AERODYNAMIC LOW FREQUENCY OSCILLATION AND NOISE REDUCTION USING A PARTIAL SHROUD IN A LOW PRESSURE TURBINE

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Abstract

The design of highly efficient low pressure gas turbines in compliance with strict emission regulations for pollutants and noise as well as long component life time requires the detailed knowledge of potential excitations in the flow. The analysis of fast response measurement data from a 1.5-stage low pressure axial turbine in the frequency domain shows multiple relevant excitation and noise sources in the flow field additional to the primary rotor-stator interaction. Especially non-synchronous low frequency modes at 10\% of the blade passing frequency occurring in the tip shroud exit cavity are found to communicate with the main flow and are amplified through the downstream stator row by a flow instability in the interaction area of passage vortex and boundary layer. Full-annular, unsteady CFD simulations are carried out in addition to the experiments to probe the origin of the cavity modes. They predict the rotor-stator interaction well, but struggle to resolve low frequency oscillations in the cavities. Out of the four tested configurations a shroud trailing edge cutback is the most efficient option to prevent the formation of the cavity modes. The trade-off for a low frequency fluctuation reduction of 12dB using a trailing edge partial shroud is a total-total stage efficiency drop of 0.7%.

Nomenclature

Variables:

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>$C_{ax}$</td>
<td>Axial chord</td>
<td>[-]</td>
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<tr>
<td>$\eta$</td>
<td>Efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$f$</td>
<td>Frequency</td>
<td>[Hz]</td>
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<tr>
<td>$\lambda$</td>
<td>Wave length</td>
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<tr>
<td>$\dot{m}$</td>
<td>Mass flow</td>
<td>[kg/s]</td>
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<td>Rotational speed</td>
<td>[rpm]</td>
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<tr>
<td>$p$</td>
<td>Pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$\Pi$</td>
<td>Pressure ratio</td>
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<tr>
<td>$Re$</td>
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<tr>
<td>$T$</td>
<td>Temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>$v$</td>
<td>velocity</td>
<td>[m/s]</td>
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Sub- and superscripts:

<table>
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<th>Symbol</th>
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<tr>
<td>$0$</td>
<td>stagnation flow quantity</td>
</tr>
<tr>
<td>$1.5$</td>
<td>One-and-one-half stages</td>
</tr>
<tr>
<td>$in$</td>
<td>Turbine inlet</td>
</tr>
<tr>
<td>$max$</td>
<td>maximum</td>
</tr>
<tr>
<td>$rel$</td>
<td>Rotor-relative</td>
</tr>
<tr>
<td>$RMS$</td>
<td>Root mean square</td>
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<tr>
<td>$stage$</td>
<td>Single stage</td>
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<td>$tt$</td>
<td>Total-to-total</td>
</tr>
<tr>
<td>$\bar{x}$</td>
<td>Time mean of $x$</td>
</tr>
<tr>
<td>$\hat{x}$</td>
<td>Engine order part of $x$</td>
</tr>
<tr>
<td>$x'$</td>
<td>Stochastic part of $x$</td>
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Abbreviations:
BPF Blade passing frequency
EO Engine order
IR Purge flow injection rate
LE Leading edge
PS Pressure side
R1 Turbine rotor
R1ex Rotor exit
S1 First stator
S1ex First stator exit
S2 Second stator
S2ex Second stator exit
SS Suction side
TE Trailing edge

Introduction

Jet engines are amongst the most sophisticated and multidisciplinary devices built today. The temperature and stress levels in different parts of the engine have been pushed in order to achieve more efficient aircraft propulsion. Typically, the interaction of different optimization goals in aerodynamics, thermodynamics and structural mechanics results in a trade-off. The superior aerodynamic efficiency and the mitigation of blade flutter are two main reasons for the wide use of shrouded blades in low pressure turbines [1]. However, for the new generation of high speed low pressure turbines the shroud aerodynamics has a significant impact on fatigue life. The quadratic increase in mean stress for higher rotational speeds results in lower allowable stress amplitudes and therefore the first representatives of this engine family feature unshrouded low pressure turbine blades ( [2], [3]). Especially for high pressure turbines a big variety of literature has been published on tip treatment, since the cooling fluid required to keep the shroud within acceptable temperatures has a major impact on the overall efficiency. Harvey [4] presents the most comprehensive overview on blade tip design. It is shown qualitatively that the resulting loss as a function of tip gap is twice as high for unshrouded turbines compared to shrouded designs. Squealer geometries have been studied as alternatives to shrouds by Kaiser et al. [5], Camci et al. [6], Mischo et al. [7] and others. Similar to the winglet geometries studied by Dey et al. [8] and Zhou et al. [9], the tip squealers do not provide additional stiffness to a blade row.

