Figure 14 for Baseline, CEW13-12 and CEW15-12 near the surge condition OP1 ($m = 4.27$ kg/s). On the one hand, almost no difference is observed in the magnitude and variation trend of absolute velocity and absolute flow angle among the three cases, which indicates the impeller outlet absolute flow is identical at the same operating condition, and the effect of widening surge margin comes from the contouring vaned diffuser. On the other hand, the absolute velocity distribution and the absolute flow angle distribution along the spanwise direction display a strong nonuniformity due to the change in flow direction from the axial inlet flow to the radial discharge flow in the centrifugal impeller passage. Especially under the interaction of secondary flow and tip leakage flow in the shroud region, absolute velocity decreases sharply from 0.25 to 0 span and increases near the hub side. The absolute flow angle becomes more tangential near the shroud side but more radial near the hub side.

Figure 15 shows the comparisons of incidence angle and radial velocity near the diffuser vane LE in the spanwise direction for the design point and the near stall point. As shown in Figure 15, the nonuniform flow distribution from the hub side to the shroud side at the impeller exit remains in the vaneless and semi vaneless space of the vaned diffuser. Compared with the design point, the radial velocity distribution at the near stall point decreases, and the reversed radial flow is observed from 0.2 to 0 span near the shroud region. Consequently, as the mass flow rate decreasing, the positive incidence angle at the hub side of the vaned diffuser LE decreases, whereas the incidence angle at the shroud side is more negative. Therefore, the negative incidence angle and reversed radial flow at the vane LE near the shroud side cause the diffuser to approach the stall condition.

To observe flow patterns in the vaned diffusers with baseline and different contoured endwalls, Mach number contours at 15% span and 95% span are shown in Figures 16 and 17 for the three cases at the near stall conditions. For Baseline, the higher Mach number core
fluid from the impeller exit expands rapidly in the semi vaneless space. As shown in Figure 16A, flow separation occurs at the vane pressure side due to the positive incidence angle on the vane LE in the hub region and the wake vortices caused by the wide trailing edge of the wedge diffuser. The low kinetic energy fluid in the shroud region suddenly accelerates along the round LE in the semi vaneless space when the flow stagnation
point deviates to the pressure side due to the large negative incidence angle near the shroud side. Consequently, a locally high Mach number zone 1 appears on the vane LE of the suction side. At the same time, a low velocity recirculation zone B and the flow separation C, as shown in Figure 17A, on the suction side is presented as a result of the interaction between boundary layer separation and the locally high Mach number zone 1. With the decrease of mass flow rate, the whole semi vaneless space is occupied by the locally high Mach number fluid, and flow separation C grows stronger and brings more blockages at the throat, which finally induces rotating stall.

For CEW15-12, flow separation A on the pressure side slightly increases compared with Baseline near the hub side, as shown in Figure 16B. The flow separation C on the suction side near the shroud side is alleviated compared with the Baseline case, as shown in Figure 17B. The smaller local Mach number region 1 at the vane LE decreases the disturbance to the mainstream. This finding implies that the contouring endwall of CEW15-12 increases the flow losses near the hub side directly but indirectly decreases the flow separation on the suction side near the shroud side, which is beneficial to delaying the rotating stall.

For CEW13-12, flow separation A on the pressure side slightly decreases compared with Baseline, as shown in Figure 16C. In each Mach number contour, different-level Mach number regions are divided by the solid black contour line reflecting the flow uniformity. Unexpectedly, the distribution of mainstream entering the vane channel near the hub side becomes more uniform compared with Baseline and CEW15-12, as shown in Figure 16C. Near the shroud side, the flow separation C on the suction side has a prominent reduction compared with Baseline and CEW15-12, as shown in Figure 17C. No flow separation is induced by the local Mach number 1 but only wall boundary layer separation. As mentioned before, the reduction of flow separation on the suction side near the shroud side contributes to delaying the rotating stall; thus, CEW13-12 has a remarkable effect on the improvement in stall margin compared with CEW15-12, as depicted in Figure 10.

