GT2011-45-\$)

NUMERICAL INVESTIGATION OF A SINGLE CIRCUMFERENTIAL GROOVE CASING TREATMENT ON THREE DIFFERENT COMPRESSOR ROTORS

F. Heinichen, V. Gümmer

Rolls-Royce Deutschland Dahlewitz, Germany

H.-P. Schiffer

Technische Universität Darmstadt Chair of Gas Turbines and Aerospace Propulsion Darmstadt, Germany

ABSTRACT

In axial compressors, casing treatments represent a passive method to increase the working range without the need to modify the blade geometry. The majority of the open literature on the topic considers one or several casing treatment variants on the same compressor. The question how one casing treatment and its basic mechanisms can be transferred to a different compressor is only covered in a small number of publications. This paper tries to fill this gap by applying a single circumferential groove type casing treatment to three different transonic compressor rotors. It is demonstrated numerically that the casing treatment is able to improve the aerodynamic performance of all three rotors. Detailed investigation of the flow field near the rotor tip shows that the single circumferential groove works by influencing the interaction between the tip clearance vortex and the shock. Hence, this type of casing treatment can be generalized to transonic rotors with a stall mechanism that is based on tip clearance vortexshock interaction.

NOMENCLATURE

Abbreviations

- CFD Computational fluid dynamics
- FLFN Inlet flow function
- H Enthalpy
- IGV Inlet guide vane
- MA Relative Mach number
- PR Pressure ratio

RANS Reynolds-averaged Navier-Stokes

U Blade speed

Subscripts

- CT Casing treatment
- NS Near stall
- ref Reference
- rel Relative
- p Polytropic
- WL Working line

INTRODUCTION

For more than six decades, research on the topic of casing treatments for axial compressors has been conducted. The design goal for a casing treatment is to increase the surge margin of a given compressor without compromising efficiency. Other criteria, such as low aerodynamic impact at operating conditions away from the surge line, small installation space or a geometric shape that allows for low cost manufacturing methods and mechanical robustness, could be added. A tremendous amount of open literature is available that deals with the different types of casing treatments, their design parameters and working principles as well as the effect that they have on the aerodynamic behavior of the compressor under consideration. Since this paper uses a circumferential groove as well as for reasons of brevity, only a selection of recent publications covering circumferential groove type casing treatments is mentioned in the following literature review.

Circumferential grooves have been applied successfully, meaning with a beneficial effect on compressor operating range, to both subsonic as well as transonic axial compressors. For subsonic compressors, publications are available that comprise solely numerical [1] - [4], largely experimental [5] - [8] or both numerical and experimental [9] - [11] investigations. The compressor surge margin extensions achieved range from approximately 10% to 30% of the surge margin without casing treatment.

A large number of the open literature on circumferential groove applications to transonic rotors utilizes NASA Rotor 37 [12] -[16]. The fact that the surge margin increases reported in these numerical investigations show a spread from almost negligible to approximately 40% relative to the case with smooth casing shows that besides the rotor with its inherent stability-limiting mechanism and the type of casing treatment, there is a considerable influence of other parameters that determine the effectiveness of the casing treatment. The publications covering both experimental and numerical studies on transonic rotors with circumferential grooves [17], [18] show that the interaction of the passage shock and the vortex driven by the rotor tip leakage plays an important role in the flow breakdown in transonic compressors. This interaction can be influenced by a circumferential groove casing treatment, thus delaying the onset of stall.

In summary, the studies published by Houghton and Day [6], [8] are the only recent ones where a casing treatment is subjected to what can be seen as a thorough robustness analysis by changing various relevant parameters such as solidity, rotor stagger and even the entire compressor. This is astonishing because a measure for the general applicability of a given casing treatment represents a critical attribute for its introduction into a real jet engine, almost at the same level of relevance as the working range and efficiency change demonstrated in a research compressor environment. After all, the design of a multi-stage compressor is a very complex and interdisciplinary process with many, sometimes conflicting requirements and a certain level of risk. While numerical methods and increased computing resources represent a powerful tool to mitigate some of this uncertainty, it is still not enough to cover every possible operating condition at any given level of deterioration or manufacturing accuracy. Besides that, there is the question how reliable today's CFD methods really are when it comes to accurately predicting certain flow details. Therefore, one could argue that a casing treatment which is known to be aerodynamically robust in the sense that its influence can be easily and reliably predicted will be preferred to one that may offer a greater working range extension, but with a lower level of confidence in a new and hence largely unknown compressor environment.

