Active Flow Control on a Highly Loaded Compressor Cascade with Periodic Nonsteady Outflow Conditions

vorgelegt von
M. Sc.
Marcel Staats
ORCID: 0000-0003-4338-4406

an der Fakultät V -Verkehrs- und Maschinensysteme
der Technischen Universität Berlin
zur Erlangung des akademischen Grades

Doktor der Ingenieurwissenschaften
-Dr.-Ing.-
genehmigte Dissertation

Promotionsausschuss:

Vorsitzender: Prof. Dr. Ing. Robert Liebich
Gutachter: Prof. Dr. Ing. Julien Weiss
Gutachter: Prof. Dr. Ing. Wolfgang Nitsche
Gutachter: Prof. Dr. Ing. Volker Gümmer

Tag der wissenschaftlichen Aussprache: 28. April 2021

Berlin 2021
## Contents

1 Introduction 1

1.1 Research Thesis ................................................. 2
1.2 State of the Art Research on Flow Control and Non-Steady Aerodynamics ................................................. 5

1.2.1 Flow Control in General ........................................... 5
1.3 Closed-Loop Active Flow Control ..................................... 11
1.4 State of the Art Flow Control in Compressors ....................... 13

1.4.1 Passive Flow Control in Compressors .............................. 14
1.4.2 Active Flow Control in Compressors ............................. 15
1.5 Research Beyond the State of the Art .................................. 18

2 Accepted Papers 21

2.1 Staats2016a ............................................................. 21
2.2 Staats2016b ............................................................. 31

3 Submitted Paper 45

3.1 Staats2020 ............................................................. 45

4 Methodology and Discussion of Results 57

4.1 Methodology ............................................................ 57
4.2 Summary of Results .................................................... 65
4.3 Discussion of Results ................................................... 66

List of Figures 79

Bibliography 80
1. Introduction

A research thesis will be defined within this chapter which identifies the research gap that this dissertation is based on. An overview of state of the art research in unsteady aerodynamics with emphasis on active and passive flow control methods will be given. Research activities in the field of closed-loop active flow control are highlighted in this chapter. Furthermore, a closer view at flow control opportunities in the compressor of a gas turbine is provided. At the end of this section the contributions beyond the state of the art are provided which were published or submitted in technical journals as part of the research work carried out.

Figure 1.1: Development of specific fuel consumption of engines operated on civil airliners

In the past 70 years research on jet engines led to major increases of overall efficiency. This resulted directly in reductions of specific fuel consumption (SFC)
of jet engines or stationary gas turbines. Figure 1.1 shows the evolution of the specific fuel consumption (at cruise condition) of jet-engines operated on commercial airliners. The SFC was reduced by approximately 20% compared to the conventional turbojet engine with the introduction of the turbofan engine in the 1960’s. Higher bypass ratios combined with improvements as optimization of blading shape led to further enhancements in the specific fuel consumption and has almost halved this value since the 1950’s. The implementation of pressure gain combustion concepts, as a new technology, can lead to further reductions of a gas turbine’s fuel consumption. The benefits of the pulse detonation engine were pointed out in a number of research projects [2, 3]. The technical application of pressure gaining combustion processes defines new boundary conditions for many components of a gas turbine. Two pressure gaining combustion concepts are currently investigated in the collaborative research centre (CRC) 1029 at TU Berlin [4, 5, 6]. Both, the pulsed detonation and the shockless explosion combustion introduce highly dynamic phenomena in gas turbine aerodynamics. The combination of a pulse detonation combustion (PDC) and a turbine was tested by Rouser et al. [7]. Interactions between the pressure pulses of PDC and the first turbine stage were discussed in detail by George et al. [8] in terms of performance degradation of the turbine. The turbine efficiency degradation caused by the amplitudes (corrected to the inlet conditions) of the incoming shock waves was predicted by Fernelius et al. [9]. They pointed out that incidence angle oscillations between $-13^\circ$ and $+7^\circ$ occur at the turbine inlet guide vane. With increasing frequency of the combustion pulses these incidence oscillations were reduced and the efficiency of the turbine was regained, but remained below the level of the steady combustion case.

1.1 Research Thesis

A research gap can be identified regarding the resulting non-steady compressor aerodynamics, when a periodic operating pulsed detonation combustion is implemented downstream the continuously operating axial compressor. The stable operation of the compressor is crucial for gas turbine application. Within the compressor the fluid continuously streams against a positive pressure gradient to generate the desired total pressure ratio. Fluctuations in the back pressure, as they are evident when pressure gaining combustion is applied, may lead to local flow separations. This can disturb the stable operation of the whole gas turbine. In previous research, active flow control (AFC) in compressors was motivated by the objective to reduce the size and weight of the component [10, 11, 12]. Implementing stator blades that feature high flow turnings is in alignment with that objective, but leads to stronger secondary flows. They have shown that active flow control is a promising tool to reduce secondary flow losses and increase the pressure recovery of such configurations. When a gas turbine is operated in a non-steady combustion regime the boundary condition at the exit of the compressor changes and impacts the flow field in at least the last stages. The compressor is one of the most critical components of a gas turbine and likely to
suffer stall problems under an oscillating outflow condition \[^{[13]}\]. Lu et al. investigated the flow conditions at the compressor outlet in a four tube pulse detonation combustor in terms of time resolved pressure measurements \[^{[14]}\], \[^{[13]}\]. The experimental setup they used is presented in fig. 1.2. Furthermore, some key findings of this work are shown. In the experiments the combustion tubes were operated simultaneously and sequentially. When they were fired in the given sequence $1 \rightarrow 2 \rightarrow 3 \rightarrow 4 \rightarrow 1 \rightarrow \text{etc.}$, a circumferentially propagating disturbance was measured in the pressure signals at the inlet of the combustion tubes. The test facility was operated by a centrifugal compressor and equipped with a connection section, a transition section and a common air inlet, forming a plenum before entering the combustion section. At the compressor outlet, upstream of all those components fluctuations in the pressure signals were still evident at a reasonable amplitude in both cases, when the tubes were fired sequentially and simultaneously. Oscillating back pressure at the compressor outlet will lead to instantaneous oscillations of the operating point as depicted in the compressor map shown in fig. 1.3 (a). Reduced back pressure shifts the operating point to lower aerodynamic loading as indicated by the blue circle, whereas increased back pressure moves the operating point towards and even beyond the surge line of the compressor. This mechanism is highly critical for the compressor in a gas turbine driven by
pressure gaining combustion and needs further investigation. Active flow control can positively affect the compressor stall behaviour. It can shift the surge line towards higher pressure ratios and thereby add stall margin as it is depicted by the modified surge line with AFC in fig. 1.3 (a).

![Figure 1.3: (a) Sketch of a compressor map; (b) Secondary flow structures in stator vanes as suggested by Kang [15]](image)

The passage flow field in an axial compressor stator is dominated by secondary flow structures as presented in fig. 1.3 (b). The passage vortex forms the most dominant flow feature. The loss structures are enhanced the higher the loading of one stage is [16]. Flow separation first occurs in the low energy fluid at the end wall regions and is referred to as the corner stall. It spreads towards the centre of the passage with increasing blade loading. The resulting blockage of the passage limits the operating range of the component, even at steady state operating conditions. Secondary flow structures with oscillating strength will also contribute to performance degradations in the non-steady operating regime of the compressor, which counteracts the higher efficiency generated by pressure gain combustion. In previous dissertations it was shown that pulsed blowing active flow control at three spots inside every stator passage stabilizes the flow field of the highly loaded stator in a steady state operating regime [11, 12] with applying only a low amount of jet momentum.

Based on the identified research gap, the following key objectives can be derived for this dissertation:

- Controlling oscillating flow separation inside an axial compressor stator cascade. Especially the dominating secondary flow features will be reduced to increase the efficiency of the stator passage. This includes to successfully mimic the outflow condition of an axial stator with a downstream located pulsed detonation combustor. The conducted experiments describe relevant phenomena that are critical for a stable compressor operation in this specific context.
1.2 State of the Art Research on Flow Control and Non-Steady Aerodynamics

- The applied active flow control approach will restore the operability of the stator. Due to the high control authority of pulsed blowing actuation, periodically working actuators will be designed and integrated in the stator passage. Additionally, the continuous pulsed blowing actuation will be compared to periodically mass-flow modulated blowing in closed-loop control experiments. The comparison of different closed-loop control approaches, aiming at the reduction of load oscillations in the compressor stator by actuation mass-flow modulation, are documented by Steinberg [17].

- The investigated active flow control concept will be evaluated in terms of its effectiveness and efficiency. For this purpose, the static pressure recovery and loss reduction of the stator passage will be considered.

Therewith, the present dissertation contributes to compressor research in general and to active flow control in non-steady compressor stator aerodynamics in particular. The type of flow field investigated features similarities to dynamic stall scenarios of airfoils, which are well documented in the literature. Most of the research investigated 2D single wing configurations. It will be shown that key features from this research are transferable to the periodic compressor stator stall mechanism investigated within the present work.

1.2 State of the Art Research on Flow Control and Non-Steady Aerodynamics

Since the discovery of laminar turbulent transition and flow separation by Ludwig Prandtl in the early 20th century [18], an increasing interest in these new topics in the field of aerodynamics was prevalent. The first flow visualization experiments by Prandtl, presented in [19], revealed a fundamental insight to the flow field around simple aerodynamic shapes. These results contributed to the basic understanding of laminar and turbulent boundary layer flows. Furthermore, Prandtl showed that sucking in fluid at the separation point of the flow around a cylinder shifts the point of separation farther downstream. Additionally, it dampens the development of unsteady flow separation phenomena, such as the Karman vortex street. After this early stage of flow control experiments, many research projects were launched to further investigate effects of flow control by active and passive measures. By means of flow control the desired state of flow can be adjusted [20].

1.2.1 Flow Control in General

Often aerodynamic components are designed for a limited number of operating points. In conventional aircraft design the wing, for example, is mainly designed for cruise flight conditions. Another example is the compressor stator in a gas turbine that is designed to deliver a certain flow turning at a given mass-flow rate. An operation beyond the specified conditions can lead to performance
1. Introduction

losses. The installation of passive or active flow control devices can lead to performance enhancements at off-design conditions and extend the operating range of the aerodynamic component, i.e. by reducing its drag, increasing the lift or triggering/delaying boundary layer transition \[20, 21\].

Flow separation is a limiting factor for aerodynamic designs and can occur at a sharp edge in the contour of an aerodynamic body or by means of a strong adverse pressure gradient \[22, 23\]. The opportunity to increase the operating range of an aerodynamic configuration makes separation flow control a topic of high interest. Often the application of active flow control features higher control authority compared to the passive one. The actuation input momentum is the standard measure for the active flow control effort. It is usually expressed by the momentum coefficient \[21\]. In case of pulsed blowing actuation, the momentum coefficient is defined as:

\[
c_{\mu} = \frac{\dot{m}_{jet} \cdot U_{jet,RMS}}{q_{ref} \cdot S_{ref}}.
\] (1.1)

This parameter relates the input momentum through the actuation to the reference momentum of the flow. The input momentum is calculated using the effective jet velocity \(U_{jet,RMS}\).

Active flow control can be applied by means of multiple devices. In earlier work, steady blowing actuators were used, i.e. to force reattachment on a separated flap \[24\] by means of circulation control or on a highly loaded compressor cascade to reduce trailing edge flow separation \[25\]. It was found that periodic excitation of air jets is a promising way to increase jet momentum at constant mass flow, or to reduce the actuation mass flow, compared to steady blowing actuators, and thereby strengthen the control authority of the active flow control device \[26, 27, 28\]. The highest efficiency of active flow control is reached at low momentum coefficients. Suggested values range from \(c_{\mu} < 3\% \[20\] to \(c_{\mu} < 5\% \[27\], where boundary layer control is performed. In terms of lift increase the highest values of \(dc_L/dc_{\mu}\) are achieved within the stated ranges of momentum coefficient. At higher momentum coefficients the benefit decreases and circulation control is performed, which manifests in a less effective slope \(dc_L/dc_{\mu}\). This context is depicted in fig. 1.4 for two different types of airfoil trailing edges.

In accordance with Greenblatt \[30\], the frequency that low momentum air jets are injected with is responsible for the formation of large coherent structures which establish a mixing layer for momentum transfer to force reattachment. If the dimensionless frequency is below a certain value (usually \(F^+ < 1\)) higher values of momentum are required to reach a comparable flow control effect \[31\].

In this dissertation active flow control is applied to a non-steady flow field in a compressor stator cascade with periodic occurring turbulent flow separations. Such a flow topology is referred to as dynamic stall in the literature. In the following the mechanism of the dynamic stall is introduced and a brief discussion on flow control measures for dynamically stalling airfoils is provided.
1.2. State of the Art Research on Flow Control and Non-Steady Aerodynamics

\[
C_L = 60.80 C\mu + 0.71
\]

\[
\Delta C_L = 16.25 C\mu
\]

\[
C_L = 41.39 C\mu + 0.62
\]

\[
\Delta C_L = 13.57 C\mu
\]

\[
\alpha = 0.0
\]

\[
h/C = 0.00106
\]

\[
\text{CIRCULAR TE}
\]

\[
r/C = 2\%
\]

\[
\text{ELLIPTIC TE}
\]

\[
r/C = 1\%
\]

\[
\text{SEPARATION CONTROL}
\]

\[
\text{SUPER-CIRCULATION CONTROL}
\]

**Figure 1.4:** Lift gains of an airfoil with respect to the applied actuator momentum as suggested by Jones [29]

1.2.1.1 Flow Control and Non-Steady Aerodynamics

A dynamic stall occurs e.g. in the presence of angle of attack oscillations beyond the static stall angle [32]. Such flow characteristics are found on axial wind turbines, where the blade experiences an unsteady wake disturbance by interfering with the tower. These flows are well documented in [33]. Further technical examples are blade passings in gas turbine flows, rapidly manoeuvring aircraft and helicopter blades [34, 35]. In the last case the non-steady flow is forced by the pitching and plunging motion of the rotor blades and the superposition of the rotating motion with the air speed. The region of the dynamic stall area is schematically marked in fig. 1.5 for a helicopter in horizontal flight.

In [36] a categorization of the phenomenon in light and deep dynamic stall is suggested. The first category includes stall situations, where the stall cell does not reach the leading edge of the profile. In the second category leading edge stall occurs at a range of phase-angles and vortex shedding at the leading edge becomes a dominant feature of the flow along the separated surface. In both cases

**Figure 1.5:** Flow patterns on helicopter blading, adapted from [35]
1. Introduction

all aerodynamic forces are affected. The reduced frequency oscillating airfoils are excited with is often calculated as follows \[37\]:

\[
k = \frac{\omega_{exc} \cdot c}{2 \cdot u_\infty}.
\]  

For that the circular frequency of the excitation \(\omega_{exc}\) is used. McCroskey found out that the value of the reduced frequency is important for the separation behaviour of the airfoil \[36, 37\]. Further criteria are the operational Reynolds number, the airfoil or wing shape, the oscillation amplitudes and mean incidence angle, etc. Sharma investigated a piching NACA-0015 airfoil with varying Reynolds Numbers and reduced frequencies in the range of \(Re = 200000\) to \(Re = 700000\) and \(k = 0.001\) to \(k = 0.5\), where the flow showed quasi-steady behaviour for reduced frequencies of \(k < 0.002\) \[38\]. Further studies on unsteady flows of pitching airfoils are given in \[39\] on an experimental basis and in \[40\] numerically. Detailed discussions on the physics of dynamically stalling airfoils are given in \[36, 41\]. The process of a dynamic stall is shown in fig. 1.6. The dashed line represents the static stall case. Lift coefficients depicted by the solid line occur when the airfoil is operated in the oscillating regime, with the angle of attack oscillation given by \(\alpha = 15 + 10 \sin(\omega t)\) \((\alpha_{mean} = 15^\circ\) and \(\Delta \alpha = \pm 10^\circ)\) at \(k = 0.1\). Thereby, higher lift coefficients are reached compared to the static case. In the early stage of the deep dynamic stall, flow reversals occur beyond the static maximum lift angle of attack fig. 1.6 (1). From that point on the lift is further increased with increasing flow reversal and the formation of large eddies on the suction surface of the airfoil fig. 1.6 (2). When the stall cell reaches the leading edge, the formation of the leading edge vortex begins fig. 1.6 (3) and it travels downstream the surface towards the trailing edge. Thus a strong increase in lift slope occurs fig. 1.6 (4). Later the vortex breaks down and the stall cell increases in size until the airfoil fully stalls fig. 1.6 (5). After reaching this condition the flow attaches from the leading edge on fig. 1.6 (6) until the flow is fully reattached.

In the past decades researchers investigated multiple flow-control methods to enhance the performance of dynamically stalling airfoils. Lift increases and drag reductions were reached by means of various mechanisms \[42, 43\]:

![Figure 1.6: Phenomenon of dynamic stall, adapted from 35](image-url)
1.2. State of the Art Research on Flow Control and Non-Steady Aerodynamics

- enhancing the turbulent mixing in the boundary layer
- airfoil dynamic shape optimization
- momentum flux addition to critical boundary layer flows
- adding shear layer instabilities by pulsed blowing

The enhancement of turbulent mixing and dynamic shape optimization can be categorized into passive flow control measures, whereas the latter belong to active flow control measures.

**Passive flow control on non-steady aerodynamic flows**

It was shown that a back-flow flap on a dynamically stalling airfoil’s suction surface reduced the pitching moment peak related to the deep dynamic stall by up to 25% and generated higher lift simultaneously [14]. This concept of passive flow control was also proven by Meyer et al. [45] in flight testing environment on the Stemme S10 motor glider. It was shown that even a self-adjusting flap is capable of delaying airfoil stall towards higher angles of attack with a $\Delta c_{a,max} = 10\%$ of maximum lift increase with respect to the uncontrolled reference condition. A pitching airfoil equipped with different leading edge vortex generators was investigated by Geissler et al. [46], indicating a comparable, positive effect. These examples involve solid devices interacting with viscous or non-viscous regions of the flow field. In [47] it was pointed out that the concept of nose-drooping reduces the maximum pitching moment peak by 50% and decreases the produced aerodynamic drag without reducing the lift during the dynamic stall, but it does involve complex mechanics to realize the nose-droop movements.

**Active flow control on non-steady aerodynamic flows**

A stronger control authority can be achieved by means of active flow control. The challenging task is to positively influence the dynamically stalling airfoil, wing or aerodynamic body at reasonable cost. Yu et al. [48] performed a comparison of various flow control possibilities on dynamically stalling airfoils by means of numerical and experimental investigations. They pointed out that upper-surface blowing on a pitching airfoil entails higher reductions in lift hysteresis and drag magnitudes than passive means, like leading-edge slats or nose-drooping. Gardner et al. [49] investigated steady blowing from round holes, as well as pulsed blowing. In their investigations steady blowing had a comparable effect to pulsed blowing, which seems to be due to the actuator system they used. Greenblatt [50, 51] conducted experiments investigating active flow control under light- and deep-stall regimes. In these cases a rectangular outlet orifice was used. It was found that oscillatory blowing introduced to a periodically stalling airfoil was far superior compared to steady blowing. The latter was even detrimental in certain cases. A further finding was that blowing farther downstream the leading edge was superior compared to leading-edge excitation. Lift hysteresis was decreased and momentum excursion was minimized. Furthermore, active flow control acted
1. Introduction

beneficial in terms of drag reduction. A detailed discussion on the mechanism of the dynamic stall control by periodic excitation is given in [52]. One key finding is that the dynamic stall accompanied with oscillatory airfoil pitching does not affect the advection of the large coherent structures generated by the active flow control device. The actuator type used for the active flow control task and the placement of the device are crucial in terms of energy efficiency and active flow control effectiveness. The effect of the actuation position on a dynamically stalling airfoil was investigated by Asgari and Tadjfar [53, 54] by means of CFD simulations. The slot position of a continuous working actuator was varied along the suction surface. Drag and lift curves were determined from the calculations and indicated that actuation at a wrong position (too close to the leading edge or too far downstream the leading edge) can be detrimental w.r.t. drag production. The lift generation of the airfoil was not as sensitive to the actuation position. Flow separation and hysteresis was dramatically reduced using a local velocity ratio of $v/u_\infty = 5$ in the optimum configuration. Key results from that work are shown in fig. 1.7. The figure also depicts the general effect of successfully applied actuation on a non-steady flow field operating under dynamic-stall condition. Taking the uncontrolled case into account the high hysteresis in lift and drag become evident. Increasing the jet momentum (here represented by the local velocity ratio $v/u_\infty$), reduces the area of hysteresis. In terms of drag lowest values are already reached at the velocity ratio $v/u_\infty = 3$ and increase for the higher jet velocity at $v/u_\infty = 5$. The lift hysteresis was further reduced with increasing jet velocity. The airfoil operates with comparably low hysteresis in lift and drag at high actuating momentums and velocity ratios. A lower drag was reached with higher lift simultaneously.

The approaches discussed above may also be suitable for the active load control of non-steady operated airfoil sections or wings. In this case, a varying inflow condition to an aerodynamic system, such as varying incidence angles or inflow speeds occurring due to wind gusts, are controlled in order to reduce dynamic

![Figure 1.7: Effect of actuation on a dynamically stalling NACA 0012 airfoil](image-url)
loads on the mechanical structures. Such examples were investigated by Coop-
erman et al. [55]. The load oscillations occurring on a 2D-airfoil due to random
inflow conditions were damped by active trailing edge deflection, minimizing the
load oscillations. An active approach was performed by Baleriola [56], where
three plasma actuator elements were distributed around the trailing edge of a 2D
wind turbine airfoil. It was feasible to reduce or increase the lift at a given inflow
angle in the study and thereby shifting the lift polar into the desired direction.
By that means, a constant airfoil load under unsteady inflow conditions can be
produced when a suitable control algorithm is applied. The potential of such ap-
lications on realistic wind turbine flows was presented by Niether et al. [57], on
a numerical basis, and a reduction in the standard deviation of load oscillations
by 32% was stated.

1.3 Closed-Loop Active Flow Control

The flow control methods described in the previous chapter are powerful tools to
increase operating ranges of aerodynamic components. Time-dependent forces
and moments occur in the field of non-steady aerodynamics. Disturbances and
changing boundary conditions may also lead to local flow separations. In these
cases constant operating active flow control might not always be the best solu-
tion in terms of energy efficiency. To save actuation mass flow or to reduce load
oscillations on an aerodynamic body, an adapted/time varying trajectory of the
actuation mass flow may increase the efficiency of the AFC system. In order to
reach those goals, closed-loop active flow control is a promising approach when
applying AFC to a non-steady flow problem. In contrast to steady operating
active flow control, the field of closed-loop active flow control research has de-
veloped more recently and thus is not as comprehensive yet. An overview of the
application of close-loop active flow control is provided by Brunton [58]. The ar-
ticle reviews applicable methods to mathematically model the flow and discusses
the control strategies that can be used to fulfil the control task.

Closed-loop active flow control was applied to control a turbulent flow separation
by Allan et al. [59]. The separation was generated in the wake of a hump. In
[60] a survey is given about the application of closed-loop active flow control in
three different wind tunnel experiments. It was feasible to reduce the separation
length on a backwards facing step, a half-diffusor and a flap-like configuration.
Furthermore, closed-loop active flow control was applied to a high lift configura-
tion [61], where the controller was able to track a given trajectory of the control
variable. In this application a non-model based control strategy was used, ap-
plying a gradient-based extremum-seeking scheme.