Partial shrouds therefore represent a potential solution providing a compromise between aerodynamic benefits and structural drawbacks. Aerodynamic effects of partial shrouds have been studied by Nirmalan et al. [10] in a cascade with three different scallop designs. Similarly to Porreca et al. [11], they found aerodynamic efficiency reductions in the order of 1% compared to a full shroud. The aerodynamic behavior of the four shroud geometries under consideration in this paper have been studied in a previous publication by Rebholz et al. [12]. The main findings relevant for this paper were the interaction of the over-tip leakage with the main flow and the propagation through the downstream stator row. For both leading edge (LE) and trailing edge (TE) cutbacks the total amount of leakage has been estimated numerically to remain constant, but the fluid exchange in the according cavities was enhanced by factors of up to two. Since the pumping is determined by the rotor pressure field also the flow field in the cavity is influenced more by the rotor motion than the dynamics of toroidal vortex systems as in the fully shrouded case. Tracking fluid with enhanced total temperature allowed identifying the dominant outflow locations from the shroud exit cavity in the middle of the second stator passage. The leakage fluid appeared to accumulate on the
suction side of the second stator down to 40% span.
All of the previously cited studies focus mainly on performance. However, potential high cycle fatigue sources as well as strict noise emission regulations require an analysis beyond integral quantities like efficiency. Fortunately, the noise signature of turbofans improves along with the increasing bypass ratio, since the emitted jet noise scales to the eighth power of the exhaust velocity as shown by Bushell [13] already in 1971. However, tonal noise like the blade passing frequency of the rows is still critical especially during landing and take-off, when the engines are pointed towards the ground. The highly unsteady nature of various other flow features creating tonal noise is difficult to capture experimentally and computationally. The cavities required for the integration of shrouded blades need to be designed primarily for clearance. Additionally to the forcing created by the interaction of the leakage and main flow shown by Gezork et al. [14], such enclosed volumes are susceptible to acoustic resonance and exhibit highly unsteady vortex dynamics, which have been analyzed by Barmpalias et al. [15] and others. The presence of highly swirling flows and vortical structures can also introduce instabilities which have been studied analytically and numerically (Blackburn et al. [16], Globulev et al. [17]). The complexity of this problem, which is derived from the curl of the Navier-Stokes equations, typically leads to a mathematical treatment and a simplification of the engine environment.
Due to the relation to the blade position, the rotor-stator interaction is typically well captured by numerical methods that are integrated in the design process. The underlying dynamics for the previously mentioned instabilities can result in non-synchronous fluctuations, which are not captured by standard industrial numerical methods. Also, periodic boundary conditions act as high-pass filters in circumferential direction and therefore may prevent the build-up of non-synchronous dynamics as in cavities as shown by Basol et al. [18]. Especially noise generation depends on the non-linear nature of the Navier-Stokes equations and typical acoustic sources like the Reynolds stresses have to be modelled accordingly. Karbasov et al. [19] treat them as source terms to determine the evolution of jet noise through an exhaust nozzle. More recently hybrid RANS/LES codes are used like by Nebenfuehr et al. [20].
Given the aforementioned challenges in fatigue and noise related problems, this paper presents an extended analysis of experimental results and a numerical approach to capture flow oscillations related to structural excitation and noise generation in cavities. The experimental data of four different shroud geometries has been transformed into the frequency domain at multiple measurement points in order to capture the whole frequency content of the signal rather than engine order components only. Using this approach in combination with tip shroud cavity instrumentation, a non-synchronous low frequency oscillation generated in the shroud exit cavity can be detected and traced through the turbine. The low damping characteristics through a stage as well as potential excitation of other flow instabilities may lead to significant low frequency vibrations and noise emissions. The non-acoustic nature of this cavity mode gives rise to a full-annular, unsteady RANS simulation that does not suppress the build-up of such a mode in pitchwise direction. However, the dominant impact of first engine order fluctuations in the proximity of the shroud seals is not modelled in the simulation and
leads to a deviation in the cavity frequency. Partial shrouds emerge from the analysis as a viable option to reduce mean blade stress compared to a full shroud while still benefitting from the superior aerodynamics and the robustness against flutter compared to unshrouded blades. The openings in the shroud platform lead to a dominance of the rotor pressure field rather than more independent cavity dynamics as for a full shroud. A shroud trailing edge cutback proves to be a viable option to reduce blade stress in high speed turbines as well as to suppress the generation of low frequency oscillations in the cavities by up to 12dB at the exit of the turbine compared to a full shroud.

**Experimental Method**

The experimental investigation was performed in the “LISA” research turbine at the Laboratory for Energy Conversion (LEC) at the Swiss Federal Institute of Technology in Zurich. A detailed description is presented by Behr et al. [21].

**Research Turbine Facility**

The research turbine shown in Figure 1 is a quasi-closed loop facility. The inlet pressure is generated by a radial compressor. The inlet total temperature $T_{0,in}$ is controlled to ±0.2K with a two-stage water to air heat exchanger and the mass flow is measured with a calibrated Venturi nozzle. A homogeneous flow field is created by a 3m flow conditioning stretch before the flow enters the test section. The acceleration in the contraction helps reducing flow non-uniformities. The flow undergoes a sub-atmospheric expansion through the 1.5 stages. After pressure is recovered to atmospheric level, the air loop is open to atmosphere downstream of the turbine. The recovery of the static pressure with a tandem de-swirl vane row is required due to the compressor’s limited compression of $\Pi_{c,max}=1.4$. The rotational speed of the turbine of 2700rpm is controlled by a DC generator to an accuracy of ±0.5rpm. The turbine torque is measured by a torquemeter. The first vane row exit flow is compressible with a Mach number of 0.52.

**Figure 1: Overview of the “LISA” test facility at the Laboratory for Energy Conversion (LEC) at ETH Zurich**

**Operating conditions**

The total-to-static pressure ratio across the 1.5 stage test section is kept constant for all measurements at $\Pi_{1.5}=1.65$. The inlet total temperature is also kept constant at $T_{0,in}=328K$. A constant amount of purge flow of 0.8% of the main mass flow (typically 11.8kg/s) is injected at the hub between the first vane and the rotor. The injection system is described in more detail by Schuepbach et al. [22]. Due to the opening to atmosphere at
the exit of the turbine all thermodynamic flow quantities are normalized by the inlet stagnation conditions. This procedure allows for an accurate comparison between different measurement days.

