Acoustic Optimization of Rotor-Stator Interaction Noise by Trailing-Edge Blowing

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The paper deals with a novel approach for minimizing tonal rotor-stator interaction noise. A stage of a low-pressure axial compressor with an axial rotor and a downstream stator was designed and investigated experimentally. Seven discrete orifices at the rear end of each rotor blade were fed with pressurized air via separate internal passages and pressure control valves and allow the generation of arbitrary spanwise trailing edge blowing profiles from hub to tip. 3D hot wire measurement revealed that the upwash velocity fluctuations downstream of the rotor are caused by (i) the wakes from the rotor blades, (ii) vortex structures at hub and tip, (iii) the potential flow field of the rotor. Former studies proposed that perfect blade wake filling will lead to maximum reduction of rotor/stator interaction noise. In this study some reduction could be achieved by this strategy, but the utilization of an evolutionary optimization algorithm in the experiments proved to be much more effective: Targeting directly the far field sound pressure level we identified a spanwise trailing edge flowing distribution that eliminates the wake related tone completely. Spectral analysis proved that the blade wakes cause the acoustic rotor/stator interaction tone at twice the blade passing frequency whereas the fundamental tone is predominantly due to the vortex structures at hub and tip and the potential flow field interacting with the rotor. This explains, why with trailing edge blowing only the second harmonic of the blade passing frequency could be eliminated but not the fundamental tone.

Nomenclature

- BPF = blade passing frequency
- c₂ = absolute velocity
- c₂, upwash = upwash velocity to the stator leading edge
- cₘ₂ = axial velocity
- cᵣ₂ = radial velocity
- Dᵢ/o = inner/outer diameter
- fₛ = sampling frequency
- h = blade height
- lᵦ = axial rotor chord length
- Lᵦ,c₂,upwash = power spectral density level upwash velocity
- Lᵦ,pₚ,ₚₚ = power spectral density level acoustic pressure
- ᴘᵦᵥ/ᵦᵩ = inlet/outlet mass flow rate
- ᴘᵦᵥ = blowing mass flow rate
- n = rotational speed
- r = radius
- s = rotor-stator distance
- Sₚₚ = acoustic pressure power spectral density
- T = sampling time
- u₂ = circumferential velocity
- ᴘ = volume flow rate

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\( w_2 \) = relative velocity  
\( z \) = number of blades  
\( \varepsilon_r \) = relative combined standard uncertainty  
\( \phi \) = dimensionless flow rate coefficient  
\( \theta \) = circumferential angle  
\( * \) = dimensionless

I. Introduction and Motivation

Since the start of commercial aviation, the development of jet-powered commercial aircraft has progressed steadily. The development of aero engines evolved from single-spool turbofans to modern turbofan engines with high-bypass ratios. This technological progress shifted the primary sources of sound generation from high jet velocities to noise resulting from the rotating fan at the intake of the engine. Recently, several research projects have addressed this issue through the development of innovative concepts based on adaptive flow control technologies. One of the noise reduction techniques to be examined is called trailing-edge blowing (TEB). It targets the noise produced by the interaction of a fan with stator vanes positioned downstream (rotor-stator interaction noise, RSI noise). Due to losses in the boundary layer, a spatially and temporarily non-uniform flow velocity is generated by the rotor blades, \( w_{2, \text{wake}} \), in Fig. 1. This flow impinges with \( c_2 \) on the leading edge of the stator vanes. The unsteady upwash component \( c_{2, \text{upwash}} \) is responsible for a time varying angle of attack and hence pressure and force fluctuations at the surface of the stator. This is an essential source of sound. The sound emitted can be divided into two parts:

- The tonal RSI noise resulting from the periodic interaction of the mean upwash velocity in the wake.
- The broadband RSI noise produced by the turbulence in the wake.

