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EFFECT OF THE AXIAL COMpressor TIP CLEARANCE SIZE: PERFORMANCE AND TRANSITION TO ROTATING STALL

Clémence Rannou
DAAA, ONERA, Université Paris Saclay
F-92190 Meudon - France
Univ. Lille, CNRS, ONERA
Arts et Metiers Institute of Technology
Centrale Lille Institut, UMR 9014-LMFL
Laboratoire de Mécanique des Fluides de Lille
Kampé de Fériet F-59000, Lille, France
clemence.rannou@onera.fr

Antoine Dazin
Univ. Lille, CNRS, ONERA
Arts et Metiers Institute of Technology
Centrale Lille Institut, UMR 9014-LMFL
Laboratoire de Mécanique des Fluides de Lille
Kampé de Fériet F-59000, Lille, France
antoine.dazin@ensam.eu

Julien Marty
DAAA
ONERA
Université Paris Saclay
F-92190 Meudon - France
julien.marty@onera.fr

Geoffrey Tanguy
Univ. Lille, CNRS, ONERA
Arts et Metiers Institute of Technology
Centrale Lille Institut, UMR 9014-LMFL
Laboratoire de Mécanique des Fluides de Lille
Kampé de Fériet F-59000, Lille, France
geoffrey.tanguy@onera.fr

Lionel Castillon
DAAA
ONERA
Univ. Lille, CNRS, ONERA
Arts et Metiers Institute of Technology
Centrale Lille Institut, UMR 9014-LMFL
Laboratoire de Mécanique des Fluides de Lille
Kampé de Fériet F-59000, Lille, France
lionel.castillon@onera.fr

Joseph Moubogha
Univ. Lille, CNRS, ONERA
Arts et Metiers Institute of Technology
Centrale Lille Institut, UMR 9014-LMFL
Laboratoire de Mécanique des Fluides de Lille
Kampé de Fériet F-59000, Lille, France
joseph.moubogha@gmail.com

ABSTRACT

The leakage flow that is developed between the rotor and the casing of an axial compressor is usually associated with losses and can promote the arising of rotating stall. During its life, an axial compressor can be subject to a variation of the tip clearance due to the aging of the machine, which can modify the leakage flow and affect its performance. The prediction of the tip leakage flow and of its evolution with gap size is thus essential to estimate correctly the performance and operability of a compressor stage. The support of this study is a single stage compressor for which three different gap size ratios ($R = \frac{\tau}{c} = 3.4\%$, defined by the tip gap size $\tau$ over the axial chord $c$) are considered ($R = 0.6\%, 1.2\%, 2.4\%$). Reynolds Averaged Navier-Stokes (RANS) simulations are performed on the three configurations. The analysis is performed with two types of computational domains (a single blade-channel and one tenth of the full annulus geometry). The aim is to identify the configuration which provides reliable results at the most reasonable computational cost. Then, a numerical study on the effect of the tip gap size on the performance and flow topology of the compressor is conducted, especially at near stall operating conditions.

Keywords: Rotating stall, tip clearance, tip leakage vortex.

NOMENCLATURE

$A$ Annulus area at the rotor inlet ($A = 0.1029~m^2$)
$c$ Axial chord (mm)
\[ \rho \quad \text{Density} \ (\rho = 1.225 \text{ kg/m}^3) \]
\[ \Delta P_s \quad \text{Static pressure variation} \ (\text{Pa}) \]
\[ \lambda \quad \text{Skin friction magnitude} \ (\text{kg/m/s}^2) \]
\[ LE \quad \text{Leading Edge} \]
\[ \phi \quad \text{Flow coefficient} \text{ defined as } \frac{\dot{m}P_A}{\rho U} \ (-) \]
\[ Pr \quad \text{Relative Pressure} \ (\text{Pa}) \]
\[ Ps \quad \text{Static Pressure} \ (\text{Pa}) \]
\[ Pt \quad \text{Total Pressure} \ (\text{Pa}) \]
\[ \Pi \quad \text{Total Pressure Ratio} \ (-) \]
\[ \dot{m} \quad \text{Mass Flow} \ (\text{kg/s}) \]
\[ \eta \quad \text{Isentropic efficiency} \ (-) \]
\[ R \quad \text{Ratio defined by the tip gap size over the axial chord} \]
\[ RANS \quad \text{Reynolds-Averaged Navier-Stokes} \]
\[ S \quad \text{Entropy} \ (J/kg/K) \]
\[ SBC \quad \text{Single-Blade Channel} \]
\[ \tau \quad \text{Tip gap size} \ (\text{mm}) \]
\[ TE \quad \text{Trailing Edge} \]
\[ U \quad \text{Blade velocity} \ (\text{m/s}) \]
\[ URANS \quad \text{Unsteady Reynolds-Averaged Navier-Stokes} \]
\[ V_x \quad \text{Absolute axial velocity} \ (\text{m/s}) \]
\[ x, y, z \quad \text{Cartesian coordinates} \]

