FULL ANNULUS SIMULATIONS OF A TRANSONIC AXIAL COMPRESSOR STAGE WITH DISTORTED INFLOW AT TRANSONIC AND SUBSONIC BLADE TIP SPEED

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ABSTRACT
It is reported on systematic numerical studies on an axial compressor stage with distorted inflow. Four operating points at two speedlines have been simulated with an inflow distortion generated by a 120°-sector segment with a wedge type cross section. With this setup the interaction between the distorted inflow and the rotor flow is studied. The focus is put on the differences of the interaction between the distorted inflow and rotor flow as well as on the compressor behaviour at subsonic and transonic blade tip speed, as the general mechanisms are analyzed in more detail in previous publications. The distorted flow itself is not influenced by the blade tip speed but the interaction phenomena depend strongly on the spool speed and operating point. Also, the blade tip speed influences the circumferential sector of the compressor stage exit which is affected by the distorted flow. The impact reaches from a small sector at 65% spool speed, peak efficiency operating point up to nearly the entire annulus at 100% spool speed, near stall operating point.

KEYWORDS
DISTORTED INFLOW, AXIAL COMPRESSOR, CFD

NOMENCLATURE

Latin Symbols

\[ DC_{60} \] distortion coefficient
\[ p \] pressure [Pa]
\[ S \] entropy [J/K]
\[ V \] velocity [m/s]

Greek Symbols

\[ \Delta \] difference
\[ \gamma \] angular extent of sector
\[ \lambda_2 \] criterion for identifying vortices
\[ \omega \] rotational speed

Subscripts

avg average value
max maximum value at respective case
t total value
undist undistorted
x in x-direction

Abbreviations

BTV blade tip vortex
DG distortion generator
DTC Darmstadt Transonic Compressor
N65 65% design spool speed
N100 100% design spool speed
NS near stall operating point
PE peak efficiency operating point
PS pressure side (of blade)
SS suction side (of blade)
St. station (see fig. 2)
INTRODUCTION

The design process of new aircraft engines heavily depends on reliable simulation environments, due to demanding and ambitious design goals as defined for instance by the ACARE group in Europe by the European Commission (2011). The requirements on aircraft concerning lower specific fuel consumption and less noise emissions are challenging. Today’s modern compressors need to provide higher pressure ratios to achieve higher thermal efficiency while the number of turbomachinery stages and the weight is reduced by the usage of lightweight materials. To obtain this goal with less turbomachinery stages the remaining stages become higher loaded which makes the compressor stages prone to deviations from their designated operating range. Flow distortions induced by the integration and the intake of the jet engine also impact the performance and surge margin of the compressor as investigated by Schnell et al. (2015). As the deviations from the designated operating range are likely to occur in flight situations like high angle of attack or strong crosswinds, the compressor stages have to withstand them without significant loss in surge margin for safety reasons. As the surge margin of compressors decreases with inlet distortions, investigations on the behavior of compressors with distorted inflow are still necessary. Hah et al. (1998) have shown the loss in surge margin in their experimental and numerical studies on the effects of an inflow distortion on an axial compressor stage. While their approach with distortion screens and modified numerical boundary conditions gives insight into the reaction of the compressor on the inflow distortion, the upstream influence of the compressor on the distorted flow cannot be investigated. Requirements for the design process of low pressure compressors and fans are simulation tools capable of resolving strong secondary flow phenomena within turbomachines, the interaction between secondary flow phenomena and inflow distortions, and handling non-uniform inlet boundary conditions. Numerical simulations of a multistage fan were published by Yao et al. (2010a,b), demonstrating the capability of 3D-Navier-Stokes solvers to capture the effects of the transport mechanisms of total pressure distortions.