Table 1: Operating conditions and geometrical characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Π</td>
<td>1.65 ± 0.4 %</td>
</tr>
<tr>
<td>T₀,in (K)</td>
<td>328 ± 0.3</td>
</tr>
<tr>
<td>IR</td>
<td>0.8%</td>
</tr>
<tr>
<td>( \frac{m \sqrt{T_{m,0}}}{p_{0,0}} )</td>
<td>152 ± 0.2</td>
</tr>
<tr>
<td>( \frac{N}{\sqrt{f_{m,0}}} )</td>
<td>2.48 ± 0.05</td>
</tr>
<tr>
<td>Mach</td>
<td>0.52/0.28/0.48</td>
</tr>
<tr>
<td>Re</td>
<td>7.1/3.8/5.1x10⁵</td>
</tr>
</tbody>
</table>

Shroud Cutback Designs

The turbine configuration is based on the design used by Jenny et al. [23]. The end walls of the first vane row are profiled at hub and tip whereas the second vane has cylindrical end walls. The hub end wall profiling as well as the blade geometry of the rotor are identical to previous experiments and are the same for all cutback designs. For the baseline the shroud features a cylindrical end wall to allow for modifications. To provide more space for the shroud leading edge modification the first sealing fin is placed 5% of the axial cavity length (7% of the rotor tip axial chord) away from the cavity inlet, increasing the inlet cavity volume by 17%. The shroud leading edge platform is then reduced to the fillet of the blade, leaving a semicircular platform. This is an engine representative design and is referred to as LE cutback in this paper. For this case 11.6% of the total shroud material is removed. The geometry is shown in Figure 2 (a). A separate rotor featuring a partial shroud at the trailing edge, referred to as TE cutback, is designed based on the findings in Porreca et al. [24]. As shown in Figure 2 (b) the shroud trailing edge platform is cut back to the maximum extent while leaving the throat area of the turbine and the fillet unaltered. The overall material reduction equals 5.3% for this design, i.e. less than half compared to the LE cutback. The goal for the combined cutback was to have the same weight and therefore also the same stress reduction as for the LE cutback. The shroud trailing edge modification is identical to the TE cutback, but the shroud leading modification has an offset compared to the LE cutback.

Figure 2: The three tested shroud cutback geometries (a) leading edge (LE) cutback, (b) trailing edge (TE) cutback and (c) combined cutback

Measurement Planes

The traverse data presented in this paper was acquired downstream of the rotor and downstream of the second stator as shown in Figure 3. The spatial resolution of the measurement grid covered 42 radial and 41 equally spaced circumferential points covering one stator pitch. The radial resolution is refined close to the end walls. Pneumatic tappings and pressure transducers were installed on the outer tip shroud cavity wall. In the inlet cavity (1) and the exit cavity (3) in Figure 4 the axial resolution of the tappings is 3% of the axial cavity dimension and 6% for the transducers respectively.
Measurement Technology

The performance of the different cutback configurations is derived from the steady flow field which has been measured with a cobra-shaped pneumatic five hole probe (5HP) with a head diameter of 0.9mm. The unsteady flow field is captured with a two sensor Fast Response Aerodynamic Probe (FRAP), which has been developed at ETH Zurich (Kupferschmied et al. [25], Pfau et al. [26]) and has a head diameter of 1.8mm. The probe resolve flow field oscillations with frequencies up to 48kHz for the FRAP. The FRAP is operated in a virtual four sensor mode allowing the measurement of the three dimensional and time-resolved flow field. Table 2 shows the relative uncertainty of the 5HP and FRAP relative to the calibration range of ±30° for the yaw angle, ±20° for the pitch angle and as a percentage of the dynamic head for the total and static pressure. Three consecutive rotor blade passings are considered in the post-processing and phase-lock averaged 85 times. The data is acquired at a sampling rate of 200kHz over a period of 2s, which results in a frequency resolution in the frequency domain of 0.5Hz. For the frequency domain analysis only the pressure signal from the yaw sensor in the probe position aligned to the main flow is considered. The signal therefore qualitatively similar to the total pressure and contains the full frequency spectrum rather than engine orders only in the phase-lock averaged case. Yaw angle variations are also recorded as pressure fluctuations on the yaw sensors. Based on the phase-lock averaged results, deviations from the total pressure due to flow angle variations are estimated to be at most 1.5% of the inlet total pressure, but typically less than 0.2% for all measurement planes.

The uncertainty of the pneumatic tapping measurements of the static wall pressure on the outer shroud cavity wall is estimated to be 0.02% of the inlet total pressure. The expanded uncertainty for the time-resolved wall pressure measurements has been estimated to be 0.1% of the inlet total pressure by Behr [27]. The unsteady wall pressure is recorded with a sampling rate of 100kHz for 3s resulting in a frequency resolution of 1/3Hz.

<table>
<thead>
<tr>
<th></th>
<th>Yaw</th>
<th>Pitch</th>
<th>P₀</th>
<th>P</th>
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<tbody>
<tr>
<td>5HP</td>
<td>0.5%</td>
<td>0.8%</td>
<td>0.6%</td>
<td>1.0%</td>
</tr>
<tr>
<td>FRAP</td>
<td>0.8%</td>
<td>2.3%</td>
<td>1.0%</td>
<td>1.2%</td>
</tr>
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</table>

Figure 3: Sketch of the measurement plane locations upstream and downstream of the rotor and downstream of the second stator

Figure 4: Sketch of the tip shroud cavity. (1) Shroud inlet cavity, (2) intermediate cavity and (3) exit cavity. All dimensions are normalized by the rotor tip axial chord and the gap between sealing fins and outer casing is 0.7mm