A further prominent example of closed-loop active flow control application is
the suppression of gust loads on aircraft wings. Experimental investigations,
performed by Kerstens et al. [62, 63], showed that non-steady inflow speeds
to a wing model lead to non-steady lift-force productions of the wing. It was
feasible to suppress the load oscillations occurring due to the gusts by means of
the implementation of a closed-loop controller for a pulsed blowing active flow
control system. The experimental setup is shown in Fig. 1.8 (a). The shutter downstream of the wind tunnel model induced the free stream velocity oscillations that mimicked the gusts. Figure 1.8 (b) shows the measured forces, generated by the wing with and without control. The wing generated $F_L = 1.8$ N of lift force under undisturbed conditions. The non-steady inflow degraded the lift force and introduced oscillations of approximately $\Delta F_L = \pm 0.2$ (purple curve). The red curve was obtained by switching on the controller. Thereby, the effect of the gust was almost fully suppressed and the original lift force of $F_L = 1.8$ N was restored and remained constant over time.

Culley et al. [64] demonstrated a closed loop active flow control approach in a multistage compressor test rig. Throttling the test rig leads to higher loadings and flow separations at some point. They used pulsed blowing actuators arranged on the suction surface of the stator blade to reduce regions of separated flow. The state of flow (attached or separated) was detected by means of static pressure measurements on the casing. It was feasible to keep the flow attached by the suggested closed-loop approach.

Wiederhold [65] applied robust control approaches as well as classical decoupling controllers to the flow in a highly loaded cascade with an AFC system. By that the corner separation was controlled and wake disturbances were mitigated. Furthermore, extremum-seeking controllers were applied to identify efficient actuation parameters. Moreover, he applied $H_\infty$-controller to delay the rotating stall in an axial fan.

Within the scope of the collaborative research centre (CRC) 1029 at Technische Universität Berlin further control approaches were developed and applied to highly non-steady compressor flows [66] and non-steady combustion test rigs [3]. The investigations regarding the non-steady compressor flow field were conducted on the compressor cascade presented within this dissertation. The com-
State of the Art Flow Control in Compressors

Compressor cascade is based on the one presented by Hecklau and Zander [28, 67] and was extended by a device that imposes the non-steady boundary condition: the choking-device. The development and implementation of the controllers used for the compressor experiments within the CRC 1029 and within this dissertation were performed by Steinberg [17, 66, 68, 69]. Thereby the focus was set to mitigate the imposed disturbances to the compressor test rig. The measured pressure distribution of the centre stator blade was used as a control variable. The blowing amplitude of the actuators (momentum coefficient) was the output variable of the controller. As the imposed disturbances had a repetitive pattern, iterative learning controllers were used to predict the control law. Learning from cycle to cycle led to increasing effectiveness of the actuation after a low number of iterations. In [66] the controller was extended and it allowed for multiple outputs, namely, the side-wall actuation momentum and the blade actuation momentum. The learned actuation trajectories were successfully applied to three neighbouring passages of the compressor cascade.

1.4 State of the Art Flow Control in Compressors

Axial compressors in gas turbines or aero engines are highly complex systems that decisively influence the engine operation in terms of efficiency and stability [70, 71]. Modern compressors in state of the art aero engines consist of 15 stages [72] and take up approximately 50% to 60% of the length of the engine. Furthermore, it contributes 40% to 50% of the overall weight, 35% to 40% of the production costs and 30% of the maintenance costs are related to compressor components [73]. Reducing the length and the weight of the compressor has a big potential affecting the overall engine performance. Many research projects aimed at those goals in the past. In terms of introducing the PDC concept to a gas turbine, the compressor has to be adapted to the new boundary conditions, such as total pressure ratio and the ability to operate in highly non-steady conditions.

A detailed understanding of compressor flow features has also been a topic of high interest in order to develop tailored measures of flow control that, for example, may increase stage load to reduce the compressor length. Increasing the total pressure ratio of a stage intensifies secondary flow structures, hence losses are massively increased [74]. In terms of the compressor stator an increasing flow turning has to be achieved, if the stage load is increased. These circumstances lead to a significant growth of the vortex structures in the passage and increase the risk of a hub stall, which negatively affects the stator performance and its aerodynamic stability [75, 76]. In a one-dimensional compressor model, the stability limit is given by an airfoil stall followed by the onset of a compressor surge [77]. In more detailed two- or three-dimensional models, more advanced flow features, such as rotating stall occurring just before a compressor stall, and rotating instabilities can be captured [78]. The latter occur prior to rotating stall and were subject to investigation in [79]. Detailed investigations on secondary flow structures in axial compressors were performed by Beselt [80]. Typically the flow field in a compressor/turbine stator is governed by unsteady flow conditions.
induced by rotors located upstream and downstream. This wake-passing effect contributes to unsteady loads, flow topologies, and causes acoustic noise \[87\]. In linear cascades, the effect of wake passing of the upstream rotor is usually simulated by running bars featuring comparable aerodynamic drag. They are traversed upstream the measured stator \[88\]. In \[89\], a parametric study on different bar geometries was performed. The rotor wakes in a compressor/turbine stage generate a local total pressure and velocity deficit that periodically affects the downstream located stator row, usually with a reduced frequency of \(Sr = 0.75\) to \(Sr = 1.5\) \[90\]. The periodic disturbance leads to oscillations of the incidence angle of the downstream stator \[91\] and oscillations of the intensity of the turbulent fluctuations (turbulent kinetic energy). The latter may impair the development of the boundary layer on the stator blades. Due to higher mixing rates the increased turbulent kinetic energy can positively affect the separation dynamics occurring in the periodic unsteady flow, as it was shown in \[92\]. In this study it was found that if the wake-passing disturbance is higher, compared to the time-averaged velocity fluctuations, boundary layer separation was at least partially suppressed.

### 1.4.1 Passive Flow Control in Compressors

Recent research projects deal with passive or active flow control in compressors to increase their operating range and stability at high loadings. De Haller introduced the deceleration ratio \((DH = u_2/u_1)\), whose decreasing value indicates increasing secondary flows and end-wall blockage. In conventional designs, the stator geometry is such that the de Haller criterion is met \((DH > 0.72)\). The compressor stator airfoil is highly loaded and operates with the risk of stall for lower values. Measures of passive/active flow control to suppress a rotating stall at the rotor tip region were performed by Koch et al. \[93\] and by D’Andrea \[94\]. The operating range is typically limited due to flow separation in compressor stator flows. A prominent region of flow separation is located at the blade-wall junction, as shown in fig. \[1.3\] (b). Low energy fluid from the end-wall boundary layer rolls up and forms the passage vortex. This structure collapses at a certain position and a region of separated flow with increasing flow separation intensity at high airfoil loadings spreads far into the stator passage. Hergt et al. \[95, 96\] found that vortex generators located at the passage inlet rather than on the suction surface of the stator blade can positively influence those flow structures. They investigated the effect on a high-speed linear compressor cascade, on which higher flow turnings and lower losses were observed in the controlled cases. Additionally, the concept was transferred to an annular test setup (1 stage compressor test rig), where it was feasible to increase the isentropic efficiency of the stage with reducing flow losses in a large range of blade loadings. A study of the vortex generator placement at the stator inlet plane and its shape was performed in \[97, 98\]. Performance improvements by passive means can also be achieved by end-wall contouring, which forces an additional pressure gradient that reduces secondary flow losses. Brennan et al. \[99\] discuss the redesign of the Royce Rolls Trent 500 HP turbine with contoured end walls with the objective
to reduce the magnitude of secondary kinetic energy (SKE), and with that, flow losses associated with the whirl angle at the stator outlet. In the conventional loss theory, the drag coefficient is proportional to the square of the lift coefficient \[^{100}\]. It can be shown that the SKE is also proportional to the lift square and can be used to account for aerodynamic drag \[^{101}\]. By that measure, it was predicted to reduce the secondary loss of the nozzle guide vane by 0.24% of stage efficiency. Contoured end walls are also implemented in the Royce Rolls Trent 900 low pressure turbine \[^{102}\].

1.4.2 Active Flow Control in Compressors

Continuously operating active flow control is another possible application for flow control in compressors. It was investigated in many research projects. By means of circulation control, such as the jet flap, slots located close to the trailing edge were used to form a coanda jet. By that means the flow turning of the stator can be increased, when approximately 0.5% to 2% of the passage mass flow is applied through the jet flap \[^{103}\] \[^{104}\]. Numerical investigations indicated that the higher flow turning (blade loading) and the more homogeneous total pressure field at the stator outlet reveals the potential to reduce the number of stator vanes by 25% \[^{105}\] \[^{106}\].

By means of boundary-layer suction viscous fluid is extracted from the boundary layer. Therefore, less entropy is produced in the compression process, leading to higher compressor efficiencies \[^{107}\]. This active flow control approach was further investigated in the past \[^{108}\] \[^{109}\] \[^{110}\]. A combination of out blowing and suction on the suction side of a compressor airfoil was tested by Cater \[^{111}\] in transonic flow regimes. In his case the actuator system was located at \(x/c = 0.8\) to \(x/c = 0.9\) with the suction holes positioned upstream the blowing holes. Significant loss reduction resulted from the use of moderate actuating mass flow with a suction to blowing ratio of 1 : 3.6. Numerical investigations on boundary layer suction at multiple positions on the blade and the end wall, aiming to reduce the secondary flow structures, showed that a combination of end-wall and blade actuation acted most beneficial in terms of loss reduction \[^{112}\].

Actuation in the end-wall region of a compressor stator cascade was performed by means of vortex generator jets (VGJ) and revealed loss reductions of up to \(\Delta \zeta = 9.5\%\) at subsonic Mach numbers \[^{113}\]. operating the cascade at \(M=0.71\) achieved higher loss reduction of up to 14.5%. The actuation momentum shifted the high-loss regions (inside the passage vortex) towards the end walls and reshaped the loss distribution in the wake of the stator blades towards a favourable configuration. Synthetic jet actuators (zero-net mass flux), generating pulsed VGJs, were applied on the suction surface of a compressor stator blade and led to loss reductions of \(\Delta \zeta = 22.8\%\) at the inlet Mach number of \(Ma = 0.3\) \[^{114}\]. At higher inflow Mach numbers, the synthetic jet actuator performed less effective due to the lower jet velocity ratio and loss reductions of only \(\Delta \zeta = 5.1\%\) were feasible. Variations of the blowing frequency of pulsed air jets have shown that the flow losses are sensitive to that excitation parameter. Highest loss reductions can be achieved at excitations Strouhal numbers in the order of \(F^+ = O(1)\) \[^{115}\]. This finding
also agrees to the results shown in [28], in which experimental and numerical data sets were compared.

Furthermore, the velocity ratio of the injected jets is a driving parameter for active flow control effectiveness. In [116], a variation of the duty cycle of the PJA system installed on the suction surface of a compressor stator blade (linear cascade) was conducted. It indicated increasing loss reductions with decreasing duty cycles, when the actuation mass-flow-rate was kept constant. The reduction of the duty cycle caused increasing velocity ratios of the actuation jet, hence the momentum coefficient increased as well. The actuation of the corner separation by means of a sweeping jet actuator was investigated by Meng et al. [117]. It was shown that the total pressure loss could be reduced by 5.6% of its original value.

In Table 1.1 some key results from numerical and experimental investigations with active flow control applied to compressor cascades are summarized. It emphasises the loss reductions ($\Delta \zeta$) and static pressure rises ($\Delta C_p$) achieved by means of AFC, as stated in the individual publications.

<table>
<thead>
<tr>
<th>author (year)</th>
<th>setup</th>
<th>inflow angle</th>
<th>Re</th>
<th>Ma</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carter [111] (2001)</td>
<td>2D cascade with 7 blades</td>
<td>$69^\circ$</td>
<td>$2 \cdot 10^6$</td>
<td>0.79</td>
</tr>
<tr>
<td>type of AFC</td>
<td>$\Delta C_p$</td>
<td>$\Delta \zeta$</td>
<td>$\dot{m}_j/\dot{m}_1$</td>
<td>$c_\mu$</td>
</tr>
<tr>
<td>suction holes and blowing holes</td>
<td>-</td>
<td>65% (ref.: .12)</td>
<td>0.016</td>
<td>-</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>author (year)</th>
<th>setup</th>
<th>inflow angle</th>
<th>Re</th>
<th>Ma</th>
</tr>
</thead>
<tbody>
<tr>
<td>Culley [116] (2005)</td>
<td>compressor test-rig with 58 blades</td>
<td>$42^\circ$</td>
<td>$5 \cdot 10^5$</td>
<td>0.146</td>
</tr>
<tr>
<td>type of AFC</td>
<td>$\Delta C_p$</td>
<td>$\Delta \zeta$</td>
<td>$\dot{m}_j/\dot{m}_1$</td>
<td>$c_\mu$</td>
</tr>
<tr>
<td>pulsed blowing</td>
<td>-</td>
<td>25% (ref.: -)</td>
<td>0.009</td>
<td>-</td>
</tr>
</tbody>
</table>
## 1.4. State of the Art Flow Control in Compressors

<table>
<thead>
<tr>
<th>author (year)</th>
<th>setup</th>
<th>inflow angle</th>
<th>Re</th>
<th>Ma</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fischer [104] (2008)</td>
<td>2D cascade with 5 blades</td>
<td>58°</td>
<td>5 \cdot 10^5</td>
<td>0.146</td>
</tr>
<tr>
<td>type of AFC</td>
<td>ΔC_p</td>
<td>Δζ</td>
<td>\dot{m}_j/\dot{m}_1</td>
<td>c_μ</td>
</tr>
<tr>
<td>coanda jet</td>
<td>0.03</td>
<td>0.04</td>
<td>0.015</td>
<td>-</td>
</tr>
<tr>
<td>Hecklau [118] (2010)</td>
<td>2D cascade with 7 blades</td>
<td>60°</td>
<td>8.4 \cdot 10^5</td>
<td>0.1</td>
</tr>
<tr>
<td>type of AFC</td>
<td>ΔC_p</td>
<td>Δζ</td>
<td>\dot{m}_j/\dot{m}_1</td>
<td>c_μ</td>
</tr>
<tr>
<td>pulsed jet</td>
<td>0.09</td>
<td>0.13</td>
<td>0.005</td>
<td>0.0144</td>
</tr>
<tr>
<td>Nerger [25] (2012)</td>
<td>2D cascade with 5 blades</td>
<td>60°</td>
<td>1.63 \cdot 10^6</td>
<td>0.2</td>
</tr>
<tr>
<td>type of AFC</td>
<td>ΔC_p</td>
<td>Δζ</td>
<td>\dot{m}_j/\dot{m}_1</td>
<td>c_μ</td>
</tr>
<tr>
<td>steady jet</td>
<td>0.1</td>
<td>-0.1</td>
<td>0.003</td>
<td>0.0656</td>
</tr>
<tr>
<td>Zander [67] (2013)</td>
<td>2D cascade with 7 blades</td>
<td>60°</td>
<td>6 \cdot 10^5</td>
<td>0.07</td>
</tr>
<tr>
<td>type of AFC</td>
<td>ΔC_p</td>
<td>Δζ</td>
<td>\dot{m}_j/\dot{m}_1</td>
<td>c_μ</td>
</tr>
<tr>
<td>synthetic jet</td>
<td>-</td>
<td>10%(ref.: .106)</td>
<td>-</td>
<td>0.0015</td>
</tr>
</tbody>
</table>
1. Introduction

<table>
<thead>
<tr>
<th>author (year)</th>
<th>setup</th>
<th>inflow angle</th>
<th>Re</th>
<th>Ma</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feng (2015)</td>
<td>2D cascade numerical</td>
<td>40.2°</td>
<td>5 · 10⁵</td>
<td>0.71</td>
</tr>
<tr>
<td></td>
<td>type of AFC</td>
<td>ΔCₚ</td>
<td>Δζ</td>
<td>m/j/m1</td>
</tr>
<tr>
<td>pulsed blowing</td>
<td>-</td>
<td>14.8%(ref.: ≈0.16)</td>
<td>0.009</td>
<td>-</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>author (year)</th>
<th>setup</th>
<th>inflow angle</th>
<th>Re</th>
<th>Ma</th>
</tr>
</thead>
<tbody>
<tr>
<td>Meng (2015)</td>
<td>2D cascade numerical</td>
<td>32.123°</td>
<td>4.58 · 10⁵</td>
<td>0.21</td>
</tr>
<tr>
<td></td>
<td>type of AFC</td>
<td>ΔCₚ</td>
<td>Δζ</td>
<td>m/j/m1</td>
</tr>
<tr>
<td>sweeping jet</td>
<td>-</td>
<td>5.6%(ref.: ≈0.0394)</td>
<td>0.0008</td>
<td>-</td>
</tr>
</tbody>
</table>

**Table 1.1:** Summary of the results of active flow control applied to compressor stator test rigs

1.5 Research Beyond the State of the Art

The research of applying a pulsed jet active flow control system to a compressor stator flow is given in the framework of this dissertation. The flow features periodic non-steady phenomena (see chapter 2: [Staats2016a], [Staats2016b], [Staats2020]). The mimicked flow conditions are expected under the operation of a pulsed detonation combustor located downstream of the considered compressor stage. A highly loaded low speed compressor cascade with a design flow turning of Δβ = 60° was used for conducting all experimental research presented within this dissertation.

In [Staats2016a] the periodic non-steady base-flow is introduced. The mechanical implementation of the choking device (device that generates the periodic disturbance downstream the compressor cascade) is discussed, and the impact on the physics of the passage flow is investigated. Therefore, wake pressure measurements that evaluated the periodic stator performance degradation, as well as particle image velocimetry at multiple positions inside the stator passage, were performed. These techniques gave a first insight into the passage flow field and allowed to investigate the periodic stall on the stator airfoils. The paper discusses the laminar-to-turbulent transition mechanism, which is also subject to
periodicity. Additionally, the impact of the variation of the choking frequency to the cascade performance is discussed. Furthermore, the flow field was actuated by means of side-wall actuation at two different positions. A detailed discussion of the behaviour of the actuated flow is also included in this paper. The side-wall actuation was performed by means of a fluidic actuator featuring two outlet orifices, of which the second orifice is located farther downstream of the first one. The two blowing positions distribute the actuation mass flow and account for the periodicity of the position of the separation point.

The second paper, [Staats2016b], describes an approach to account for the periodic non-steady flow by means of closed-loop active flow control. The results of [Staats2016a] indicated a favourable actuator position for side-wall actuation of the periodic non-steady flow field that was consequently used for the experimental investigations presented in [Staats2016b]. A constrained iterative learning controller was implemented with an objective function set to mitigate disturbances measured on the suction surface of the stator blade. It was feasible to reduce the amplitude of the disturbance generated by the choking device. Optimized actuation trajectories (modulation of momentum coefficient) were used for the closed-loop experiments and the stator flow was evaluated in terms of static pressure recovery downstream of the blade row. The results were compared to a non-modulated actuation case (steady working PJA), and it was shown that the momentum modulation of the PJA by closed-loop control has the potential to strengthen the control authority of the actuator at certain phase angles.

The flow field in the highly loaded compressor cascade suffers from strong secondary flow features that can be manipulated by means of side-wall actuation, and leads to overall loss reductions and gains in static pressure recovery. Nevertheless, the flow separation on the centre blade could not be reached by the side-wall actuator. Therefore, a blade actuator was designed and the two PJA actuator systems were operated simultaneously. Experimental results are documented in [Staats2020]. The passage flow field was resolved with high resolution particle image velocimetry at seven light sheet positions inside the stator passage. Those data revealed information about the dynamic stall behaviour of the stator airfoil. Additionally, the impact of the combined AFC approach was investigated. Furthermore, pressure measurements on the stator blade and in the wake of the blade row are discussed. The AFC efficiency was evaluated by means of two individual figures of merit stated within this paper.
1. Introduction
2. Accepted Papers

2.1 Staats2016a


DOI: [10.1115/1.4031934](10.1115/1.4031934)

version: published version
Active Control of the Corner Separation on a Highly Loaded Compressor Cascade With Periodic Nonsteady Boundary Conditions by Means of Fluidic Actuators

This paper discusses the impact of a nonsteady outflow condition on the compressor stator flow that is forced through a mimic in the wake of a linear low-speed cascade to simulate the conditions that would be expected in a pulsed detonation engine. 2D/3C-PIV measurements were made to describe the flow field in the passage. Detailed wake measurements provide information about static pressure rise as well as total pressure loss.

The stator profile used for the investigations is highly loaded and operates with three-dimensional flow separations under design conditions and without active flow control. It is shown that sidewall actuation helps stabilize the flow field at every phase angle and extends the operating range of the compressor stator. Furthermore, the static pressure gain can be increased by 6% with a 4% loss reduction in time-averaged data.

[DOI: 10.1115/1.4031934]

Introduction

Due to the rising economic pressure on airlines, many research projects were launched that aimed for a general improvement of airliner performance. Reducing the specific fuel consumption (SFC) would result in an overall increase of airplane efficiency. Regarding the jet engine, the SFC can be reduced by increasing the engine’s overall efficiency. The compressor efficiency of modern aero-engines is roughly 90%, which means that more than 15 stages are used to generate the required total pressure rise [1]. To increase the work per stage, the stall margin has to be reduced, which would lead to an additional pressure rise per stage, but the compressor is not stable at all operating points. In the design process of turbomachinery, the de Haller criterion, which describes the deceleration ratio of one stator, has to be met to guarantee a satisfactory stall margin (DH > 0.72). Compressors with lower de Haller numbers are highly loaded and operate with the risk of stall under design conditions. Steady blowing can enhance the operating range of a highly loaded stator [2]. In earlier work, it was shown that the flow separations on a highly loaded compressor stator profile can be eliminated using active flow control (AFC) [3,4]. The corner separation was influenced using pulsed air jets for sidewall actuation. Blade actuators were used to eliminate separation on the blade itself. The efforts resulted in a static pressure gain of 8% with a reduction in total pressure loss of 13% simultaneously. Loss reductions of 10% were achieved by the use of synthetic jet actuators on a highly loaded compressor cascade [5,6]. The highest losses in a compressor stator originated from the corner vortex [7,8].

In the two-dimensional test rig, the boundary layer that builds up at both sidewalls causes a nonconstant, but still symmetric, velocity profile from one sidewall to the other. Taking two streamlines into account, one is located in the shear layer (close to the sidewall) and the other somewhere in the center passage, it can be found that the one at the sidewall bends farther toward the blade’s suction surface, which is a result of the lower centrifugal force at this position. The curvature of the blade’s surface remains the same but the flow speed decreases, due to the boundary layer profile. This phenomenon establishes a strong secondary flow at the region of the blade–wall junction, where fluid streams pitchwise from the pressure to the suction side within one passage. This effect is enhanced by the pitchwise pressure gradient within the passage. The extra mass is then transported toward the center of the passage. A secondary flow system is formed that is known as the passage vortex. Due to the heavy adverse pressure gradient along one passage, the flow separates from the sidewall at a certain point. The low-energy fluid then creates the corner vortex, which rotates in the same direction as the passage vortex [7]. These two vortices cause high flow losses and block the passage. In a three-dimensional compressor, which is found in gas turbines, the vortex system becomes more complex due to nonsymmetric conditions and wake passings of upstream blade rows.