**INTRODUCTION**

To achieve the constraints which are imposed to aeronautical engine manufacturers in terms of \(CO_2\) and pollutant emissions, the performance and efficiency of each engine component have to be improved. Numerical simulation is thus an essential tool to allow these improvements.

For axial compressors, the clearance size between the casing and the rotor plays an important role in the performance both at nominal operating point and off-design conditions. As an example, according to the work of Cumpsty [1], increasing the ratio \( R = \frac{\tau}{c} \), defined by the tip gap size \( \tau \) over the axial chord \( c \), from 0.9% to 3.4%, considerably reduces the isentropic efficiency and the operating zone due to a reduction of almost 10% of the stall safety margin. In a recent experimental study, Doukelis et al. [2], focused their research on a cascade of a ring compressor for 4 different clearance sizes. They observed a rise in losses in the tip gap region with the increase of the clearance size. This effect was mainly due to the increase in the size of the tip leakage vortex. This vortex is defined as a highly unsteady phenomenon formed by the interaction between the tip leakage flow and the main flow. It initiates losses generated by the flow recirculation which result in a decrease of the local work provided by the rotor. Finally Doukelis et al. [2] showed that the tip leakage vortex separation point was moving toward the trailing edge for larger tip gap sizes. Inoue et al. [3], also noted this behavior with lighter variations than observed by [4] with large clearance sizes \( R > 4\% \). For clearance sizes between 1% and 4% of the chord length, Chen [5] demonstrated that the decrease in efficiency and that the blade loading vary linearly with the clearance size. Smith [6] showed that the peak pressure rise of a blade is inversely proportional to the tip gap size with 4% additional pressure losses for every percent of increase in the tip gap size.

From an operability point of view, Hewkin-Smith [7] showed that there is an optimal tip gap size, on the basis of numerical simulations. Thus, the maximum stall safety margin is obtained for a small, optimal and non-zero clearance size. From a case with no gap to one with an optimal size, the surge margin decrease would be due to the existence of flow separation at the casing which increases as the clearance size is reduced. In this area, the tip leakage vortex energizes the flow and reduces flow edge separation at casing, and thus improves the flow incidence conditions. For clearance sizes larger than the optimal one, the vortex becomes more intense (which is consistent with the work of Storer and Cumpsty [8]). To sum up, Hewkin-Smith et al [7], characterized: \( i \) cases with large tip clearance sizes, defined by a “tip leakage jet” mechanism where blockages of the tip leakage flow are mitigated by the leakage momentum, providing possible backflows. \( ii \) cases related to small tip gap sizes, dominated by a casing corner separation. For the latter, rotating stall arising is due to the absence of flow spillage and hence, by blockages near trailing edge, inducing high incidence of the flow. Additionally, numerical studies of Vo et al [9] focused on larger tip gap sizes for axial compressor at low speed. They have highlighted two criteria related to tip leakage flow based on single-blade channel configuration and multiple blades passages. This allows to describe the formation of a spike disturbance, precursor of rotating stall. By performing comparison between the two configurations, the authors underlined the need of the SBC configuration to understand compressor stall. Recently, Brion [10] experimentally studied the clearance flow via an isolated NACA0012 profile. The aim of the study was mainly to investigate vortex instabilities. For clearance sizes less than 2%, the clearance flow is practically perpendicular to the chord i.e. the vortex forms away from the rotor blade, close to the hub. For tip gap sizes greater than 2%, the vortex remains close to the profile. The clearance size therefore influences the behavior of the clearance vortex but also the entire flow topology. Kang and Hirsch [4] analyzed clearance sizes of 1%, 2% and 3.3% with experimental observation on NACA65 type blades. The horseshoe vortex was detected only for small tip gap sizes. Accordingly, the impact of various flow topologies affects losses due to different tip gap sizes directly.