The investigation presented in this paper is based on the work of Lesser and Niehuis (2014) and Lesser (2015). The distortion and the interaction with the flow phenomena present in the compressor at full spool speed (N100) is extensively analyzed by Barthmes et al. (2015) and Haug et al. (2015). In this paper, the focus is put on the behaviour of the distorted inflow and the interaction of itself with the rotor flow at full (transonic) spool speed (N100) and reduced (subsonic) spool speed (65% spool speed, N65). In order to evaluate the influence of the spool speed on the distortion and the affected distortion on the rotor flow, full annulus unsteady numerical simulations on a transonic axial compressor stage with an inflow distortion were conducted at two operating conditions, peak efficiency (PE) and near stall (NS) at each speedline.

METHOD

Compressor

The compressor chosen for the numerical and experimental investigations is the TU Darmstadt Transonic Compressor (DTC). The DTC in its “rotor 1 / stator 2” configuration is an axial transonic one stage compressor. It contains 16 rotor blades and 29 single stator vanes. At its design spool speed of 20,000 rpm (N100) it delivers an overall total pressure ratio of 1.5. The DTC is a test vehicle especially designed for the investigation of 3D-, secondary- and stall flow phenomena. Design features similar to turbofans of jet engines are exhibited by rotor 1, therefore the interactions with inlet distortions are assumed to be transferable to real engines. The compressor itself is well-known in open literature, detailed information on the compressor is published by e.g. Schulze (1996), Bergner (2006), Biela et al. (2008), and Müller et al. (2008).
Since the DTC test facility is not suited to integrate a typical jet engine intake duct a different approach had to be chosen. To mimic the flow situation present in typical intake configurations a particular distortion generator (DG) for the feeding duct had to be designed.

**Distortion Generator**

The design of the distortion generator started with a type of DG that was providing unsteady cross flow vortices of particular frequency to simulate the vortex shedding of a stalled intake duct, see Niehuis et al. (2013). Based on the results of Vunnam and Hoover (2011) the second type of distortion generator should provide a flow distortion directly at the casing in order to generate a kind of distortion with stronger impact on compressor stability since the DTC is known as a tip staller. In the case under consideration here the distortion also has to be comparable to an inlet distortion, generated by a nacelle or jet engine intake at high angle of attack or crossflow. As shown by e.g. Colin et al. (2007) and Hall and Hynes (2006), this type of distortion can reach an angular extent up to 180° for the investigated distortion pattern.

![Distortion generator](image)

![Compressor rotor and Distortion generator](image)

**Figure 1:** Distortion generator in detail and mounted at the test rig (green)

Since the interaction of the compressor rotor and the distortion is investigated experimentally and numerically, a generic distortion by a modified numerical boundary condition as well as the typical total pressure distortion screen, which is hard to transfer into a realistic numerical model, was no option. Instead the design of the distortion generator had to suit both the numerical investigation and the experimental investigations. These requirements forced the development of a new distortion generator, which resulted in a 120° segment of a circular ring with a wedge type cross section and a height of 10% duct height. The distortion generator is mounted with the sharp edge towards the inflow boundary at the casing as shown in Fig. 1, comparable to a backward facing step. The three mounting struts which are necessary to mount the distortion generator onto the traversing unit are omitted in the numerical model, because no relevant influence was expected.

With an extent over a 120° segment, the size of the distortion generator is large enough to prevent the interaction of the undistorted fluid at the entry of the distortion with the undistorted fluid from the exit of the distortion. As the critical spoiling sector according to Reid (1969) is reached, a strong distortion with a $DC_{60} = 0.43$ at PE100 and $DC_{60} = 0.5$ at NS100 occurs, which does not mix out upstream of the rotor entry plane. Although the separated flow is rea-
tached upstream of the rotor blade’s leading edge, there are strong interactions of the distorted inflow with the rotor flow as intended (Wartzek et al. (2016)).