It was shown that passive flow control can reduce total pressure losses [9] by influencing the corner stall at the blade–wall junction. In this paper, an active approach is introduced using a linear low-speed cascade consisting of seven highly loaded stator blades, in which sidewall actuation was used to control the corner separation. Various actuators can be used for active flow control. An overview about these actuators is given in Ref. [10]. In the presented paper, actuators based on the principle of fluidic amplification are used for active flow control purposes.

Nowadays, combustion takes place under nearly isobaric conditions. One approach to enhance the efficiency in turbomachinery is to use isochoric combustion instead. This would result in a pulsed detonation engine. One of the main advantages of the pulsed combustion is that the detonations cause the same amount of enthalpy and thereby generate less entropy compared to regular combustion used in current turbomachinery [11]. One big challenge is to manage the nonsteady outflow condition on the compressor. The last compressor stage, in particular, is under impact of the pulsed detonation and the compressor operates under risk of stall. A detailed analysis of this nonsteady...
Experimental Setup

The experimental data were obtained from a low-speed linear compressor cascade. The test rig consisted of seven highly loaded stator blades to guarantee symmetric conditions at the measurement passage. The key data are given in Table 1 and the basic setup is shown in Fig. 1. The blades were mounted on a rotatable disk that allowed an adjustment of the inflow angle of up to Δα = ±5 deg. The design inflow angle was α = 60 deg. A boundary layer suction was installed at the inlet of the cascade. The static pressure of the inflow was measured through static pressure taps located at c/3 upstream of the blade row. The total pressure was measured upstream of the cascade inlet. Furthermore, the test rig was equipped with a choking device in the wake of the cascade, consisting of 21 choke blades. This device was driven by a geared motor. On the shaft, 21 phase-angle shifted tappets were installed. Each was in contact with one choke-blade and caused its movement. By running the choking device, a periodic disturbance to the stator was generated. The motion choked the passages in the following sequence: 1-2-3-4-5-6-1, etc. This led to a maximum blocking for any one passage of 90%, by a constant 13% blocking of the cascade’s wake area. Further details regarding the choking device’s operation are found in Ref. [12]. The apparatus was capable of reaching frequencies of up to f_c = 3.5 Hz. All experiments presented here were carried out at a Reynolds number of Re = 6 × 10^5, with an average inflow speed of v_1 = 25 m/s. Taking the Strouhal number

\[ S_r = \frac{f_c \cdot L}{v_1} \]  

it is found that S_r = 0.0525 was reached using this apparatus. In engine scale compressors, this would result in choking frequencies of up to approximately f_c = 400 Hz, assuming an inflow Mach number to the stator of M_a = 0.7 and a chord length of c = 0.04 m.

Measurement System. The heavily disturbed stator flow was investigated using two independent measurement systems. In an earlier work, a traversable blade was used to provide data of the surface pressure distribution. Results obtained from the surface pressure data are documented in Ref. [12]. Furthermore, 2D/3C-particle image velocimetry (PIV) measurements were undertaken using the setup shown in Fig. 2. A pair of CMOS cameras (PCO: pco.edge 5.5) were used, offering a maximum resolution of 2560 × 2160 pixels. The laser planes were arranged in the midsection of the measurement blade. Two camera setups were used to examine the trailing edge separation (camera setup I) as well as the inflow velocity field (camera setup II). A five-hole probe was used for wake measurement. Using this probe, the static pressure rise coefficient was determined using

\[ C_p = \frac{p_{s2} - p_{s1}}{q_1} \]  


<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chord length c</td>
<td>375 mm</td>
</tr>
<tr>
<td>Blade pitch P</td>
<td>150 mm</td>
</tr>
<tr>
<td>Blade height H</td>
<td>300 mm</td>
</tr>
<tr>
<td>Inflow angle α</td>
<td>60 deg</td>
</tr>
<tr>
<td>Stagger angle γ</td>
<td>20 deg</td>
</tr>
<tr>
<td>Mach number M_a</td>
<td>0.07</td>
</tr>
<tr>
<td>Reynolds number Re</td>
<td>600,000</td>
</tr>
<tr>
<td>Choke-blade height d</td>
<td>50 mm</td>
</tr>
<tr>
<td>Choke-blade pitch b</td>
<td>50 mm</td>
</tr>
</tbody>
</table>

Fig. 1 Experimental setup: (a) cascade test rig and (b) measurement section

Fig. 2 PIV planes
The total pressure loss coefficient was calculated using
\[ \zeta = \frac{p_{0,1} - p_{0,2}}{q_1} \]
for any one actuating configuration in order to reveal the actuating efficiency. This measurement device was traversed in the wake of the measurement passage at a distance of \( L/3 \); only half of the passage was covered due to the fully symmetric flow condition.

**Actuator System.** A fluidic actuator based on the principle of fluidic amplification was used for the sidewall actuation. This kind of actuator represents a bistable system, as depicted in Fig. 3. Two pneumatic circles were used to provide fluid for the main and the control mass flow.

A well-designed fluidic amplifier is capable of switching a main mass flow between these two states in a fully modulated manner by using only \( 10 - 20\% \) of the main mass flow for the control mass flow. By adding a splitter, two output orifices were formed and switching from one state to the other at a given frequency resulted in a pulsed output behavior. The main mass flow was accelerated until the narrowest cross section was reached and caused entrainment on the control ports. The high-speed air jet is referred to as the powerjet. In state I (Fig. 3(a)), the control mass flow was led into the right control port, whereas the left control port was closed and, due to entrainment, a low-pressure region in the left control port was formed.

Coupled with the Coanda effect, this caused the powerjet to attach to the left wall and an output signal was generated on the left orifice. State II, as shown in Fig. 3(b), was achieved in an equivalent way. By closing the right control port, the low-pressure region forced the powerjet to attach on the right sidewall. Control mass flow streamed through the left control port. A fast switching valve (Festo: MHE2-MS1H-3/2G-QS-4-K) was used to switch the control mass flow between the two control ports. The two outlet orifices were arranged one after the other, as shown in Fig. 4(a). The key measurements of the actuator are to be obtained from this figure as well. The blowing angle of the outlets was \( \theta = 45 \text{ deg.} \)

Total pressure measurements were made using a pitot probe in the middle of one outlet to determine the switching characteristics of the actuator. The time-resolved pressure signal of the left orifice is shown in Fig. 5.

It can be seen that the actuator switching was fully modulated at the investigated frequency of \( f = 160 \text{ Hz}. \) In the case shown, the mass flow rate of the main mass flow was set to \( \dot{m} = 1.57 \times 10^{-3} \text{ kg/s}. \) The maximum switching frequency of the actuator was limited to the operating range of the fast switching valves. The actuator was designed for producing jet velocity peaks of up to \( \nu_{\text{jet}} = 200 \text{ m/s}, \) which would correspond to a velocity ratio of \( \nu_{\text{jet}}/\nu_1 = 8 \) in relation to the test rig's inflow condition. In this paper, two different positions of sidewall actuation were investigated and compared with each other, as shown in Fig. 4(b). In one case, the actuator was located at a relative suction surface coordinate of \( (s/S)_{\text{actu}} = 14.5\% \) and in the other at \( (s/S)_{\text{actu}} = 26.5\%. \)

**Baseflow**

The periodic nonsteady outflow condition forced a highly dynamic flow to the compressor stator. Figures 6 and 7 show the velocity field obtained from the midsection of the measurement blade, using the 2D/3C-PIV measurement technique. The frequency of the chocking device was set to a corresponding Strouhal...
number of Sr = 0.04. To illustrate the formation of the suction peak, an isovelocity line is shown at \( v = 27 \text{ m/s} \) (black-dashed isoline) in Fig. 7. The relative position of the choking device to the measurement blade is also indicated. For all of the phase angles, a velocity peak formed near the leading edge. Due to a positive pressure gradient, the flow subsequently decelerated. A trailing edge separation was found in all cases, but the separation point varied depending on the phase angle. The flow field started with a laminar boundary layer from the leading edge on, which was stabilized due to the distinct negative pressure gradient until the suction peak was reached. The low-pressure region that formed past the leading edge resulted in the velocity peak found in the flow field data. The transition from the laminar to the turbulent boundary layer took place through a laminar separation bubble (LSB), as shown in Fig. 6. On the ordinate, the distance perpendicular to the blade’s surface is found and the abscissa displays the relative suction surface coordinate. In the contour plot, the relative tangential velocity is displayed. The variable \( U_{\text{max}} \) corresponds to a certain \( s/S \) coordinate, and thus describes the local maximum tangential velocity found at every position for a range of relevant relative suction surface coordinates. Figure 7(a) depicts the phase angle \( \varphi = 360 \text{ deg} \), where the measurement passage flow field experienced the lowest possible disturbance through the choking device, as it was farthest away. Hence, the formation of the trailing edge separation was found to be at a minimum, compared to the other phase angles discussed. The trailing edge separation point was located at \( s/c = 0.8 \). A relatively small velocity peak was found between \( x/c = 0.1 \) and \( x/c = 0.2 \). After the suction peak, the flow field was destabilized by the positive pressure gradient and laminar boundary layer separation occurred, as shown in Fig. 6(a). The shape of the LSB is outlined by the dashed white line, which represents the isovelocity line for \( U/U_{\text{max}} = 0 \). It can be seen that the flow was attached until reaching \( s/S = 0.25 \). From there on a laminar separation occurred and a recirculation area formed until the flow attached again at \( s/S = 0.3 \). Behind the reattachment point, the flow formed a turbulent boundary layer. The thickness of the boundary layer increased rapidly after the point of reattachment.

In Fig. 7(b), the phase angle \( \varphi = 140 \text{ deg} \) is shown, where a greater trailing edge flow separation was noticeable. The choking device was disturbing the passage just before the measurement passage, so the separation point moved to \( s/c = 0.7 \). Taking positions before the leading edge (\( s/c \leq 0 \)) into account, it was also found that the incidence angle rises and so the suction peak appeared more distinct. The maximum velocity magnitude is found to be \( v = 33 \text{ m/s} \). Due to the fact that the choking device blocked passage 3, the mass flow rate for the discussed phase angle increased in the neighboring passages and thus caused the increased incidence angle and the increased velocity peak. Examining Fig. 6(b), it is found that the laminar–turbulent transition occurred at an earlier stage. The LSB was now stretched from \( s/S = 20\% \) to \( s/S = 25\% \). Due to the extended velocity peak, the laminar flow was destabilized closer to the leading edge. The higher mass flow rate caused a slightly increased inflow speed and also established an earlier transition.

Figure 7(c) represents the phase angle \( \varphi = 210 \text{ deg} \). As the choking device directly disturbed the measurement passage, the flow field was the most destabilized and a critical trailing edge flow separation occurred. For this phase angle, the outflow condition forced a compressor flow field, in which the trailing edge separation point moved forward to \( s/c = 0.6 \). Taking the flow characteristics at the leading edge into account, it was found that the velocity peak that was indicated by the black-dashed line was damped and formed less distinctly, but the inflow angle stayed at a near constant level, compared to Fig. 7(b). Considering Fig. 6(c), the length of the laminar flow was shortened and the transition in this case appeared from \( s/S = 17\% \) to \( s/S = 23.5\% \) for the phase angle \( \varphi = 210 \text{ deg} \).

Fig. 6 Velocity profile normal to blade surface at different relative suction surface coordinates; white dashed line represents isoline for \( U/U_{\text{max}} = 0 \): (a) \( \varphi = 360 \text{ deg} \), (b) \( \varphi = 140 \text{ deg} \), (c) \( \varphi = 210 \text{ deg} \), (d) \( \varphi = 280 \text{ deg} \)
When phase angles were reached, in which the choking device was moving away from the measurement section, as is shown in Fig. 7(d) for the phase angle $\varphi = 280$ deg, a stabilization of the flow field developed and the trailing edge separation point moved toward higher $x/c$ values. In the depicted case, the trailing edge separation point appeared at $x/c = 0.7$. The incidence angle decreased so that the size of the suction peak reached its minimum. The laminar–turbulent transition was shifted downstream, as shown in Fig. 6(d), and the LSB now formed at relative suction surface coordinates ranging from $s/S = 21\%$ to $s/S = 26\%$. The unsteady outflow condition that was forced by means of the choking device generated an oscillating trailing edge separation, combined with a moving transition. This caused phase-dependent losses as well as pressure rises, as will be seen in the following wake measurement data.

Figure 8 shows the time-averaged static pressure rise in the wake of the measurement passage derived using Eq. (2). The isolines show the local total pressure loss coefficient, which was calculated in accordance with Eq. (3). In time-averaged data, cyclic boundary conditions were formed at the suction side (ss) and the pressure side (ps) of the investigated stator passage. The right-hand case ($Sr = 0.04$) represents the time-averaged flow field with the choking device turned on. The undisturbed flow is shown on the left ($Sr = 0$). In general, the greatest losses were generated where flow separation and strong secondary flow structures occurred. As the investigated stator blade was aerodynamically highly loaded, trailing edge flow separation appeared under design conditions and the corner stall at the blade–wall junction caused the corner vortex, where farther losses were generated. The typical footprint of the corner stall can be seen in the wake data, for both cases. The lowest pressure rises were to be found in the range of the corner vortex (region I). This is the area where the highest pressure loss coefficients also emerged, with total pressure losses of up to $\zeta = 0.4$. The corner stall and the separation on the blade also limit the pressure rise a stator can deliver. Within the center of the passage (region II), relatively intense pressure rises were found. The loss coefficients in this region were zero; thus, this area was not influenced directly by any flow separation. The overall highest pressure rise was detected close to the sidewall (region III), between region I and the pressure side (ps) of the blade, that was situated on top. It is striking that the time-averaged static

![Figure 7 Phase-averaged velocity field at midsection with $v = 27$ m/s isoline (dashed black line) and area of recirculation (dashed white line): (a) $\varphi = 360$ deg, (b) $\varphi = 140$ deg, (c) $\varphi = 210$ deg, (d) $\varphi = 280$ deg](image1)

![Figure 8 Static pressure rise obtained from wake measurements and loss coefficient isolines](image2)
pressure rise throughout the passage was higher for the nondisturbed flow. Observing region I, it became evident that periodical choking resulted in a greater area of low pressure rises. Due to lower static pressure rises in the time-averaged data, the total pressure loss coefficient was lower as well. In regions II and III, the static pressure rises were lower in the choked case. The total pressure losses in these two regions were identical (zero) in both investigated cases. Region III was pushed toward the sidewall when choking was applied.

For the following investigations, the values of the static pressure rise coefficient and the total pressure loss coefficient were averaged over half of the passage for all investigated phase angles. Figure 9 shows data, in which the static pressure rise coefficient and the total pressure loss coefficient were phase-averaged. In this case, the total pressure loss coefficient was mass flow averaged. Three different Strouhal numbers were compared to the undisturbed flow field coefficients. An inconsistent distribution of both coefficients was observed. The shaded segments correspond to the regions of the measurement passage closed; hence choking to that passage occurred. Taking Fig. 9(a) into account, it emerges that the value of the static pressure rise coefficient strongly depended on the phase angle. For phase angles ranging from \( \phi = 10 \) deg to \( \phi = 210 \) deg, the periodic disturbance caused static pressure rise coefficients that were generally lower than in the undisturbed case. On the other hand, the static pressure rise coefficient increased for phase angles outside of this range. The overall impact of the choking was higher, as the lower the Strouhal number got. For the undisturbed case, the static pressure rise through the passage was \( C_p = 0.57 \). The lowest value of this coefficient was found for phase angles ranging from \( \phi = 140 \) deg to \( \phi = 170 \) deg, which in this case equalled \( C_p = 0.53 \) and was found for a Strouhal number of \( Sr = 0.015 \). The highest pressure rise was found at \( \phi = 265 \) deg and equalled \( C_p = 0.62 \) for the same Strouhal number. All other values were inside these boundaries.

Considering the total pressure loss coefficients plot in Fig. 9, it was found that this value equalled \( \xi = 0.105 \) for the undisturbed stator flow. The periodic unsteady flow also caused this value to oscillate. The losses that were found in the wake of the passage were mainly generated by the highly three-dimensional flow structures within one passage. Inspecting the total pressure regime, it was apparent that the total pressure losses can drop beyond those found in the steady flow case. The boundary condition that was imposed through the choking device changed the flow around the airfoil and caused phase angle dependent lift distribution, and thus changed the formation of the corner stall as well as the trailing edge separation and the position of the laminar–turbulent transition. For phase angles where the measurement passage was under direct influence through the choking device (\( \phi = 180 \) deg to \( \phi = 250 \) deg), the total pressure losses were relatively low. Shortly after \( \phi = 250 \) deg, the losses rise in all cases. The disturbance was highest for \( Sr = 0.015 \) compared with the two other non-steady cases, whereas the lines for \( Sr = 0.0225 \) and \( Sr = 0.03 \) were even identical for certain phase angles. This means that the total pressure loss coefficient had an asymptotic regime regarding rising Strouhal numbers. For higher Strouhal numbers, the highest total pressure losses occurred phase-shifted to the highest static pressure rises, which was an effect of the fluid inertia. The blocked passage reduced the outflow velocity; hence, the static pressure rises and this forced the dynamic change of flow structures and extended flow separation phenomena in the passage.

**Actuated Flow**

The highly three-dimensional flow structures in a compressor can be affected using active flow control. The corner stall that appears in the stator passage can be influenced by sidewall actuation. Here, two different actuating positions were compared, one with the sidewall actuators from Fig. 4(a) placed at a relative suction surface coordinate of \( (s/S)_{rel} = 14.5\% \) directly at the blade–wall junction. The other was located farther downstream at \( s/S = 25.6\% \). In general, the corner vortex was re-energized by the pulsed sidewall actuation; hence, the total pressure loss decreased and the static pressure rise increased, as will be shown later on.

Figure 10 shows the total pressure loss coefficient distribution for the actuated flow compared with the nonactuated flow field, which is represented through the isolines of the total pressure loss that was found for the baseflow case, by the use of the same choking frequency corresponding to a Strouhal number of \( Sr = 0.03 \). To take the additional mass flow through the pulsed jets into account, a method presented in Ref. [14] was applied, where the inflow total pressure was corrected. For the following investigations, the momentum coefficient was calculated by Hecklau et al. [4] as

\[
C_p = \frac{m_A \cdot \sqrt{\delta_{RMS}}}{\varphi \cdot A_t} \tag{4}
\]

The total pressure loss coefficients were obtained for the two investigated actuating positions. In both cases, the momentum coefficient was set to \( C_p = 1.4\% \). The total pressure loss

---

*Figure 9 (a) Phase-averaged static pressure rise coefficient and (b) phase-averaged total pressure loss coefficient*
coefficient distribution was similar for both cases. The region where high total pressure losses were generated in the nonactuated flow was pushed toward the sidewalls with no regard to the actuation position and ranged from \(y/H = 0.3\) to \(y/H = 0.4\). The maximum total pressure loss coefficients found in this region equaled \(c_{p,\text{nonact}} = 0.38\), which resulted in an improvement of 2\% compared to the nonactuated maximum loss coefficient. Considering the actuator position \((s/S)_{\text{actu}} = 26.5\%\) (Fig. 10, right), it was found that a spot with high total pressure losses was established in the time-averaged data spreading from \(y/H = 0.05\) to \(y/H = 0.15\). This spot was not found when the actuating position was \((s/S)_{\text{actu}} = 14.5\%\). In addition, the area of high total pressure losses reached farther into the flow field regarding the actuator position \((s/S)_{\text{actu}} = 26.5\%\). As the actuator position influenced the formation of the corner vortex, the blocking of the passage was affected as well and thus led to differences in static pressure rise, which is shown in Fig. 11 for the two actuator positions. The isolines represent the static pressure rise of the nonactuated flow field data. Higher static pressure rises were obtained all over the measurement field, compared to Fig. 8, which represents the baseline case. With the corner vortex pushed toward the sidewall, the fields of minimum pressure rise were shifted toward this direction too, as it was mentioned above. The minimum static pressure rises were found in the center of the corner vortex. It is striking that the static pressure rise generated in the center of the passage was higher for the left-hand case (actuating position: \((s/S)_{\text{actu}} = 14.5\%\)). The highest static pressure rises still occurred close to the sidewall, where the flow field was not under the corner stall’s impact, which is consistent to the nonactuated flow field.

In Fig. 12(a), the static pressure gain was averaged along the \(z/P\) coordinate and plotted for the two actuating positions. The static pressure gain was defined by

\[
\Delta C_p = (C_{p,\text{actu}} - C_{p,\text{base}})
\]

It was found that different values of pressure gain were achieved at different blade height coordinates. The two curves showed a very similar progression, but the pressure gain was approximately 0.8\% higher for the forward actuating position \((s/S)_{\text{actu}} = 14.5\%) compared to the rear one \((s/S)_{\text{actu}} = 26.5\%)\). The highest gains were found at \(y/H = -0.14\). At this coordinate, local achievements resulting in \(\Delta C_p = 6.4\%\) were feasible regarding the forward actuating position. At the position \((s/S)_{\text{actu}} = 26.5\%)\), the pressure gain was limited to a local value of \(\Delta C_p = 5.6\%)\), but at the same relative height coordinate.

Taking Fig. 12(b) into account, the changing integral total pressure loss generation became evident, which in this case was mass flow averaged. It emerges that high losses in total pressure occurred close to the sidewall, which was due to the shifted position of the corner vortex. The highest losses were found at \(y/H = -0.4\). Further toward the center of the blade, the most improvements regarding the losses were found, compared to the nonactuated flow. Comparing the two actuating positions with each other by means of the internal total pressure loss, it is striking that in the center of the passage the losses were more reduced using \((s/S)_{\text{actu}} = 14.5\%)\) for the actuating position, whereas close to the wall the actuator at position \((s/S)_{\text{actu}} = 26.5\%)\) resulted in the lowest flow losses.

The losses generated in the passage correlated with the vorticity found in the wake of the measurement passage. Figure 12(c) shows the vorticity that was averaged pitchwise. It is noticeable...
that the vorticity at coordinates ranging from $y/H = -0.47$ to $y/H = -0.38$ was higher for the two discussed actuated flow fields. This corresponds to the coordinates, where relatively high total pressure losses were found in Fig. 12(b), but which were still lower in the actuated cases. The reduction in the size of the corner vortex resulted in higher vorticity close to the sidewall. Furthermore, the intensities of the losses were arranged as the intensity of the vorticity found in Fig. 12, regarding the actuation position. Higher vorticity was found for the actuation position of flow separation at the center of the blade created the losses at the area, ranging from $y/H = -0.47$ to $y/H = -0.38$. From the coordinate $y/H = -0.35$ to the midsection of the passage, this behavior changed and the vorticity was lower for the two investigated actuating positions compared to the nonactuated flow field. Close to the center of the passage, the vorticity was close to zero and the losses occurred in this region, but the vorticity was not subject to change. Due to the heavy disturbance that was produced through the choking device, the effect of actuation could not stay constant throughout one phase. Figure 13 shows the dependency of the static pressure gain with respect to the phase angle and actuating frequency. The Strouhal number of the choking, as well as the momentum coefficient, remained the same as those for the data presented in Fig. 12. Figure 13 shows the pressure gain achieved with actuation at $(s/S)_{act} = 14.5\%$. It stands out that positive percentages in pressure gain were achieved over every phase angle with any actuating frequency. Relatively low AFC achievements resulted for the phase angles ranging from $\phi = 60\,\text{deg}$ to $\phi = 180\,\text{deg}$. For high actuating frequencies, the pressure gain dropped to minimum values of 2%, whereas low actuating frequencies resulted in higher values of pressure gain. This behavior regarding the actuating frequency was observed all over the investigated data field.