This paper focuses on the reduction of tonal RSI noise by TEB. Although several authors described this technique as a mean for appropriate wake manipulation, it never has been put into practice. Based on our earlier own investigations the objective of this paper is the identification of the mass flow rate distribution along the blades trailing edges of a generic low pressure compressor impeller resulting in a maximum reduction of tonal noise. For that we describe and test an experimental optimization method.

II. Background

Brookfield and Waitz [1, 2] investigated a compressor stage with two different spanwise blowing rate distributions along each rotor blade. Basically the target was a momentumless wake. With tip wake oriented blowing strategy, a momentumless wake was only achieved at approx. 80% relative blade height. The tip wake was overblown in the tip region. Using this TEB blowing distribution, an 85% reduction in the first three blade-passing frequency wake harmonic amplitudes as compared to the unblown case was achieved. The stator unsteady loading, i.e. the fluctuating pressure on the guide vanes was reduced by 10 dB through TEB. Note that these authors did not present detailed acoustic measurements. Sutliff et al. [3] focused on the acoustic benefit of the TEB technique. Blowing was applied on a low-speed fan (16 blades and 14 stator vanes) by implementing 19 blowing slots per fan blade. The blowing mass flow entered the rotating system through the drive shaft and was delivered to the hollow blades. Inside of the blades, the flow was guided by 18 internal vanes. With an optimum blowing mass flow of 1.8% of the overall fan mass flow rate, a substantial tone reduction of around 10 dB was achieved. Further investigations of Sutliff et al. [4] focused on the reduction of broadband noise by the use of TEB. It was assumed that minimizing the wake velocity deficit by TEB reduces velocity gradients in the wake, and hence the turbulence generated in the wake and eventually broadband noise due to wake interaction with a downstream stator as well. The blowing...
distribution at optimum blowing rate was not perfect. At the tip region, the mean wake profile was overfilled while the hub span was underfilled. This resulted in a non-uniform wake filling and a smaller reduction in turbulence intensity (25 - 50\%) at lower relative blade heights. The overblown tip wake resulted in stronger turbulence production, but it was pointed out that a uniform wake modification might lead to a more effective reduction. Surface pressure measurements at the stator leading edge showed an averaged reduction of broadband pressure fluctuations of 2 - 3 dB due to the turbulence decrease. However, acoustic far field measurements revealed that the sound reduction was negligible despite the reduced pressure fluctuations.

The goal of the aforementioned studies was to smooth the aerodynamic wake quantities assuming that this will result in the best reduction of RSI noise. In contrast to the traditional wake filling approach this present paper focuses on an evolutionary optimization of the emitted tonal RSI noise by variation of the TEB mass flow rates and their spanwise distribution (TEB configuration). This requires an enhanced TEB rotor design and a test facility, which allows an automated control of the blowing mass flow rates during the measurements.

### III. Methodology

#### A. Axial Compressor Stage Design

A stage of a low-pressure axial compressor was designed. In contrast to any design rule, here the number of rotor and stator blades is chosen as being equal. The idea is to enhance blade passing frequency sound as much as possible. For the rotor blades, NACA 6512-63 airfoil sections are used, the the stator vanes are streamlined by the standard 4digit NACA thickness distribution . Further design parameters are given in Tab. 1 and Fig. 2.

In all experiments shown here the compressor is operated at its design flow rate coefficient

\[
\phi_n = \frac{\dot{V}_n}{\pi D_{n}\varphi} = 0.20. \tag{1}
\]

A reference plane that corresponds to the stator leading edge is located at a non-dimensional distance from the rotor trailing edge

\[
x^* = \frac{s}{l_{s}} = 0.4, \tag{2}
\]

where \(l_{s}\) is the axial projection of the rotor blade chord length at \(D_{n}\). For spanwise wake velocity analysis a relative blade height

\[
h^* = \frac{h}{(D_{n}-D_{t})/2} \tag{3}
\]

is used. The angular distribution of flow field data is given in terms of a relative circumferential angle

\[
\theta^* = \frac{\theta}{2\pi / z}. \tag{4}
\]