Table 2: Relative uncertainty of the 5HP and the FRAP
Computational Methods

Grid and boundary conditions

In order not to restrict the flow behavior by periodic boundary conditions the computational domain of the baseline covers the full annulus of a 1.5 stage configuration. The mesh of the time-resolved simulations has been created in the commercial mesh generator “Ansys IcemCFD” and consists of approximately 100 million nodes. It is refined in the areas of interaction in the tip shroud cavity and uses matching interfaces in order to minimize interpolation errors both at the stage interfaces as well as at the shroud. The simulations do contain the hub rim seal upstream of the rotor without purge flow injection. The impact of the purge flow on the tip shroud cavity is marginal, since it only affects the flow field up to 60% span [23]. The boundary conditions at the inlet and the outlet of the turbine are extracted from performance measurements. At the domain inlet the total pressure and total temperature are specified accordingly. The turbulence intensity is set to 0.5% in the main flow upstream of the first stator. The exit static pressure is fixed at the hub and follows a radial equilibrium distribution along the span.

Solver

The current numerical study is conducted using LEC’s in-house, GPU-accelerated compressible URANS solver “MULTI3”. The numerical algorithm is based on the Ni’s Lax-Wendroff explicit time marching scheme which has second order accuracy both in space and in time. Turbulence closure is obtained using the Wilcox’s k-ω turbulence model. The solver is described in detail by Basol et al. [18]. A full revolution is discretized by 540 physical time-steps using 150 sub-iterations. The initial condition for the full-annular simulation is taken from a converged sector model of the domain. The computation has been carried out on 18 GPUs.

Validation

The convergence of the simulation has been judged based on the periodic correlation between two blade passings at midspan in the main flow at the exit of each stage. However, the presence of non-periodic flow features in the cavities prevents a conclusive statement on complete convergence.

Figure 5: Comparison between CFD and experiment at rotor exit for the baseline, circumferentially mass-averaged a) normalized relative total pressure and b) relative flow yaw angle.

Figure 6: Comparison between CFD and experiment of normalized static wall pressure at shroud casing.
Figure 5 shows a comparison between experiment and computation of the mass-averaged relative total pressure and relative yaw angle at rotor exit for the baseline. Since the flow field has been found to be affected by the purge flow injection up to 60% span only the tip part of the curves is considered for the validation. Here the relative total pressure shows good agreement and deviations of less than 0.5%. The underturning caused by the rotor tip passage vortex at 85% span is less than 2° above 60% span. In the hub region below 30% span the impact of the higher purge flow from the experiment can be noticed clearly. The pressure drop across the first shroud fin is captured well by CFD. However, the rotor inlet static pressure is under-predicted by approximately 0.8% as shown in Figure 6. This offset is also seen in the intermediate cavity, so the pressure drop across the second fin is under-estimated by approximately 1%.

Results and Discussions

The general flow field and the aerodynamic impact of different tip shroud cutbacks has been discussed in previously by Rebholz et al. [12] and presented at ISABE 2013 in Busan, Korea. The scope of this paper focusses less on the aerodynamics, but the impact on noise generation and potential structural excitation which is typically difficult to predict accurately, especially for non-synchronous flow excitations. Experimentally, however, the Fast Response Aerodynamic Probe (FRAP) technology enables highly accurate flow field measurements with a bandwidth of 48kHz. The high temporal resolution of the probe relative to machine characteristics like the blade passing frequency in combination with fine measurement grids of approximately 1700 points per measurement plane has proven to capture unsteady flow features in great detail. Due to the large amount of data and due to usage of the probe in a virtual four sensor mode it is required, however, to apply a phase-lock averaging in order to obtain the main flow quantities like total and static pressure, flow angles and Mach number. Using the rotor rotation as a reference therefore leaves all fluctuations in the data which are integer multiples of the rotational speed (engine orders). However, for noise and structural considerations the whole spectrum and especially the low frequency content are of interest. The signal decomposition [28] into a time-mean pressure $\bar{p}$, a periodic part $\tilde{p}(t)$ and a stochastic part $p'(t)$ according to equation (1) is suitable to identify flow regions containing fluctuations not related to the rotational speed. The periodic part of the signal is obtained by dividing the signal into $N$ revolutions at the same rotor/stator position and then ensemble average the pieces.

$$p(t) = \bar{p} + \tilde{p}(t) + p'(t) \quad (1)$$

In order not to lose any frequency content the raw signal of the yaw sensor $p_i(t)$ is used to calculate the difference to the phase-lock averaged signal. In this position the yaw sensor is aligned to the mean flow direction according to pneumatic probe measurements. The value of $p_i(t)$ is therefore qualitatively representative of the unsteady total pressure. As indicated by equation (2) the root mean square $p^2_{\text{RMS}}$ of the resulting difference $p_i'(t)$ returns increased values if there are fluctuations other than engine orders present in the signal. The values for $p^2_{\text{RMS}}$ are typically calculated for three rotor blade passing periods and the root mean square is calculated from $N=85$ instants in time where the rotor is
at the same relative position to the stator, i.e. once per revolution.

\[ p_{1,RMS} \left( 0 \leq t < \frac{3}{f_{\text{blade passing}}} \right) \]

\[ = \sqrt{\frac{1}{N} \sum_{i=0}^{N-1} \left( p(t + t/f_{\text{rot}}) - \bar{p} - p(t + t/f_{\text{rot}}) \right)^2} \]  

(2)

Non-synchronous flow perturbations

The most apparent fluctuations in turbomachines are inherent to their working principle: Tonal noise and structural excitation are caused due to the superposition of moving potential fields as well as periodic flow perturbations like wakes or shocks and their interaction with rotating or stationary parts. These features associated to the respective blade passing frequencies, i.e. the number of blades multiplied by the rotational speed, and the governing aerodynamics can be captured well in experiments and standard 4D RANS codes in periodic flow domains. Due to the vast amount of research on these topics in the past the main interactions of stationary and rotating blade rows have been understood well. Low frequency excitation orders and low harmonics of the rotational speed are typically associated to upstream struts or manufacturing and assembly tolerances like rotor whirl, axial run-out and so on.