The aim of this paper is to evaluate the ability of a numerical methodology based on RANS and URANS simulations to predict the performance curve of an axial compressor for three different tip gap sizes. The modification of the flow induced by the variation of the clearance is also evaluated. This numerical study was also conducted to identify a new larger tip gap size in order to support an experimental campaign on the CME2 test bench. The present work is thus the first step of an joint experimental and numerical study which aims at evaluating the effect of the tip gap
size on the performance, stall inception and efficiency of active flow control for surge margin improvement. Compared to the work of [9], the present study focuses on smaller tip gap sizes, closer to the optimal clearance defined by Hewkin-Smith [7]. The paper is organised as follows: first, the methodology developed for capturing flow behavior within a CFD analysis is described. Then, the flow mechanisms and their effect on the performance are investigated. Finally, the results obtained with two different computational domains will show the benefit of each in order to effectively implement a simulation to capture tip gap size effect.

MATERIALS AND METHODS

CME2 Setup

The considered test-case is the compressor CME2 (Figure 1) located at the Lille Fluid Mechanics Laboratory (LMFL). It is a single stage low speed axial compressor. The rotational speed is 3200 RPM. The numbers of blades of the rotor and the stator are respectively 30 and 40, giving a periodicity of $2\pi/10$. Hence, a configuration based on this periodicity has been chosen for simulations as it is of intermediate complexity between a single blade channel and full annulus configuration. It is able to capture many of the first order flow phenomena, such as rotor/stator interaction, but at a reasonable computational cost. At nominal conditions, the mass flow is 5.3 kg/s. Table 1 summarizes the main characteristics of the compressor. The tip gap of the experimental rotor is 0.5 mm. More details are given in [11]. The impact of the tip gap size on the compressor performance will also be evaluated for a 1 mm and 2 mm gap size.

<table>
<thead>
<tr>
<th>TABLE 1. CHARACTERISTICS OF THE CME2</th>
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<tbody>
<tr>
<td>Casing diameter</td>
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<tr>
<td>Hub-tip ratio, LE</td>
</tr>
<tr>
<td>Rotor tip chord</td>
</tr>
<tr>
<td>Rotor tip stagger angle</td>
</tr>
<tr>
<td>Rotor tip gap</td>
</tr>
</tbody>
</table>

Numerical Setup

A first computational domain consists of a single-blade channel composed of one rotor blade and one stator blade (Figure 2). This configuration is referred as 'Single Blade Channel' (SBC). The mesh is composed of $3.1 \times 10^6$ nodes. The $y^+$ values at the blade wall are lower than 1 in the entire domain. The $2\pi/10$ configuration is obtained by duplication of the SBC one. For the two configurations, when the tip gap size is changed, the number of point has been adjusted to maintain the cells aspect ratio. RANS (SBC and $2\pi/10$) and URANS ($2\pi/10$) calculations are performed with the elsA solver [13], developed at ONERA and co-owned by ONERA and SAFRAN. This code relies on a cell-centered finite-volume discretization on structured multi block meshes. A convergence study has been conducted with $10^6$ and $2.10^6$ nodes in order to study the compromise between computational cost and reliability in previous papers as quoted in [14]. Reynolds-Averaged Navier-Stokes equations are solved for compressible flows. Ideal gas law characterizes the fluid with the Sutherland’s law for the viscosity. The heat fluxes are computed with the Fourier’s law and diffusive fluxes are calculated with a classical second-order-centered scheme. Turbulence is modeled with the Spalart–Allmaras model [15]. The Roe solver is used as second order centered space discretization in order to compute the transport equations of turbulence models. Time integration is solved by a backward Euler scheme. Local time stepping is also applied to enhance the convergence rate in steady flow conditions. The boundary conditions that have been implemented are summarized in Table 2. A mixing plane is used in order to resolve the rotor-stator interface and connect the rows, in RANS calculations.