![Fig. 2: Schematic drawing of the TEB low pressure compressor stage: meridional cut (left), coaxial section (right).](image)

<table>
<thead>
<tr>
<th>Tab. 1: Characteristic quantities</th>
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<tr>
<td>(z_{\text{rotor}})</td>
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<tr>
<td>(z_{\text{stator}})</td>
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<tr>
<td>(l_{s})</td>
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<tr>
<td>(D_{n})</td>
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<td>(D_{t})</td>
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<tr>
<td>tip clearance</td>
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<td>(n)</td>
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All velocities are non-dimensionalized by the mean axial velocity at the compressor outlet

$$\frac{V_{\text{ax}}}{\pi / 4(D_2^4 - D_1^4)}.$$  \hspace{1cm} (5)

Note that $V_{\text{ax}}$ is the sum of both, the volume flow rate through the compressor and the extra amount blown from an external reservoir through the blade trailing edges.

As the circumferential velocity and hence the flow velocity around the blade increases from hub to tip, the wake deficit increases as well. This requires a spanwise distribution of the blowing mass flow rates. For that each blade’s slot is divided spanwise into seven discrete orifices which are fed separately via internal passages. The position of the orifices was chosen to be 1 mm upstream of the trailing edge. The blowing angle is determined by the orifice geometry and was aimed to equal the flow angle at the trailing edge. The internal passages are shaped carefully to avoid excessive pressure losses. They are connected to seven small plenums in the hub providing the air to all orifices at a specific value of $h^*$ (Fig. 3). The outer part of the rotor including the six blades with their internal passages was manufactured employing a selective laser sintering technique.

**B. Test Facility and Experimental Setup**

The TEB test facility and the air supply circuit are shown in Fig. 4. Based on mass flow meter readings the seven driving pressures are adjusted by proportional pressure valves according to the desired flow rates. This process requires an automated switching of the 5/2 directional control valves, an automated measurement and control of the blowing mass flow rates as well as an automated adjustment of the rotational speed of the impeller. This is realized with a LabView® program including several PID controllers. The hoses to the rotary air transducer in the rotor hub are hidden in the hollow stator vanes. The compressor stage takes air from an aero-acoustic wind tunnel [5], which is equipped with an auxiliary fan to realize the operation point. The wind tunnel is equipped with a muffler and several screens and honeycombs to produce a homogeneous velocity profile with low turbulence at the inlet of the stage. The flow passes the compressor stage and exits into a semi-anechoic chamber. The sound pressure is measured by three microphones (Brüel & Kjaer type 4190) with a distance of 1.5 m from the rotor-stator stage outlet and an angular spacing of -60°, -30° and 30° from the axis of rotation. All time signals of the sound pressure are captured with a sampling frequency $f_s = 25.6$ kHz. The power spectral density (PSD) is obtained by signal analysis via the MATLAB function pwelch which with

$$\text{window} = \text{hann(nfft)}, \quad noverlap = 0, \quad nfft = f_s.$$

The spectra from the windows are averaged with a final frequency resolution of $\Delta f = 1$ Hz. The tonal noise is analyzed by an integration of the spectral power $S_{pp}$ over a small bandwidth of $f \pm BPF \pm 7$ Hz with the blade passing frequency $BPF = n \cdot z$.