Unfortunately, the topic of aerodynamic robustness of a given type of casing treatment has not been at the center of attention.



FIGURE 1. SINGLE GROOVE CASING TREATMENT MESH AND GEOMETRY.

This paper is intended as a contribution to this particular aspect of casing treatments. A single skewed circumferential groove for transonic compressor rotors is presented. This casing treatment is applied to three different rotors, all transonic but with different levels of relative inlet Mach number, aerodynamic loading as well as geometric parameters such as solidity and tip clearance. The working range of each rotor is evaluated at design speed for both a reference case with smooth casing and a variant with casing treatment. Finally, an attempt is made to correlate the change in working range to certain flow features in order to explain the differences between the three compressors under consideration.

1 NUMERICAL SETUP

The Rolls-Royce in-house Hydra CFD suite was employed as the numerical tool. Besides the RANS-solver itself, the package consists of various pre- and postprocessing tools. The software is used extensively within Rolls-Royce and among university and other research partners. It has been validated for a wide range of turbomachinery applications [19]. Furthermore, it has been successfully used to assess the qualitative influence of different casing treatments on the stability and efficiency of axial compressors [20]. This means that even though uncertainty remains about the absolute values of a given compressor's stability margin derived with the type of CFD model used in this study, it can be applied with some confidence to compare different compressors and/or casing treatments on a qualitative basis. In other words, the authors would not expect to find the absolute values of working range presented in this paper in a rig test, neither with nor without casing treatment. However, the question if a compressor is sensitive to the presence of a casing treatment or not can be answered correctly, provided sufficient care was taken to ensure that the numerical model captures the relevant flow details.

The numerical setup for the present study used the steady flow



FIGURE 2. COMPRESSOR 1 SIDE VIEW.

solver with mixing planes between the rotating and non-rotating, single blade passage domains of each compressor. The calculations were run with fully turbulent boundary layers and used the Spalart-Almaras turbulence model with wall functions. The mesh featured a hybrid O-H topology, with several layers of Omesh around the aerofoil and H-grid cells in the blade passage, near the periodic boundaries and in the inlet and outlet sections. Clustering of mesh nodes was present towards the hub and casing, in the rotor tip gap as well as towards the blade surface. The mesh density of the rotors presented in this paper was within the range commonly used for steady viscous CFD models. Depending on the rotor, the radial number of nodes was between 65 and 95. Between 10 and 20 of these nodes were located in the rotor tip gap. The overall number of nodes for the baseline rotor meshes was in the range from approximately 600,000 to 1.2 million. The additional casing treatment meshes were based on the mesh density of the surrounding rotor meshes in axial and circumferential direction and had 30 nodes in radial direction. An example is shown in Fig. 1. Towards the walls as well as at the interface with the main rotor mesh, mesh clustering was applied in order to resolve the viscous effects with appropriate accuracy. At the inlet of the computational domain, radial profiles of absolute total pressure, absolute total temperature, absolute whirl angle as well as radial flow angle and turbulence properties were prescribed. A radial equilibrium boundary condition, where the static pressure is prescribed at one mid-height radial position only, was applied at the compressor outlet. The extent of the computational models for the three compressors as well as the positions of the inlet and exit planes can be seen in the side views provided in Fig. 2 through 4.

For each compressor characteristic, the inlet boundary conditions were kept constant and the static pressure at the outlet was increased gradually. When no converged solution could be obtained even with a two-fold decrease in step width, the characteristic was considered to have reached its numerical stall limit. The last converged point of each characteristic is referred to as "near stall" in this paper. For this investigation, only characteristics at design speed were evaluated.