**Meshing Procedures**

The structured meshes within the compressor domain are generated with the DLR G3DHexa meshing tool. The rotor and stator mesh consist of an OCH topology. Based on a comprehensive mesh independency study published by Ciorciari et al. (2012), the meshes used for the simulations were chosen. The inlet domain with the distortion generator uses a mesh generated with ANSYS ICEMCFD with a grid resolution adapted to the downstream rotor grid as shown in Fig. 2 to minimize interpolation errors and mesh generated numerical diffusion between the two domains. The grid mismatch between the shear layer downstream of the distortion generator in the inlet domain (A) and the casing region of the rotor domain (B) does not cause significant interpolation errors or strong smearing of the flow field, as shown by the Mach number plot in Fig. 2. The calculations were carried out with wall functions and a $y^+ \geq 10$ due to the high demands on computational resources for these full annulus simulations. This results in a mesh with 480 structured blocks with about 65 million cells.

![Figure 2: The structured mesh used for simulations with TRACE](image)

**Flow Solver**

The simulations were conducted using the flow solver TRACE developed by the DLR Cologne. TRACE is specialized on turbomachinery applications and internal flows with structured grids. It solves the unsteady Reynolds-averaged Navier-Stokes equations (URANS) with an implicit finite volume approach. Second order non-reflecting boundary conditions are used as well as different options to model rotating frames of reference. Further information on TRACE can be found in Kozulovic et al. (2004), and Marciniak et al. (2010).

The implementation of the well-known Wilcox $k-\omega$ turbulence model has been used in the studies presented in this paper. Based on experience with several other compressor simulations, conducted with TRACE by Ciorciari et al. (2012), a similar set of numerical parameters is chosen. As boundary conditions total pressure and total temperature are prescribed at the inlet. The boundary condition at the outlet is set by a static pressure radial equilibrium at midspan. Spatial and time discretisation was set to second order, after obtaining an initial solution with first order
spatial discretisation. The simulations were conducted with 1,800 timesteps per revolution, and 25 subiterations with the residual dropping about 1.5 magnitudes, six to eight revolutions were needed for convergence after switching to second order spatial discretisation. All in all about 280,000 CPUhours were needed for each operating point.

The post processing was performed using DLR POST, TecPlot 360ex, TecPlot Chorus, and several self-implemented scripts.

Validation of the numerical simulations against experimental data, taken at the TU Darmstadt with the same configuration was performed by Barthmes et al. (2015) and Wartzek et al. (2016).

RESULTS

In this section of the paper the results of the conducted simulations are presented. A brief description on the distortion in general is given, after which the distorted flow upstream of the rotor entry plane is analyzed. Afterwards the interaction of the distorted inflow with the flow phenomena present in the rotor at N100 and N65 is discussed.

![Normalized static pressure variation around the annulus at station 2 and 4](image1)

**Figure 3:** Normalized static pressure variation around the annulus at station 2 and 4

![Absolute X-vorticity at station 4, N100 NS](image2)

**Figure 4:** Absolute X-vorticity at station 4, N100 NS

The distortion in general

A region of recirculation with a significantly decreased static pressure is caused by the distortion generator between itself and the rotor entry plane as plotted for an instantaneous snapshot in fig. 3 for two different stations (see fig. 2) between the distortion generator and the rotor entry plane. At N100 (pink and blue lines) the pressure spikes caused by the transonic operating condition of the rotor are superimposed with the general static pressure field around the annulus. The co- and counter-rotating swirls at the entry and the exit of the distorted sector, as depicted in fig. 4 by the x-component of the vorticity, are caused by the pressure gradient between the lower static pressure inside the distorted flow, which is marked with “A” in fig. 3 and the higher pressure inside the undistorted flow. The asymmetry of these swirls is caused by the upstream influence of the rotor on the flow field. These swirls influence the shape of the distorted flow in a way that its circumferential extent decreases in the region between the distortion generator and the rotor entry plane. This decreasing size is recognizable when comparing the differences of the low pressure region at two different stations between the distortion generator.
and the rotor entry plane (dash-dotted line, near the DG; solid line, near the rotor entry plane) between “I” and “II” in figure 3. Before entering the rotor, the flow is completely reattached. At all calculated operating points the following features are exhibited by the distortion, as also stated by Lesser and Niehuis (2014) and Wartzek et al. (2015). They are also described by Gunn et al. (2012), who analyzed another fan geometry operating with a continuous inlet stagnation pressure distortion, although their distortion is generated on a different way by a 60° sector gauze.