Figures 18 and 19 provide the limit streamlines and the static pressure recovery coefficient contours on suction and pressure surface, respectively. For Baseline, in Figure 18, the static pressure distribution in the semi vaneless space shown in the black box is nonuniform in the spanwise direction. Therefore, the flow separation C on the suction side in Figure 17 is a recirculation region located at the throat plane, which is caused by the large adverse pressure gradient. For CEW15-12, the uniformity of static pressure is improved, and the recirculation region is decreased. For CEW13-12, the recirculation region disappears in the same mass flow rate condition. On the pressure side, a large flow separation vortex is clearly visible at the hub side for Baseline, as shown in Figure 19. For CEW15-12, the peak location of convex curvature on the pressure side locates in the initial position of flow separation to accelerate the flow deterioration, and the start point of flow separation is advanced. For CEW13-12, flow separation vortex is weakened primarily because the convex peak of CEW13-12 is closer to the vane LE than CEW15-12, and it improves the momentum of the low-energy fluid in advance to delay flow separation.

To present a more explicit understanding of the flow pattern in vaned diffusers with different endwall, the circumferentially averaged flow angle distribution along the spanwise direction at the vane LE is displayed in Figure 20 for the three cases near the stall conditions. In Figure 20, the applications of contouring endwalls make the flow angle of core flow decrease for the two CEW cases, which worsens the flow separation on the pressure side, as shown in Figure 19. At the shroud side, the flow angle is over deflected, and the flow angle is slightly increased for the two CEW cases. The above analysis shows that the core flow entering the diffuser channel turns to flow in a more radial direction, as the endwall shape of convex surface on the pressure side and concave surface on the suction side diverts the flow from pressure side to
suction side. At the same time, the two CEW cases have the same influence on the distributions of flow angle at vane LE, which indicates CEW13-12 alters the flow within the semi vaneless space to alleviate the flow separation at the diffuser vane suction side near the shroud side.

Further determining the stability enhancement mechanism for CEW13-12 is attributed to altering the flow within the semi vaneless space, and comparisons of static pressure recovery coefficient distribution along the streamwise direction at 10% and 90% span are displayed in Figure 21. A substantial adverse static pressure gradient on the suction side from 0% to 40% streamwise location in the diffuser channel easily causes recirculation flow. The adverse static pressure gradient at 10% span for CEW13-12 on the suction side is decreased compared with Baseline in the streamwise direction because the concave surface on the suction side increases the static pressure recovery by reducing the flow velocity before the throat plane. Similarly, the static pressure recovery coefficient distribution for CEW13-12 at 90% span on the suction side becomes more uniform compared with Baseline. Moreover, adverse static pressure gradient is greatly cut down near the throat, which is beneficial to weakening the flow separation on the suction side near the shroud side to delay the onset of rotating stall.

Figure 22 presents the contour plots of radial velocity at the diffuser LE for Baseline and CEW13-12, and the radial velocity difference between CEW13-12 and Baseline. A plus sign indicates an increase in radial velocity, and a minus sign indicates a decrease in radial velocity. In Figure 22B, the radial velocity at the suction side increases, and the radial velocity at the pressure side decreases. Therefore, CEW13-12 raises the radial velocity of core flow, and the core flow is deflected toward the suction side. At the same time, the momentum of low-energy fluid on the suction side is improved, which stabilizes the wall boundary layer on the suction surface and suppresses flow separation.

Figure 23 presents the contour plots of radial velocity at the diffuser throat plane for Baseline and CEW13-12, and the radial velocity difference at the diffuser throat plane between CEW13-12 and Baseline. In the same manner, CEW13-12 improves the momentum of low-energy fluid on the suction side by increasing the radial velocity at the hub corner and the shroud corner. The radial velocity on the pressure side is decreased, and the shock wave at the LE on the pressure side is weakened. The combined action reduces the aerodynamic blockage at the throat plane near the stall condition. The radial

![Image](image_url)

**Figure 20** Circumferential-average spanwise distribution of flow angle at vane LE.

![Image](image_url)

**Figure 21** Comparison of static pressure recovery coefficient distribution. (A) 10% span and (B) 90% span.
velocity variations from the LE to the throat plane indicates CEW13-12 improves the momentum of low-energy fluid on the suction side to increase the flow stability in semi-vaneless space.