FIGURE 1. CME2 TEST BENCH FROM [12]

FIGURE 2. CME2 COMPUTATIONAL DOMAIN
TABLE 2. BOUNDARY CONDITIONS

| Upstream | - Subsonic inlet condition, with prescribed total pressure | - Axial flow direction | - $P_t = 101325$ Pa, $T_t = 288.15$ K |
| Blades, casing, hub | - Adiabatic wall condition | - Fixed wall condition for casing and a part of the hub | - Mobile wall condition for hub and blades |
| Downstream | - Subsonic outlet condition with radial equilibrium using a valve law on static pressure |

RESULTS AND DISCUSSION

Validation case

The reference case corresponds to the CME2 configuration. The nominal flow rate is 5.3 kg/s. Stall is triggered experimentally by throttling progressively a valve. The critical flow for which the compressor encounters stall is generally between 4.1 kg/s and 4.2 kg/s. The simulations investigate a range of mass flow rates from 2.8 kg/s to 6.2 kg/s. Among the conditions considered, several operating points are identified as follow:

- Nominal conditions ($\phi = 0.53$)
- Near stall conditions ($\phi = 0.43$)
- Stall conditions ($\phi = 0.41$)
- Post-stall conditions ($\phi < 0.41$)

The simulations of the single blade channel (SBC) and the $2\pi/10$ configurations are compared with experimental data from the LMFL [11]. In Figure 3, the performance curve from the URANS $2\pi/10$ simulations provides excellent results when compared with the experimental data. The discrepancies are less than 1% at nominal or stall conditions. RANS numerical results from the $2\pi/10$ and SBC configurations are also in good agreements with experimental data. Indeed, at the nominal conditions ($\phi = 0.53$), the numerical pressure ratio computed for the full compressor stage is under-predicted compared with the experimental data, which corresponds to about 1.5% difference. The largest deviation from experimental data is for the conditions close to stall ($\phi = 0.43$) with a difference of about 5%. However, the critical mass flow corresponding to $\phi = 0.41$ for which the onset of stall condition is observed, is well predicted by the numerical results.

For a mass flow corresponding to $\phi = 0.43$, at near stall operating conditions, the numerical static pressure field for the SBC and $2\pi/10$ configuration is compared with experimental data obtained at casing from [11] based on phase lock averaged measurement (Figure 4). It was observed experimentally that at this high incidence, a separation occurred at the rotor leading edge (Figure 4: a). Thereby it was noted that the tip leakage vortex became parallel to a line joining the rotor leading edges. At this low flow
rate it was also observed that the vortex could split in two, as evidenced by the dashed lines in Figure 4: a. The numerical pressure fields present characteristics very close to the experimental ones whatever the numerical configuration considered (Figure 4: b and c): the tip leakage vortex is well displayed by a characteristic low relative pressure region at the leading edge whose orientation is similar to the experimental one. In addition, the vortex splitting observed in the experimental data (Figure 4: a) is also identified in the numerical fields, as shown by the two dashed lines for Figure 4: b and Figure 4: c.

The results obtained for the reference case based on SBC or 2π/10 configurations demonstrate a reasonably good agreement with the experimental data in terms of performances and flow topology prediction. In addition, the stall and close to post-stall conditions are well captured by the numerical data with acceptable convergence and massflow rate stabilization.

**Impact of the tip gap size**

The CME2 compressor has a clearance size (R = 0.6% or a ratio of 0.006) smaller than what is usually found in the literature. For example, the compressors investigated by Hewkin-Smith [7] and Deppe et al [16] have clearance ratios of respectively 0.018, and 0.015. A second clearance ratio for the CME2 geometry, with a value of 0.012 (R = 1.2%), is thus considered in the present study.