3D hot wire measurements at the stator leading edge are carried out with a triple hot wire probe TSI 1299. The probe is operated in a constant-temperature mode, using the Streamline unit from Dantec Dynamics. A temperature-correction is applied to the measured signals. The probe is aligned to the absolute velocity vector $c_z$ and positioned in radial direction with an accuracy of 0.02 mm using a three-axes traverse system. The sampling rate is chosen to be $f_s = 24$ kHz with a measured time signal of $T = 6$ s per radial position in the duct. The spatial radial resolution is $\Delta h^* = 1.5\%$ starting from $h^* = 7.5\%$ and ending at $h^* = 92.5\%$. For a single hot-wire probe, the relative combined standard uncertainty, including the effect of calibration curve-fitting, A/D conversion, probe positioning, temperature variation, ambient pressure and humidity variations, is estimated as $u_r = 1.5\%$ of the measured velocities [6]. The angle calibration was performed only for one reference velocity, even though it is expected to be a function of $Re$. Wittmer et al [7] showed that this effect can be neglected. Since the flow angle in the wake is only changing by less than 15°, the flow impinges on the wires of the probe within the measurement cone. Hence, the probe must not be realigned during the measurements. The rotational speed is triggered using a tachometer with an optosensor. Hereby, the measured absolute velocity components $c_{u2}$, $c_{v2}$ and $c_{z2}$ could be analyzed by the phase-locke averaging.
(PLA) technique [8]. These ensemble averaged velocities are transformed into turbomachinery velocities and angles. With that the upwash velocity $c_{2,\text{upwash}}$ at the stator leading edge were evaluated.

![Diagram of the TEB test facility](image)

**Fig. 4:** Schematic drawing of the TEB test facility (top) and compressor stage with pneumatic circuit (bottom): A centrifugal fan, B Vibration Isolator, C Muffler, D Screens/Honeycomb, E Contraction, F Compressor stage, G Microphones, H Opening for Flow Recirculation; 1 pressure source, 2 compressed air reservoir, 3 5/2 directional control valve, 4 mass flow meter, 5 proportional pressure valves, 6 manometer, 7 DC motor, 8 rotary air transducer, 9 stator vanes, 10 TEB impeller.

### C. Optimization Strategy

The optimization strategy implemented here for RSI noise reduction is based on evolutionary algorithms summarized e.g. by Thévenin and Janiga [9]. Each blowing configuration is an 'individual' with seven 'genes' - the spanwise distributed mass flow rates at the blades trailing edges. The first generation is obtained by random assignment of arbitrary blowing rates (within a reasonable range) to the individuals. The performance of each configuration with respect to the target function (minimization of tonal noise at a chosen frequency) is evaluated by acoustic far field measurements. Based on the emitted noise, the individuals are ranked. A better ranking increases the probability of an individual to participate as a parent in the reproduction procedure for the next generation. The probability of reproduction by means of averaging the genes of two parent individuals is set to 20% while 80% of the next generation is obtained from crossover, i.e. mixing the genes from two parent individuals. This reproduction procedure is enhanced by random mutations of selected genes. Eventually, the new generation is measured and ranked offering the basis for the production of a further generation. This loop of reproduction, evaluation and ranking is repeated several times resulting in generations with less production of tonal RSI noise.

### IV. Results

#### A. Traditional wake filling

In most previous studies the aim of TEB was to fill the wake velocity deficit as good as possible. In an earlier study we employed numerical RANS to obtain the spatial distribution of the TE blowing flow rates (Kohlhaas et al. [10]). Experiments with this distribution yields results as depicted in Fig. 5. Here the ensemble averaged upwash velocities in the reference plane (stator removed) at several relative blade heights are shown. Ideally the upwash velocity should be zero at any angular position at any blade height. Obviously this is not the case. Careful analysis allows to identify the wake which in the diagrams of Fig. 5 are indicated by arrows. Reasonably good wake filling has been achieved for $h^* \geq 0.3$. In the hub region (i.e. at $h^* = 0.1$) most probably the wake structure is displaced by
the well-known hub vortex system and in the tip region a second vortex appears due to leakage flows. From these graphs it becomes clear that with TEB one is more or less able to eliminate the wake induced upwash velocity. However, between the wakes the upwash velocity is non zero and, depending on height, varies considerably. This is attributed to secondary vortex systems in the wall regions and the potential flow field still existent downstream of the blade channels.