- Co-rotating swirl at the entry of the distortion
- Counter-rotating swirl at the exit of the distortion
- Alteration and change of shape of the distortion on its way downstream through the compressor
- Variation of incidence and blade loading
- Mass flow redistribution

It was found here that neither the speedline nor the operating point on the particular speedline have a significant influence on the circumferential extent of the distorted flow upstream of the rotor entry plane. Also, the shape of the graph in region “A”, figure 3, with the spikes at the entry and exit region of the distorted flow is comparable for all simulated operating points, despite the amount of the pressure drop inside the distortion. This behaviour of the distortion matches the assumption of the critical spoiling sector to be reached by the DG at all analyzed operating conditions.

**Interaction of the distorted flow and the rotor**

The local swirl areas influence the flow angle at the rotor entry plane in a way that the incidence is lowered in the entry region (I at figures 3, 6, and 8) and raised in the exit region of the distorted flow (II). Due to these variations of the incidence around one revolution of the rotor, the aerodynamic loading on the blades varies accordingly with one revolution in a way as already reported in Barthmes et al. (2015). Figure 5 visualizes the averaged static pressure from 70% to 100% span height from the leading edge to the trailing edge of the blade on the pressure side (PS, solid lines) and on the suction side (SS, dashed-dotted lines) around one rotor revolution. The static pressure is normalized with the static pressure on the respective blade side in the undistorted flow region of the same simulation. The difference between the normalized pressure on both blade surfaces (dashed lines in lower part of fig. 5) is an indicator for the aerodynamic loading of the blade in its outer part. The region marked with “A” indicates a lowered blade loading compared to the undistorted region, as the pressure on the pressure side drops prior to the pressure on the suction side. Region “B” indicates a region, where the aerodynamic loading is comparable to the blade loading in the undistorted region. At the exit of the distorted flow, region “C” marks the region, where the aerodynamic loading rises above the level of the the undistorted flow region. The mechanisms driving this change in aerodynamic blade loading are explained in more detail by Haug et al. (2015). As the congruence of the trend of both speedlines is fairly good, the general mechanism of the unloading and loading of the blades travelling through the distorted flow is almost independent on the rotational speed of the rotor. This also agrees with the distortion in general upstream of the rotor to be independent of the speedline.
Figure 5: **Normalized pressure on the blades averaged over 70% to 100% rel. blade height, Pressure Delta at PE**

As the aerodynamic loading of the blades has an impact on the flow field, variations of it result in a varying flow field in the compressor stage from the rotor entry plane throughout the entire compressor stage down to the flow field at the stage exit. Figure 6 shows the entropy distribution in the blade tip region at about 98% span for the operating points PE and NS at both speedlines. At all four cases the lowered blade loading at the entry of the distortion (I) results in a reduced entropy production (A’) in the stator region. The raised entropy production originating from the area with higher blade loading (II) is visible in the stator region (C’) of the compressor as well.

**Stretching and transportation of the distorted flow in circumferential direction**

The rotational speed of the rotor differs between the two speedlines, as well as the time period the fluid resides in the rotor - expressed by the axial velocity - varies between all operating points under consideration here. Due to these differences between all four operating points, the amount of the annulus, for which the distorted fluid is transported in circumferential direction, depends on the operating point (PE and NS) and the speedline (N65 and N100). As described by Haug et al. (2015), the distortion is not only transported around the annulus but also stretched in the direction of the rotor rotation around the annulus. This again depends on the rotational speed and the axial velocity of the fluid, so this stretching depends on the operating point and speedline, too. This behaviour is also observed by Wartzek et al. (2016) in the experimental test data for this setup. The peculiarity of the transportation and stretching are the factors which define the amount of the circumference which is affected by the distortion at the stage exit. Figure 7 visualizes these differences in the circumferential extent at all four operating points with the
aid of the entropy ratio, where \( \gamma \) denotes the sector influenced by the distortion. The entropy ratio is defined as the local entropy related to the maximum entropy of the respective case.