The surge margin definition suggested by Cumpsty [21] was em-



FIGURE 3. COMPRESSOR 2 SIDE VIEW.



FIGURE 4. COMPRESSOR 3 SIDE VIEW.

TABLE 1. COMPARISON OF SEVERAL RELEVANT AERODY-NAMIC AND GEOMETRIC PARAMETERS AT ROTOR TIP.

Parameter	Compressor 1	Compressor 2	Compressor 3
Rel. inlet Mach number	1.3	1.1	1.05
Diffusion factor	0.4	0.5	0.5
Clearance (% blade height)	1.9	1.0	0.7
Solidity	1.2	1.5	1.4

ployed to quantify the compressor working range. It takes into account both the variation of flow and pressure ratio between the near stall and a reference point:

$$SM = 1 - \left(\frac{FLFN_{NS}}{FLFN_{ref}}\frac{PR_{ref}}{PR_{NS}}\right) \tag{1}$$

For each of the three compressors, the peak efficiency operating condition of the baseline configuration without casing treatment was taken as the reference point in order to create a certain degree of consistency.

2 COMPRESSORS

Three different compressors were selected for the present study. All three have a single transonic rotor. Apart from this commonality, they differ in many other aerodynamic and geometric features. This section gives a short overview about each compressor and summarizes some of the differences most relevant for the present investigation. In addition, the casing treatment is described.

The first compressor, termed Compressor 1, consists of two blade rows: the rotor and a downstream stator. A sketch of the compressor can be found in Fig. 2. It is a research vehicle and was designed in the early 1990's, therefore representing the oldest design in this study. While the hub radius increases in downstream direction, the casing annulus line is at constant radius due to facility and other constraints. Compressor 2, which is shown schematically in Fig. 3, also features a casing with constant radius. In addition to the rotor and downstream stator row, it also has an IGV row. Like Compressor 1, it is a research vehicle and the rotor was designed to be aerodynamically representative for a front stage of a jet engine high pressure compressor. However, it differs from Compressor 1 not only because it is a more recent design. The diffusion factor at the rotor tip, which is another parameter with great influence on the working range of the blade row, is higher. It also has a higher aerodynamic loading and tip solidity as well as a smaller rotor tip clearance and lower relative inlet Mach number at rotor tip. Figure 4 shows Compressor 3, which, like Compressor 1, consists of only two blade rows. It is an embedded front stage of a multi-stage compressor. Its solidity is situated between those of Compressor 1 and Compressor 2 and the diffusion factor matches the value for Compressor 2. While it has the lowest tip Mach number, its tip loading $\frac{\Delta H}{U^2}$ is the highest of the three compressors. Furthermore, it is the most engine-like compressor for a number of reasons such as the thickness of the hub and casing boundary layers at compressor inlet or the mechanical criteria the rotor was designed to meet. Besides other geometric parameters, especially the falling casing annulus line of Compressor 3 resembles a typical front stage more closely than the other two compressors. Therefore, it can be considered the most relevant compressor from a "real world application" perspective. Table 1 provides a short overview of the different compressors and how some of their most relevant parameters compare to each other.

The casing treatment applied to all three compressors is a single circumferential groove. It is inclined in upstream direction by an angle of 45 degrees. For application to the different compressor rotors, its main geometric parameters, namely the width and height of the groove, were scaled with the axial chord length of the rotor tip. It is important to note that no parameter studies were performed in order to optimize the casing treatment for the different aerodynamic environments of each compressor. This "blind read-across" approach was chosen because the goal of the study is to investigate the robustness of the casing treatment towards different aerodynamic environments. If it can be successfully transfered from one compressor to another with a simple scaling rule that is only based on geometric



FIGURE 5. DESIGN SPEED CHARACTERISTICS FOR COM-PRESSOR 1 WITH AND WITHOUT CASING TREATMENT.

parameters and some knowledge about the flow field of the compressor under consideration, it can indeed be considered aerodynamically robust.