Since ensemble averaged time records of wake velocities are of limited value [11], the upwash velocity component is analysed by spectral decomposition at every specific blade height. This yields energy spectra as in Fig. 6. From that the spanwise distribution of amplitudes at blade passing frequency (1st BPF) and its harmonics can be derived. Focusing on the 1st BPF (Fig. 7, left) wake filling affects the upwash velocity fluctuations only marginally or even increases their levels, e.g. at \( h^* = 0.3 \). On the other hand, at 2nd BPF the impact of TEB is remarkable (Fig. 7, right): Throughout the complete blade region \( 0.3 \leq h^* \leq 0.9 \) the upwash velocity levels are reduced up to some 10 dB.

![Fig. 5: Ensemble averaged profiles of upwash velocity \( c_{2,\text{upwash}} \) without and with traditional TEB; arrows indicate upwash of wakes from two adjacent rotor blades; hot wire measurements in reference plane with stator removed.](image)

![Fig. 6: Traditional wake filling: Energy spectra of upwash velocity without and with TEB at midspan; hot wire measurements in reference plane with stator removed.](image)
B. Optimized TEB

In contrast to the traditional wake filling approach now the evolutionary optimization algorithm is employed to minimize directly the measured RSI tones by variation of the TEB mass flow rates and their spanwise distribution. Due to constraints in the pressurized air supply, each of the seven TEB mass flow rates $\dot{m}_{\text{TEB}}$ can vary in a range between 0.037% and 0.755% of the overall mass flow through the rotor-stator stage. In a first optimization experiment the target function was the measured sound level at 1st BPF. As with traditional wake filling nearly no effect on the tone at 1st BPF could be observed. In a second experiment we targeted at 2nd BPF. Fig. 8 shows the results of the optimization. The PSD level of acoustic pressure is plotted over the accumulated TEB rate $\sum \dot{m}_{\text{TEB}}$, which is an indicator of the overall blowing energy required. It is obvious that the TEB individuals lead to a reduction of the targeted tone continuously from generation to generation. A global optimum is found at an overall TEB rate of approx. 2% $\dot{m}_{\text{out}}$. Here the 2nd BPF tone is reduced by surprising 20.4 decibels. All points generate a clear front representing the boundary between possible, but not optimal, and infeasible solutions. Although the accumulated TEB rate is with 2% $\dot{m}_{\text{out}}$ the same as for the traditional TEB, the spanwise distribution differs significantly, Fig. 9: In contrast to the linear profile of traditional wake filling the optimized blowing profile is characterized by a higher flow rate in the hub region and significantly lower towards the blade tip. Only in midspan regions blowing rates are similar.

In analogy to Fig. 5, Fig. 10 shows angular upwash velocity profiles. Seemingly, optimization eliminates the midspan wake gaps better than traditional wake filling (Fig. 5). In the hub region ($h^* = 0.1$) the overall band width of the upwash velocity fluctuations is somewhat decreased, but in the tip region not much affected at all. The energy spectrum, Fig. 11, clearly illustrates the benefit at midspan: The peak at 1st BPF is completely eliminated, at 2nd BPF and higher harmonics to a large deal. As obvious from Fig. 12, in the whole blade region $0.3 \leq h^* \leq 0.8$, optimizations reduces upwash velocity levels considerably better then traditional wake filling - the peak reduction is some 25 dB.

![Fig. 7: Traditional wake filling: Energy spectra of upwash velocity without and with TEB; de-/increase of levels along blade height; hot wire measurements in reference plane with stator removed.](image)

**Fig. 7:** Traditional wake filling: Energy spectra of upwash velocity without and with TEB; de-/increase of levels along blade height; hot wire measurements in reference plane with stator removed.

![Fig. 8: Successive reduction of sound pressure level at 2nd BPF during ten generations of blowing configurations.](image)

**Fig. 8:** Successive reduction of sound pressure level at 2nd BPF during ten generations of blowing configurations.
Fig. 9: TEB mass flow distribution along the blade height $h^*$. 