In turbines non-synchronous fluctuations can be distinguished into tonal and broadband types. Typically, broadband fluctuations are associated to turbulence, like in boundary layers and blade wakes or secondary flows. The spectrum for this type of fluctuation varies depending on the source of the turbulence and the probe position both in bandwidth and amplitude, but generally ranges from low engine orders \((1,2,...)\) to the blade passing frequency and then decays up to the second BPF harmonic. More discrete frequencies are found in cavities, wake shedding and other instabilities, like vortex interaction, swirl instability, acoustic resonances and others. Elevated values of \(p_{1,RMS}'\) indicate the presence of such non-synchronous features and point to specific areas in the flow that will be analyzed more in detail in the frequency domain using a Fourier transform. The reference value in the free stream region at the inlet of the turbine has been determined to be 35 Pa. Since the fluctuations encountered at the exit of the first stator are of broadband nature in the wake and secondary flow regions they are not presented here. The typical subharmonic frequencies encountered at non-zero purge flow injection rates in the hub rim seal cavity have been described by Schuepbach et al. [22] for a narrower cavity opening. Due to the wider interaction area between cavity flow and main flow the dominant fluctuation is the blade passing frequency rather than the subharmonics for this cavity geometry, which is described more in detail by Jenny et al. [23].

In order to get the whole spectral information \(p_{1,RMS}'\) is plotted in the stationary frame of reference at rotor exit of the baseline case in Figure 7 a). Features associated to the rotor therefore leave a band of elevated fluctuations in pitch-wise direction, while stationary features can be located in this graph. The position of the downstream stator leading edge is indicated by the inclined dashed line. Compared to the reference unsteadiness of 35 Pa at turbine inlet the periodic passing of the rotor wake and secondary flows increases the level of unsteadiness at rotor exit by factors of 30 to 100. Their occurrence is periodic, but since the turbulence level in these flow structures is enhanced, the overall level of unsteadiness is also enhanced compared to turbine inlet. The circumferential band of elevated unsteadiness from 20-40% span is
associated to the passing of the rotor hub passage vortex. Previous research of Jenny et al. [23] has shown that the injected purge flow between the hub of stator 1 and the rotor is accumulated inside the passage vortex and therefore increases the pressure fluctuations in this area. The longer residence time in the area of the stator 2 passage and the rather quick swing of the vortex around the leading edge potential field result in the peak unsteadiness value of the hub passage vortex in region 1 of Figure 7 a). The radial migration along the pitch is also related to blade row interaction and the modulation of the purge flow injection at rotor inlet.

In a previous publication by Rebholz et al. [12] the outflow of over-tip leakage fluid from the shroud cavity close to the tip endwall was identified using unsteady total temperature measurements. The region with the peak value of $p_{1,RMS}$ coincides with this region in the mid passage of the second stator indicated as 2. Since cavities are known to exhibit resonances and since the cavity outflow, like the probe, is bound to the stationary frame of reference, region 2 is of special interest to more detailed analysis. The mean level of unsteadiness is enhanced even further at the exit of the second stator as shown in Figure 7 b). The signatures of wake and secondary flows of stator 2 show clearly as well as convected structures of the rotor. The hub passage vortex of stator 2 is located in region 1 and the parts of the wake in region 2. The total temperature measurements from the previous study [12] show an accumulation of fluid from rotor tip secondary flows as well as rotor over-tip leakage fluid in region 3 and 4. The level of unsteadiness is even higher in these regions than at the associated locations at rotor exit. Therefore the pressure fluctuations are not decaying through a stage, but accumulated convectively and potentially amplified. Interestingly the highest value of $p_{1,RMS}$ is found downstream of the second stator. In region 4, where the stator 2 tip passage vortex, the boundary layer as well as the wake are interacting also with shroud leakage fluid, values of up to 3600Pa are encountered, which is more than 100 times the level at the inlet of the turbine.