3 RESULTS

3.1 Compressor 1

The normalized design speed characteristics of Compressor 1 with and without casing treatment are plotted in Fig. 5. At operating conditions close to the working line, which is at a normalized inlet flow function value of one, there is no significant difference in pressure ratio. Polytropic efficiency is 0.51% higher for the variant with casing treatment at this operating point. With decreasing flow function, an increasing difference in pressure ratio can be observed between the two variants, the characteristic with casing treatment exhibiting the steeper slope. At the same time, there is also a growing difference in efficiency as the stall limit is approached, with the casing treatment variant showing higher values here as well. With casing treatment, the characteristic runs to both higher pressure ratio and lower flow function. The surge margin is increased from 15.3% without casing treatment to 18.4%, which is a relative increase of 20.3%.

In order to investigate the effects that the casing treatment has on the rotor flow field in greater detail, three points are selected from the two characteristics:

- The baseline configuration at its highest stable operating



FIGURE 6. RADIAL DISTRIBUTION OF ROTOR PRESSURE LOSS COEFFICIENT AT SELECTED POINTS FOR COMPRES-SOR 1.

condition. This point will be referred to as PT1.

- The operating point of the configuration with casing treatment, where the rotor exit flow function is closest to the lowest stable value achieved for the baseline configuration. This point is called PT2 and is regarded to have approximately the same overall degree of rotor throttle as PT1.
- The configuration with casing treatment at its highest stable operating condition, termed PT3.

When considering the total pressure loss coefficients at these three points, it becomes clear that application of the casing treatment increases efficiency by reducing rotor losses, since stator losses did not change by any significant amount. The comparison of PT1 and PT2 shown in Fig. 6 shows that at the same operating condition, the near-casing loss peak caused by the secondary flow in this area is reduced considerably. The region with reduced losses extends from the casing down to 40% duct height. This means that the impact of the casing treatment is not restricted locally to the area near the rotor casing, but is also visible in the majority of the main flow. This reduction of rotor losses can be seen as a general improvement in the flow field, since it is possible to increase the back pressure further than it was possible without casing treatment. At the point PT3, the near-casing loss peak has increased from PT2 until it has reached the same level as PT1. Between approximately 40% and 90% radial height, the losses are higher than PT1, suggesting that the local blade loading may also be higher at these locations. This can be taken as a clue that the working range of Compressor



FIGURE 7. DISTRIBUTION OF NORMALIZED ENTROPY AND MA=1 (WHITE LINE) AT 99% BLADE HEIGHT FOR COMPRESSOR 1 WITHOUT CT AT PT1.



FIGURE 8. DISTRIBUTION OF NORMALIZED ENTROPY AND MA=1 (WHITE LINE) AT 99% BLADE HEIGHT FOR COMPRESSOR 1 WITH CT AT PT2.

1 is indeed limited by the rotor tip region, which exhibited the greatest loss reduction at 5% of dynamic pressure.

Another hint at the mechanism that limits the stability of Compressor 1 can be found in the distribution of entropy (normalized with a reference value) at rotor tip, which is shown for PT1 in Fig. 7. The plot clearly shows the tip clearance vortex trajectory as well as other sources of entropy increase and associated total pressure loss. By also plotting the MA=1 contour into the plot, it can be observed that the vortex cannot pass through the shock without being significantly widened and deflected towards the leading edge of the adjacent blade. Furthermore, the formation of a high entropy region can be seen just downstream of the vortex-shock interaction, indicating that this is the main loss generation mechanism at the rotor tip. A second high entropy area originates at the rotor suction side. Its location downstream of the suction side shock position suggests that this loss mechanism is based on the interaction between the shock and the rotor suction side boundary layer. However, this second high entropy region is of smaller size and intensity than the one caused by the tip clearance vortex, which leads to the hypothesis that vortex-shock interaction also is the stability-limiting mechanism for Compressor 1. This goes in line with the current understanding of spike-type stall inception, which states that one indicator for the stall point is the alignment of the interface between the incoming flow and the tip clearance flow with the blade leading edge [22]. It becomes clear from Fig. 7 that at PT1, this point is almost reached, explaining why this is the last stable operating condition for Compressor 1 without casing treatment. The instability mechanism involving vortex-shock interaction is also known as "Tip Blockage Stall" and rotors exhibiting this attribute are good candidates for casing treatment applications [23].