**Global performance assessment** The analysis was conducted on the three tip gap sizes (R = 0.6%, R = 1.2% and R = 2.4%) to compare the influence of this parameter on the performance. Based on the performance curves presented in Figure 5, two main observations given by the RANS simulations can be drawn in comparison with URANS results taken as references: i) The increase of the tip clearance is affecting the compressor performance for each operating point considered. The drop of static pressure variation \( \Delta P_s \) is of the order of 6% at the nominal operating conditions and reaches 10% in near stall conditions for the largest size, on the 2\( \pi \)/10 configuration. ii) For the 2\( \pi \)/10 configuration, a reduction of the stable operating range is also observed: the last stable point is at \( \phi = 0.41 \) for the smallest tip clearance whereas it is \( \phi = 0.45 \) for the largest tip gap size. According to NASA communications [17], the Stall Margin can be defined as follow: \( SM = (\frac{\Pi_s}{\Pi_n} - 1) \times 100 \). With \( m_n \) at nominal operating point (N) and at the stall point (S) and, \( \Pi_n \), at nominal operating point (N) and at the stall point (S). This definition is applied to the present case to analyze the tip gap size effect. The smallest tip gap size is taken as a reference. Thus, the value of \( \Pi_n \) and the \( m_n \) are evaluated for the smallest tip gap size (R = 0.6%) at nominal condition. The \( \Pi_n \) and the \( m_n \) are calculated for the different tip gap sizes. Table 3 gives the value of SM for each tip gap size for the two configurations. SM is decreasing as the tip gap size is getting larger. As already observed, the SBC fails to predict the Stall Margin for the two largest tip gap sizes.

Figure 6 shows the isentropic efficiency for all the tip gap sizes: losses are increasing as the tip gap size is getting larger. Moreover, it can be noted that URANS and RANS simulations predict comparable efficiency if the nominal flow rate is excepted for the largest tip gap size. Additionally, it has to be noted that for the larger tip clearances, the performance trend is not well captured by the SBC configuration, for the operating ranges close to stall compared with nominal to stall conditions. As a consequence, a comparison between SBC and 2\( \pi \)/10 domains is provided in the last part of this paper in order to determine the origin of the discrepancies between the two configurations. The present section focuses on flow fields obtained with the 2\( \pi \)/10 configuration to discuss the origin of the differences in the performance curves described above.

**TABLE 3. STALL MARGIN FOR SBC AND 2\( \pi \)/10 CONFIGURATIONS AT RATIO = 0.6 %, RATIO = 1.2 %, RATIO = 2.4 %**

<table>
<thead>
<tr>
<th>Ratio</th>
<th>SM (SBC)</th>
<th>SM (2( \pi )/10)</th>
</tr>
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<tbody>
<tr>
<td>0.6%</td>
<td>29.4 %</td>
<td>29.4 %</td>
</tr>
<tr>
<td>1.2%</td>
<td>23.4 %</td>
<td>13.3 %</td>
</tr>
<tr>
<td>2.4%</td>
<td>13.3 %</td>
<td>12.2 %</td>
</tr>
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</table>
At near stall conditions, observations are similar to nominal conditions concerning the skin friction magnitude for R = 0.6 % and R = 1.2 %, with a higher intensity at the casing wall (Figure 7: d, e). For R = 2.4 %, the compressor is already stalled and the flow topology is completely different compared with the reference case (Figure 7: d, f) with an appearance of large vortices on the suction side. A progressive migration of these structures seems to appear towards the trailing edge from the casing to the hub, when looking at the corner vortices between the case and the tip of the blade. This larger tip gap size presents a much higher entropy production at tip, and a larger extent in the radial direction (Figure 8: f). For Ratio = 1.2 %, highest entropy fields are also observed at the tip of the blade. On static pressure fields close to casing, for Ratio = 2.4 % (Figure 9: Near Stall, c), the tip leakage vortex has completely disappeared at this flow rate (φ = 0.43), due to the large flow recirculation that covers about 45% of the blade. This is confirmed by the earlier pressure drop occurring at φ = 0.45 which implies the development of stall at larger mass flow rate than for the two other configurations. For Ratio = 0.6 % and 1.2 %, (Figure 9: Near Stall, a, b), the splitting of the tip leakage vortex is observed with dashed lines. This picture also highlights the difference of incidence of the tip leakage vortex between these two clearance sizes. For R = 1.2%, there is a slight vortex migration which tends to be aligned with the leading edges of the adjacent rows (Figure 9: b). This modification of incidence could be one of the reasons for the earlier onset of stall for this tip gap compared to the reference case. The second reason can be linked to the shift of the pressure drop of the performance curve towards the higher flow rates, associated with the larger pressure losses occurring at φ = 0.43.

