Modification of a compressor test rig for measuring the influence of inlet distortions on the compressor flow

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Abstract

Flying at high angles of attack or in crosswind typically involves the danger of distorted inflow into the turbofan engine. Reasons are normally separated flow at the intake or around the wing or fuselage. The interaction of the distortion with the flow through the engine creates numerous difficulties for the application of numerical codes e.g. the missing periodicity of the flow or the unsteady behaviour of the flow.

As validation of the numerical results is essential the transonic compressor test rig of the Technische Universität Darmstadt is modified to gain a deeper insight into the effects and to provide a data base of high quality. In this paper the results from the first measurement campaign are analysed in detail and compared to existing data with similar configurations but without inflow inhomogeneities. Doing so, effects during stall inception will be identified and if possible compared to the case with distorted inflow.

Based on these results necessary modifications concerning hardware and measurement techniques are presented to improve the understanding of the compressor reaction to inlet distortions and to improve the available data basis for the validation of the numerical results.

Nomenclature

CT	Casing Treatment
$N65$	65% of design Speed
$N100$	100% of design Speed
$3C3D$	Three component, three dimensions
TU	Technische Universität
BPF	Blade passing frequency
RF	Rotational frequency

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1 Introduction

A still challenging aspect of numerical simulations are accurate predictions of the flow near the limits of the flight envelope at high angles of attack or under crosswind. Delivering a numerical method that is not only able to simulate the aerodynamics close to these limits but also to simulate the effect of the atmosphere on the airplane and the engine is the goal of the DFG[1] research project FOR1066.

These simulations shall predict not only the behaviour of the airplane but as well the interaction of the fuselage, the wings and the atmosphere with the engines. In addition concepts of modern aircrafts increase the need in accurate simulation tools as for examples the intakes get more and more complex, see Rabe [18], Hall [9] or Hill [10]. A brief overview of typical kinds of intake distortions will be given in the next section.

Therefore two well known and tested numerical codes from the DLR, namely TAU [20] and TRACE [23], are coupled. With TAU the outer flow around the airplane and in the first part of the intake is simulated, TRACE is used to simulate the turbomachinery.

The transonic compressor test rig at the TU Darmstadt measures the effects of intake distortions on the compressor and generates data for the validation of the numerics. Distortion generators are designed producing inflow profiles typical for intakes under crosswind or at high angles of attack. During a first campaign basic measurements with a generic distortion body were performed. Some interpretation of the data is already published, concentrating on the unsteady behaviour of the vortices shed from the distortion generator. This paper, in contrast, focuses on the stall inception process of the compressor and analyses further the impact of this first distortion generator on the stability limit and the performance. For this purpose the available data is compared to results from measurements with casing treatments (CT). The idea is to show the effect of the stabilising CT to illustrate the influence needed onto the tip region flow if an effect on the stability limit is evident. Hence the results of the first distortion generator can be better understood leading to better design parameters for the next measurement period.

1.1 Intake Distortions

Increasing operational ranges in combination with highly loaded blading of the engine make the intake become more and more essential to guarantee stable operation of the engine. Purpose of the intake is to deliver an uniform, homogeneous flow to the fan, while producing only a low amount of loss.

The challenge is to fulfill these demands at all possible operating conditions, like high angles of attack, under crosswind at the ground as well as during cruise condition. Also the mounting position has to be taken into account as the phenomena differ for engines under the wing or in the tail. Because of the accompanying complex flow fields around the nacelle and the intake it can happen that the flow becomes inhomogeneous. This can be due to vortices sucked into the intake or separations on the lip of the intake.

During ground operation the formation of so called ground vortices is a typical phenomenon. One or two vortices from the ground form and are sucked into the intake, see e.g. Mishra et al. [15], Johns [11] or Shmilovich [22]. These vortices can result in an aerodynamic destabilisation of the fan and can as well lead to forced response oscillations, shown by di Mare et al. [8].

[1] German Research Foundation

The existence of crosswind can also lead to separations inside the intake that cause an inhomogeneous distribution of the total pressure at the fan face. The position of the separation area is on the lee side of the intake due to the sharp directional changes (see Colin et al. [6]).