Figure 8, which shows the same plot for PT2, makes it clear that with casing treatment, the vortex is able to pass the sudden rise in static pressure caused by the shock without being deflected or bursting. The interface between the incoming and tip clearance flow has shifted relative to PT1 and is now further away from the rotor leading edge. Furthermore, the high entropy area is less severe, only the area directly at the rotor suction side seems to be unaffected. It can be concluded that the casing treatment works by influencing the interaction between the tip clearance vortex and the shock. The fact that this leads to an increase in compressor working range is taken as another indicator that vortex-shock interaction is the most likely instability mechanism for Compressor 1.

3.2 Compressor 2

The results for Compressor 2 show a strong qualitative similarity to those for Compressor 1, even though some differences have to be pointed out. As for Compressor 1, the design speed



FIGURE 9. DESIGN SPEED CHARACTERISTICS FOR COM-PRESSOR 2 WITH AND WITHOUT CASING TREATMENT.

characteristics shown in Fig. 9 coincide at the working line point, the variant with casing treatment exhibits a steeper slope and runs to both higher pressure ratio and lower flow function than the baseline without casing treatment. The baseline surge margin amounts to 8.9%, which is a lower value than for Compressor 1. However, with casing treatment, the surge margin increases to 12.6%, which means that the casing treatment leads to a higher relative surge margin increase for Compressor 2. The efficiency for the variant with casing treatment is higher than without for the entire characteristic of Compressor 2. The increase, derived at constant flow, ranges from 0.2% at working line conditions up to 0.74% at the highest baseline point, which also exceeds the respective Compressor 1 values.

In analogy to Compressor 1, three points were selected from the two Compressor 2 characteristics for further analysis. The radial distributions for rotor loss coefficients for PT1, PT2 and PT3 are plotted in Fig. 10. As with Compressor 1, there is a region with reduced losses for PT2 compared to PT1 that extends from the casing to approximately 40% duct height. The greatest decrease is found at the high loss zone near the rotor tip. At the highest stable point with casing treatment, PT3, this high loss zone does not return to the level of PT1. Instead, the highest level of pressure loss coefficient that is found for the rotor of Compressor 2 with casing treatment is still about 5% of dynamic pressure lower than the highest loss without it. This finding is different from the observations made for Compressor 1 and serves as another indicator that even though the casing treatment



FIGURE 10. RADIAL DISTRIBUTION OF ROTOR PRES-SURE LOSS COEFFICIENT AT SELECTED POINTS FOR COMPRESSOR 2.

seems to introduce the same qualitative changes to the rotor flow field, the quantitative effect is different.

The question wether the mechanism of the casing treatment is the same on Compressor 2 as the one found on Compressor 1 can be answered by considering the rise of entropy across the rotor as well as the tip vortex without and with casing treatment, as depicted in Fig. 11 and Fig. 12, respectively. It can be seen that for the baseline case, the vortex is the primary source of entropy rise and corresponding loss production in the rotor tip region. As soon as the vortex passes through the passage shock, the vortex diameter expands and a sharp increase in entropy can be observed. With casing treatment, the vortex remains more compact even after passing the shock and the entropy rise is significantly lower for PT2 than for PT1. As for Compressor 1, the interface between the incoming and tip clearance flow is close to the leading edge at PT1 and further downstream for the configuration with casing treatment, i.e. PT2. Hence, in analogy to Compressor 1, it can be concluded that the dominating instability mechanism for Compressor 2 is also related to shock-vortex interaction.

3.3 Compressor 3

The results obtained from the calculations of Compressor 3 also show a certain resemblance to those already seen for the other compressors, but a number of distinct differences have to be noted. The design characteristics with and without casing



FIGURE 11. DISTRIBUTION OF NORMALIZED ENTROPY AND MA=1 (WHITE LINE) AT 99% BLADE HEIGHT FOR COMPRESSOR 2 WITHOUT CT AT PT1.