Figure 6: Entropy distribution at 98% span

(a) N65,PE  (b) N65,NS  (c) N100,PE  (d) N100,NS

Blade tip vortices
As the blade tip vortices (BTV) are driven by the pressure gradient between the two sides of the blade (PS to SS), they are affected by the variations of this pressure gradient (fig. 5). In figure 8 it is shown a blade-to-blade plot at 98% span, coloured with the ratio of the local axial velocity to the case’s maximum axial velocity. The gray lines in the rotor tip region surround the BTVs via the \( \lambda_2 \)-criterion, which can be used to identify these vortices. The variation of the shape of the BTVs around the annulus can easily be recognized as the difference of the axial velocity ratio, especially in the rotor tip region, can be, both features are discussed in the following.

Figure 7: Entropy distribution at the stage exit

(a) N65,PE  (b) N65,NS  (c) N100,PE  (d) N100,NS

Subsonic blade tip speed
At N65 the BTVs are not as pronounced as they are at N100, as the pressure difference between both blades sides is lower. At the operating points on the N65 speedline (figs. 8a and 8b), the BTVs are quite short and reach to about one third of the chord length of the rotor blades.
The fluid inside the BTV gets energized by the surrounding fluid and flows downstream without blocking the blade tip area. At the blades entering the distorted flow (I), the BTV decreases in size. The slower, low energy fluid from the shortened BTV stretches over a small area at the leading edge part in the blade tip region and gets then transported downstream. As soon as the blade leaves the distorted flow area (II), the BTV recovers to its original shape. At N65 NS the velocity drop of the low energy fluid of the shortened BTV is stronger than at N65 PE. The size of the area at the blade tip region with low energy fluid is larger. These differences between N65 PE and NS are identifiable by comparing the axial velocity ratio (figs. 8a and 8b) as well as the entropy ratio in the blade tip region (figs. 6a and 6b), which is an indicator for generated losses.

**Transonic blade tip speed**

For the N100 speedline the general mechanism is the same as for N65, but the BTVs are more distinctive. As N100 features transonic flow, the BTVs have additionally to withstand the pressure rise of the shockwave. While the shockwave and the BTVs are transformed by the distortion, the BTV is not able to resist the pressure rise in the core of the distorted flow and breaks down (Barthmes et al. (2015), Haug et al. (2015)). The high entropy production at transonic blade tip speeds (figs. 6c,6d; II, C') is a result from the break down of the BTVs and the high amount of low energy fluid (figs. 8c, 8d; II) which resides in the blade tip region and blocks the through flow in the passage at the blade tip region. At N100 PE the BTV recovers soon after leaving the distorted flow sector, such that the entropy level drops and the axial velocity inside the blade passage at the tip region rises quickly after the blades leave the distorted flow sector. At N100 NS the BTVs need a larger part of the circumference to recover from break down to their original shape than at N100 PE. Additionally, a larger part of the passage is blocked by the low energy fluid from the broken down BTVs in the blade tip region.

Due to the higher rotational speed at N100 compared to the N65 case, the fluid from the distortion is transported around the annulus for a larger circumferential distance and thus a larger part of the downstream flow field is influenced by the distortion. As the reduced mass flow through the compressor stage is lower at the NS operating points, the axial velocity is accord-
ingly reduced compared to PE. This results in a stronger influence of the rotational movement of the rotor on the flow field caused by the increased time period the fluid resides in the rotor. These differences not only affect the flow field directly downstream of the rotor, but also the flow field at the stage exit, downstream of the stator. As the distorted flow gets transported over sectors of different size, depending on the operating condition, the distortion affects a sector of about 90° at N65, PE (γ, fig. 7a) to almost the full annulus at N100, NS (γ, fig. 7d) at the stage exit of the compressor.