Fig. 10: Ensemble averaged profiles of upwash velocity $c_{2,\text{upwash}}$ without and with optimized TEB; hot wire measurements in reference plane with stator removed. 

Fig. 11: Optimized wake filling: Energy spectra of upwash velocity without and with TEB at midspan; hot wire measurements in reference plane with stator removed.
C. Effect on far field acoustics

A comparison of the acoustic spectra resulting from the both TEB methods is plotted in Fig. 13. Since, as already mentioned, the optimization for the 1st BPF target was not successful, the result shown here is based on the 2nd BPF as target function.

It is not surprising that traditional wake filling as implemented in our experiment did not reduce the 1st BPF tone. As obvious from Fig. 7 (left), the spectral level of the upwash fluctuations were barely decreased. On the other hand the 2nd BPF it is reduced by 12.2 dB which fully reflects the findings in Fig. 7 (right). From Fig. 12 one may expect that at least optimimzed TEB will reduce the 1st BPF tone as well. This, however, is not the case. The only reason for that is that in the hub and tip region the large structures of vortices and secondary flow regimes dominate the flow field (the wake is barely visible), interact with the downstream stator and hence produce the 1st BPF tone. TEB, wheather traditional or optimized, is not able to modify these flows. On the other hand the 2nd BPF it is reduced by 21.4 dB which is 9 dB more than by traditional wake filling and in full agreement with the reduction of upwash fluctuations in Fig. 12 (right).

TEB in any form increases the broad band sound pressure up from 2 kHz. This is attributed to self noise of the jets leaving the slots in the rear part of the blades.
Fig. 14 shows the reduction of far field sound pressure levels due to traditional wake filling when varying the operating point of the compressor and adapting the TEB flow rate accordingly (while keeping the spanwise distribution the same). As pointed out in the previous sections for the standard operating point ($\phi_{in} = 0.20$), the effect of TEB on the 1st BPF tone is marginal, whereas the 2nd BPF tone is reduced by some 12 dB. In general the 1st BPF tone is slightly reduced only for $\phi_{in} < 0.20$ and even amplified for larger flow rates. On the other hand, the 2nd BPF tone decreases by more than 5 dB throughout the operating range $0.18 < \phi_{in} < 0.24$, the maximum reduction is 14 dB. A similar diagramme but with optimized TEB rather traditional wake filling would be of interest, but the experimental effort is immense, hence we refer to future work.

![Graph]

**Fig. 14:** Characteristic curve of the relative tone reduction at 1st and 2nd BPF due to traditional wake filling.

**V. Conclusions**

In this study we investigated the flow field downstream of a generic axial low pressure compressor rotor. 3D hot wire measurement revealed that the - from an acoustic point of view - undesirable upwash velocity fluctuations downstream of the rotor are caused by (i) the wakes from the rotor blades, (ii) vortex structures at hub and tip, (iii) the potential flow field of the rotor. Spectral analysis proved that the blade wake induced upwash velocity fluctuation strongly determines the acoustic rotor/stator interaction tone at the second harmonic of blade passing frequency. On the other hand the fundamental tone at blade passing frequency is due to vortex structures at hub and tip and the potential flow field of the rotor interacting with the downstream stator. These findings explain why, by actively blowing air out of the rotor blade trailing edges, we could reduce or even eliminate the second harmonic of the blade passing frequency related interaction tone, but not the fundamental tone.

Former studies proposed that perfect blade wake filling will lead to maximum reduction of rotor/stator interaction noise. In this study some reduction could be achieved by this strategy, but the utilization of an evolutionary optimization algorithm in the experiments proved to be much more effective. Targeting directly the far field sound pressure level we identified a spanwise trailing edge flowing distribution that eliminates the wake related tone completely. From this perspective an interesting hypothesis is that with trailing edge blowing one possibly could manipulate the acoustic mode shape in the duct, i.e. convert propagating into evanescent modes by e.g. modification of the radial mode order. This, however, is left to further analysis.

**References**