Effect of the Rotor Blades Part Span Shrouds on the Transonic Axial Compressor Operation Map

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Abstract. The paper presents the results of the investigation of the part span shroud effect on the transonic axial compressor operational map. The objective of the paper is a 4.5-stage transonic low-pressure compressor of the aero-derivative gas turbine engine. The study was performed with using 3D RANS calculations via Ansys CFX commercial solver. The numerical model was validated with experimental data for 4 speedlines. The standard k-ε turbulence model demonstrated good agreement with the experimental data while SST model unexpectedly overpredicted mass flow rate. The strong effect of the mesh aspect ratio in the radial direction was observed. A significant decrease of the mass flow rate and the compressor efficiency was shown for the rotation speed range 85% – 95%. The maximum mass flow rate decreasing was estimated as 3.5%, maximum efficiency deterioration – 1.5%.

1. Introduction
Modern gas turbine engines for civil aviation are developing to fulfill strict requirements for environmental compatibility, reduction of the specific fuel consumption, and mass-size parameters. The engine thermodynamic efficiency has to be improved to satisfy these demands [1]. Usually, this could be achieved with the growth of the low-pressure compressor (LPC) pressure ratio. However, it leads the engineers to challenge aerodynamic, strength, and vibration issues since the aerodynamic loading of the LPC stages also increase [2, 3]. For this reason, the first stages of the turbofan engines LPC often have vibration problems and, therefore, the shape of the rotor blades is commonly a compromise between high aerodynamic efficiency and reliability [3, 4, 5].

The mistuning of the LPC rotor blades typically could be provided with using the part span shrouds at the rotor blades or utilizing the special wide chord blade design [6]. The first approach is traditional and in use from the early 70s. The wide chord blade design is a state-of-the-art technology that is based on the complex 3D blade shape and blisk blade/disc assembly. Using the wide chord design allows increasing the mass flow rate and a slight excess of efficiency as mentioned in [6]. Nevertheless, a wide chord design is characterized by thicker airfoils and a higher friction area compared with a common rotor blade with a part span shroud. Peters et al. [7] demonstrated that these issues significantly affect stage efficiency due to the growth of friction and secondary losses.

Wadia and Szucs [6] conducted an aerodynamic comparison of the rotor blade with part span shroud with its modification, redesigned according to wide chord principles. Original geometry was taken from [8]. They showed that at design speed the wide chord design provides significant efficiency deterioration due to the mass flow rate span reallocation. However, this difference reduces with the reduction of the rotation speed.
Many authors have shown that implementation of the part span shroud provides additional losses. Ware et al. [9], Sitaram and Sivakumar [10] experimentally investigated the transonic fan stages with part span shroud. Koch and Smith [11], Roberts [12] performed the part span shroud loss correlations in terms of a drag coefficient. The literature review demonstrated that these correlations are still relevant. Zhou et al. [13] performed a CFD investigation of the part span shroud on the aerodynamics of a transonic fan stage. The blockage effect was estimated at approximately 2.1% at single speedline. They also demonstrated growth of the stall margin for the shrouded case. Liang et al. [14] numerically estimated the part span shroud effect on the design speedline of the transonic 1.5 stage fan of aero-engine. The blockage effect of the part span shroud was estimated as 0.06 – 0.12%, efficiency deterioration in comparison with the unshrouded stage was estimated as 0.09 – 0.3%.

The similar part span shroud effect of blockage and efficiency deterioration also typical for the last stages of the powerful steam turbines. Bruggemann et al. [15] demonstrated that the blockage effect of the part span shrouds in the last stage of the low-pressure steam turbine could reach up to 2%. Sreekumar and Varpe [16] estimated efficiency decrease of the powerful steam turbine last stage due to part span shroud as 0.75 – 1.7%.