Radial flow redistribution

Due to the distortion of the inflow, the flow gets redistributed in radial direction in the flow downstream from the DG. This radial flow redistribution increases in intensity at the rotor region affected by the interaction of the distorted flow with the rotor flow, especially the BTVs (Haug et al. (2015)). As stated in the previous sections, the impact of the distortion on the flow field downstream of the rotor entry plane depends on the rotational speed as well as on the operating point on the respective speedline. This causes the radial flow redistribution to be dependent on the operating conditions as well. At N65 at the stage exit only a small sector of the circumference is affected by the distortion. In comparison to the undistorted flow this causes the altered radial flow redistribution to be limited to this small sector, too. As at N100 the distorted sector at the stage exit is of a larger extent, the flow redistribution in radial direction extends over a larger sector segment. The sectors marked with γ in fig. 7 are a good indication for the sector at which the flow gets radially redistributed at the stage exit of the compressor.

CONCLUSIONS

In this paper the focus is put on the comparison of an axial compressor stage with distorted inflow generated by a distortion generator upstream of the rotor entry plane at different spool speeds. At the lower spool speed (N65) the compressor operates at subsonic conditions, while the higher spool speed (N100) is a transonic speedline. On each speedline two operating points, PE and NS, are simulated. The flow phenomena inside the compressor and the interaction of the distorted flow and the rotor flow are compared at all four operating points.

- While the shape of the distortion upstream of the rotor entry plane is neither influenced by the operating point nor the rotational speed of the rotor, the pressure drop inside the distorted flow sector strongly depends on the rotor rpm and shows a minor dependency on the operating point at the respective speedline.

- The degree of interaction between the distorted inflow and the rotor flow depends on the speedline as well as on the operating point on the respective speedline. Especially, the behaviour of the blade tip vortex varies with the rotational speed of the rotor when moving through the distorted flow sector. As the interaction of the distorted flow with the BTV has a strong impact on the compressor behaviour and performance, it can be considered as an important part of the interaction of the two flow regimes. At both speedlines the BTV changes its shape soon after the blade enters the distorted flow, but characteristics and the recovery of the original shape of the BTVs depend on the speedline and operating point:

  - At N65 the BTV recovers quickly after the blades leave the distorted flow. Also, the passage is not fully blocked in the tip region by low energy fluid of the shortened or altered BTVs.
– At N100 the interaction is stronger, as the BTV additionally interacts with the shock wave. Also, after break down the BTV recovers not as fast as at N65. The differences between PE and NS are higher than at N65, as the BTV regains its original shape much faster after leaving the distorted flow region at N100 PE than at N100 NS.

- The general trend of the variation of the blade loading shows similarities for the particular operating point on both speedlines, but the amplitude of the variations of the blade loading at one rotor revolution are higher at N100 than at N65.

- The radial flow redistribution depends on the speedline as well as on the operating point on each speedline, as it is driven by the mechanisms which are responsible for the interaction of the distorted flow and the rotor flow.

It has been shown that the effect of a strong inflow distortion, comparable to a stalled nacelle, on the compressor stage depends strongly on the operating condition. Albeit, the distortion upstream of the rotor entry plane is only slightly affected by the speedline and operating point. The strongest impact of the distorted inflow on the compressor behaviour is on the transonic near stall operating point (N100, NS). While at subsonic operating conditions (N65) the sector segment influenced by the distorted flow sector reaches up to a maximum of about 160° at the stage exit and the flow in the rotor blade tip region recovers quickly after leaving the distorted sector. At transonic operating condition, the distorted flow interacts with the flow in the rotor blade tip region in a massive manner and nearly the entire annulus is influenced by the distorted inflow at N100 NS. As a significant part of the compressor annulus is influenced strongly by the distorted inflow, this behaviour has to be considered when designing new compressors with aerodynamically higher loaded stages to prevent operation beyond the stability margins at common distorted inflow conditions.

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