Modern concepts of integrating the engine into the airplane lead to difficulties, as stated before. A new problem arising with the usage of s-shaped ducts is the existence of swirl distortion at the engine face. Rabe [18] published experimental and numerical results from the flow through a s-shaped inlet duct and the interaction with the compressor, showing significant impacts on the stability margin. Sheoran [21] showed calculations of the effect of swirl distortion on the compressor performance. It is stated that swirl acts, depending on its type, similar to variable inlet guide vanes, leading to higher or lower incidence of the rotor. Both, Rabe [18] and Sheoran [21] state that there is the need to incorporate swirl into the distortion testing. Davis [7] even states that a time signature must be implemented to the distortion, showing results from classical steady state distortion testing as well as results using time dependent distortions.

Another reason for inhomogeneities at the fan face can be separations caused by high angles of attack of the nacelle. Schulze [19] published measurements from a through flow nacelle model showing the separation bubble, that can be highly unsteady.

2 The Transonic Compressor Test Rig

All experiments were performed on the one stage transonic compressor test rig of the Institute of Gasturbines and Aerospace Propulsion at the Technische Universität Darmstadt.

The test rig was designed for the investigation of single high-pressure compressor stages with axial in- and outflow. Driven by a DC electric motor the shaft speed is controlled and kept constant while the outlet total pressure is varied by a throttle. In figure 1 a sketch of the whole test facility is shown. Air is sucked in from ambience on the left and guided through a rectangular channel to the settling chamber. Several meshes and a honeycomb flow straightener diminish turbulences and provide an uniform inflow to the test section. The total pressure in the settling chamber and the static pressure in the "Mass flow sensor" are measured to determine the mass flow. This is done in combination with nozzle specific total pressure loss coefficients. The air passes the compressor and is led into ambience again by passing through the throttle and a radial diffuser.

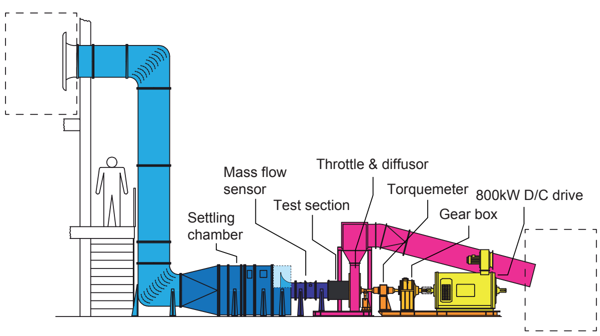


Figure 1: Sketch of the transonic test facility

On the right side of Figure 1 the torquemeter can be seen connecting the compressor rotor to the gear box and the electric motor. The torquemeter measures the rotational speed and

Outer diameter	0.38 m
Inlet hub to tip ratio	0.5
Tip gap	1.7%
Rotor blades	16
Stator blades	29
Max. Speed	20,300 rpm
Max. mass flow	16.3 kg/s
Max. power	800 kW
Axial mach number inlet	0.5
Relative mach number tip	1.35

Table 1: Parameters of the test facility - Rotor 1

the torque giving the ability to monitor the power and to use these values for calculating the efficiency of the compressor stage.

Table 1 gives a brief overview of the relevant parameters of the test rig.

2.1 Instrumentation

At the test facility many measurement techniques are available. All the techniques can be divided into permanent instrumentation and instrumentation that is temporarily incorporated if needed.

2.1.1 Permanent Instrumentation

As the operating point of the compressor stage is defined by the mass flow (see equation (1)), the shaft speed (see equation (2)) and the pressure ratio the instrumentation for these values is permanent. All these measurements are performed using steady techniques to get averaged values and to reduce the amount of measured data.