Comparing the pressure gain from Fig. 13 with the passage’s static pressure rise in Fig. 9, it emerges that the performance of the actuation was lower regarding the pressure gain for phase angles where the static pressure rise was also low and vice versa. For the actuation at the phase angle of $30\,\text{deg}$, the phase dependency of the actuation success originated from the fact that the biggest achievements were generated for phase angles where strong flow separation phenomena occurred. As AFC counteracted the flow separation, the separation point moved downstream and higher pressure rises resulted. A more distinct dependency from the actuating frequency with regard to the pressure gain was found in Fig. 13(b) than in Fig. 13(a), but the best pressure gain was still lower for the actuation position at position $(s/S)_{act} = 26.5\%$. The Strouhal number of the choking, as well as the static pressure gain with respect to the phase angle and actuating frequency. The loss reduction was calculated using

$$\Delta \zeta = \left( \zeta_{\text{base}} - \zeta_{\text{act}} \right)$$

In Fig. 14, it becomes apparent that as the momentum coefficient rises the static pressure gain rises as well. It emerges that slightly more than $\Delta \zeta = 1\%$ can be realized using as little as $C_{\lambda} = 0.002$ for momentum coefficient regarding the two actuation positions investigated. Actuation with momentum coefficients lower than $C_{\lambda} = 0.01$ showed minor differences in the overall pressure gain comparing the two actuating positions investigated with each other. As the momentum coefficient rises farther, it emerged that higher pressure ratios were achieved using the upstream-located actuator position. Regarding the reduction in total pressure loss, it can be said that it showed a very asymptotic behavior regarding rising momentum coefficients. The maximum loss reduction that was achieved depended on the position on which the actuator was installed. A better loss reduction was achieved using the upstream actuator position. As observed from Fig. 14, upstream actuators led to a $\Delta \zeta = 4\%$ loss reduction. Regarding the actuating position, $(s/S)_{act} = 26.5\%$, a loss reduction of only $\Delta \zeta = 3.4\%$ was apparent. It can be said that the total pressure loss reduction
demonstrated very asymptotic behavior for both investigated cases. The static pressure gain rises as long as higher momentum coefficients were applied.

Conclusions
In this paper, an unsteady compressor stator flow has been investigated, in which the impact to the compressor stator that is to be expected by the use of a pulsed combustion has been simulated using a choking device in the wake of a linear cascade. Several measurement techniques were used to describe the flow field within the stator passage, as well as wake characteristics. PTV measurements showed that a heavy flow separation occurred periodically due to the unsteady outflow condition. It gave evidence about the position of the laminar–turbulent transition that took place through an LSB and the fluctuating transition that resulted. Detailed wake measurements showed that the generated total pressure losses, as well as static pressure rises, were subject to change under the investigated conditions. The greatest fluctuations occurred in the static pressure rise and they slightly depended on the frequency with which the choking device was driven, as did the total pressure losses. It has been pointed out that with a rising choking frequency, an asymptotic behavior is to be expected for both static pressure rise and total pressure loss.

The comparison of two different actuation positions of sidewall actuation proved that sidewall actuation farther upstream in the passage worked better regarding pressure gain and loss reduction. Due to the unsteady flow, the total pressure loss reduction as well as the generated pressure gains were phase-dependent, where the best gain in static pressure is achieved at phase angles at which massive flow separation occurred and the flow field needed to be stabilized. The sidewall actuators mainly influenced the corner stall and only partly influenced the flow separation at the center of the blade. In the future, blade actuators will be integrated to eliminate all flow separations. Using the investigated actuation setup, total pressure loss reductions of up to $\Delta \xi = 4\%$ were achieved, with a gain in static pressure rise of $\Delta C_p = 7.5\%$ in time-averaged data.

Acknowledgment
The authors gratefully acknowledge support by the Deutsche Forschungsgemeinschaft (DFG) as part of collaborative research center SFB 1029 “Substantial efficiency increase in gas turbines through direct use of coupled unsteady combustion and flow dynamics.”

Nomenclature

- $A =$ area
- $b =$ choke-blade pitch
- $c =$ chord length
- $C_p =$ static pressure coefficient
- $C_m =$ momentum coefficient
- $d =$ choke-blade height
- $\text{DH} =$ de Hailler number
- $f =$ actuating frequency
- $f_z =$ choking frequency
- $H =$ blade height
- $m =$ mass flow rate
- $\text{Ma} =$ Mach number
- $P =$ blade pitch
- $p_s =$ static pressure
- $p_t =$ total pressure
- $p_s =$ pressure side
- $q =$ stagnation pressure
- $Re =$ Reynolds number
- $Sr =$ Strouhal number
- $ss =$ suction side
- $s/S =$ Reynolds number
- $U =$ relative suction surface coordinate
- $v =$ velocity
- $x =$ length coordinate
- $y =$ spanwise coordinate
- $z =$ pitchwise coordinate
- $\gamma =$ inflow angle
- $\Delta =$ increment
- $\zeta =$ total pressure loss coefficient
- $\phi =$ phase angle of the choking device
- $\omega =$ blowing angle of the sidewall actuators
- $a_{\text{base}} =$ index for actuated flow
- $j =$ index for baseflow
- $o =$ index for actuator jet
- $o_{\text{act}} =$ index for actuator output
- $\text{rms} =$ index for root mean square
- $i =$ index for inflow plane
- $i_2 =$ index for outflow plane

References

2.2 Staats2016b


DOI: 10.1007/s13272-016-0232-1

version: published version
Closed-loop active flow control of a non-steady flow field in a highly-loaded compressor cascade

M. Staats · W. Nitsche · S. J. Steinberg · R. King

Abstract The provision of secure compressor operation under circumstances of a pulsed detonation engine is crucial for the success of pressure gaining combustion processes for turbo machinery applications. This paper discusses active flow control as a possible solution to approach this challenge. The presented experiments were conducted on a highly loaded low speed linear compressor stator cascade operated at \( Re = 600,000 \) and \( Ma = 0.07 \). A choking-device which was located in the wake of the cascade simulated the non-steady outflow condition that is expected under the conditions of pressure gaining combustion. In the discussed experiments, the choking-device generated a periodic disturbance to every passage at a typical Strouhal number of \( Sr = 0.03 \). The flow structures of the non-steady flow field were strongly correlated to the working-phase of the choking-device. In this paper, an iterative learning controller was used to find an optimized actuation trajectory that was used for closed-loop sidewall-actuation to control the corner separation in the non-steady flow field. The iterative learning controller took advantage of the periodicity of the disturbance to calculate a non-steady actuation trajectory that optimally suppressed the impact of the choking-device on the flow. The active flow control effect was evaluated by means of static pressure rise using five hole probe measurements in the wake of one passage.

Keywords Active flow control · Linear compressor cascade · Non-steady compressor flow · Iterative learning controller

1 Introduction

Since the first turbojet powered flight in 1939 many research projects led to major improvements regarding the specific fuel consumption of turbomachinery. By the use of state-of-the-art combustion technology, the overall efficiency of single cycle gas turbines is limited to approximately 40%, using the Joule cycle. Nowadays, the combustion is an isobaric process, where only minor pressure losses have to be guaranteed to maintain film cooling on the first turbine stator. Multiple research projects are recently aiming for higher overall efficiency of gas turbines. One promising approach to reach efficiency improvements by 10% is to replace the isobaric combustion process by an isochoric one. An aerothermodynamic analysis of gas turbine engines with implemented constant volume combustion is given in [15], where 30% fuel savings are stated for a 60 kW microturbine. Potentials of saving up to 40% fuel burn for a 300-passenger long-range mission, compared to a baseline engine modeled on a Rolls Royce Tren772, are stated in [5]. In [3], an overview about opportunities to improve the thermodynamic cycle of gas turbines is given. It was found that constant volume combustion is always beneficial for the engine efficiency. In such an engine the combustion process will be pressure gaining. The combustion can be carried out as pulsed detonations or even more efficient as a shockless explosion
combustion. Either way, both combustion concepts are highly non-steady and interact especially with the neighbouring machine parts. The combustion can be processed using can-annular combustion chambers where the tubes will be closed on the compressor side sequentially when combustion is carried out, to keep the disturbances local and manageable, which results in a periodic non-steady outflow condition to the compressor. Especially the compressor stability in all operating points is crucial for the success of the non-steady combustion.

A modern compressor operates with 15 compressor stages to reach the desired overall pressure ratio (OPR) (Rolls Royce Trent1000: OPR = 50 : 1 used in B787). A pulsed detonation engine will still need to be operated with an intermediate pressure compressor to fill the combustion tubes with the required air mass flow. To further reduce the stage count, highly loaded stator blades can be used to increase the flow turning and reach higher pressure ratios per stage. Stator blades are referred to as highly loaded, when the de Haller criterion is not fulfilled and the de Haller number drops below $D_{HI} = 0.75$ [18].

Increased static pressure ratios can be reached using active flow control (AFC) [1, 13, 27]. An overview about actuators for active flow control is given in [9]. Considering compressor stator flows, the flow field is dominated by massive three-dimensional flow structures and separation phenomena especially at the blade wall junction [19]. The corner separation partially blocks the passage and limits the operating range of a compressor stator [26, 28]. It can be stated that half of the losses in compressors originate from the end wall region [20]. In [14], pulsed blowing was applied to the end walls and the suction side of the stator blades in a linear cascade, where 8–9% higher static pressure ratios and 13% reduced total pressure losses were reached simultaneously.

In the current paper, results obtained from a linear low speed cascade are shown. The non-steady outflow condition is imposed through a choking-device in the wake of the cascade that blocks every one passage subsequently. It has been shown that the flow field is heavily disturbed under the investigated conditions and active flow control can help to stabilize the flow field [17]. In [16], an energy balance for the actuator system based on the total pressure was made. Local time averaged total pressure loss reductions of 4% were achieved using moderate momentum coefficients. Due to the fact that the choking-device works periodically, the occurring flow separation phenomena are periodic as well. Iterative learning and repetitive control was applied to the periodic non-steady flow field, where it was found that the optimal AFC input highly correlates to the working phase of the choking-device [22, 24], as expected. Fluidic actuators were used for active flow control application. Fluidic devices were also used in [12] and [25] in case of loss reduction.

In this contribution, an constrained optimization-based Iterative Learning Controller (ILC) was used to find optimal actuation trajectories that were used in closed-loop experiments to control the corner separation of the non-steady operated compressor stator.

The detailed experimental setup is given in the next section. Section 3 explains the control theory used for the ILC. The optimized AFC trajectories were found using the pressure distribution on one stator blade. The effect of the optimized actuation is compared to pulsed blowing using a fixed mass-flow-rate through the actuators. Wake measurements where conducted to compare the active flow control benefits with each other. All experiments are carried out at a Reynolds of $Re = 600,000$, which is a typical design Reynold number for aero engine high pressure compressor stators, at an incompressible Mach number of $Ma = 0.07$. The relevance of the experiments is given for subsonic stator flows. The secondary flow structures that build up in the stator passage and dominate the flow field as well as the transition process are only depending on the Reynolds number, considering a given geometry of the passage [18].

### 2 Experimental setup

A two dimensional low speed linear compressor stator cascade was used for the investigations. The test rig consisted of seven highly loaded stator blades forming six passages. A choking-device was mounted at $0.71 \cdot c$ downstream of the trailing edges. Figure 1 shows two dimensional drawings of the test rig and in Table 1 the dimensions of the cascade are listed. The choking-device performs a periodic movement of everyone flap, which causes a periodic disturbance to each stator passage at a given frequency or Strouhal number. The position of the choking-device is expressed through the variable $\phi$. The phase-angle $\phi = 180^\circ$ corresponds to the case shown in Fig. 1, where the choking-blade downstream the measurement passage (passage 4) is fully closed. The phase angle $\phi = 0^\circ$ is reached, when all the centre passages are fully open. The test rig was mounted to a rotatable disk which allowed to adjust the inflow angle from $\alpha = 55^\circ$ to $\alpha = 65^\circ$. In the presented experiments, the inflow angle was $\alpha = 60^\circ$. A boundary layer suction was installed to guarantee symmetric inflow conditions to the measurement passage. Adjustable walls and tailboards helped to adjust the desired flow conditions. The inflow conditions were measured through flush mounted static pressure taps at $c/3$ upstream of the leading edges and a pitot probe before the nozzle in front of the test rig.
2. Accepted Papers

Closed-loop active flow control of a non-steady flow field...

![Cascade test rig](image)

**Fig. 1** Experimental setup

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Key data of cascade test rig</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
<td>Value</td>
</tr>
<tr>
<td>Chord length</td>
<td>(c = 375) mm</td>
</tr>
<tr>
<td>Blade pitch</td>
<td>(P = 150) mm</td>
</tr>
<tr>
<td>Blade height</td>
<td>(H = 300) mm</td>
</tr>
<tr>
<td>Inflow angle</td>
<td>(\alpha = 60^\circ)</td>
</tr>
<tr>
<td>Stagger angle</td>
<td>(\gamma = 20^\circ)</td>
</tr>
<tr>
<td>Mach number</td>
<td>(Ma = 0.07)</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>(Re = 600.000)</td>
</tr>
<tr>
<td>Choke-blade height</td>
<td>(d = 50) mm</td>
</tr>
<tr>
<td>Choke-blade pitch</td>
<td>(b = 50) mm</td>
</tr>
</tbody>
</table>

The wake measurements discussed were evaluated in terms of static pressure rise, that was measured at \(c/3\) downstream of the trailing edges of the measurement passage. The five hole probe that was used was equipped with FirstSensor HCLA12X5 differential pressure sensors. Side wall actuation was used for AFC, where only the measurement passage was actuated by one actuator on each end wall with optimized active flow control trajectories and one fixed mass flow actuation set up. In [16], two actuator positions were compared with each other. It was found that when placing the actuators at \(s/S = 14.5\%\) relative suction surface coordinate led to the best results regarding loss reduction and static pressure rise. The devices used for AFC were fluidic actuators that were based on the principle of fluidic amplification as depicted in Fig. 2.

Two pneumatic cycles were necessary to operate this actuator. One cycle fed the main mass flow and the second one the control mass flow. Last used 10–20 of the main mass flow rate. The actuator operated periodically in two stable states. State one is shown in Fig. 2a, where the left control port is closed and mass flow was led through the right control port, which caused the power jet to attach on the left wall and the left outlet orifice became active. State two is depicted in Fig. 2b and works vice versa. In the discussed experiments the switching frequency of the actuators was set to \(f_{act} = 60\) Hz. Further information regarding this actuator can be found in [16]. In the presented experiments, the actuator was designed in such way that the two outlet orifices were arranged one downstream of the other in flow direction, with a blowing angle of \(\alpha = 45^\circ\) as shown in Fig. 3. The key dimensions of the actuator can be found in this figure as well. An iterative learning controller (ILC), the working principle of which will be explained briefly in the next section, computed an optimal trajectory, i.e., time sequence, of values for the main actuation mass flow. The frequency controlling mass flow was kept at a constant level for all times. The ILC utilizes the pressure distribution along midspan of the centre blade of the compressor cascade to find the optimum solution, as it will be described in the next section.

The actuation amplitude is described through the momentum coefficient that is defined as

\[
c_u = \frac{m_{jet} \cdot \bar{u}_{jet}}{q_i \cdot A_{jet}}. \tag{1}
\]

where \(q_i\) is the inflow dynamic pressure and \(A_{jet}\) the inflow area of one passage. \(m_{jet}\) refers to the mass flow of the
actuator and $\bar{u}_{jet}$ is the mean actuator output velocity, which is calculated using the actuator geometry and the equation of continuity. By calculating
\begin{equation}
F^+ = \frac{\bar{F}_{act} \cdot l_{ref}}{u_s},
\end{equation}
the dimensionless frequency of the actuators can be found. $l_{ref}$ represents the chord length minus the length from the leading edge to the side-wall actuator. The local flow speed ($u_s$) is calculated using the pressure coefficient’s definition.

3 Iterative learning control

Previous experiments showed that the choking-device induces recurring disturbances in the entire flow field. In a next step, the goal is the mitigation of the impact of these periodic disturbance on the $c_p$-distribution through a closed-loop active flow control approach. Due to the repetitive nature of the disturbances, we decided to implement an ILC [2]. ILCs are specially designed to deal with such repetitive control tasks. By learning from one cycle to the next, the ILC successively improves the actuation trajectory. The corresponding input and output variables will be introduced in the subsequent section.

To avoid any confusion concerning the typography, it should be mentioned that variables primarily related to the field of aerodynamics are presented in the normal font, e.g., $x$, whereas control theory and signal processing-related variables are presented in italic type, e.g., $x$.

3.1 Definition of the input and output variables

In the field of control theory, dynamic systems are often described by differential equations (continuous-time) or difference equation (discrete-time) that reflect the dynamic input/output behavior of the system to be controlled. In this paper, the input $u \in \mathbb{R}$ to the cascade is the amplitude of the sidewall actuators given as the momentum coefficient $c_p$. For time step $k \in \mathbb{N}_0$, the input reads as
\begin{equation}
u(k) = c_p(k).
\end{equation}
The current status of the passage flow is monitored through a surrogate output variable $y$ which is based on the measured midspan $c_p$-profile of the middle blade’s suction surface. For time step $k$, the output is defined as
\begin{equation}
y(k) = p^T (c_{p,ref} - c_p(k)),
\end{equation}
where $c_p(k) \in \mathbb{R}^{25}$ is the measured $c_p$-profile at time step $k$, $c_{p,ref} \in \mathbb{R}^{25}$ is the reference $c_p$-profile describing the case without disturbance and without actuation, and $p \in \mathbb{R}^{25}$ with 2-norm $||p||_2 = 1$ is the direction of the $c_p$-profile that can be manipulated most effectively through the sidewall actuation. A block diagram describing this input/output system is depicted in Fig. 4. Equation (4) describes the absolute value of the projection of the current error in the measured $c_p$-profile onto the direction $p$.

The vector $p$ is the outcome of a principal component analysis (PCA) [7] which was applied to data obtained from experiments designed to reveal the coherences of actuation amplitude and $c_p$-profile. In these experiments, the input was varied in a quasi-steady fashion and the resulting change in the $c_p$-profile was recorded. The PCA then calculates the directions, the so-called principal components, that describe the data best under certain assumptions, for instance, that these directions are orthogonal to each other. For more details please refer to [22–24].

In consequence, the surrogate control variable will only detect deviations from the reference if it is possible to mitigate them effectively through the sidewall actuation.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{diagram.jpg}
\caption{Definition of system inputs and outputs for the closed-loop controller. The systems dynamic input/output behavior is described by a state space model (5), (6).}
\end{figure}
3.2 System description

To introduce the lifted system description, consider a general linear, time-invariant, single-input single-output, discrete-time state space model of the dynamic system to be controlled

\[ \dot{\mathbf{x}}(k+1) = A \mathbf{x}(k) + B u(k), \quad \mathbf{x}(0) = \mathbf{x}_0, \quad y(k+n_d) = \mathbf{c}^T \mathbf{x}(k) + d(k+n_d), \]  

(5)

where \( u(k) \in \mathbb{R} \) is the input, \( \mathbf{x}(k) \in \mathbb{R}^{n_x} \) the state, \( y(k) \in \mathbb{R} \) the output, and \( d(k) \in \mathbb{R} \) the output disturbance at time step \( k \in \mathbb{N}_0 \). The output \( y \) is assumed to be delayed by \( n_d \in \mathbb{N} \) time steps. The relative degree of the system is \( q = n_d + 1 \).

The initial condition is \( \mathbf{x}_0 \in \mathbb{R}^{n_x} \). The first \( n_d \) outputs are defined as \( y(k) = c^T \mathbf{x}_d(k) + d(k), \forall k < n_d \). \( A \in \mathbb{R}^{n_x \times n_x} \) is the discrete-time state matrix, \( B \in \mathbb{R}^{n_x} \) the discrete-time input vector, and \( c \in \mathbb{R}^{n_x} \) the output vector of the state space model. The system (5), (6) is assumed to be asymptotically stable, observable, and controllable.

Both the reference for the closed-loop operation and the disturbance are assumed to be repetitive, i.e.,

\[ r(k) = r(k_i + k), \quad d(k) = d(k_i + k), \]  

(7)

where \( k_i = i \cdot n_p, \forall i \in \mathbb{N}_0 \)

(9)
defines the point in time from where the \( i \)-th cycle, also named iteration, starts and \( n_p \in \mathbb{N} \) is the number of time steps per iteration.

A lifted vector describes the time series or the trajectory of its corresponding signal. In this context, the horizon \( n_h \in \mathbb{N} \) is the length of the time span that is described by a lifted vector. For causality reasons the later introduced ILC algorithm, the horizon must be chosen such that \( n_h \leq n_p - q \) holds. The lifted input \( \mathbf{u}(k) \in \mathbb{R}^{n_u} \) and lifted output \( \mathbf{y}(k) \in \mathbb{R}^{n_y} \) at time step \( k \) are defined as

\[ \mathbf{u}(k) = \begin{cases} \mathbf{0} \in \mathbb{R}^{n_u}, & k < n_p - 1 \\ \left( u(k-n_p+1), \ldots, u(k-n_p+n_h) \right)^T, & k \geq n_p - 1 \end{cases}, \]  

(10)

\[ \mathbf{y}(k) = \begin{cases} \mathbf{0} \in \mathbb{R}^{n_y}, & k < n_p - 1 \\ \left( y(k-n_h+1), \ldots, y(k) \right)^T, & k \geq n_p - 1 \end{cases}. \]  

(11)

respectively. The lifted disturbance \( d(k) \) and lifted reference \( r(k) \) are built up according to (11). The lifted vectors are related to each other via (5) and (6)

\[ y(k) = \mathbf{G} \mathbf{u}(k) + \mathbf{F} \mathbf{x}(k-n_p+1) + \mathbf{d}(k), \]  

(12)

\[ e(k) = r(k) - y(k), \]  

(13)

where \( \mathbf{x}(k-n_p+1) \) is calculated from (5). The matrices \( \mathbf{G} \in \mathbb{R}^{n_y \times n_u} \) and \( \mathbf{F} \in \mathbb{R}^{n_y \times n_x} \) read as follows

\[ \mathbf{G} = \begin{pmatrix} \mathbf{c}^T & \mathbf{0} & \mathbf{0} \\ \mathbf{c}^T \mathbf{A}^k & \mathbf{c}^T \mathbf{A}^{k-1} & \mathbf{c}^T \mathbf{A}^{k-2} & \mathbf{c}^T \mathbf{A}^{k-3} \end{pmatrix}, \]  

(14)

\[ \mathbf{F} = \begin{pmatrix} \mathbf{0}^T, \mathbf{0}^T, \ldots, \mathbf{0}^T \end{pmatrix}. \]  

(15)

3.3 Constrained optimization-based ILC

The ILC algorithm considered is based on ideas presented in [8]. The ILC solves a time-variant optimization problem in every time step and applies only the first element of the optimal input trajectory to the plant. The ILC uses future predictions of the lifted signal vectors. These predictions are based on measured data from the previous iteration mainly, but are corrected for changes in the initial conditions by a model-based approach.