In summary, the model of peak position located in control point R3 changes in the flow field within the semi-vaneless space. At the near stall condition, controlling the flow separation on the diffuser vane suction side near the shroud region is essential to suppressing the development of rotating stall. The design concept of endwall contouring makes the diffuser hub-side endwall have a concave surface on the suction side and a convex surface on the pressure side, which forces the fluid entering the semi-vaneless space to deflect toward the suction side. The contoured endwalls raises the radial velocity on the suction side in the semi-vaneless space to increase the momentum of low-energy fluid in the wall boundary layer of suction surface. At the same time, the concave surface on the suction side improves the static pressure near the throat plane to reduce the adverse static pressure gradient for the fluid on the suction side near the shroud, and the fluid remains attached to the vane suction side to delay flow separation until lower flow rates.

5.2.2 Near choke condition OP2

Based on the analysis above, CEW15-12 considerably improves the performance near choke conditions, whereas CEW13-12 has a negligible influence on the choke condition. Quantitative comparisons of compressor performance at operating point OP2 for Baseline, CEW15-12, and CEW13-12 at 100% rotating speed are shown in Table 3. CEW15-12 has an improvement of 8.51% and 8.27% on total pressure ratio and adiabatic efficiency, respectively, whereas the variation of CEW13-12 is negligible.

The vane loading distributions at 15% span for the three cases at operating point OP2 are shown in Figure 24. The difference in streamwise static pressure between Baseline and CEW13-12 is negligible, whereas CEW15-12 has a remarkable static pressure rise on the pressure side and the suction side compared with Baseline and CEW13-12. The streamwise distributions of static pressure on the suction side sharply decrease first and then slowly increase for the three cases. The static pressure on the suction side of Baseline and CEW13-12 goes through valleys near the throat plane location, whereas the static pressure on the suction side of CEW15-12 increases steadily.

For a better understanding of vane loading distribution, the static pressure distribution contours at 15% span near the vane LE of different diffusers are shown in Figure 25. For Baseline near the choke condition, when the fluid approaches the vane surface, stagnation point flow appears on the vane LE near the suction side, as shown in Figure 25A, along with a strong rising shock wave on the suction side at 0% stream location, as shown in Figure 24. Then, the fluid accelerates to local supersonic along with the wedge diffuser round LE, along with a strong falling shock wave on the pressure side at 0% stream location, as shown in Figure 24. According to Figure 25A, the zone where the static pressure is below 280,000 Pa is extended to the suction side, as shown in Figure 24. For CEW13-12, the intensity of low static pressure wave caused by local supersonic is evidently higher than Baseline, and the low static pressure (below 280,000 Pa) zone extends to the suction side, as shown in Figure 25C. Therefore, the valley of the shock wave on the pressure side at 0% stream location is lower than
Baseline, and the valley on the suction side near the throat location is similar to Baseline, as shown in Figure 24. A great difference is noted between Baseline and CEW15-12. The low static pressure zone induced by local supersonic has a very substantial reduction, which indicates throat blockage is relieved visibly, as shown in Figure 25B. Thus CEW15-12 achieves a better performance improvement near the choke condition.
According to the analyses above, throat blockage is a key factor in improving compressor performance near the choke condition. To reveal the mechanism of endwall contouring on improving the performance at the choke condition, Mach number distribution contours at throat planes for the three cases on operating point OP2 are shown in Figure 26. The throat areas of CEW15-12 and CEW13-12 clearly change. For CEW15-12, the concave area on the suction side is larger than the convex area on the pressure side, such that the throat area is increased by 2.28% than Baseline, as shown in Figure 26B. CEW15-12 reduces the higher Mach number region on the pressure side than Baseline. Nevertheless, for CEW13-12, the concave area on the suction side is smaller than the convex area on the pressure side, such that the throat area change is reduced by 1.209% than Baseline, as shown in Figure 26C. Importantly, the larger amplitude of convexity locates in the throat pressure side, which causes the increase of local higher Mach number area to deteriorate throat blockage.