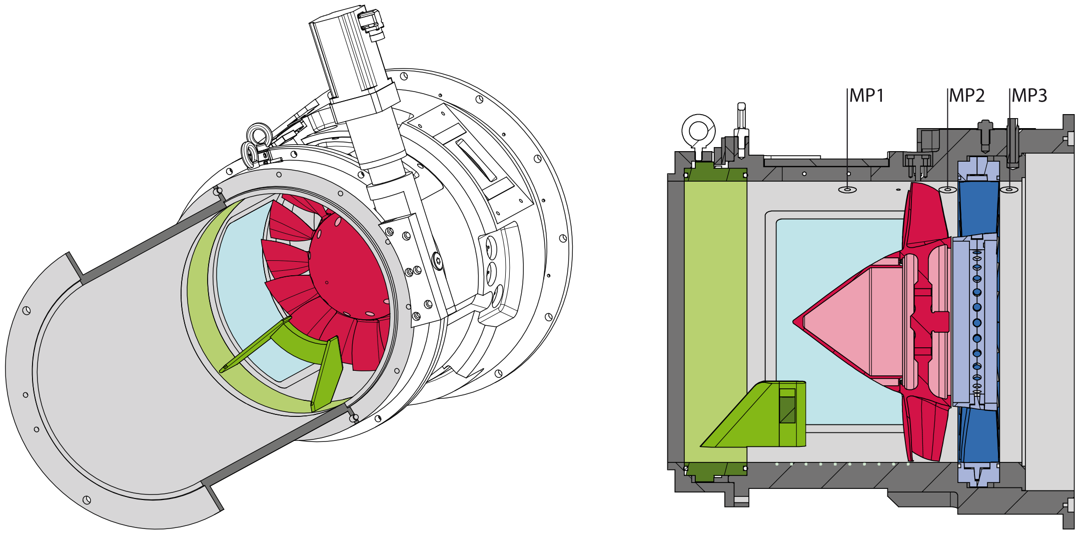
$$\dot{m}_{red} = \dot{m}\,\frac{\sqrt{T_t}}{p_t} \tag{1}$$

$$n_{red} = \frac{n}{\sqrt{T_t}} \tag{2}$$

The inflow conditions are measured, as stated before, in the settling chamber and the mass flow sensor element. Total pressure and temperature is available in the settling chamber and static pressure in the sensor element. The outflow conditions are measured with four exit rakes measuring at eleven radial positions per rake. Two rakes measure the total pressure and two the total temperature. Additionally the static pressure at the casing and the hub is measured at stage exit.

2.1.2 Supplementary Instrumentation

Usually a deeper insight into the compressor flow is needed to investigate the behaviour of the flow, especially for research applications. Additional instrumentation is available delivering steady as well as unsteady data at different axial positions throughout the whole stage.



(a) Sketch of the test section

(b) Cut through the casing

Figure 2: Test setup for the FOR1066 measurements

Probes Through the years several probe types have been developed by the researchers at the test rig. Available are common probes like 5 hole probes and boundary layer probes as well as unsteady total pressure probes.

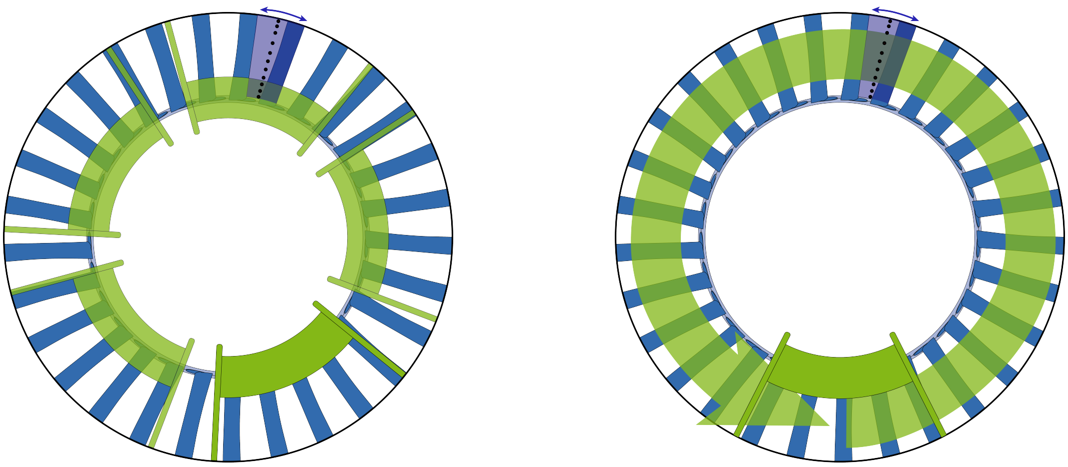
Unsteady Pressure Measurements As the effects in the rotor are due to the rotation and the nature of the flow highly unsteady there is the need to measure with a high time and pressure resolution. 16 Bit A/D sample boards with a maximum sample frequency of 500 kHz in combination with special pressure transducers (normally Kulite XCS-062) give the ability to measure for example the static wall pressure above the rotor time resolved as well as the total pressure in the passage via probes or by instrumentated stator blades.