Let \( \mathbf{e}(k+n_p|k) \in \mathbb{R}^{n_u} \) be the prediction of the future output error \( e(k+n_p) \), where information from up to the \( k \)-th time step is used to make the prediction. Furthermore, let

\[ \hat{\mathbf{u}}(k+n_p|k) = \hat{\mathbf{u}}(k+n_p|k) - \mathbf{u}(k) \]  

(16)

be the change of the actual past input trajectory \( \mathbf{u}(k) \) toward the predicted future input trajectory \( \hat{\mathbf{u}}(k+n_p|k) \). At every time instance \( k \) the ILC calculates \( \hat{\mathbf{u}}(k+n_p|k) \) by minimizing the quadratic cost function

\[ J(k) = \frac{1}{2} \left( \mathbf{e}^T(k+n_p|k) \mathbf{W}_e \mathbf{e}(k+n_p|k) \right. \]  

\[ \left. + \hat{\mathbf{u}}^T(k+n_p|k) \mathbf{W}_f \hat{\mathbf{u}}(k+n_p|k) \right). \]  

(17)

The symmetric weighting matrices \( \mathbf{W}_e \in \mathbb{R}^{n_u \times n_u} \), \( \mathbf{W}_f \in \mathbb{R}^{n_u \times n_u} \), \( \mathbf{W}_c \in \mathbb{R}^{n_u \times n_u} \) take into account the corresponding lifted vectors. They must be chosen such that the Hessian

\[ \mathbf{H} = \left( \mathbf{G}^T \mathbf{W}_e \mathbf{G} + \mathbf{W}_f + \mathbf{W}_c \right) \]  

(18)

is positive definite. Physical limitations, e.g., actuator and state saturations, can be included into the optimization problem by expressing them through a feasible set of \( n_u \in \mathbb{N}_0 \) inequality constraints and \( n_u \in \mathbb{N}_0 \) equality constraints.
M(k) \dot{\mu}(k+n_p|k) \leq m(k) \quad (19)
N(k) \dot{\mu}(k+n_p|k) = n(k) \quad (20)

with \( M(k) \in \mathbb{R}^{n(k) \times n_p}, N(k) \in \mathbb{R}^{n_p \times n_p}, m(k) \in \mathbb{R}^{n(k)}, \) and 
\( n(k) \in \mathbb{R}^{n_p}. \)

After some re-arrangements, this optimization problem can be formulated as a standard Quadratic Program (QP) [21]

\[
\min_{\dot{\mu}(k+n_p|k)} \frac{1}{2} \dot{\mu}(k+n_p|k)^T H \dot{\mu}(k+n_p|k) + f(k)^T \dot{\mu}(k+n_p|k)
\]
subject to
\[
\begin{align*}
M(k) \dot{\mu}(k+n_p|k) &\leq m(k) \\
N(k) \dot{\mu}(k+n_p|k) & = n(k)
\end{align*}
\] (21)

Various computationally effective algorithms, e.g., [6], exist to solve (21) in real time. The vector

\[ f(k) = -G^T W(\epsilon(k) - F \Delta \zeta(k+1)) + W_u u(k) \quad (22) \]

contains the measured control errors \( \epsilon(k) \) and absolute inputs \( u(k) \) which represent only data from the current and previous time steps. The term \( F \Delta \zeta(k+1) \) corrects the measured error trajectory \( \epsilon(k) \) to compensate for iteration-variant initial conditions. The change in the initial condition is based on a simulation approach

\[ \Delta \zeta(k+1) = \begin{cases} 0 & k < n_p, \\ A \Delta \zeta(k) + b \left( u(k) - u(k-n_p) \right), & k \geq n_p. \end{cases} \] (23)

using the state space model (5).

The optimal \( \dot{\mu}(k+n_p|k) \) is obtained from minimizing (21). This allows to calculate \( \dot{u}(k+n_p|k) \) through (16). Only the first element of \( \dot{u}(k+n_p|k) \) is applied to the plant at time step \( k \). In the upcoming \( (k+1) \)-th time step a new optimization problem is solved respecting the new conditions leading to \( \dot{u}(k+n_p+1|k+1) \). Repeating this procedure in every time step is referred to as the receding-horizon principle [10].

4 Results

A chart depicting the impact of the choking-device to the stator blade’s pressure coefficient distribution is shown in Fig. 5. The solid black line correspond to the \( \phi = 0^\circ \) case (all passages open) and the dashed one to the \( \phi = 180^\circ \) case (measurement passage blocked). In both cases, a laminar separation bubble (LSB) can be found, causing the laminar to turbulent transition followed by a deceleration of the passage flow field, which increases the local pressure coefficients. During one working-phase of the choking device the changing outflow boundary condition causes different inflow angles [16]. Taking the phase angle \( \phi = 180^\circ \) into account (Fig. 5: dashed black line), the increased inflow angle results in an increased suction peak with the transition taking place from \( x/c = 0.14 \) to \( x/c = 0.17 \), followed by a steep increase of the local static pressure. The decreased incidence angle at \( \phi = 0^\circ \) changes the pressure distribution on the blade’s suction surface in the way that a moderate suction peak forms up, which positively affects the transition process. The LSB now forms from \( x/c = 0.24 \) to \( x/c = 0.27 \). The decreased flow turning at this phase angle results in lower pressure recovery along the passage and a lower trailing edge pressure coefficient is reached at this phase angle compared to \( \phi = 180^\circ \). The oscillating pressure distribution caused by the periodic boundary condition was studied in detail in [17]. In this contribution, the non-disturbed pressure distribution, with all choking blades opened was chosen as the reference profile. The reference case is also shown in Fig. 5 (red line). It was found that the stator blade’s pressure distribution oscillates around this profile.

Affecting the flow field by side-wall actuation changes the pressure coefficient distribution and was used to reduce the disturbance generated by the choking device. The black line in Fig. 6 shows the norm of the disturbance over one phase of the choking device. The green line in Fig. 6 resulted when side-wall actuation was used at both endwalls in every passage with the momentum coefficient of \( c_p = 1.2\% \) and dimensionless frequency of \( F_+ = 0.5 \). The

![Fig. 5 Pressure distribution on measurement blade](image)

![Fig. 6 Disturbance generated by the choking device](image)
blue line in Fig. 6 resulted in consequence of only actuating the measurement passage, with the rest of the actuators turned off, using the same momentum coefficient and dimensionless frequency. For both curves regions can be found, where flow control has a beneficial effect on reducing the disturbance. In this contribution, only the measurement passage was actuated.

The signal processing and the ILC algorithm were implemented on a dSPACE system with a DS1005 processor board where the QP (21) was solved with qpOASES [4]. During the closed-loop experiments, the choking-device ran with a frequency of 2 Hz ($f_r = 0.03$). The controller sampling time was set to $T_s = 0.005$ s resulting in a period length of $n_p = 100$ time steps. The parameters of the dynamic model (5), (6) were identified through an experimental approach. To this end, the sidewall actuators excited the undisturbed passage flow with a pseudo random binary signal (PRBS) leading to a dynamic response of the cascade. The system response was measured using the surrogate output variable (4). Applying the prediction error method (PEM) [11] to this data leads to a dynamic model. The parameters of the (normalized) model with $n_s = 1$ are

$$A = 0.9386, b = 0.0614, c = 1, n_d = 3.$$ \hspace{1cm} (24)

These model parameters are used within (14) and (15) to build up the system matrices $G$ and $F$, respectively.

For the ILC synthesis, the weighting matrices are chosen to be $W_0 = I$, $W_p = 0.25 \cdot I$ and $W_f = 10 \cdot I$. Furthermore, three different upper input constraints $c_{p, \text{max}} \in \{0.6\%, 1.0\%, 2.0\%\}$ were considered for the closed-loop experiments. The lower bound for the input was set to be $c_{p, \text{min}} = 0.2\%$ which corresponding to a main actuation mass flow of 0 kg/s. In the particular case of $c_{p}(k) = c_{p, \text{min}}$, only the frequency controlling actuation mass flow contributes to the momentum coefficient of the actuation jet. These physical constraints were implemented by appropriate choices of $M(k), N(k), g_i(k)$, and $g(k)$. During the closed-loop experiments, the reference was set to be $p = 0$ which expresses the goal to keep the difference $(\zeta_r - \zeta_f)$ close to zero. The reference $c_{p}$-profile was chosen to be the steady-state operating point of the undisturbed and unactuated flow. The objective of the controller is thus to retain these flow conditions by expending a reasonable amount of actuation energy when the disturbance generator is operating.

Figure 7, left shows the evolution of the normalized error norm over the iterations for different values of $c_{p, \text{max}}$. Depending on the choice of the controller parameter $c_{p, \text{max}}$, the ILC converges within approximately 10 iterations for the $c_{p, \text{max}} = 0.6\%$-case and within approximately 45 iterations for the $c_{p, \text{max}} = 2.0\%$-case. We find that for larger values of $c_{p, \text{max}}$ smaller values of the converged normalized error norm are achieved. However, the results imply that the actuator is more effective at small actuation amplitudes.

The polar diagram in Fig. 7, right, shows the optimal, converged $c_{p}$-trajectories for the different $c_{p, \text{max}}$-cases. It can be seen that the ILC works with very small actuation amplitudes around $c_{p, \text{min}}$ for approximately 30% of the cycle. The actuation amplitude is only increased if it is necessary, i.e., if an increase of $c_{p}$ leads to a decrease of the cost function (17). We can conclude that utilizing an optimized actuation trajectory has great potential when it comes to saving actuation energy compared to an actuation strategy of constant actuation amplitude.

Figure 8 shows some dedicated snapshots from the time series of a converged, ILC controlled midspan $c_{p}$-profile (white, rectangle) compared to the case without flow control (black, circle) for the $c_{p, \text{max}} = 1\%$-case. The reference profile $C_{p, \text{ref}}$ (solid, red) represents the case without disturbance and without actuation. As can be seen, the controller keeps the profile closer to reference than in the uncontrolled case.

To evaluate the performance enhancements due to the actuation, the results of wake measurements are discussed in the following. Therefore, the increment of the static pressure rise coefficient

$$\Delta C_p = 2 \frac{H}{p} \int C_{p, \text{act}}(y, z) - C_{p, \text{non-act}}(y, z) \, dy \, dz,$$ \hspace{1cm} (25)

was used. $C_{p, \text{act}}$ refers to the actuated and $C_{p, \text{non-act}}$ to the non-actuated flow field. The static pressure rise coefficients were measured in the wake of the measurement passage and were calculated using the following equation.

$$C_p = \frac{p_{\text{stat.2}} - p_{\text{stat.1}}}{q_1},$$ \hspace{1cm} (26)

The values $p_{\text{stat.1}}$, $q_1$ were measured in front of the cascade and $p_{\text{stat.2}}$ was taken at the wake measurement plane, as
depicted in Fig. 1b. The optimal actuation trajectories, found through the ILC, regarded phase-dependent mass flow through the actuators. In the experiments, the control mass flow of the side wall actuators was fixed to $m_{control} = 4.3 \cdot 10^{-4}$ kg/s per actuator. The actuator’s main mass flow was modulated, which resulted in the specific output behaviour as shown in Fig. 9a, where the dynamic pressure at one output orifice is plotted. With emphasis to the periodic character of the flow field, polar coordinates were chosen. The main mass flow was zero for phase-angles between $\phi = 30^\circ$ to $\phi = 150^\circ$. In these cases, only the control mass flow causes the output signal, hence it was constant for the three investigated constrained optimization-based ILC results. It is clear to see that 30 actuation pulses are generated per working phase of the choking device and per actuator orifice.

Figure 9b depicts the increments in static pressure rise that were phase-averaged and integrated over the whole passage (25). The three curves $c_{p,\text{max}} = 0.6\%$, $c_{p,\text{max}} = 1.0\%$ and $c_{p,\text{max}} = 2.0\%$ correspond to the optimal solution found through the ILC algorithm. The dashed blue line depicts the result obtained with steady state actuation, where the actuation mass flow was kept constant at $m = 10.42 \cdot 10^{-4}$ kg/s. This is exactly the average mass flow used in the $c_{p,\text{max}} = 2.0\%$ case. The actuator’s switching frequency was $f_{\text{act}} = 60$ Hz in this case as well. Due to the fact that just the measurement passage is influenced by the side wall actuators, only moderate but still positive increments in static pressure rise occurred with every investigated set of parameters.

Taking the line for the steady state actuation setup in Fig. 9b into account, it emerges that for phase-angles $0^\circ < \phi < 90^\circ$ the lowest static pressure rises occurred compared with all other investigated cases. Looking at phase-angles ranging from $90^\circ < \phi < 180^\circ$ indifferent increments in static pressure rise were reached, whereas the pressure output of the actuator had a significant higher amplitude compared to the three other investigated cases. At $180^\circ < \phi < 240^\circ$ a small band of phase-angles was found, where the results of the steady state actuation delivered the highest increase in static pressure rise. In between $240^\circ < \phi < 330^\circ$ the $\Delta c_p$ values leveled off with the $c_{p,\text{max}} = 0.6\%$ and $c_{p,\text{max}} = 1.0\%$ cases. In between
330° < φ < 360° the pressure recovery of the passage decreased again and lower increments in static pressure rise were found in the steady state actuated case.

The phase-dependant modulation of the actuators main mass flow helped to increase the positive effect of the actuation to the static pressure recovery of the stator passage for a large range of phase-angles. Comparing the actuation amplitudes of the steady state actuation mass flow in Fig. 9a with the $c_{u,\text{max}} = 2.0\%$ case, it becomes evident that for phase-angles ranging from φ = 10° to φ = 190° these amplitudes were highest for the steady state actuation case. Looking at the remaining phase-angles, mass flow modulated cases had higher amplitudes. Almost identical actuation amplitudes were found for the $c_{u,\text{max}} = 0.6\%$ case regarding these phase-angles.

The highest values of Δ$C_p$ are located at phase-angles that do not necessarily correspond to the one with higher actuation amplitudes. Taking phase-angles in-between 0° < φ < 90° into account, it emerges that the curve for the $c_{u,\text{max}} = 2.0\%$ case converges with the Δ$C_p$ line for the $c_{u,\text{max}} = 1.0\%$ one at φ = 30°. At φ = 90° these two curves converge with the $c_{u,\text{max}} = 0.6\%$ and steady state actuation cases. Comparable Δ$C_p$ results were reached from there on until approaching φ = 210° regarding all actuation cases. The highest increment in static pressure rise was found at φ = 285° for the $c_{u,\text{max}} = 2.0\%$ actuation case. Comparing the cases for $c_{u,\text{max}} = 0.6\%$ and $c_{u,\text{max}} = 1.0\%$ with each other shows that comparable static pressure rise was reached when 90° < φ < 330°. Leaving this range, higher Δ$C_p$ occurred for the $c_{u,\text{max}} = 1.0\%$.

The detailed passage flow field is shown in Fig. 10 regarding three different phase-angles (φ = 0°, φ = 120°, φ = 285°). In this plot, the non-actuated case (Fig. 10a–c) is compared with two actuated ones. In Fig. 10d–f the steady state actuation mass flow was used. For the cases depicted in Fig. 10g–i, the optimized actuation trajectory found for $c_{u,\text{max}} = 2.0\%$ was applied. On the abscissa, the relative blade height coordinate is shown and on the ordinate the relative pitch-wise coordinate is plotted. The vector plots depict the in-plane velocity components. In the contour plots the static pressure rise, calculated from (26), becomes evident. To clearly visualize the position of the corner vortex, the Q-criterion was calculated. In general, the Q-criterion is a common tool to visualize vortex cores in three dimensional and also in two dimensional flows. Positive Q values are an indicator for vortex cores. The dashed line is an iso-Q-line for Q = 0.2 and it surrounds areas where $Q > 0.2$ values are found. The Fig. 10j–l show the position of the choking-device that corresponded to the discussed phase-angles.

Considering Fig. 10a, where the phase-angle was φ = 0°, a low pressure region was formed close to the suction sides (ss) wake. This region originated from the strong secondary flow structures forced at the blade-wall junction. In the centre passage, moderate static pressure rises were found. As the steady state actuation was switched on, as it is depicted in Fig. 10d, the vector plot significantly changed and a more dominant vortex structure was formed as it can be seen in the vector plot in between y/H = −0.45 to y/H = −0.3 and z/P = 0 to z/P = 0.3. Looking at the iso-Q-line, it emerges that the indicated vortex formed closer to the end wall. The side wall actuation had a stabilizing effect to the corner vortex, hence the blockage of the passage was reduced and the static pressure rise was increased. All over the centre passage higher static pressure rises were found compared to Fig. 10a. In Fig. 10g the optimized actuation trajectory was applied. In this case, the highest static pressure rises were achieved compared to the non-actuated and steady state actuated case. The static pressure inside the area surrounded by the iso-Q-line is also higher compared to the steady-state actuation.

In both actuated cases, the effect of the actuation to the shape of the corner vortex is similar.

When the phase-angle φ = 120° is investigated as it is shown in Fig. 10b, e, h the static pressure rises were lower in contrast to the related cases for the phase-angle φ = 0°. At this phase-angle, the blade experienced very high aerodynamic loading, which forced the trailing edge separation to become larger, hence static pressure rises were limited. Figure 10b shows a large area of low static pressure rises starting at z/P = 0.1 and reaching up to z/P = 0.6. The highest static pressure rises were still found at the end-wall region in the centre passage. Due to the larger trailing edge separation the vortex indicated through the Q-criterion was shifted in pitch wise direction, further into the passage. Regarding the actuated cases, it becomes apparent that for this phase-angle the effect actuation had to the integral static pressure rise was independent from the actuation trajectory applied to the flow field. Still, there are some differences found in the detailed flow field’s data. Taking Fig. 10e with steady state actuation applied into account, it was found that the isolated region of low static pressure rises was significantly smaller as was the area circled by the iso-Q-line. Comparing this state with the case shown in Fig. 10h, where the optimized actuation trajectory was applied, it is found that the higher actuation mass flow used in the Fig. 10e forces a regions of very low pressure rises ranging from y/H = −0.4 to y/H = −0.3 and z/P = 0.2 to z/P = 0.4, that is not found in Fig. 10h. From Fig. 9a it is seen that the actuation mass flow was lower in the optimized case in Fig. 10h and caused the corner vortex to be larger as it is seen by the iso-Q-line. But in this case the higher mass flow for the steady state actuation case (see Fig. 9a) does not influence the flow field in a more effective way.
2.2. Staats2016b
Closed-loop active flow control of a non-steady flow field...

Taking the phase-angle $\phi = 285^\circ$ into account, generally the highest static pressure rises were found. The non-actuated case showed a large area of static pressure coefficients with $C_p > 0.6$ in the centre passage. At pitch wise positions of $z/P < 0.6$ lower static pressure occurred. The iso-Q-line reaches the most into the passage compared to the other two phase-angles discussed. In this case, the secondary flow originating from the corner separation spreads out the most over the passage and its actuation promises the most positive effect. As depicted in Fig. 9b, the highest $\Delta C_p$-values were found at this specific phase-angle regarding all actuated cases. Figure 10f shows the flow field as it developed for the steady state actuated case at the phase-angle $\phi = 285^\circ$. Again the dimension of the area surrounded by $Q = 0.5$ became smaller due to the actuation. In the centre passage a more homogeneous pressure field forms. When the optimized actuation trajectory was applied, as it is shown in Fig. 10i, the core of the corner vortex was shifted pitch wise towards higher $z/P$ values. A rather uniform pressure distribution developed between $z/P = 0.4$ until the end of the passage at $z/P = 1$. Considering values of the pitch-wise coordinate ranging from $z/P = 0$ to $z/P = 0.25$ a low pressure region forms, which could not be influenced by the side-wall actuators at the discussed phase angle of the choking-device.

5 Conclusions

In the presented paper a compressor stator flow was discussed, in which a two dimensional low speed compressor cascade was operated under highly non-steady conditions that would be expected under the regime of a pulsed detonation engine. The non-steady outflow condition was imposed by a choking-device situated in the wake of the seven blades, causing a periodic choking of every passage. Due to the fact that the disturbances were periodic, a constrained optimization based iterative learning controller was used to modulate the input of two side wall actuators situated in the measurement passage to control the corner separation in this one passage. The static pressure distribution along the blade at midspan was used to calculate the output variable the ILC used. Wake measurements gave evidence about the effectiveness of the optimized actuation trajectories. The results were compared to a steady state actuation, which showed that the mass flow modulated actuator output helped to affect the corner separation more effectively in terms of static pressure rise in contrast to the steady state actuation trajectory. Local increments in static pressure rise of $\Delta C_p = 2.5\%$ were reached, while the steady state actuation only reached $\Delta C_p = 1.9\%$. The experiments showed that the chosen approach in the ILC algorithm led to positive effects regarding the compressor performance in terms of static pressure rise. In upcoming experiments the stator-flow will be influenced by additional actuators located at the suction side of the stator blade. The controller will also be applied to neighbouring passages to investigate its performance under more relevant conditions.

Acknowledgements The authors gratefully acknowledge support by the Deutsche Forschungsgemeinschaft (DFG) as part of collaborative research center SFB 1029 “Substantial efficiency increase in gas turbines through direct use of coupled unsteady combustion and flow dynamics”.

References


2. Accepted Papers
3. Submitted Paper

3.1 Staats2020

Experimental Investigations on the Efficiency of Active Flow Control in a Compressor Cascade with Periodic non-steady Outflow Conditions

Marcel Staats
PhD student
Chair of Aerodynamics
Technische Universität Berlin
Marchstraße 12, 10587 Berlin
Email: marcel.staats@tu-berlin.de

Wolfgang Nitsche
Professor
Chair of Aerodynamics
Technische Universität Berlin
Email: wolfgang.nitsche@tu-berlin.de

We present results of experiments on a periodically unsteady compressor stator flow of the type which would be expected in consequence of pulsed combustion. A Reynolds number of $Re = 600,000$ was used for the investigations. The experiments were conducted on the two-dimensional low-speed compressor testing facility in Berlin. A choking device downstream the trailing edges induced a periodic non-steady outflow condition to each stator vane which simulated the impact of a pressure gaining combuster downstream from the last stator. The Strouhal number of the periodic disturbance was $Sr = 0.03$ w.r.t. the stator chord length and the inflow speed. Due to the periodic non-steady outflow condition the flow-field suffers from periodic flow separation phenomena, which were managed by means of active flow control. In our case, active control of the corner separation was applied using fluidic actuators based on the principle of fluidic amplification. The flow separation on the centre region of the stator blade was suppressed by means of a fluidic blade actuator leading to an overall time-averaged loss reduction of 11.5%. The static pressure recovery was increased by 6.8% while operating in the non-steady regime. Pressure measurements on the stator blade and the wake as well as PIV data proved the beneficial effect of the active flow control application to the flow field and the improvement of the compressor characteristics. The actuation efficiency was evaluated by two figures of merit that were introduced in this contribution.