The literature review demonstrates very limited information concerning the effect of part span shrouds on a compressor stage operation map. Moreover, investigations of its effect on the whole LPC operation map are absent. The present paper presents such analysis for a transonic LPC. It includes a methodology section with a description of the numerical model and its validation procedure. The results section provides an analysis of the flow fields and the LPC operation map. An example of General Electric LM2500 + aero-derivative engine shows that implementation of the part span shroud is still applicable [17] and, hence, the scope of the paper is relevant.

2. Methods

2.1 The Object of the Investigation

The object of the investigation is a 4.5-stage transonic low-pressure compressor of the aero-derivative gas turbine engine. The compressor flow path is represented in figure 1, its main geometric parameters and operation conditions are provided in table 1. The inlet guide vane has a variable geometry part to control the mass flow rate through the compressor. The first and second stages have part span shrouds due to vibration issues (see figure 2).

The radial gap size was estimated on the base of steady-state thermal and structural simulations at the maximum speed design point. The real part span shrouds geometry was simplified in the first part of the numerical model validation to provide a structural hex-mesh generation with adequate quality. The simplified (ideal) shrouds have constant cross-section taken from the real shrouds middle section as illustrated in figures 2b, 2d.

![Figure 1. Representation of the compressor flow path (the part span shrouds are hidden).](image-url)
Table 1. The main geometric parameters and operation conditions of the compressor.

|  | Z  | $\Delta_{rad}$, mm | $D_{m}$, m | $p_{in}$, Pa | $T_{in}$, K | $u_{tip\ max}$, m/s | $n_{max}$, rev/min |
|---|---|---|---|---|---|---|---|
| IGV | 19 | - | 0.591 | 101325 | 288.15 | - | - |
| R1 | 34 | ~0.5 | 0.620 | 448.9 | 10200 |
| S1 | 44 | - | 0.620 | - | - |
| R2 | 42 | 0.5 | 0.635 | 422.6 | 10200 |
| S2 | 74 | - | 0.635 | - | - |
| R3 | 57 | ~0.5 | - | 405.8 | 10200 |
| S3 | 84 | - | 0.635 | - | - |
| R4 | 38 | ~0.5 | 0.626 | 390.4 | 10200 |
| S4 | 82 | - | 0.626 | - | - |

Figure 2. The geometry of real R1 / R2 rows shrouds ((a) / (c)) versus the ideal R1 / R2 rows shrouds ((b) / (d)).

2.2 Numerical Model

The numerical model is provided in Ansys CFX commercial solver. Steady-state Reynolds Averaged Navier-Stokes (RANS) simulations were performed. 1 blade channel per each passage was modelled with rotational periodicity boundary condition instead of 360°-degree modelling. The stator and rotor domains were coupled with using Stage interface with the velocity averaging option. The summary of the solver settings is presented in table 2.

Table 2. The summary of the solver settings.

| Parameter            | Setting                                  | Parameter         | Setting         |
|----------------------|------------------------------------------|-------------------|-----------------|
| Approach             | Steady-state RANS                        | Heat transfer     | Adiabatic walls |
| Advection scheme     | Adaptive 2$^{nd}$ order (High resolution)| Energy option     | Total energy    |
| Timescale            | Physical, 2e-5 s                         | Turbulence model  | various         |
| Working fluid        | Air (ideal gas): Cp – zero-pressure polynomial function, $\mu$ – Sutherland equation, $\lambda$ – const. |

The grid convergence study was performed for the structural hex-meshes for all blade rows with using SST turbulence model with reattachment modification (SST-RM). Cornelius et al. [18] demonstrated sufficient accuracy of this approach. The radial gaps have 17 elements with mesh refinement near the walls as shown in figure 3a. Mesh resolution near the leading edges is illustrated in figure 3b. The first near-wall cell height was set to fulfill the conditions $y_{max}^+ < 4$, $y_{avg}^+ < 3$. The results of the grid convergence study are presented in figures 4a, 4b for two compressor speedlines. The obtained results demonstrate that a medium grid resolution of ~550k nodes per blade row provides sufficient accuracy.
in terms of the mass flow rate and compressor efficiency which correlates with the results of Zheng and Yang [19].