Figure 7: Time-averaged $p_{1,RMS}$ at a) rotor exit and b) stator 2 exit, both in the stationary frame of reference for baseline case. The free stream $p_{1,RMS}$ at turbine inlet equals 35Pa.
The transformation in the frequency domain reveals sources of the enhanced unsteadiness levels described in the previous section. The pressure signal in the area of the rotor hub passage vortex in Figure 8 a) contains only two distinct amplitude peaks between 0–5000Hz: At the rotor blade passing frequency and its second harmonic. The increased unsteadiness level in Figure 7 originates in a broadband perturbation, which is more than four times higher in this area than in the free stream at midspan. The dynamics of the ingested purge flow as well as the passage vortex behavior lead to a redistribution of energy from the blade passing frequency and its harmonics to the whole spectrum. While the free stream reference contains significant fluctuations up to the fifth BPF harmonic, the hub passage vortex area amplitudes at these frequencies are reduced by factors of 2.5 above the second harmonic. This type of signal behavior is expected for the comparison with a highly periodic flow region, like the wake passing at midspan. Contrary to the blade wakes the vortical structures close to the hub (and tip) need to be analyzed individually per blade passage to distinguish enhanced turbulence from non-periodicity. However, in order to resolve the vortex dynamics more in detail an unfeasibly high sampling frequency would be required (>50MHz). The frequency analysis of the shroud leakage is shown in Figure 8 b). Compared to the free stream and hub area several new features can be identified: 1) The amplitude peak at EO1 goes up to 25% of the BPF amplitude at midspan. Rotor whirl and out of roundness of the casing result in an estimated shroud sealing gap variation of 2–5%, which is suspected to be the main driver of a first engine order fluctuation inside the shroud cavity. 2) A non-synchronous excitation appears around 244Hz in the region which is associated to shroud leakage. The peak strength of this mode is 30% of the blade passing amplitude at midspan and therefore represents a significant noise and excitation source. The maximum amplitude of the non-synchronous excitation is found at the same location as the maximum unsteadiness values $p_{L,RMS}$. The depicted point is chosen off center of the extremum because the features described in the next paragraph are not as pronounced as in the border region of the shroud leakage. 3) Both the EO1 mode and the non-synchronous mode appear to interact in a non-linear way with the rotor blade passing frequency, since mirror images of the difference frequencies can be found at blade passing plus and minus 244Hz. Since the low frequency excitation of the flow appears only in flow regions which are strongly influenced by the shroud exit cavity, a detailed analysis of the cavity excitation is presented in a later section.
Figure 9: Fourier transform of yaw sensor pressure signal at stator 2 exit for baseline configuration in a) accumulation area of rotor tip leakage and rotor secondary flow and b) stator tip passage vortex and boundary layer interaction area. The amplitudes are normalized by the inlet total pressure.

Similar to the broadband fluctuations found in the hub passage vortex region at rotor exit, regions 1 (stator 2 hub passage vortex) and 2 (stator 2 wake) in Figure 7 b) show elevated pressure amplitudes over the whole spectrum with similar magnitudes at rotor exit (five times the free stream turbulence at rotor exit). Parts of the rotor blade passing fluctuations are smoothed out by the convection through the second stator, since significant amplitudes of the rotor blade passing frequency only appear up to the second harmonic (fifth harmonic at rotor exit). Since the tip secondary flows and shroud leakage is accumulated around midspan on the suction side of the second stator, the presence of the low frequency fluctuation around 244Hz in Figure 9 a) can be related to the upstream leakage fluid. The amplitudes of both the first engine order as well as the non-synchronous mode are not reduced significantly through the vane row and therefore also represent a noise or excitation source for downstream rows. Part b) of Figure 9 shows the amplitude spectrum in the interaction region of stator 2 tip passage vortex and the tip endwall boundary layer at stator 2 exit. The peak amplitude in this area can be found at 233.5Hz and is approximately 1.5 times higher than the rotor blade passing amplitude at midspan at rotor exit. This mode therefore represents one of the highest fluctuations found in the turbine flow field. Zooming in on the frequency band from 0-500Hz also shows multiple distinct peaks in the vicinity of the resonance at 233.5Hz at 20-25Hz intervals. The cavity resonance at rotor exit has a comparably smooth amplitude distribution around the resonance. Since the leakage oscillation frequency covers the frequency band of this instability, it is apparent that the non-synchronous fluctuation of the shroud leakage flow excites a flow instability in the area of the stator 2 tip secondary flow formation. A qualitative time-frequency analysis using the short time Fourier transform shows that the peak shape of the shroud leakage oscillation varies significantly over time and therefore results in the rather broad amplitude distribution around 244Hz. The frequency associated to the peak at 233.5Hz in Figure 9 b) is found to be more constant over time. This supports the conclusion that the shroud leakage oscillations excite a separate instability in the tip secondary flows of the second stator. Further, the fast Fourier transformation applied to the signal is expected to underestimate the amplitudes in the non-stationary fluctuations in the shroud leakage area.
Figure 10: Circumferentially averaged Fourier transform of the unsteady shroud cavity wall pressure at 91.3% axial distance for the three shroud designs. The amplitudes are normalized by the inlet total pressure.

Low frequency noise reduction using a partial shroud

In order to verify the origin of the non-synchronous, low frequency oscillations to be within the shroud cavity, the unsteady wall pressure at the shroud casing is analyzed for different shroud geometries. Figure 10 shows the over 40 points circumferentially averaged amplitude spectrum at 91.3% axial cavity length for the baseline, the LE cutback and the TE cutback. The described 244Hz oscillation as well as the difference frequency to the blade passing frequency measured with the FRAP probe downstream of the rotor and the second stator can be clearly observed inside the cavity for baseline as well. Due to the circumferential averaging the signal looks cleaner and the peaks are better distinguishable, but the peak amplitude of the resonance is damped. Compared to the fluctuations in the main flow the maximum fluctuations at the outer casing wall are approximately half as high. The signature of the low engine orders is enhanced significantly compared to the main flow as well, since their mirror images around the blade passing frequency are also visible.