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AFC</td>
<td>active flow control</td>
</tr>
<tr>
<td>PIV</td>
<td>particle image velocimetry</td>
</tr>
<tr>
<td>BA</td>
<td>blade actuator</td>
</tr>
<tr>
<td>SWA</td>
<td>side-wall actuator</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds Number based on the non-steady outflow condition</td>
</tr>
<tr>
<td>Sr</td>
<td>Strouhal number based on the non-steady outflow condition</td>
</tr>
<tr>
<td>$F^+$</td>
<td>Strouhal number based on the actuation parameters</td>
</tr>
<tr>
<td>A</td>
<td>area</td>
</tr>
<tr>
<td>B</td>
<td>distance between output orifices</td>
</tr>
<tr>
<td>L</td>
<td>length of output orifices</td>
</tr>
<tr>
<td>D</td>
<td>depth of output orifices</td>
</tr>
<tr>
<td>FM</td>
<td>figure of merit</td>
</tr>
<tr>
<td>H</td>
<td>blade height</td>
</tr>
<tr>
<td>P</td>
<td>energy</td>
</tr>
<tr>
<td>b</td>
<td>choke-blade pitch</td>
</tr>
<tr>
<td>c</td>
<td>chord length</td>
</tr>
<tr>
<td>$c_\mu$</td>
<td>momentum coefficient</td>
</tr>
<tr>
<td>$c_p$</td>
<td>blade pressure coefficient</td>
</tr>
<tr>
<td>$c_p$</td>
<td>pressure coefficient measured with five-hole-probe</td>
</tr>
<tr>
<td>d</td>
<td>choke-blade height</td>
</tr>
<tr>
<td>m</td>
<td>mass-flow-rate</td>
</tr>
<tr>
<td>p</td>
<td>pressure</td>
</tr>
<tr>
<td>q</td>
<td>dynamic pressure</td>
</tr>
<tr>
<td>s</td>
<td>coordinate along the blade’s surface</td>
</tr>
<tr>
<td>t</td>
<td>time</td>
</tr>
<tr>
<td>v</td>
<td>velocity</td>
</tr>
<tr>
<td>x</td>
<td>length coordinate</td>
</tr>
<tr>
<td>y</td>
<td>span wise coordinate</td>
</tr>
<tr>
<td>z</td>
<td>pitch wise coordinate</td>
</tr>
</tbody>
</table>
### 1 INTRODUCTION

Introducing pressure gaining combustion concepts in gas turbines has the potential to increase the overall efficiency of the engines by up to 20% compared to state-of-the-art single cycle gas turbines [1]. Using constant volume combustion (CVC) instead of constant pressure combustion increases the thermal efficiency of the gas turbine cycle. Air-breathing engines usually operate with continuous combustion. A pulse detonation engine (PDE) is one type of a propulsion concept that uses CVC and has a highly non-steady characteristic [2]. Implementing the PDE concept in a gas turbine can be managed by the use of can-annular combustion chambers, where the tubes close sequentially when combustion is carried out. The upstream located last compressor stage then suffers from a non-steady outflow condition, leading to the risk of compressor stall. However, a stable compressor operation is crucial in gas turbine applications.

In recent decades many research projects have investigated passive and active flow control methods to increase the flow turning and reduce the losses of a stator vane [3], [4], [5]. The vortex structures at the blade-wall junctions generate most of the losses in a compressor stator vane [6]. Synthetic jet actuators reduce the losses of the secondary flow structures in a two-dimensional stator cascade by 10% [7]. In Hecklau et al. [8], pulsed blowing on the end-walls was combined with suction side actuation, where trailing edge flow separation on a highly loaded stator blade was suppressed and secondary flow structures were reduced, leading to 5% efficiency increase of the stator vane. Fluidic actuators have a high potential to be used for flow control applications also due to their robust design and high output jet velocities [9], [10].

In this contribution, a compressor stator flow of the type that is expected in a PDE was investigated. The two-dimensional low-speed compressor test rig in Berlin was used for the investigations. It consists of seven highly loaded stator blades and was operated at a Reynolds number of $Re = 600,000$. A periodic non-steady outflow condition was imposed by a choking device mounted downstream from the trailing edges, simulating the impact of a pulse detonation combustor to the compressor stator. An active flow control (AFC) approach was used that helped to stabilize the compressor stator flow field. Therefore, fluidic actuators were used as side-wall- and blade actuators. Recently, similar devices were investigated in terms of separation flow control in a number of research projects [11], [12].

Various measurement techniques were used for the experimental investigations. The beneficial effect of AFC was rated by means of the static pressure recovery and the total pressure loss reduction. Two suitable figures of merit were introduced which were inspired by the work of Seifert et al. [13].

### 2 EXPERIMENTAL SETUP

The cascade test-rig shown in Fig. 1 was used for the experiments. It consists of seven highly-loaded stator blades forming six two-dimensional passages. Table 1 introduces the key dimensions of the test-rig. The relevance of the Reynolds number of $Re = 600,000$ is given by the last compressor stage of a high pressure compressor in a gas-turbine, where high subsonic Mach-numbers are present with temperatures around $T = 500 \, K$ at the inlet plane. In the last compressor stage low aspect-ratios of the blades are typically used. In the described setup an aspect-ratio of $H/c = 0.8$ was installed. This leads to intense secondary flow effects that strongly depend on the Reynolds number the test-rig is operated with.

The non-steady outflow condition imposed by the constant volume combustion was simulated by a choking-device mounted at $0.71c$ behind the trailing edges. This device was designed to generate a sequential blocking of every stator

---

**Symbols**

- $\alpha$: flow angle
- $\gamma$: stagger angle
- $\Delta$: increment
- $\zeta$: total pressure loss coefficient
- $\phi$: phase angle of the choking device
- $\omega$: blowing angle of the side-wall actuators

**Indices**

- $1$: inflow plane
- $2$: outflow plane
- $actu$: actuator
- $ax$: axial
- $jet$: actuator jet
- $RMS$: root mean square
- $s$: total
- $st$: static
- $BA$: blade actuator
- $SWA$: side-wall actuator
Table 1. KEY DATA OF THE TEST RIG

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>chord length c</td>
<td>0.375 m</td>
</tr>
<tr>
<td>blade pitch P</td>
<td>0.15 m</td>
</tr>
<tr>
<td>blade height H</td>
<td>0.3 m</td>
</tr>
<tr>
<td>inflow angle $\alpha_1$</td>
<td>60°</td>
</tr>
<tr>
<td>stagger angle $\gamma$</td>
<td>20°</td>
</tr>
<tr>
<td>inlet velocity $v_1$</td>
<td>25 m/s</td>
</tr>
<tr>
<td>Reynolds number Re</td>
<td>600,000</td>
</tr>
<tr>
<td>choke-blade height d</td>
<td>0.05 m</td>
</tr>
<tr>
<td>choke-blade pitch b</td>
<td>0.05 m</td>
</tr>
<tr>
<td>Strouhal number $Sr_{choking}$</td>
<td>0.0525</td>
</tr>
</tbody>
</table>

In the current publication, results obtained from multiple measurement techniques are presented. For wake measurements a five-hole-probe was used. By that, the static pressure recovery as well as the total pressure losses of one passage was evaluated. The wake measurement plane was located at $c/3$ downstream the trailing edges. The five-hole probe was equipped with five First Sensor: HCLA 12X5 differential pressure sensors. Furthermore, the pressure distribution on the mid-section of the measurement blade was measured simultaneously. Thirty-two First Sensor: HCL 12X5 differential pressure transducers were installed around the centre-line on the measurement blade. Twenty-one of them were
mounted on the suction side of the profile and the rest were distributed along the pressure side. Additionally, the 2D/3C-PIV technique was used to clarify the internal flow structures in the stator vane focusing on the periodic flow separation phenomena. In Tab. 2 the positions of the PIV planes are given. A Litron: nanoPIV L 200-15 PIV laser and two PCO: pco.edge 5.5 cameras were used for the stereoscopic setup. The standard PIV system was triggered by a light barrier mounted to the choking device, allowing phase locked PIV snapshots. At every phase-angle, \(n = 50\) pictures were evaluated and averaged. Figure 3 depicts the location of the measurement planes that were positioned inside (PIV) and downstream (five-hole probe) from passage four (measurement passage). In the experiments, one side-wall actuator was installed at \(s/s_{max} = 14.5\%\) on each end-wall of the six passages. The blade actuator outlet orifices were located at \(s/s_{max} = 52.29\%\) on the blade suction surface in the passages two, three, four and five. Both actuator-systems were based on the principle of fluid amplification and follow the design rules presented in [15].

The flow channels of the actuators are shown in Fig. 4. The side-wall actuators, shown in Fig. 4 (a), generate pulsed air jets under a blowing-angle of \(\theta_{SWA} = 45^\circ\) with two outlet orifices on each device that were arranged in stream-wise direction. The two outlet orifices were \(D_{SWA} = 20\) mm apart and measured \(D_{SWA} = 0.4\) mm in width and featured a length of \(L_{SWA} = 20\) mm. Further information regarding the geometry and the working principle of the actuator is given in [14]. An \(\theta_{BA} = 30^\circ\) blowing-angle was used in the design of the blade actuator. The device’s flow channels are shown in Fig. 4 (b). It can be operated as a fluidic oscillator by connecting port 1a with 2a and 1b with 2b. In our case, the ports 1a and 1b were plugged and a solenoid valve was connected to the ports 2a and 2b in order to vary mass-flow-rate and switching frequency independently. The six outlet orifices of the actuator featured a slot width of \(D_{BA} = 0.4\) mm and a length of \(L_{BA} = 13\) mm. Furthermore, the slots were installed with a spacing of \(B_{BA} = 13\) mm. Extended explanations regarding the blade actuator are given in [16]. The outlet signals of the actuators that were measured using a Pitot-probe are plotted in Fig. 5. It was shown that the optimized switching frequency of both actuators corresponds to a Strouhal number of \(F_{SWA} = F_{BA} = 0.5\) [16] earlier work. The Strouhal number is defined by

\[
P^+ = \frac{f_{actuator}}{U_{ref}}
\]

The variable \(f_{actuator}\) represents the switching frequency of the active flow control device and \(U_{ref}\) describes the distance from the actuator outlet orifice to the trailing edge of the stator blade. The local velocity \(U_{local}\) was calculated from the static pressure coefficient measured at the \(x/c\) position of the actuators. Figure 5 (a) shows the total pressure at the outlet of the side-wall actuator. The switching frequency of the actuator was set to \(f_{actuator,SWA} = 40\) Hz \((F_{SWA} \approx 0.5)\) and the mass-flow-rate was \(\dot{m}_{actuator,SWA} = 6.344 \cdot 10^{-4}\) kg/s. Figure 5 (b) shows a corresponding plot regarding the blade actuator. In this case the switching frequency was set to \(f_{actuator,BA} = 60\) Hz, which also corresponds to the Strouhal number of \(F_{BA} \approx 0.5\). The mass-flow-rate was \(\dot{m}_{actuator,BA} = 1.106 \cdot 10^{-3}\) kg/s. It is necessary to take the power of the jets into account in order to evaluate the efficiency of the actuation. The jet-power is calculated using:

\[
P_{jet} = \dot{m}_{actuator} \frac{v_{jet,max}^2}{2}.
\]

The mass-flow-rate of the actuators \(\dot{m}_{actuator}\) was measured through Festo: SFAB-200 flow-meters. The variable \(v_{jet,max}\) refers to the maximum velocity at one actuator orifice and was calculated using the equation of continuity. Figure 6 shows the mass-flow-rate of the actuator on the abscissa and the maximum velocity on the right hand side axis. On the
ordinate the actuator jet-power is plotted. Taking the side-wall actuator into account it is found that the jet velocity and the jet-power are always higher for a given mass-flow-rate compared to the blade-actuator. This is due to the areas of the outlet orifices. The side-wall-actuator has a jet outlet area of \( A_{\text{outlet,SWA}} = 0.8 \cdot 10^{-5} \, \text{m}^2 \), whereas the blade actuator jet outlets cover an effective outlet area of \( A_{\text{outlet,BA}} = 1.56 \cdot 10^{-3} \, \text{m}^2 \). The input actuation momentum is taken into account using the following definition of the momentum coefficient \( c_\mu \) [14].

\[
c_\mu = \frac{m_{\text{actuator}} \cdot v_{\text{jet},\text{RMS}}}{q_1 \cdot A_1}.
\]

In this equation, the RMS value of the jet velocity is used. The pulsed jet signal is approximated with a square wave to allow the RMS value of the jet velocity to be calculated as \( v_{\text{jet},\text{RMS}} = \sqrt{0.5 \cdot v_{\text{jet},\text{max}}} \) [8]. The variable \( q_1 \) represents the inflow dynamic pressure in the cascade experiments and \( A_1 \) refers to the inflow area of one passage.

**Measurement uncertainties**

The manufacturer of the flow meters used in the experiments states an accuracy of ±0.8 % with respect to the full scale span, and the inflow dynamic pressure was measured with an error of \( \Delta q_1 = \pm 1.25 \, \text{Pa} \), as specified for the First Sensor: HCLA 12X5 differential pressure sensor. Based on these values, the deviation from the quoted momentum coefficient was always less than \( \Delta c_\mu,\text{SWA} = \Delta c_\mu,\text{BA} \leq \pm 0.064 \% \). The static pressure on the stator blade was measured with a maximum error of \( \Delta c_p \leq \pm 0.0052 \). All five-hole-probe data were evaluated in terms of static pressure rise \( c_p \) and total pressure loss \( \zeta \) behind the measurement passage, which were calculated with combined errors for the integrated values of \( \Delta c_p \leq 5.53 \cdot 10^{-4} \) and \( \Delta c_\zeta \leq \pm 3.03 \cdot 10^{-5} \) with respect to repeatability.

### 3 REFERENCE FLOW

Figure 7 shows results obtained from 2D/3C-PIV measurements in the stator cascade. The periodic motion of the choking device forces highly dynamic and recurring flow structure in the passages that were resolved using a triggered standard PIV system. In the contour plots of Fig. 7 (a), the local axial velocity ratios regarding the phase-angle
$\phi = 360^\circ$ are shown. At this phase angle no passage was blocked (see Fig. 2 (a)). At this phase-angle the flow-field was least disturbed. The strong structures of the end-wall flow are indicated by the streamlines. The suction-side leg of the horse-shoe vortex already forms at the leading edge of the stator blade. The vortex is located at the blade-wall junction. This vortex structure breaks down and mixes with the passage vortex, which represents the strongest secondary flow structure in the stator vane. It establishes due to the incoming boundary layer of the inlet flow. A streamline that is located within the end-wall boundary layer is bent farther toward the suction side of the stator blade compared to one outside the boundary layer. Thereby, a secondary flow from the pressure side toward the suction side is formed at the end-wall region. The added mass is then transported towards the centre passage and a compensating flow results that forms the passage vortex structure [17]. Toward the trailing edge, the passage vortex core was shifted closer to the suction surface. The resulting corner separation took up much space and generated major losses at the outlet of the stator vane. The streamlines on the suction surface are shown in Fig. 7 (b). The yellow dashed line indicates the vortex separation line. At $x/c = 1$ the corner separation reached into the passage until $y/H = 0.2$.

When the choking device blocked passage two as shown in Fig. 7 (c) and (d), the inflow angle was increased and all secondary flow structures were enhanced. The passage vortex core hit the suction surface farther upstream and took up more space in the passage. Lower axial velocity occurs close to the blade surface. The surface streamlines indicate a grown corner separation that is already reaching the midsection of the measurement blade. An iso-line for $v_{*a} = 0$ m/s shown in purple indicates the flow separation line. Along the vortex separation line a separation bubble started to form from $x/c = 0.5$ on.

Increasing flow separation was observed as the choking device blocked passage number three as shown in Fig. 7 (e) and (f). The surface streamlines strongly bent from the end wall towards the midsection of the measurement blade and the separation bubble broke down and reached the trailing edge, resulting in a large pressure induced trailing edge flow separation.

The flow separation extended to its maximum when passage five was blocked as shown in Fig. 7 (g) and (h). The data show an extended area with backflow, indicated by the purple iso-line in Fig. 7 (h). The separated flow reaches from $x/c = 0.75$ to the trailing edge for the blades centre section. From $y/H = 0$ to $y/H = 0.2$, a two dimensional flow separation was observed. Between $y/H = 0.2$ and $y/H = 0.5$, a strong impact of the passage vortex became evident. The contour plot in Fig. 7 (g) shows the three dimensional structures of the flow-field when passage five was blocked. The trailing edge flow separation reached far into the passage flow-field displacing the indicated streamlines away from the blade’s suction surface. The stator vane recovers from the strong flow separation phenomena when passage six was blocked. Figure 7 (k) shows the pressure induced flow separation from $x/c = 0.8$ until the trailing edge. The reduced flow separation also becomes evident in the passage flow-field shown in Fig. 7 (j).

4 ACTUATED FLOW

The compressor stator flow-field showed a dynamic corner stall and periodic pressure induced trailing edge flow separation. Two actuator systems were installed to prevent both flow separations. Experimental results of the dynamic flow field with actuation only at the side walls are presented in [14] and [18].

Figure 8 depicts the pressure recovery of the stator vane at a critical phase-angle of $\phi = 250^\circ$ with and without active flow control. At this phase-angle passage five was fully blocked. The static pressure coefficient $c_p$ was defined by

$$c_p = \frac{p_2(y,z) - p_1}{q_1}, \quad (4)$$

where $p_2$ represents the measured static pressure at a certain $y$ and $z$ position in the five-hole-probe measurement plane. The amount of $N = 240$ positions were evaluated in the wake-measurement plane. The variables $p_1$ and $q_1$ represent the inflow conditions. Considering Fig. 8 (a), no flow control was applied. The corner separation and the pressure induced flow separation on the stator blades suction surface, as found in Fig 7 (g) and (h), limits the pressure recovery of the stator passage. The lowest values of $c_p$ were found in the direct wake of the centre blade flow separation. Applying AFC through the side-wall- and the blade actuators, an overall increase in pressure recovery was observed as shown in Fig. 8 (b). In this case, the overall momentum coefficient was set to $c_\mu = 1.5\%$, where the side-wall actuators
operated at a momentum coefficient of $c_{\mu,SWA} = 1\%$ and the blade actuators with $c_{\mu,BA} = 0.5\%$. The dimensionless actuating frequency of both actuator systems was kept to the corresponding Strouhal number of $F_{SWA} = F_{BA} = 0.5$. Similar to the non actuated case, the data field shows the highest pressure recoveries in the centre of the passage for pitch wise values from $z/P = 0.5$ to $z/P = 1$. Below that pitch wise coordinate, the three dimensional flow structures in the passage still led to lower local pressure recoveries.

Adjusting the mass-flow-rate through the actuators results in a variation of the momentum coefficient $c_\mu$. The benefits of the actuating momentum were evaluated in terms of static pressure recovery as presented in Eq. 4 and total pressure loss coefficient measured downstream from passage four. The total pressure loss coefficient was corrected by means of the total pressure input of the actuators and was calculated using the following equation:

$$\zeta = \frac{p_{11,\text{corrected}} - p_2(y,z)}{q_1} \quad [19]$$

The corrected inflow total pressure $p_{11,\text{corrected}}$ was calculated as follows:

$$p_{11,\text{corrected}} = \frac{p_{11} \cdot \dot{m}_{1} + p_{1,jet} \cdot \dot{m}_{jet}}{\dot{m}_{1} + \dot{m}_{jet}}. \quad [6]$$

The inflow total pressure $p_{11}$ and the total pressure at the actuator outlet orifices $p_{1,jet}$ were mass-flow averaged. In order to evaluate the global effect of the AFC approach on the flow-field plane- and time-averaged values were used as presented in Eq. 7 and 8.

$$\overline{c_P} = \frac{1}{N} \sum_y \sum_z c_P(y,z) \quad [7]$$

$$\overline{\zeta} = \frac{\sum_y \sum_z \zeta \cdot \dot{m}(y,z)}{\sum_y \sum_z \dot{m}(y,z)} \quad [8]$$

Figure 9 shows the integral static pressure recoveries and total pressure losses with respect to the actuation momentum. On the abscissa the sum of the momentum coefficients is plotted. Every investigated combination of momentum through the side-wall actuator and the blade actuator is shown in two data points in this plot. The blue markers represent the static pressure recovery (from Eq. 7) and the red markers indicate the total pressure loss (from Eq. 8) of the stator vane. The red dashed line indicates the reference total pressure loss coefficient of the stator vane. Without any AFC, the total pressure loss coefficient was $\zeta = 0.12$. Taking the momentum coefficient variation into account, it was found that the flow losses were reduced for a certain range of actuation momentums. When only the side-wall actuator was operative, as indicated in the plot, it was possible to reduce the total pressure losses by $\Delta \zeta = 0.012$ to a value of $\zeta = 0.1079$ using a momentum coefficient of $c_{\mu,SWA} = 0.61\%$. For increasing momentum coefficients, higher total pressure losses resulted due to the high actuator output velocities. When the momentum coefficient was kept below $c_{\mu,SWA} = 1.5\%$, reductions in total pressure losses were measured. When the actuation was applied only through the blade actuator, no decreasing losses were observed. The combination of the two actuator systems led to the best results with respect to loss reduction with values of up to $\Delta \zeta = 0.014$ using a combined momentum coefficient of $c_{\mu} = 0.91\%$. In this case a total pressure loss coefficient of $\zeta = 0.1059$ was measured, when a momentum coefficient of $c_{\mu,SWA} = 0.61\%$ was applied through the side-wall actuator and the blade actuator was operated with a momentum coefficient of $c_{\mu,BA} = 0.3\%$.

The static pressure recovery of the stator vane under reference conditions was $\bar{c}_P = 0.565$, shown by the blue dashed line in Fig. 9. It was found that increasing momentum of the actuation always led to rising pressure recoveries. The major increase was caused by the side-wall actuator. If the actuation only took place through the blade actuator, less increase was measured. Applying $c_{\mu,BA} = 1.5\%$ at the blade actuator led to a pressure recovery of $\bar{c}_P = 0.597$. When the same amount of momentum was applied to the side-wall actuator this value increased to $\bar{c}_P = 0.627$. For the combined actuation, higher values in terms of pressure recovery were measured only regarding momentum coefficients exceeding $c_{\mu} = 1.5\%$.  

---

52
In order to evaluate the AFC approach in terms of efficiency, two figures of merit are introduced. The beneficial effect of AFC was outlined in terms of total pressure loss reduction and increasing static pressure recovery. The static pressure at the exit plane of the stator correlates to the trailing edge pressure of the stator blade and is used to calculate the first figure of merit. It is defined by:

\[
FM_1 = \frac{\Delta P_{T,E, AFC}}{\Delta P_{T,E, noAFC} + (P_{jet, BA} + P_{jet, SWA})} \tag{9}
\]

The variable \(\Delta P_{T,E}\) describes the physical energy conversion through the stator, based on the static pressure difference at the trailing edge of the stator blade to the inflow static pressure and is expressed by:

\[
\Delta P_{T,E} = c_{p,T,E} \cdot \frac{\dot{m}}{2} \cdot \frac{v_f^2}{2}.
\]

The static pressure coefficient \(c_{p,T,E}\) was measured at the position \(x/c = 0.956\) in the mid-section of the measurement blade. In Eq. \(9\) the energy conversion of the stator passage with AFC \(\Delta P_{T,E, AFC}\) is related to the energy conversion of the stator vane without AFC \(\Delta P_{T,E, noAFC}\) and the jet energies injected by the side-wall and blade actuator \(P_{jet, BA}\) and \(P_{jet, SWA}\). Figure 10 (a) shows the evaluation of Eq. \(9\) in the contour lines with regard to the momentum coefficients of the side-wall and the blade actuator. The contour line for \(FM_1 = 1.0\) depicts the break-even point, where the amount of input actuation energy equals the increase of the energy conversion that was measured at the trailing edge. Increasing values indicate more efficient AFC set-ups. The optimum became apparent for momentum coefficients of the side-wall actuator of \(c_{\mu,SWA} \approx 1%\) and \(c_{\mu,BA} \approx 0.4%\) regarding the blade actuator.