At stall conditions, vortices are observed for the clearance at R = 1.2% as stall happened. Similar to R = 2.4 %, the production of entropy is more developed at the tip of the blade (Figure 8: h, i). This is linked to the higher intensity of the tip leakage vortex which is also observed on the pressure static field at this case (Figure 9). This intensification of the tip gap flow is at the origin of the pressure drop observed on the performance curve for Ratio = 1.2 % at this operating point. Similar observations drawn from R = 2.4 % at near stall conditions can be also be extended to R = 1.2 % at stall conditions. On Figure 9: Stall, b, for R = 1.2 %, it can also be noticed that the flow is due to a non periodic solution : all the blade passages present almost no pressure rise. These two blade passages are also presenting large region of highest entropy (Figure 8: g) and a larger extension of the entropy production in the radial direction as well as for Ratio = 2.4 % (Figure 8: f, h). It is undoubtedly due to a blockage of the flow on a large part of the height of the inter-blade channel.

At post-stall condition, stall occurred for R = 0.6 %, and vortices appeared linked to a highest production of entropy and the pressure drop occurring at φ = 0.41.

Finally, the previous observations lead to the conclusion that the tip leakage vortex seems to influence the performance of the compressor and the onset of stall. The analysis of the fields given by the simulations, especially those of the static pressure or entropy, reveals a delay in the development of the stall phenomenon as the size of clearance decreases. Besides, the tip gap size has also an influence on the losses and thus on the performance curves as seen with the change in $\frac{\Delta P}{\rho U^2}$ observed between small and large tip gap sizes.
Comparison between RANS $2\pi/10$ and SBC simulations

RANS calculations have been conducted on different operating conditions for a SBC and a $2\pi/10$ configuration. This section aims at discussing the results obtained with these two configurations for the three tip clearances investigated, $R = 0.6\%$, $R = 1.2\%$, and $R = 2.4\%$.

Performance comparison

Figure 10:a presents a comparison of the performance curves for the tip gap size of reference. The curves for each configuration are likely the same for the rotor, the stator, and the whole stage. In terms of performance, it suggests that using a SBC or a $2\pi/10$ configuration will not make any difference, from nominal point to stall conditions.

Figure 10: b and Figure 10: c, present the results for the larger tip gap sizes. The two configurations are in good agreements on a large range of operating points. Nevertheless, at near stall conditions for $R = 1.2\%$, the pressure rise is higher and the stall is delayed for the SBC configuration and the stall. Similarly, for $R = 1.2\%$, before near stall conditions, the observations are the same. Moreover, according to Figure 11, the comparison in terms of performance can be done between the two configurations for the isentropic efficiency at the rotor at different conditions (Nominal case, Near Stall, Stall and Post-stall): essentially, all the curves show that stall onset occurred at the blade tip for all the tip gap size but slight differences exist above 0.6\% of blade height with overestimation of performance for the SBC design for the operating points closed to stall for each tip gap size. After stall occurred, the isentropic efficiency is still higher for SBC design. The differences in compressor performance are clearly due to the flow behavior in the rotor which is at the origin of the major part of the differences observed between the two numerical configurations.

Therefore, the flow field has to be investigated to understand why degrading the computational domain leads to a performance mis-prediction for the larger tip clearances.

Flow field comparison

The flow field can be investigated in terms of axial velocity variation for the two designs at the rotor inlet. Figure 12 presents the different operating conditions for the two configurations at each tip clearance. It appears that at nominal conditions, few backflow is visible for the $2\pi/10$ configuration but not for the SBC one. This variation is increasing at the near stall conditions, where larger backflow is observed.
at the tip, highlighting some areas of flow recirculation for the $2\pi/10$ configuration. Moreover, the SBC design seems to underestimate the intensity of the leakage flow at the tip, with less backflow compared to the $2\pi/10$ configuration which presents some areas of highest velocity decreases.