The cavity geometry as well as the static pressure at rotor exit has not changed from the baseline to the LE cutback test case. Therefore, the observed change in resonance frequency for the baseline (244Hz) and the LE cutback (155Hz) has to be associated to the changed static pressure drop across the seals. It has been reported in [12] that the pressure drop across the first shroud fin increases by 37% and decreases for the second fin by 13.5% for the LE cutback. Additionally, the shroud inlet cavity flow field is dominated by the rotor flow field. The excitation of the exit cavity flow structures is therefore changed for the LE cutback and leads to a shift of 90Hz for the oscillation frequency and the difference frequency close to the rotor blade passing. Similarly to the inlet cavity for the LE cutback, the flow field in the exit cavity of the TE cutback is enforced by the dominating rotor blade passing. The rotor-bound fluid exchange with the main flow has also been described in [12]. Due to the dominance of the rotor blade passing frequency no non-synchronous excitations are present in the shroud exit cavity for the TE cutback. However, the enhance fluid exchange with the main flow compared to the baseline leads to an increased broadband amplitude level up to the blade passing frequency. According to numerical estimations in [12] the tip shroud seals are not choked. In the subsonic flow across the seals acoustic modes would be able to communicate between the three cavities. Acoustic resonances of the cavities would therefore be equally visible along the whole shroud and possibly even upstream in the main flow at rotor inlet. Typical mode shapes would be axial standing waves as well as pitchwise rotating modes. Due to the complex boundary conditions at the interfaces to the main flow and due to the presence of low frequency modulation by the lowest engine orders especially close to gaps, the
accurate prediction of the frequency of such modes is unfeasible with commercial tools. As the non-choked flow across the seals Figure 11 a) and b) indicate the presence of the low frequency oscillations at 244Hz for the baseline and 155Hz for the LE cutback case only in the exit cavity downstream of the second seal, an acoustic resonance can be excluded as source for the oscillation. Rotor whirl and eccentricity of the rotor casing dominate the pressure oscillation in vicinity of the shroud seals and could presumably be related to the onset of the low frequency oscillations in the shroud exit cavity. Despite the presence of high first engine order fluctuations across the second seal for the TE cutback in part c) of Figure 11, no frequency content is detected in the exit cavity. The amplitude of the rotor blade passing frequency is more than twice as high than for the other test cases and extends upstream from the cavity exit almost up to the second seal and therefore enforces a more periodic, rotor bound flow field inside the cavity. The remaining frequency content in the two bands from 90-190Hz and 180-280Hz is marginal and can be attributed to enhanced broadband fluctuations. Additionally to the altered pressure drop across the two shroud sealing fins the potential excitation mechanism in the LE cutback case is different compared to the baseline. The rotor blade passing frequency is dominant at the first fin due to the rotor-locked fluid injection and therefore it is also enhanced in the intermediate cavity as well as across the second fin. Contrary to the TE cutback the oscillations at the rotor blade passing frequency do not originate at cavity exit, but are transmitted through the seals.

**Figure 11:** Axial evolution of normalized and circumferentially averaged power spectrum of shroud casing wall pressure fluctuations for a) baseline, b) LE cutback and c) TE cutback for EO1, BPF and the band of the cavity frequency band.

**Figure 12:** Non-linear coupling frequency maximum for different rotor pressure drops for the LE cutback case.
Figure 12 shows the evolution of the difference frequency as a function of rotor pressure drop. Due to the entanglement with the first to third engine order the low frequency oscillations are difficult to extract separately at various measurement points. Therefore the non-linear coupling frequency, which is the difference of the low frequency oscillations and the blade passing frequency, is depicted here for the LE cutback. A pressure drop of one indicates the nominal operating point. Since the data is not averaged circumferentially and recorded at a different axial location than the data in Figure 10 the frequency deviates by approximately 20Hz. Figure 12 shows a linear dependency of the non-linear coupling frequency and therefore the low frequency oscillation on the pressure drop across the rotor. Compared the change in pressure drop across the rotor, the exit static pressure at the rotor tip and therefore the shroud exit cavity static pressure are relatively constant for the different test cases. With the same inlet total temperature to the turbine the density in the exit cavity remains also approximately constant, i.e. an acoustic resonance cannot explain the frequency shifts. Overall, swirl and vortical instabilities are therefore the most likely sources for the low frequency mode, since they are closely related to leakage mass flow and seal jet velocity. The consequences of all shroud cutbacks at turbine exit for the two most prominent oscillations, the cavity frequency and the blade passing frequency, are summarized in Figure 13. The peak pressure fluctuation intensity change relative to the baseline is shown at stator 2 exit for the cavity frequency range of 100-350Hz in part a) of the figure. Close to the hub none of the shroud modifications shows a significant impact, since this area is dominated by the stator 2 and rotor hub flow field, which is not affected significantly by the shroud modifications. The absence of the cavity mode starts showing for the shroud trailing edge modifications in the accumulation zone of rotor tip secondary flows and over-tip leakage on the suction side of the second stator above 30% span. While the LE cutback remains close to the baseline intensity level, the TE and the combined cutback show a reduction of approximately 7dB at midspan. Due to the excitation of an instability in the area of the stator 2 tip passage vortex and boundary layer interaction at 233Hz the intensity peaks for the baseline close to the endwall and therefore even the LE cutback shows a reduction in the peak fluctuation intensity of 8dB at the tip endwall. The TE cutback and the combined cutback feature identical modifications to the shroud trailing edge platform and show a reduction in fluctuation intensity of 12dB for the TE cutback and 14dB for the combined cutback compared.

Figure 13: Span-wise peak fluctuation intensity change at stator 2 exit compared to the baseline for a) the frequency band from 100-350Hz containing the cavity frequency and b) the blade passing frequency.
Part b) of Figure 13 shows the span-wise evolution of the other dominant tonal fluctuation component, the blade passing frequency. On average all cutbacks create higher fluctuations due to the alteration of the rotor secondary and leakage flows. The fluctuation intensity increases on average by less than 3dB and peaks close to the tip endwall at 5dB for the two cases featuring a trailing edge modification.

Figure 14: Change in pressure fluctuation intensity rel. to the baseline at stator 2 exit compared to stage efficiency loss for the rotor BPF and the cavity frequency range 100-350Hz.