![Illustration of the mesh resolution at the radial gap (a) and leading edge hub region (b)](image)

**Figure 3.** Illustration of the mesh resolution at the radial gap (a) and leading edge hub region (b)

![The results of the grid convergence study (hex-meshes, ideal shrouds, SST-RM)](image)

**Figure 4.** The results of the grid convergence study (hex-meshes, ideal shrouds, SST-RM).

Further grid investigations with using tetra-meshes with prism layers ($y_{max}^+ < 4$, $y_{avg}^+ < 3$) demonstrated the tendency of decreasing the discrepancy between CFD and experimental results in terms of mass flow rate and efficiency as illustrated in figure 5a. The comparison of the static and total pressure span variations (see figure 5b) shows a significant difference in the region where tetra-mesh has lower aspect ratios (less than 1000). The possible reason for this situation is an occurrence of a significant radial velocity component since the cells with the maximum aspects in this area are stretched in the radial direction. For this reason, the tetra-meshes with prism layers were used for final simulations.

### 2.3 The Speedline Calculation Procedure

The speedline calculation was performed automatically with a sequenced step reduction of the outlet boundary condition. Total parameters (pressure, temperature) at the inlet and corrected mass flow rate at the outlet were used as the boundary conditions. When the absolute value of the pressure ratio fall exceeded a prescribed value (near-stall point), the simulation was interrupted by the specified interrupt condition. The whole procedure is illustrated in figure 6. It is noteworthy that the mass flow rate stabilization from the preceding outlet boundary condition change significantly delaying versus the pressure ratio behaviour.
2.4 The Effect of the Turbulence Model

Results of the turbulence model variation study are presented in figures 7a, 7b in terms of $\pi_{t-t}(G)$ and $\pi_{t-t}(\eta_{t-t})$ curves for hex-meshed rows with ideal part span shrouds geometry. SST model without reattachment modification demonstrated a significant underprediction of the stall point. The realizable $k-\varepsilon$ model demonstrated unstable behaviour. For these reasons, both results are omitted. The standard $k-\varepsilon$ model was tested both with and without curvature correction modification, but it does not demonstrate a significant difference.

SST-RM model shows a significant mass flow rate and pressure ratio overprediction (see figure 7a). It also provides too optimistic efficiency level at 70% and 80% rotation speeds while the $k-\varepsilon$ model slightly overpredicts efficiency at 90% and 100% (see figure 7b). It should be outlined that high discrepancies with experiment for both models at 90% speed are related to the grid issues since this part of the validation was performed with using hex-meshes with high aspect ratios.

Generally, the standard $k-\varepsilon$ model shows lower discrepancies between CFD and experimental data. This correlates with the results of Piovesan et al. [20] for the transonic compressor cascade. Therefore, the standard $k-\varepsilon$ model was used for further investigations.
Figure 7. Comparison of the compressor speedlines ($\pi_{t-t}(G)$ (a) and $\pi_{t-t}(\eta_{t-t})$ (b) curves) obtained with using different turbulence models (hex-mesh, ideal shrouds).

3. Results

The numerically obtained compressor speedlines for both with and without part span shrouds configurations are presented in figures 8a, 8b in terms of $\pi_{t-t}(G)$ and $\pi_{t-t}(\eta_{t-t})$ curves. The results show that the part span shroud effect becomes significant at rotation speeds of approximately higher than 85%. At 88.6% speed the mass flow rate decrease compared with unshrouded blades is estimated as 2.1% (see figure 8a), compressor efficiency deterioration – as 0.5% (see figure 8b). At 90% rotation speed the mass flow rate fall exceeds 3%, compressor efficiency deterioration – 1.3%.