FIGURE 12. DISTRIBUTION OF NORMALIZED ENTROPY AND MA=1 (WHITE LINE) AT 99% BLADE HEIGHT FOR COMPRESSOR 2 WITH CT AT PT2.



FIGURE 13. DESIGN SPEED CHARACTERISTICS FOR COMPRESSOR 3 WITH AND WITHOUT CASING TREAT-MENT.

treatment plotted in Fig. 13 do exhibit the coinciding working line point and the steeper slope with casing treatment, but with an amount of 0.01% at working line and 0.03% at baseline near stall conditions, the magnitude of the efficiency increase is almost negligible. Furthermore, even though the peak pressure ratio is increased with casing treatment, the minimum obtainable flow is higher than for the baseline calculation. The associated surge margin changes from 8.91% for the baseline to 8.94% with casing treatment, which is a relative change of 0.3%.

This result is rather surprising, because the rotor of Compressor 3 is known to be tip critical from earlier studies and it is also known that shock-tip clearance vortex interaction contributes to its instability mechanism. Further investigation showed that not the rotor tip, but rather the downstream stator hub is limiting the numerical stability of this compressor. Hence, the calculations with and without casing treatment were repeated for the isolated rotor. The results of these calculations termed "rotor only" can be found in Fig. 14.

It becomes apparent that without the stator, applying the casing treatment to the rotor of Compressor 3 yields all qualitative compressor performance changes observed for the other two compressors. The corresponding characteristic now runs to both higher peak pressure ratio and lower flows than the baseline. In relative terms, the surge margin is increased by 12%. Finally, polytropic efficiency at the flow of the baseline near stall point is increased by 0.31%. It should be noted that this efficiency



FIGURE 14. DESIGN SPEED CHARACTERISTICS FOR COMPRESSOR 3 ROTOR ONLY CALCULATIONS WITH AND WITHOUT CASING TREATMENT.

increase is also present in the full stage calculations, but is offset by the stator to result in the small efficiency change reported earlier.

The radial pressure loss distributions for the rotor given in Fig. 15 show a decrease of rotor tip loss by more than 5% for PT2 compared to PT1. The radial extend of the loss reduction is also of similar magnitude as that seen for the other cases. However, a comparison between PT3 and PT2 shows that further throttling of the rotor with casing treatment does not result in any significant increase in rotor tip loss, as seen for Compressor 1 and 2. Rather, the largest change in loss is found in the region between 60% and 80% radial height. Here, the loss level reached at PT3 is the same as that for PT1.

The tip entropy contour plots at PT1 and PT2 given in Fig. 16 and 17 reveal other key differences between Compressor 3 and the other two cases. Even though for PT1, there is an entropy increase where the vortex passes through the shock, the other high entropy regions located further downstream at both the rotor suction and pressure side are more severe. At PT2, these regions appear to have been improved more than the mid-passage one directly downstream of the shock.

Overall, the loss and entropy plots suggest that even though shock-vortex interaction also plays a role in the flow field of this rotor, there may be other instability mechanisms involved which are not necessarily located directly at the rotor tip. The role of the non-constant casing annulus line should be pointed out,



FIGURE 15. RADIAL DISTRIBUTION OF ROTOR PRES-SURE LOSS COEFFICIENT AT SELECTED POINTS FOR COMPRESSOR 3.

TABLE 2.SUMMARY OF RESULTS WITH AND WITHOUTCASING TREATMENT FOR ALL THREE COMPRESSORS. COM-PRESSOR 3 VALUES ARE FROM ROTOR ONLY CALCULATION.

Parameter	Compressor 1	Compressor 2	Compressor 3
SM _{Baseline}	15.3%	8.9%	9.2%
SM _{CT}	18.4%	12.6%	10.3%
ΔSM_{rel}	20.3%	41.6%	12.0%
$\Delta\eta_{p,WL}$	0.51%	0.2%	0.01%
$\Delta\eta_{p,NS-Baseline}$	0.70%	0.74%	0.31%

since it helps to locally decrease the diffusion at the rotor tip, thus reducing the aerodynamic loading in this region compared to the other radial heights of the blade.