Focusing on the slices of static pressure at 97% of blade height (Figure 13) at near stall for $R = 1.2\%$, it can be noticed some variation in the intensity between the two configurations. This suggests that the location of tip leakage vortex has changed between SBC and $2\pi/10$ configurations, and seems to be more aligned with the LE of the adjacent rows in the latter case. This is confirmed by the higher intensity of entropy in the radial direction (Figure 14) in this case. Moreover, at nominal conditions, the periodicity of the flow is obtained for each design. A slight difference of intensity is shown, especially at the tip of the rotor blade where the entropy is higher for the $2\pi/10$ design. This could explain the variation of performance at nominal condition between the two configurations. At near stall conditions, a variation between the two configurations is also illustrated by the periodicity of the flow which is not observed with slight discrepancies of entropy along the rotor side.

Therefore, it seems that SBC configuration delays and mitigates stall phenomenon at larger tip gap sizes compared to the $2\pi/10$ computational domain. Even if a single blade channel can predict correctly the performance for narrow tip clearance, it seems that the $2\pi/10$ is the best configuration to implement simulations for larger tip gap.

**CONCLUSION**

This paper allows to compare the ability of RANS numerical methods to investigate the impact of the tip gap size on the performance of the CME2. The reference geometry ($R = 0.6\%$) was also compared with experimental data and URANS calculations were used as validation cases.

The rise in the tip clearance from $R = 0.6\%$ to $R = 2.4\%$ clearly leads to an increase of the tip leakage vortex size which conducts to additional pressure losses at the rotor stage. This affects the pressure rise through the whole compressor stage. These additional pressure losses are increased before near stall operating conditions. In addition, at this flow rate ($\phi = 0.43$), especially for Ratio = 1.2 % for which observations can be done,

![Figure 8: Entropy (J/kg/K) Field for R = 0.6 % (A), D), J), R = 1.2% (B), E), H), k)), R = 2.4% (C), F), I), L)) at Nominal Case, Near Stall, Stall and Post-Stall for a Transversal Slice at the Rotor Outlet for $2\pi/10$ Design](image)
FIGURE 9. STATIC PRESSURE FIELD (PA) FOR A) R = 0.6 % , B) R = 1.2 %, AND C) R = 2.4 % AT NEAR STALL AND STALL CONDITIONS AT 97% OF BLADE HEIGHT FOR 2Π/10 DESIGN

the tip leakage trajectory is shifted upstream and tends to interact with the adjacent blade leading edge. Besides, the pressure drop happens at higher flow rates for the larger tip clearances. These two points lead to an anticipated transition to stall in the configuration close to stall.

For the smaller tip gap (R = 0.6%), the Single Blade Channel and 2π/10 configurations are giving very similar results from nominal to stall conditions which are very close to experimental data. For the larger tip gaps (R = 1.2% and R = 2.4%), the SBC simulation is also giving satisfactory prediction of the performance on a large operating range but it fails to predict i/ the additional losses observed before and at near-stall operating conditions and ii/ the anticipated stall transition.

Full annulus calculations using URANS combined with experimental measurements of the three tip gap sizes on the CME2 should provide further performance comparisons and flow topology assessments.

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REFERENCES


FIGURE 10. COMPARISON OF PERFORMANCE MAPS AT DIFFERENT ROWS (ROTOR/STATOR/FULL DOMAIN) BETWEEN SBC AND 2Π/10 SIMULATIONS AT THE TIP GAP SIZE: A) R = 0.6%, B) R = 1.2%, AND C) R = 2.4%

FIGURE 11. COMPARISON OF ISENTROPIC EFFICIENCY (-) OF THE ROTOR ROWS AT NOMINAL CASE, NEAR STALL, (POST) STALL BETWEEN 2Π/10 AND SBC SIMULATIONS FOR R = 0.6%, 1.2%, 2.4%

FIGURE 12. COMPARISON OF AXIAL VELOCITY (M/S) AT NOMINAL CASE, NEAR STALL, STALL, (POST)-STALL BETWEEN 2Π/10 AND SBC SIMULATIONS, UPSTREAM OF ROTOR
FIGURE 13. COMPARISON OF STATIC PRESSURE (Pa) AT NEAR STALL BETWEEN SBC A) AND 2Π/10 B) SIMULATIONS FOR R = 1.2%, AT 97% OF BLADE HEIGHT

FIGURE 14. COMPARISON OF ENTROPY (J/Kg/K) AT NOMINAL CASE AND NEAR STALL BETWEEN SBC A) AND 2Π/10 B) SIMULATIONS FOR R = 1.2% AT ROTOR OUTLET


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