In order to calculate the efficiency of the AFC by taking static pressure recovery and loss reduction into account, a second figure of merit is introduced.

\[
FM_2 = \frac{c_{P, AFC}/\zeta_{P, AFC}}{c_{P, noAFC}/\zeta_{P, noAFC}} \tag{11}
\]

High pressure recoveries combined with low total pressure losses are desirable configurations of a compressor stator. The quotient \(c_{\mu}/\zeta\) is used to calculate an aerodynamic efficiency of the stator. Equation \(11\) relates the aerodynamic efficiency of the stator measured in the five-hole-probe plane with AFC to the reference case without AFC. The value for \(\Delta AFC\) was calculated using Eq. \(8\). In the contour plot of Fig. 10 (a) the result of the second figure of merit is shown. Only values regarding \(FM_2 > 1\) are shown in the contour plot. The optimum AFC setup is now found for \(c_{\mu,SWA} \approx 0.6%\) and \(c_{\mu,BA} \approx 0.3%\). The increase in the total pressure loss coefficients after reaching \(c_{\mu,SWA} + c_{\mu,BA} \approx 1%\) leads to decreasing values regarding the second figure of merit \(FM_2\). Outside the optimum, effective AFC setups still occur regarding a large range of momentum coefficients.

Figure 10 (b) shows the pressure distribution around the mid-section of the measurement blade for the reference case compared to two actuating cases for the phase-angle \(\phi = 250^\circ\). In all cases the transition takes place through a laminar separation bubble at the position \(x/c = 0.17\). For the reference case it was found that a large pressure induced trailing edge flow separation occurred at \(x/c = 0.6\) in the mid-section of the measurement blade that limited the pressure recovery from that point on.

When the actuation was operated corresponding to the optimum found in the second figure of merit, the pressure distribution shown by the green line in Fig. 10 (b) resulted. Higher pressure recovery on the trailing edge was measured and the region with flow separation was reduced but not fully suppressed. A continuously increasing static pressure was measured from the suction peak to the trailing edge when the optimum found by the first figure of merit was used for the AFC momentum coefficients.

The two different AFC setups led to static pressure recoveries that are depicted in Fig. 11 (a). In the polar plot, phase-averaged static pressure recoveries are shown. Comparing the reference case to the actuated ones it becomes apparent that an actuation with the optimum found in the first figure of merit and the second figure of merit led to increases in static pressure recovery at all phase-angles.

The green line depicts the case when the second figure of merit was at its maximum. In this case the overall momentum coefficient of the actuators was set to \(c_{\mu,BA} + c_{\mu,SWA} = 0.9%\). Increasing the momentum to \(c_{\mu,BA} + c_{\mu,SWA} = 1.4%\) led to a further increase regarding the static pressure recovery and depicts the case, where the first figure of merit had its maximum.

In Fig. 11 (b) the phase-averaged total pressure losses become evident. The equation used to calculate the second figure of merit accounts the total pressure losses in the stator vane. When the optimum momentum coefficients based on the second figure of merit were applied as seen in the green
3. Submitted Paper

Fig. 11. IMPACT OF THE AFC TO THE FLOW-FIELD REGARDING THE OPTIMUMS OF THE FIRST- AND THE SECOND FIGURE OF MERIT. (a) PHASE-AVERAGED STATIC PRESSURE RECOVERY OF THE STATOR VANE EVALUATED IN THE FIVE-HOLE-PROBE PLANE. (b) PHASE-AVERAGED TOTAL PRESSURE LOSS IN THE FIVE-HOLE-PROBE PLANE.

The higher momentum used in the optimum based on the first figure of merit led to loss reductions for a wide range of phase-angles but between $\phi = 36^\circ$ to $\phi = 180^\circ$ increasing total pressure losses occurred due to the higher actuation power input.

5 CONCLUSIONS

Periodically occurring flow separations in a compressor stator vane induced by a non-steady outflow condition were investigated. The wake of a two-dimensional low speed stator cascade was periodically disturbed to simulate the impact of a non-steady operating combustion to the last compressor stator in a gas turbine. 2D/3C-PIV measurements revealed the three dimensional flow structures and the structures of the flow separation phenomena. Phase-averaged data showed a separation bubble along the passage vortex separation line that breaks down and forms the trailing edge flow separation with changing intensity w.r.t. the working-phase of the choking device. The flow separations were suppressed by means of side-wall and blade actuation. Time averaged loss-reductions of $\Delta \bar{\tau} = 11.5\%$ were feasible by the combined AFC setup with increased static pressure recoveries by $\Delta \bar{\tau} = 6.8\%$ using an overall momentum coefficient of $\mu_p = 0.9\%$. The energy input of the actuators were taken into account in order to formulate two figures of merit that evaluate the AFC setup in terms of efficiency. It was shown that the optimized actuation setup led to major reduction of the flow separation phenomena and a strong increase in static pressure recovery regarding all phase-angles of the choking-device.
Acknowledgements

The authors gratefully acknowledge support by the Deutsche Forschungsgemeinschaft (DFG) as part of collaborative research centre SFB 1029 "Substantial efficiency increase in gas turbines through direct use of coupled unsteady combustion and flow dynamics".

References


4. Methodology and Discussion of Results

This chapter will comprehend the setup of the test-section and the actuator system that was used for the experiments. A summary and a discussion of the results presented in the research papers will be provided.

4.1 Methodology

Test-rig and instrumentation

The compressor stator cascade test section was equipped with seven highly loaded, controlled diffusion airfoils and was attached to an open wind tunnel with a maximum input power of 60 kW. A boundary layer suction was installed at both ends of the blade row, and adjustable tail boards were used to regulate symmetric flow conditions. The key features of the compressor stator test-rig are shown in fig. 4.1.

The blade actuator insert allows to equip the airfoils with different actuator systems without replacing the entire blades. In this work a fluidic actuator insert was used for the investigations. A similar actuator was applied to influence the corner separation and was mounted to both end walls in every passage. The choking device at the exit plane mimics the boundary condition of the pulsed combustion to the compressor. It is powered by an electric motor that drives a shaft, onto which twenty one tappets are arranged. They convert the rotating motion of the shaft into the movement of each individual choking flap. The device blocks the passages with a continuous motion in a fixed sequence, always blocking the outlet area of one passage at a time. In all presented experimental investigations, the passages were choked in the following sequence: 1 - 2 - 3 - 4 - 5 - 6 - 1 - 2 - etc. In order to apply phase-locked measurement techniques, such as phase-locked standard particle image velocimetry (PIV), the choking devices position is measured by an electric light barrier mounted to the top end.
4. Methodology and Discussion of Results

Figure 4.1: Compressor stator test section

of the shaft. A second electric light barrier was installed at the lower end to quantify the shaft’s rotational speed. Figure 4.2 illustrates the schematic drawing of the cascade test arrangement with the key-measures listed in tab. 4.1. Operating the choking device leads to a blocking of every passage with respect to the phase-angle, as shown in fig. 4.3. All measurements were conducted inside or downstream passage number four. The blocking of the measurement passage begins at phase-angle $\phi = 152^\circ$ and reaches its maximum at $\phi = 200^\circ$ with 93% of the passage being blocked.

The frequency of the choking-device was adjustable in the range of $f_{\text{choking}} = 0$ Hz to $f_{\text{choking}} = 3.5$ Hz. This corresponds to a maximum Strouhal number with respect to the chord length and inflow velocity of

$$S_{r_{\text{max}}} = \frac{f_{\text{choking,max}} \cdot c}{v_1} = 0.0525 \ . \quad (4.1)$$

In the presented experiments, the test rig was operated at a Reynolds number of $Re = 6 \cdot 10^5$. This condition leads to the inflow Mach number of $Ma_1 = 0.07$, at which no compressibility needs to be taken into account. The inflow speed was measured with a pitot probe upstream the test section and a static pressure tap located 1/3 $c$ upstream the centre blade. A row of static pressure ports was used to monitor and guarantee symmetric inflow conditions in every stator passage.

2D/3C particle image velocimetry measurement technique (PIV) was used for the experiments conducted in [Staats2016a]. The laser light sheet was led into the passage through a slot in one of the throttling blades. The slot was of a size that did not impact the non-steady effect of the choking device to the compressor.
4.1. Methodology

Inflow

$\frac{v_1}{\alpha_1}$

$\gamma$

$0.71 \ c$

$P$

$\frac{1}{3} c$

Pressure taps (SWA)

(BA)

Choking-device

Boundary layer suction

Adjustable tailboard

Outflow

Figure 4.2: Drawing of the test-rig

Figure 4.3: Blockage of the stator passages

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>chord length</td>
<td>$c = 0.375 \text{ m}$</td>
</tr>
<tr>
<td>blade pitch</td>
<td>$P = 0.15 \text{ m}$</td>
</tr>
<tr>
<td>blade height</td>
<td>$H = 0.3 \text{ m}$</td>
</tr>
<tr>
<td>inflow angle</td>
<td>$\alpha_1 = 60^\circ$</td>
</tr>
<tr>
<td>stagger angle</td>
<td>$\gamma = 20^\circ$</td>
</tr>
<tr>
<td>inlet velocity</td>
<td>$v_1 = 25 \text{ m/s}$</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>$Re = 6 \cdot 10^5$</td>
</tr>
<tr>
<td>choke-blade height</td>
<td>$d = 0.05 \text{ m}$</td>
</tr>
<tr>
<td>choke-blade pitch</td>
<td>$b = 0.05 \text{ m}$</td>
</tr>
<tr>
<td>Strouhal number</td>
<td>$Sr_{choking} = 0...0.0525$</td>
</tr>
<tr>
<td>diffusion factor</td>
<td>DF = 0.67</td>
</tr>
<tr>
<td>De Haller number</td>
<td>DH = 0.5</td>
</tr>
</tbody>
</table>
cascade. Measurements were conducted at four light sheets sequentially to reconstruct the flow field along the centre line of the measurement blade. One of the PIV planes was arranged in the area of the laminar-to-turbulent transition and boundary layer PIV was performed. Furthermore, five-hole probe measurements were applied. A cranked probe was connected to a traversing system and led into the cascade through the left-hand side end wall. The distance of the exit plane of the stator vane to the measurement plane was always constant at 1/3 c. The same five-hole-probe set up was applied in [Staats2016b] and [Staats2020]. For the 2D/3C-PIV measurements in [Staats2020] seven PIV measurement planes were investigated. The light sheets were led into the cascade through either the left- or the right end wall and positioned perpendicular to the suction side surface. To measure the pressure distribution on the stator blade, twenty one differential pressure transducers were connected to flush-mounted pressure taps on the suction surface and ten on the pressure side of the measurement blade.

**Actuator system**

Each passage was actuated by two side-wall-actuators and one blade actuator. Their designs were based on the principle of fluidic amplification. It was developed to serve as an active flow control system by Bauer [123]. Bauer applied such types of actuators to a two dimensional wing [124] and a complex outer wing geometry [125, 126].

![Diagram of fluidic actuators](image)

**Figure 4.4:** Drawing of the fluidic actuators used for side-wall and blade actuation, as adapted from Bauer [123]

The actuators used within the current work were driven by separated pneumatic cycles. Each was controlled by one **Bronkhorst: F-203AV** mass-flow regulator. Drawings of the flow channels of the actuators are presented in fig. 4.4. For each
4.1. Methodology

For the side-wall actuator, the two outlet orifices were arranged in flow direction. One was active at a time and a blowing angle of $\omega_{SWA} = 45^\circ$ was used. In fig. 4.5, the formation of the shear layer vortices, triggered by the side-wall actuator for the phase angles $t/T = 0$ and $t/T = 0.5$, is shown. The axis are normalized by the slot distance and the mean velocity is subtracted for the vector plot. In the top plot of fig. 4.5 the mass flow is led through the first outlet orifice and the initial vortex forms. When the first actuator outlet is inactive a second vortex is formed by the starting jet from the second outlet orifice of the actuator, as shown in the bottom plot of fig. 4.5. The second vortex seems to be weaker compared to the initial one but still contributes to higher mixing rates of the boundary layer with the inviscid flow. The injection at the two locations accounts for the periodicity of the flow separation, as will be explained later. The modulation

![Figure 4.5: Velocity field of the side-wall actuator with cross flow](image-url)
was used to rate the quality of the actuators outlet signal. This value is calculated using \[ \text{mod} = \frac{u_{\text{jet,max}} - u_{\text{jet,min}}}{u_{\text{jet,max}}} \] (4.2) and relates the amplitude of the jet pulse \((u_{\text{jet,max}} - u_{\text{jet,min}})\) to the maximum velocity of the jet \(u_{\text{jet,max}}\). The value of \(\text{mod} = 1\) indicated a perfect jet pulse. 

In fig. 4.6 this value is plotted for the side-wall actuator. The mass-flow rate through one actuator is shown on the abscissa. The ordinate shows the switching frequency. The value of the modulation is visualized in the boundaries from \(\text{mod} = 0.99\) to \(\text{mod} = 1.01\) in the contour plot. The switching frequency of this device is limited by the operating range of the solenoid valves. The chart shows a uniform distribution of the signal quality with respect to the mass-flow-rate and switching frequency. Hereinafter the output velocity will be calculated using the equation of continuity, for which a square-wave signal is assumed. The outlet velocity then calculates from 

\[
 u_{\text{jet,max}} = \frac{\dot{m}}{\rho \cdot A_{\text{out}} \cdot DC},
\] (4.3) 

where \(\dot{m}\) describes the mass-flow through the actuator and \(A_{\text{out}}\) corresponds to the entire outlet area of the actuator orifice. The actuator operates with a fixed duty cycle of \(DC = 0.5\). The left-hand side plot in fig. 4.7 shows the maximum outlet velocity with respect to the actuator mass-flow rate and compares the calculated value to measurements acquired with a pitot probe. The area of the square shaped outlet orifices of the side-wall actuator is \(A_{\text{out,SW,A}} = 2 \cdot 0.4 \cdot 20 \text{ mm}^2 = 1.6 \cdot 10^{-5} \text{ m}^2\). Data points of frequencies ranging from \(f = 20 \text{ Hz}\) to \(f = 240 \text{ Hz}\) are shown in the figure. Taking mass-flow rates below \(\dot{m} = 1.4 \text{ g/s}\) into account, a small variance of the measured data points becomes evident. Beyond that mass-flow rate, the maximum outlet velocity is a function of the switching frequency of the actuator and cannot be estimated with the presented approach.

A comparison of measured time-series of the total pressure at one outlet orifice is shown in figure 4.7 on the right-hand side. One chart contains measurements of two different switching frequencies \((f = 40 \text{ Hz} \text{ and } f = 240 \text{ Hz})\). Taking the upper right chart into account, with the mass-flow rate set to \(\dot{m} = 0.86 \text{ g/s}\), it becomes evident that the two presented output signals still reach the same maximum and minimum total pressure. When the minimum is reached, no mass-flow is led through the measured outlet, and the total pressure equals the ambient pressure.
pressure in both investigated cases. Both signals are estimated as square waves and satisfy the approach in eq. 4.3. In the lower right corner of fig. 4.7, a higher mass-flow rate of \( \dot{m} = 1.94 \, \text{g/s} \), which is located in the off-design regime of the actuator, was applied. The two investigated switching frequencies show major differences in their signal shape. At \( f = 40 \, \text{Hz} \) the assumption of a square wave is still valid, whereas for the switching frequency of \( f = 240 \, \text{Hz} \) this requirement is not fulfilled. During one cycle at high switching frequencies, the diverted power-jet does not attach to either wall in the inside of the actuator. This disables the coanda effect and the fully modulated switching characteristic \[127\], thus leading to the fact that both outlet orifices are always active and the device does not operate properly. The definition of the efficiency of the actuation, as presented later in this work, uses the jet power to account for the input energy. The power of the fluid jet is defined by:

\[
P_{\text{jet}} = \dot{m}_{\text{jet}} \cdot \frac{U_{\text{jet,max}}^2}{2}.
\]

\[4.4\]
Based on the assumption of the fully modulated square-wave pulsed air jet at the outlets with a duty-cycle of $DC = 0.5$, the jet-power is calculated as follows:

$$P_{jet} = \frac{2 \cdot \dot{m}_{jet}^3}{(\rho \cdot A_{out})^2}.$$  \hfill (4.5)

In fig. 4.7 (left-hand side) the red curve shows the evaluation of eq. 4.5 and in comparison, the estimation of the jet-power based on the pitot-probe measurements calculated from eq. 4.4. Unsurprisingly, the measurement data fit the theoretical curve of the jet power well for mass-flow rates ranging from $\dot{m} = 0 \text{ g/s}$ to $\dot{m} = 1.4 \text{ g/s}$.

**Blade actuator**

The blade actuator has six outlet orifices, each is of rectangular shape and has an area of $A_{out,BA} = 5.2 \cdot 10^{-6} \text{ m}^2$. These outlets feature a blowing angle of $\omega_{BA} = 30^\circ$. Three outlets are active at a time with a fixed duty-cycle of $DC = 50\%$. The output characteristic of the blade actuator is shown in fig. 4.8. Again,
orifices $2-4-6$ are inactive. The total pressure equals the ambient pressure level. The situation changes when the outlets $2-4-6$ are active. All positions covered by the outlet orifices $1-3-5$ show ambient total pressure levels (no blowing), while the remaining orifices show a maximum amplitude at compatible values. Again, the jet-power and the momentum coefficient of the blade actuator will be calculated based on the presented assumptions with the equation of continuity.

4.2 Summary of Results

The results within the three relevant publications contribute to the understanding of the response of the flow field in a compressor when introducing a periodic non-steady outflow condition to the last compressor stator. This specific condition represents the worst case scenario in the operation of a pulsed detonation combustor, being just downstream the last compressor stage.

In [Staats2016a] it was found that the periodic wake disturbance manifests in every flow feature of the stator passages flow field. Starting at the passage inflow plane, the incidence angle oscillates with an amplitude of $\Delta \alpha \approx \pm 2^\circ$. The exact amplitude depends on the Strouhal number the choking device is operated with. For the investigated configuration, the laminar-to-turbulent transition takes place through a laminar separation bubble. It is shifted in position between $s/S = 18\%$ to $s/S = 26\%$ with respect to the operating phase of the choking device by maintaining the length and height of the separation bubble during one cycle. Turbulent flow separation occurs on the suction surface of the stator blades with its extent being a function of the working phase of the choking device as well. All these flow features lead to time dependent distributions of the static pressure coefficient and total pressure loss in the wake of the stator row. Those values showed decreasing oscillation amplitudes with increasing choking frequencies. In the first publication the highest choking frequency corresponded to a Strouhal number of $St = 0.03$. Active flow control experiments by means of side-wall actuation indicated a sensitivity of the flow field to the actuator position. Placing the side-wall actuator at $s/S = 14.5\%$ had most beneficial effects in terms of loss reduction and static pressure recovery. The time averaged static pressure rise of the stator passage (measured at the centre line $c/3$ downstream the trailing edges) could be increased by 7.5% with decreasing the losses by 4%. The phase-resolved data showed that the effect of flow control is depending on the working phase of the choking device. It has the highest impact when strong flow separation phenomena occur.

An approach for finding an optimized actuation trajectory (mass-flow modulation) for the side-wall actuator is presented in [Staats2016b]. The objective in this contribution was the mitigation of the impact of the choking device to the static pressure distribution at the centre line of the measurement blade (suction side only). Mass-flow modulated side-wall actuation was used to maintain the reference pressure distribution that was measured under undisturbed condition. The chosen iterative learning controller found the optimized trajectory after less than 100 iterations and managed to reduce the error norm by 50%. Applying
the optimized actuation trajectory in closed-loop experiments allowed for wake measurements where the stator performance in terms of static pressure recovery was evaluated and compared to the non-mass-flow-modulated actuation (pulsed blowing with steady state mass-flow rate). The mass-flow modulation led to 0.6% higher pressure recoveries compared to the non-modulated mass-flow actuation.

High detail 2D/3C PIV results presented in [Staats2020] revealed that all of the flow separation phenomena can be avoided by applying a combination of side-wall and blade actuation. The investigation of the base flow case (no AFC) gave evidence about the phase-dependent separated flow structures during one cycle of the choking device. In this paper two figures of merits were introduced that help to evaluate the efficiency of the active flow control approach by taking the jet powers through the actuators into account. The first figure of merit \( FM_1 \) can be applied in the absence of loss measurements, whereas the second figure of merit \( FM_2 \) also uses the total pressure losses to calculate the efficiency related value. Both figures of merit predict the most efficient actuator momentums in the same order of magnitude. A parameter study indicated the optimized momentum share between the side-wall and the blade actuators. Higher momentum applied through the side-wall actuator and less momentum applied through the blade actuator have the most beneficial effect to the stator flow. Applying those optimized momentums massively reduces the trailing edge flow separation on the suction surface of the stator blade by applying as little as \( c_p = 0.9\% \).

4.3 Discussion of Results

The impact of a periodic non-steady outflow condition to the flow field in a linear compressor stator cascade is investigated in this dissertation. Such a boundary condition is expected in the last compressor stator row in a gas turbine with a downstream located pulsed detonation combustor. Furthermore, active flow control as a tool to prevent the occurring dynamic stall of the compressor airfoil is demonstrated and the potential of it, to stabilize the flow field, is shown in all three relevant publications.

In order to tailor active flow control to the occurring flow features, such as laminar to turbulent transition and point of turbulent flow separation, the reference flow field (non-steady flow field without AFC) needed to be analysed. Based on the PIV results presented in [staats2016a], the data shown in fig. 4.9 can be derived. This plot illustrates the dynamic stall case in terms of the turbulent separation location vs. the angle of incidence for one cycle of the choking device with the choking frequency being set to \( f_{choking} = 2 \) Hz. The Strouhal number \( Sr = 0.03 \) corresponds to that choking frequency. When calculating the reduced frequency from eq. 1.2, the value \( k = 0.094 \) is derived. As \( k \) is in the order of magnitude of \( k = O(0.1) \), the airfoil does not behave as in steady conditions, and a dynamic stall scheme is evident, as it was observed by Carr [35]. The incidence angle of the compressor stator blade varies in between \( \Delta \theta \approx \pm 2^\circ \), whereas the trailing edge turbulent flow separation oscillates from \( x/c \approx 1 \) (no flow separation) to \( x/c \approx 0.75 \). At the beginning of one cycle, the separation point is located close to
the trailing edge and the stator blades incidence angle reaches its lowest values. As the phase angle increases, the inflow angle and the separation length increase, until reaching a phase angle of \( \varphi = 150^\circ \). From there on, the incidence angle decreases but the separation point keeps moving towards lower \( x/c \) values. The most upstream located separation point is found at the phase angle \( \varphi = 220^\circ \), from which on the flow field recovers and the flow separation length decreases. A comparison of the results to the ones presented by McCroskey \[37\] is attempted in fig. [4.10] McCroskey used a single airfoil (VR-7) that oscillated with a given reduced frequency and an incidence variation of \( \Delta \alpha = 5^\circ \). The trailing edge separation point vs. the working phase and the angle of attack is given in fig.