3.4 Comparison of all compressors

The parameters listed in Tab. 2 give a quick overview of the results described earlier. The values for Compressor 3 are taken from the rotor only calculations. The comparison of all three compressors shows that the casing treatment leads to the highest relative surge margin increase on Compressor 2. Moreover, it also has the highest near stall efficiency increase. Compressor 3, which is regarded as the case with the closest resemblance to an actual stage of a jet engine compressor, shows the smallest



FIGURE 16. DISTRIBUTION OF NORMALIZED ENTROPY AND MA=1 (WHITE LINE) AT 99% BLADE HEIGHT FOR COMPRESSOR 3 WITHOUT CT AT PT1.



FIGURE 17. DISTRIBUTION OF NORMALIZED ENTROPY AND MA=1 (WHITE LINE) AT 99% BLADE HEIGHT FOR COMPRESSOR 3 WITH CT AT PT2.

efficiency and relative surge margin increase.

Deriving specific rules that could be used to predict the working range extension for other rotors is difficult due to the number of potentially relevant parameters and their specific combination found in the three compressors under consideration. A larger tip clearance, for example, has been demonstrated to promote higher surge margin increases due to casing treatment [11], [24]. According to this, Compressor 1 should have exhibited the highest surge margin increase. It also has the highest inlet Mach number as well as lowest solidity, which should also promote the effectiveness of any measure that aims to counteract the instability due to tip clearance vortex breakdown at the passage shock. On the other hand, Compressor 2 has higher diffusion across the blade row, which could also be an indicator. Similar considerations based on tip loading, incidence angle or static pressure distribution at the rotor tip section also do not lead to clear answers. It can be stated that for the cases considered in this paper, the casing treatment works best on the compressor with what appears to be the most pronounced vortex-shock based instability, i. e. Compressor 2. However, this flow feature is difficult to predict during the early stages of a compressor design, where high fidelity CFD results may not be available. Furthermore, there is the challenge of translating the susceptibility of a given rotor to vortex-shock instability into a potential working range increase, which to the knowledge of the authors currently cannot be reliably quantified. That is also not the intention of this paper, which only aims to demonstrate a casing treatment that can be transferred from one compressor to another without any aerodynamic design iterations while providing the desired influence on compressor performance on a qualitative level.

4 SUMMARY AND CONCLUSIONS

It was predicted that the single circumferential groove casing treatment presented in this paper can be successfully applied to different axial compressor rotors without further design iterations. For all three rotors investigated in this study, the treatment lead to increases in both working range and efficiency. At design point conditions, the impact on the aerodynamic performance parameters is negligible.

Despite the fact that the rotors are different in terms of loading, Mach number, solidity and other key parameters, the detailed flow analysis revealed that the rotors also had some relevant commonalities:

Besides obviously being transonic, all rotors

- 1. are tip critical, meaning their working range is limited by the tip region of the blade and more precisely
- 2. have a stall mechanism based on shock-vortex interaction.

The treatment works by local flow redistribution which leads to a smaller blockage build-up across the shock, enabling the tip clearance vortex to pass the shock without bursting. It is therefore concluded that the most important criterion for the successful application of the circumferential groove is the presence of a shock-vortex interaction instability mechanism and that other parameters may have an impact on the magnitude of the working range extension and change in efficiency, but are not critical in determining if the treatment will work at all or not.

Overall, although other types of casing treatment may offer greater working range extensions for selected compressors, the inclined single groove is attractive due to its simplicity (both geometrically and from a design effort point of view), absence of negative impact on compressor efficiency and the possibility to be read across from one compressor to another.

ACKNOWLEDGMENT

The authors would like to thank Rolls-Royce Deutschland Ltd & Co KG for the permission to publish this paper. Furthermore, Georg Reichstein is acknowledged for his contribution.

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