Considering the maximum fluctuation amplitude at stator 2 exit as a potential noise source in the last stages of a low pressure turbine, Figure 14 illustrates the trade-off between total-total stage efficiency drop and reduction in tonal noise generation. The maximum increase in intensity is rather constant for all cutbacks at around 5-6dB. However, the shaded area indicates that the cavity fluctuation reduction outweighs the increase at the blade passing frequency. If the design priority is put on noise emission, like in the last stages of a low pressure turbine, a shroud trailing edge cutback therefore represents an effective tool to reduce low frequency noise generation. As presented in [12] this particular TE cutback design reduces the total-to-total stage efficiency by 0.7% resulting in the best trade-off of all cutbacks.

**Numerical prediction of non-synchronous cavity modes**

With increasing computational power computational fluid dynamics has become an integral part of the engine design process. In order to resolve all relevant flow features for a reasonably accurate prediction of turbine efficiency using unsteady Reynolds-averaged Navier-Stokes equations the flow domain still has to be divided into periodic sectors with conventional methods. The periodicity in circumferential direction acts physically as a high-pass filter for pitch-wise fluctuations, i.e. only pressure waves with wavelengths of $\lambda = \frac{2}{n \cdot \text{pitch}}$, $n = 1, 2, \ldots$, for standing waves and $\lambda = \frac{1}{n \cdot \text{pitch}}$ for travelling waves can be sustained. Making use of the computational acceleration of up to 30x [18] on GPUs allows simulating a full 360° model without any periodic assumption.

The buildup of low frequency modes requires more simulation time than the inherent oscillation like blade passing events. Figure 15 shows the Fourier transform of the pressure signal in the shroud exit cavity for the baseline case after 2.7 full rotor revolutions for the latest 300 time steps. Next to the rotor blade passing frequency an amplitude peak between 1500Hz and 2000Hz can be observed. The frequency does not match the dominant mode observed in the experiment. Also, the non-linear coupling with the blade passing frequency does not show. The analysis of the three-dimensional flow field does not point out a clear source of this oscillation. In comparison with the experiment also the absence of the first to engine orders is apparent, since assembly tolerances are not modelled.
Typical convergence criteria like the periodic correlation for two blade passing events do not apply necessarily to cavities in full-annular simulations. Although the main flow is well converged judging from the periodic correlation, Figure 16 clearly indicates that the amplitude variation in the cavity has not stabilized even after 3.25 simulated rotor revolutions. Interestingly this is also true for the blade passing frequency. Since the amplitudes in the frequency band from 1500Hz to 2000Hz do not show a decaying trend and since no peaks start to build up around 244Hz the simulation has been stopped at this point. From the initial guess, which was taken from periodic simulations, the amplitudes of both the rotor passing frequency as well as the frequency band around 1.75kHz decrease rather steadily for about one full rotor revolution. Afterwards the amplitudes exhibit a more fluctuating character. The variation in amplitude does, as pointed out previously, not necessarily mean that flow is not converged yet.

However, the general trend of the pressure amplitudes is rising over several revolutions and since the experimental data does not indicate any significant oscillations below the first engine order, a convergence issue cannot be excluded in this case. The shorter periods of these fluctuations are not constant in the observed time, but in the order of a rotor revolution. This suggests that the presence of the first and second engine order with comparable amplitudes would drastically alter the behavior in shroud cavity.

From the extensive probing of four different tip shroud configurations the fully shrouded rotor emerges as the aerodynamically most efficient one. Since the optimization of noise emissions and part life time have become an integral part of the design procedure of modern aero-engines, an extended analysis of the turbine flow field has been carried out in the frequency domain. In addition to the rotor-stator interaction, which is well captured in standard processing methods, a non-synchronous instability in the shroud exit cavity at 244Hz has been identified. The propagation into the main flow results in amplitudes of 30% of the rotor blade passing frequency at rotor exit. The perturbations are convected almost without attenuation through the
downstream vane row and excite an instability at 233Hz in the interaction area between stator tip secondary flows and the boundary layer. The order of magnitude of the resulting fluctuations at turbine exit exceeds the blade passing frequency amplitude at rotor exit by up to 50%. The dependency on the pressure drop across the second rotor shroud seal results in a variation of the cavity frequency and therefore excludes acoustic resonances as excitation source rather than swirl or vortex instabilities. This is also supported by the observation that the modes occur only in the exit cavity and do not propagate in the adjacent cavities, although the seals are not choked. Due to the reduced cavity frequency (155Hz) with the LE cutback case the instability in the second stator is not triggered.

Although the constraint of periodic boundaries is omitted for a full-annular unsteady CFD simulation the low frequency dynamics could not be resolved numerically. The dominance of the first engine order excitation in the seal gaps cannot be modelled with conventional CFD methods. However, due to the proximity to the cavity modes, the first engine order might well be responsible for the excitation of the non-synchronous mode. The dominance of the rotor flow field suppresses the creation of any cavity dynamics completely for shroud trailing edge cutbacks. The absence of the cavity modes result in a reduction of low frequency peak amplitudes by 12dB for the TE cutback at a total-total stage efficiency reduction of 0.7% compared to the baseline. Since the prediction of the cavity modes is found to be difficult, the application of shroud trailing edge cutbacks appears as a viable measure to suppress low frequency oscillations, which are typically associated to combustion chambers and therefore not budgeted in the acoustic design of a low pressure turbine. The high amplitude of these fluctuations and the low attenuation through blade rows represents a considerable excitation source both for structural vibrations as well as the emission of additional tonal noise from the exhaust of a jet engine.

Acknowledgements

The work leading to the results of this paper was carried out within the joint industrial and academic research program which is part of the “Luftfahrtforschungs-programm LuFo IV” (reference number 2070810) supported by the German Federal Ministry of Economics and Technology. The first author would also like to thank P. Plagowski for his support during the data analysis.

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