The mass flow rate spanwise shift leads to velocities reallocation because of the part span shrouds. Figures 9a, 9b illustrate the span variation of the diffusion factor for the first and second rotor blades. The strong peaks correspond to the shrouds aerodynamic wakes. It can be seen that the shrouded blade rows have significantly lower diffusion factors at the tip sections. This positively affects the stall margin – for the shrouded case it tends to significantly increase, mostly because of the second stage (see figure 9b).

Figure 8. The influence of the part span shrouds on the compressor speedlines ($\pi_{t-t}(G)$ (a) and $\pi_{t-t}(\eta_{t-t})$ (b) curves) (NS – without shrouds, RS – real shrouds).
Figure 9. The Lieblein diffusion factor for shrouded and unshrouded R1 (a) and R2 (b) blade rows (design point at 88.6% speed, NS – without shrouds, RS – real shrouds).

Figure 10. The axial velocity span variation at 80% speed (a) and 88.6% speed (b) design points (NS – without shrouds, IS – ideal shrouds, RS – real shrouds).

Axial velocity spanwise profiles performed for 80% and 88.6% speeds design points show close to identical distributions for both shrouded and unshrouded cases excluding the part span shroud region as illustrated in figure 10. The strong velocity slumps in figure 10 are instigated by the shrouds aerodynamic wakes similarly to figure 9.

The shroud wake width in the radial direction has negligible differences at 80% and 88.6% speeds. Nevertheless, the slump of the axial velocity is slightly displaced towards the hub at 88.6% speed because of the intensification of the pressure side wake as illustrated in figure 11. An enhancement of the part span shroud effect with rotation speed growth is related to the intensification of its aerodynamic wake as illustrated in figures 10, 11. The axial velocity slump at 88.6% rotation speed is estimated as 28% of mean value while at 80% it is only 18.5%.

Figures 10, 11 also demonstrate no significant difference between ideal and real part span shrouds cases. At 88.6% speed design point real geometry provides stronger wake at the blade pressure side. Its effect on the speedline is negligible (see figures 10, 11) and, therefore, the speedlines for ideal shroud configuration are omitted in figures 8a, 8b.
The part span shroud effect eliminates at rotation speeds of approximately higher than 95% because of the first stage choking while at the lower speeds the second stage defines the mass flow rate through the compressor. This phenomenon is illustrated in figure 7 where 100% speedline is vertical.

![Ideal and real part span shrouds](image)

**Figure 11.** The wake of the 2nd stage part span shrouds at design points for 80% speed (a) and 88.6% speed (b)

4. Conclusion
The strong effect of the part span shroud on the 4.5-stage transonic LPC operation map was shown for the rotation speed range 85% – 95% on the base of 3D RANS calculations via Ansys CFX commercial solver. The numerical model was validated with experimental data for 4 speedlines. The standard k-ε turbulence model demonstrated good agreement with the experimental data while SST model unexpectedly overpredicted mass flow rate. The maximum mass flow rate decreasing was estimated as 3.5%, maximum efficiency deterioration – 1.5%. The main reason for the strong part span shroud effect is an intensive aerodynamic wake. Nevertheless, its role eliminates at rotation speed lower than 80% (low wake intensity) and higher than 95% (first stage choking).

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Nomenclature

- $C_p$: specific heat capacity at constant pressure (kJ/(kg·K))
- $l_{norm}$: span normalized coordinate
- $n_{max}$: maximum rotation speed (rev/min)
- $u_{tip\ max}$: maximum shroud circumferential velocity (m/s)
- $\eta_{corr,\ norm}$: normalized corrected mass flow rate
- $\pi$: normalized total-to-total pressure ratio
- $\eta_{t-t\ norm}$: normalized total-to-total efficiency
- $C_{ax}$: axial velocity (m/s)
- $D_m$: mean diameter (mm)
- $p$: pressure (Pa)
- $Z$: number of blades
- $\Delta_{rad}$: radial clearance (mm)
- $\lambda$: thermal conductivity (W/(m·K))
- $\mu$: dynamic viscosity (Pa·s)
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