![Figure 4.9: Incidence angle vs. trailing edge separation point](image)

![Figure 4.10: Separation point vs. working phase; left: for the VR-7 airfoil \[37\]; right: for the compressor stator blade](image)
The plot shows three different reduced frequencies, the airfoil was oscillated with. Taking the case with \( k = 0.025 \) (lowest reduced frequency) into account, the highest incidence is reached at \( \omega t = \pi/2 \). For the lowest reduced frequency of \( k = 0.025 \) the separation point is located at the most upstream position and occurs with no phase shift to the highest angle of attack, which is indicated on the second y-axis. Taking higher reduced frequencies into account reduced trailing edge separation lengths were obtained, and the phase angle of highest separation length is shifted to a higher phase angle. Comparing these findings to the compressor cascade flow, some similarities are identified. The right-hand side plot in fig. 4.10 shows the phase dependent separation point of the periodic non-steady stator passage flow. Here, the separation length also depends on the working phase and the reduced frequency that the choking device is operated with. Higher \( k \)-values increase the phase angle at which the largest separation length occurs and decreases the maximum reverse flow region.

The flow topology in the compressor cascade presented in this dissertation is different from the one investigated by McCroskey. A choking device induces a periodic wake blocking that reduces the mass flow through one passage. This leads to variations of the axial velocity component of the inflow velocity triangle, hence to oscillations of the inflow angle. The dynamic mechanism of the cascade flow w.r.t. the transition and turbulent flow separation is presented in fig. 4.11. The reduced frequency of the periodic disturbance was set to \( k \approx 0.1 \). In the plot on the right-hand side of fig. 4.11 two curves are shown. The red curve represents the current incidence angle, and the black one, with the shaded area, shows the current separation length on the stator profile. It is found that the largest trailing edge separation length is phase shifted by \( \Delta \varphi \approx 60^\circ \) compared to the phase angle of highest incidence angle. A surprising observation is that the phase angle shift is in the identical order of magnitude as observed by McCroskey (fig. 4.10 left), even though the boundary conditions are different. Additionally, this plot visualizes the formation of the flow field at the centre section of the measurement blade. The left column indicates the region around the transitional flow ranging from \( s/S_{\text{max}} \approx 0.15 \) to \( s/S_{\text{max}} \approx 0.3 \). The white line indicates the separation line and depicts the formation of the laminar separation bubble during one phase. The bubble is shifted in position but remains nearly constant in length and height. The laminar turbulent transition shifts upstream when the incidence angle is increased. The most upstream position is reached, when the blockage of the passage (passage 4) reaches its maximum (see fig. 4.3). The high pressure gradients after reattachment lead to an early point of trailing edge flow separation. The trailing edge flow is depicted in the second column of fig. 4.11 and is discussed in detail in [Staats2016a]. For the successful application of active flow control, the actuator placement is a crucial parameter. The sensitivity of positioning the actuator slot at different locations on the suction surface of a NACA0012 wing section is shown in fig. 4.12. Here, the airfoil was operated in a deep dynamic stall regime with constant blowing active flow control. The reduced frequency was set to \( k = 0.15 \), and the angle of attack oscillations were a sinus motion in between \( \alpha = 5^\circ \) and \( \alpha = 25^\circ \). The left plot indicates the lift...
4.3. Discussion of Results

Figure 4.11: Flow field measured at the transition location and the trailing edge for different phase angles of the choking device
Figure 4.12: Impact of actuator position to the aerodynamic loads on a dynamically stalling airfoil, as observed numerically by [54].

It is seen that the drag coefficient is more sensitive to the actuator position than the lift coefficient. Any investigated actuator position turned out to be beneficial in terms of lift increase, whereas placing the actuator too far downstream (here at \( x/c = 10\% \)) drag increases resulted. In this case, the optimum position was found at \( x/c = 6\% \), where the highest drag reductions occurred and hysteresis was most reduced. In [Staats2016a] a variation of the actuator placement for the side-wall actuators was performed. Two actuator positions were compared to each other (\( s/S_{\max} = 14.5\% \) and \( s/S_{\max} = 26.5\% \)). The findings from 4.12 are not fully transferable to the side-wall actuator placement as the type of flow separation is different. Tadjfar [54] investigated a two-dimensional dynamically stalling airfoil. Here, the low aspect ratio of the staggered airfoils introduce strong secondary flow structures that are affected by the side-wall actuation. The dynamic stall in the compressor cascade led to oscillations of the passage vortex break down position. By applying active flow control on the end walls, the passage vortex is kept stable, and the vortex breakdown is shifted farther downstream the stator passage. Hence, having the actuator placed too far downstream, beyond the point of vortex breakdown, reduces the impact of the active flow control device, at least at specific phase angles.

Phase-dependent total pressure losses and static pressure recoveries of the stator passage with two different actuation positions (\( s/S_{\max} = 14.5\% \) and \( s/S_{\max} = 26.5\% \)) for the side-wall actuators and the reference configuration, without AFC (“AFC OFF”) are shown in Fig. 4.13 left. In both actuated cases the momentum coefficient was kept constant at \( c_\mu = 1.4\% \). Shifting the curves to the lower right corner in the plot indicated the most favourable actuation setup. Actuating at \( s/S_{\max} = 14.5\% \) leads to the lowest pressure losses and even shows less hysteresis compared to the downstream actuating position. In terms of static
4.3. Discussion of Results

Pressure recovery, both cases reach similar values. Figure 4.13 right shows the time-averaged total pressure loss reduction and the static pressure increase of the stator passage measured with the two investigated side-wall actuator setups (line and time averaged at the centre section, $x/c = 1/3$ downstream the measurement blade). Results, also presented in [Staats2016a], revealed different actuation frequencies to work best for the two actuator positions. For the actuator placement at $s/S_{\text{max}} = 14.5\%$, the frequency $f = 30$ Hz ($Sr \approx 0.38$) revealed the highest pressure recoveries. At $s/S_{\text{max}} = 26.5\%$ it was $f = 60$ Hz ($Sr \approx 0.33$). It can be stated that both excitation frequencies match in terms of the Strouhal number ($Sr \approx 0.35$). The total pressure loss produced in the stator passage is also a measure for the drag generated by the airfoil. Higher static pressure recoveries also indicate higher lift generation by the airfoil. In the discussed case the time-averaged total pressure loss reduction is more impacted by the actuator position than the static pressure recovery. That behaviour is similar to the observations reported by Tadjfar [54].

The highly unsteady nature of the process investigated in the compressor stator cascade suggested the use of some control approach. In this order the input momentum of the actuators was more effectively distributed with respect to the working phase of the choking device. An iterative learning control (ILC) approach was applied to find an optimized actuation trajectory in order to account for the repetitive occurrence of the disturbances. The optimized trajectory was used for closed-loop control in experiments as discussed in [Staats2016b]. Therefore the constrained optimization based ILC, implemented by Steinberg, found an actuation trajectory that minimized the disturbance of the choking device to the compressor stator flow. The objective function of the ILC was to minimize the deviation of the suction-side pressure distribution to a reference profile by

Figure 4.13: Impact of actuator position on the aerodynamic coefficients of the compressor stator airfoil; left: phase resolved; right: time averaged.
using a pre-defined maximum amount of actuation momentum. The reference $c_p$ profile was taken from the undisturbed compressor stator flow. In these experiments, the measurement passage of the stator cascade (passage 4) was equipped with active flow control devices at both end walls. All other passages remained unactuated. The results indicated the potential to reach higher pressure recoveries with the ILC-optimized actuation trajectory compared to steady state actuation, as seen in fig. 4.14 (a). Here, steady state actuation refers to pulsed blowing actuation without mass-flow-rate modulation. Figure 4.14 (a) compares

![Figure 4.14: Comparison of stator passage flow with and without close-loop active flow control; a) Integrated pressure rise increment of the stator passage generated by the AFC; b) Actuation trajectories](image-url)

the integrated static pressure recovery measured in the wake of the stator cascade with and without close-loop active flow control. Therefore the increase of the static pressure recovery was calculated as follows: $\Delta C_P = C_{P,AFC} - C_{P,ref}$. The value for $C_{P,ref}$ was taken from the disturbed stator flow, without AFC. For steady state actuation the mass-flow rate used for the close-loop experiment was time averaged and applied constantly through the side-wall actuators. In the plot, a controlled case, where the momentum coefficient constraint was set to $c_{\mu,max} = 2\%$, is shown as well. It is striking that higher pressure recoveries were reached for this case. Only in-between $\phi = 180^\circ$ and $\phi = 240^\circ$, the steady-state actuation led to higher increases of the static pressure recovery. The two actuation trajectories used in the experiments are shown in fig. 4.14 (b). The plot shows the dynamic pressure measured at one outlet orifice of one side-wall actuator. It should be kept in mind that the mass-flow modulated actuation of all passages would lead to different actuation trajectories found by the ILC and may also result in different pressure recoveries. In the scope of this work, the two actuation concepts were compared with respect to the static pressure recoveries of one stator passage.

In [69], a constrained repetitive model predictive control (RMPC) approach was used for side-wall actuation. Here, the compressor stator cascade was also operated in the non-steady regime and the passages were throttled with a Strouhal
number of $Sr = 0.03$. In this work, the results obtained with the controller were also compared to steady state actuation. The momentum coefficient from the RMPC-trajectory was time-averaged and applied through the side-wall actuators for the non-mass-flow-modulated active flow control. In these cases, the actuation was also applied to the centre stator passage of the compressor cascade. The obtained wake measurement data were evaluated in terms of static pressure recovery and total pressure loss of the measurement passage. Figure 4.15 shows time-averaged data comparing the RMPC actuation with the constant momentum actuation. In the left-hand side plot, the total pressure loss coefficient is indicated. For both cases, the total pressure loss of the stator passage is reduced with increasing momentum coefficients. A growing static pressure recovery is obtained as well, as indicated by the right-hand side plot. Only minor differences exist between the constant momentum actuation and the RMPC active flow control approach. In the time-averaged data, the RMPC approach always leads to slightly higher pressure recoveries as it was also the case when the actuation trajectory was calculated with the ILC approach. This was the case in [Staats2016b].

![Figure 4.15: Time averaged total-pressure loss coefficient and static pressure rise coefficient of the unsteady compressor flow with active flow control](image)

Applying active flow control through the side-wall actuators is not sufficient to control the periodically occurring centre-blade flow separation, as it is shown in [Staats2020]. Within this work the flow structures at the blade-wall junction were resolved at high detail by the applied measurement technique. A composition of multiple 2D/3C-PIV sheets indicated that the passage flow around the centre line is nearly fully attached at phase angles where no stator passage is blocked. This corresponds to the findings from [Staats2016a], where 2D/3C-PIV measurements were conducted in the centre section of the measurement passage. The region of reverse flow drastically increases when passages close to the measurement passage are blocked. This leads to highly dynamic loads of the stator and causes periodically oscillating static pressure recoveries and loss productions. In fig. 4.11 the trailing edge flow separation reached $x/c = 75\%$ to $x/c = 80\%$ in between the phase angles $\phi = 200^\circ$ to $\phi = 250^\circ$. That extent of trailing edge
separation was confirmed in [Staats2020], as depicted in fig. 4.16. In this plot, the static pressure distribution, measured at the centre line of the measurements blade is shown. The trailing-edge separation found in the static pressure data match with the surface streamlines on the measurement blade (The lower plot in the figure). The surface streamlines were extracted from the phase-resolved 2D/3C-PIV composition. The purple line represents the separation line and indicates the separation point for the depicted phase-angle of $\phi = 250^\circ$ to be at $x/c \approx 0.72$. Downstream of that position, a pressure gradient found in the pressure distribution along the suction surface was zero when no active flow control was applied (reference case), as shown in the figure as well.

In the third contribution, a blade actuator was installed in each stator passage, where the outlet slots were located at $x/c \approx 55\%$. At this position, the flow is fully attached to the stator blades suction surface during the whole working phase of the choking device, as depicted in fig. 4.11.

![Image of flow separation on the measurement blade at a critical phase angle.](image)

**Figure 4.16:** Flow separation on the measurement blade at a critical phase angle: comparison between static pressure data and PIV data.

Switching on the AFC system with high momentum coefficient leads to full re-attachment of the flow. Figure 4.17 shows the flow field in the stator passage at a critical phase angle with and without AFC. In the non-actuated case the
4.3. Discussion of Results

Figure 4.17: Impact of the AFC on the passage flow at $\phi = 250^\circ$

dominating flow separations are illustrated by the $v_{ax} = 0$ and the $Q = 0$ iso-surfaces. When the AFC-system is switched on the vortex structures are kept close to the side wall and the flow stays attached over the entire stator blade. In this case the overall momentum coefficient equalled $c_{\mu} = 3.2\%$.

Within the skope of [Staats2020] two different evaluation methods for active flow control of the investigated compressor stator flow were presented. The first figure of merit is calculated based on the pressure recovery of the stator passage and the actuation jet power. For the second Figure of Merit ($FM_2$), the stator is considered a device to convert kinetic energy into static pressure with low loss production. The static pressure recovery, as well as the total pressure loss and actuator jet momentum are used for its calculation.

When flow losses are not to be considered, but the efficient increase of the static pressure recovery is evaluated, the first figure of merit, $FM_1$, as presented in eq. 4.6 can be used.

$$FM_1 = \frac{\Delta P_{TE,c}}{\Delta P_{TE,b} + P_{jet}} = \frac{c_{p,TE,c} \cdot \frac{v_1^2}{2} \cdot \dot{m}_1}{c_{p,TE,b} \cdot \frac{v_1^2}{2} \cdot \dot{m}_1 + v_{afc}^2/2 \cdot \dot{m}_{afc}}$$ (4.6)

A portion of the passage inflow kinetic energy ($E_{kin,1} = \frac{\rho}{2} \cdot v_1^2 \cdot \dot{V}_1$) is converted into potential energy downstream the stator passage. The increment of the potential energy conversion can be expressed by: $(\Delta E_{pot} = (p_{s,2} - p_{s,1}) \cdot \dot{V}_1)$. The quotient of these two terms reveal the portion of energy conversion through the stator and is expressed by the static pressure coefficient. In eq. 4.6 the conversion of potential energy through the stator passage with active flow control ($\Delta E_{pot,c} = \Delta P_{TE,c}$) is related to the conversion in the reference case ($\Delta E_{pot,b} = \Delta P_{TE,b}$). The fluid jet power of the actuator is added ($P_{jet} = \dot{m}_{jet} \cdot U_{jet,max}^2 / 2$). The value of $FM_1 = 1$ indicates the break even, where input jet power to the system is equivalent to the increased static pressure energy conversion of the stator. In [Staats2020], the trailing edge pressure of the stator profile ($\Delta P_{TE,b}$ and $c_{p,TE}$) in the measurement passage was considered to be representative for the overall pressure recovery of the whole stator passage. Fluidic actuators were used that generate pulsed blowing. The jet mass flow always fed two outlet channels in an alternating pattern with a fixed duty cycle of $DC = 50\%$. Therefore, the maximum outlet velocity of one cycle was used to calculate the overall jet power...
of the actuator. The jet velocity was calculated using the equation of continuity. By evaluating the AFC results using the first figure of merit, values of up to $FM1 = 1.2$ were reached for the combined AFC approach. The calculation of the FM1 is also valid, when the space and time averaged static pressure recovery from downstream the stator passage is taken into account. These values can be derived from five-hole-probe measurements.

The second figures of merit, introduced in this work is oriented on the aerodynamic first figures of merit (AFM1) suggested by Seifert [128]. The AFM1 relates the uncontrolled lift to drag ratio of an aerodynamic body to the one in the controlled case (see eq. 4.7). The input of the actuation is taken into account by adding its supply power to the drag in the controlled case. The AFM1 reaches highest values when the lift to drag ratio is increased with low use of actuator input power.

$$AFM1 = \frac{U_\infty L_c}{(U_\infty D_c + Pa)} (L/D)_b$$ [128] (4.7)

In [129], values of $AFM1 = 1.4$ were reported with active flow control on a three dimensional wing. In this case the flow control was a piezo driven flap on the suction surface of the wing. The first aerodynamic figure of merit reached a value of $AFM1 \approx 2.05$ [126] by means of pulsed blowing active flow control on a wind tunnel model of an outer wing segment.

The second figure of merit (FM2) uses comparable quantities to evaluate the flow control measures in terms of their efficiency. On the one hand, the aerodynamic drag produced in the compressor stator passage leads to flow losses which are expressed through the total pressure loss coefficient $\zeta$ measured in the wake of the stator passage. Thereby, the input power of the actuator is considered by the total pressure of the fluid jets. The mass-flow weighted inflow and jet total pressures correct the total pressure at the inflow plane as follows:

$$p_{t1,\text{corrected}} = \frac{p_{t1} \cdot m_1 + p_{tjet} \cdot m_{jet}}{m_1 + \dot{m}_{jet}}.$$ (4.8)

On the other hand, high lift generation of the compressor airfoil leads to high static pressure recoveries that are expressed by the static pressure coefficient $C_p$. The two quantities were measured downstream the stator passage. The quotient of the two values, static pressure coefficient and total pressure loss coefficient, is considered as the aerodynamic efficiency of the stator passage. The value of FM2 relates the aerodynamic efficiency in the controlled case to the one under reference conditions and is calculated as follows:

$$FM2 = \frac{(C_p/\zeta)_c}{(C_p/\zeta)_b} = \frac{C_{p,c} \cdot \dot{q}_1/(p_{t1,\text{corrected}} - p_{t2})}{(C_p/\zeta)_b}.$$ (4.9)

Values of up to $FM2 = 1.07$ were feasible in the investigated cases with combined side-wall and blade active flow control. Low jet velocities, leading to low jet total pressures, reveal highest positive effects expressed by the second figure of merit.

Figure 4.18 shows the evaluation of the two figures of merit for the combined
4.3. Discussion of Results

AFC approach. The results indicate that the range of momentum coefficient with efficient application of active flow control is comparable. For both evaluation methods, low momentum coefficients lead to most favourable effects. Even the two optimums are located at comparable momentum coefficients. The optimum of FM1 is located at slightly higher momentum coefficients. In figure 4.16 the static pressure distribution in the centre section of the measurement blade with the optimum AFC settings for the best FM1 and FM2 values are shown. Both lead to strong increases of the trailing edge pressure. The flow seems fully re-attached when the momentum coefficients of the side-wall and blade actuators were set to $c_{\mu,BA} \approx 0.4\%$ and $c_{\mu,SWA} \approx 1\%$ (optimum value for FM1).

Concluding, this dissertation contributes to research undertakings in the field of highly non-steady compressor stator flows with active flow control. Here, the non-steady flow was forced by periodic disturbances in the wake of the cascade that mimicked the impact of a pulsed detonation combustion downstream the last stator row. Active flow control was applied at two locations in the stator passage. Side-wall and blade actuators were designed to prevent the airfoil from periodic stall. They were based on the principle of fluidic amplification. A new concept was investigated for side-wall actuation, as the rectangular slots were placed in a staggered arrangement, where the second outlet was located downstream of the first one. This accounted for the oscillating point of flow separation. The blade actuator consisted of one fluidic diverter with three outlets attached to each flow channel. It allowed to distribute the actuation mass flow sufficiently along the passage height. With the two actuators, the periodically occurring flow separation could be fully suppressed.

This work showed the successful application of AFC in a compressor stator with periodic non-steady boundary conditions was shown in this work. Thereby the technology readiness level (TRL) was relatively low. In order to demonstrate the AFC approach at higher TRL, experimental investigations on test rigs featuring realistic Mach and Reynolds numbers are necessary. Furthermore, the miniaturization of the actuators to fit full scale compressor blades is a challenging future

![Figure 4.18: Comparison of the two suggested figures of merit from [Staats2020] (FM1 is shown in the contour lines)](image-url)
4. Methodology and Discussion of Results

task. It has to be ensured that the actuators still feature their fully modulated switching characteristic at the scaled frequency of $F^+ = 0.5$. Additionally, the actuators will have to produce pulsed exit jets at supersonic condition to ensure the desired control authority.
List of Figures

1.1 Development of specific fuel consumption of engines operated on civil airliners [1] ......................................................... 1
1.2 Test-rig of a four-tube pulse detonation combustion engine with pressure measurements taken at the compressor outlet and combustion tube inlets as investigated by Lu et al. [13], [14] ........ 3
1.3 (a) Sketch of a compressor map; (b) Secondary flow structures in stator vanes as suggested by Kang [15] ............................... 4
1.4 Lift gains of an airfoil with respect to the applied actuator momentum as suggested by Jones [29] .................................................... 7
1.5 Flow patterns on helicopter blading, adapted from [35] .............. 7
1.6 Phenomenon of dynamic stall, adapted from [35] ....................... 8
1.7 Effect of actuation on a dynamically stalling NACA 0012 airfoil [53] 10
1.8 Suppression of gust loads as performed by Kerstens [62]; (a) Experimental setup; (b) Measured and modelled forces ............... 12
4.1 Compressor stator test section ................................................ 58
4.2 Drawing of the test-rig ......................................................... 59
4.3 Blockage of the stator passages ............................................. 59
4.4 Drawing of the fluidic actuators used for side-wall and blade actuation, as adapted from Bauer [123] ........................................ 60
4.5 Velocity field of the side-wall actuator with cross flow ............... 61
4.6 Modulation of the side-wall-actuators output signal .................. 62
4.7 Side-wall actuator output characteristic .................................. 63
4.8 Blade actuator outlet velocity, jet power and phase-averaged results of measurements in space and time ................................ 64
4.9 Incidence angle vs. trailing edge separation point ..................... 67
<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.10</td>
<td>Separation point vs. working phase; left: for the VR-7 airfoil; right: for the compressor stator blade</td>
<td>67</td>
</tr>
<tr>
<td>4.11</td>
<td>Flow field measured at the transition location and the trailing edge for different phase angles of the choking device</td>
<td>69</td>
</tr>
<tr>
<td>4.12</td>
<td>Impact of actuator position to the aerodynamic loads on a dynamically stalling airfoil, as observed numerically by</td>
<td>70</td>
</tr>
<tr>
<td>4.13</td>
<td>Impact of actuator position on the aerodynamic coefficients of the compressor stator airfoil; left: phase resolved; right: time averaged</td>
<td>71</td>
</tr>
<tr>
<td>4.14</td>
<td>Comparison of stator passage flow with and without close-loop active flow control; a) Integrated pressure rise increment of the stator passage generated by the AFC; b) Actuation trajectories</td>
<td>72</td>
</tr>
<tr>
<td>4.15</td>
<td>Time averaged total-pressure loss coefficient and static pressure rise coefficient of the unsteady compressor flow with active flow control</td>
<td>73</td>
</tr>
<tr>
<td>4.16</td>
<td>Flow separation on the measurement blade at a critical phase angle: comparison between static pressure data and PIV data</td>
<td>74</td>
</tr>
<tr>
<td>4.17</td>
<td>Impact of the AFC on the passage flow at $\varphi = 250^\circ$</td>
<td>75</td>
</tr>
<tr>
<td>4.18</td>
<td>Comparison of the two suggested figures of merit from Staats2020 (FM1 is shown in the contour lines)</td>
<td>77</td>
</tr>
</tbody>
</table>
Bibliography