The reason why the throat area for contouring diffuser changes than Baseline originates from the method of endwall contouring. The method of endwall contouring defines a sinusoid function starting from the camber line on the suction side to the adjacent camber line on the pressure side to ensure the equal area change between the pressure side and the suction side in the circumferential surface. In Figure 27, the concave area on the suction side is equal to the convex area on the pressure side in the circumferential direction. However, the area change in the throat plane is decided by the endwall profile parameters. As described above, the different combinations of peak height, peak radial position, and frequency coefficient may cause changes in the throat area. Furthermore,
changes in the throat area lead to an increase or decrease in throat blockage.

6 | CONCLUSION

The paper proposed an endwall contouring technology on the hub-side wall of a radial vane diffuser to investigate the effect of endwall contouring on the performance and stable operating range for the NASA CC3 centrifugal compressor stages. Numerical simulations were performed on compressor stages with baseline and contoured diffusers. The specific flow field and mechanism of stability enhancement were analyzed, and the main conclusions are as follows:

1) Endwall contouring technology is an effective method to improve compressor stability and consequently increase the stable operating range of centrifugal compressor stages with vaned diffusers. Moreover, the stable operating range extends toward higher and lower mass flow rates. The most successful endwall contouring diffuser can improve the stall margin by up to 40%, and no evident reduction is observed in stage performance at the design condition. At the choke condition, choke margin is increased at most by 10%, and remarkable improvements in compressor performance are achieved.

2) A parameterized endwall contouring guideline is exploited to produce different forms of contoured endwalls, considering peak height, peak radial position, and frequency coefficient. In the contoured region, the peak radial position near the vane LE changes in the flow field within the semi vaneless space, which causes a substantial improvement in stall margin. The contoured cases with a frequency coefficient of 1 have more excellent stability performance than those of with a frequency coefficient of 2. The optimal peak height is 12% diffuser vane height. And the peak height is beyond 12% diffuser vane height, the contoured cases continuously decrease the stage performances as peak height increases.

3) At the near surge conditions, the flow separation on the diffuser vane suction side near the shroud region induces the development of rotating stall due to the negative incidence on the vane LE near the shroud side. The flow field in the diffuser passage is redistributed with the contouring endwalls, and the influence of contoured endwalls on the flow field extends from the hub side to the shroud side to delay the formation and development of rotating stall.

4) The endwall contouring diffuser forces the fluid entering the semi vaneless space to deflect toward the suction side, and the contoured endwalls raises the radial velocity on the suction side in the semi vaneless space to increase the momentum of low-energy fluid in the wall boundary layer of the suction surface. Then, the concave surface on the suction side improves the static pressure near the throat plane to reduce the static pressure gradient. The combined action improves the ability to overcome the adverse static pressure gradient for the fluid on the suction side near the shroud region, and the fluid remains attached to the vane suction side to delay the onset of rotating stall.

5) At the near choke conditions, the stable operating range toward higher mass flow is limited by throat blockage due to the local supersonic area on the throat plane. The endwall contouring ensures the equal area change between the pressure side and the suction side in the circumferential plane, but the area change in the throat plane is not equal. The throat area of CEW15-12 increases to weaken the throat blockage clearly, which causes CEW15-12 to achieve a higher performance improvement near the choke condition.
NOMENCLATURE

\( (r, \theta, z) \) cylindrical-coordinate system

\( m \) mass flow rate

\( H \) vane height

\( c \) absolute velocity

\( y^+ \) nondimensional first cell height

\( \alpha \) absolute flow angle (from radial direction)

\( \lambda \) frequency coefficient

\( \eta \) adiabatic efficiency

\( \pi \) total pressure ratio

\( \Delta \) change in property

\( SR \) stable operating range

\( SM \) surge margin

\( CM \) choke margin

\( P \) static pressure

\( Pt \) total pressure

\( C_L \) total pressure loss coefficient

\( Cp \) static pressure recovery coefficient

SUBSCRIPTS

\( A \) attachment angle

\( b \) blade

\( c \) choke point

\( s \) surge point

\( \text{peak} \) amplitude of curve

\( 3 \) diffuser inlet

\( 4 \) diffuser outlet

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