Particle Image Velocimetry A commercial Nd:YAG system with two laser cavities, each emitting a pulse with a wavelength of 532 nm and 200 mJ maximum energy is available. Two cameras are available to allow the reconstruction of all three velocity vector components in the light sheet plane.

3 Experimental Setup

The idea for the measurements was to use a well known setup, namely the so called "Rotor 1 - Stator 1" stage. A newly designed casing features large optical accesses upstream of the rotor as well as ports for inserting probes, see also Lieser et al. [13]. In Figure 2 a sketch of the test section is shown. The positions of the holes for inserting probes are marked with MP1, MP2 and MP3 in Figure 2b.

The rotor is marked red, the stator blue. The green part is the distortion body and the traversing unit. Traversing the distortion in circumferential direction is essential as the position of the instrumentation is fixed at the circumference. Results of the full annulus are achieved by moving the distortion to a circumferential position and traverse the stator one passage further. Afterwards the stator is moved back and the distortion traversed to the next circumferential position. Following this routine always the same passage of the stator is measured but with all relative positions between stator, measurement position and distortion. To illustrate this procedure in Figure 3 the two used measurement principles are shown. In blue again the stator and in green the distortion generator are shown. The black dots represent the eleven radial measurement positions. Figure 3a shows the low resolution approach. The distortion generator is positioned at five equally distributed circumferential positions. The stator traverses always one passage. Figure 3b illustrates the high resolution approach. The distortion generator is positioned at 29 equally spaced positions (according to 29 stator passages) providing all possible relative positions between distortion, stator and instrumentation. This was necessary as it is not possible to traverse the stator 360° and has to be considered for the future measurements. All distortions have to be traversable relating to their eventual periodicity.



(a) 5 passage measurements (b) 360° measurements

Figure 3: Measurement principles for distorted inflow

3.1 The Distortion Generator

During the first measurement campaign a generic distortion body, shown in Figure 4, was designed with the intention to generate an unsteady distortion with a clearly identifiable vortex system. With respect to the investigated rotational speeds of 20,000 rpm (100% of design speed) and 13,000 rpm (65% of design speed) the frequency of the vortex shedding should be higher than the rotating frequency of the rotor so that more than one vortex per revolution hits the rotor. On the other hand the frequency of the vortices should not be too high to guarantee the ability of simulating the distortion numerically.

The final design of the distortion generator aimed at a frequency 30% higher than the rotational frequency at 100% of design speed leading to a vortex shedding frequency of 442 Hz. More details about the design of the distortion generator and the numerical and experimental validation can be found in Lesser et al. [12].

4 Results

The results of the measurements with the total pressure probe and the laser measurement techniques have been published already and are available by Lieser et al. [14]. The focus in

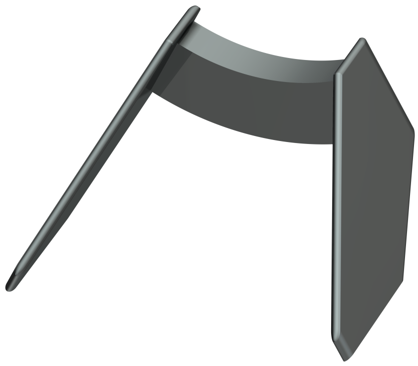


Figure 4: The distortion generator used for the experiments

this paper is to investigate the impact of distorted inflow on the compressor stage with respect to the stability limit.

4.1 Speed Lines

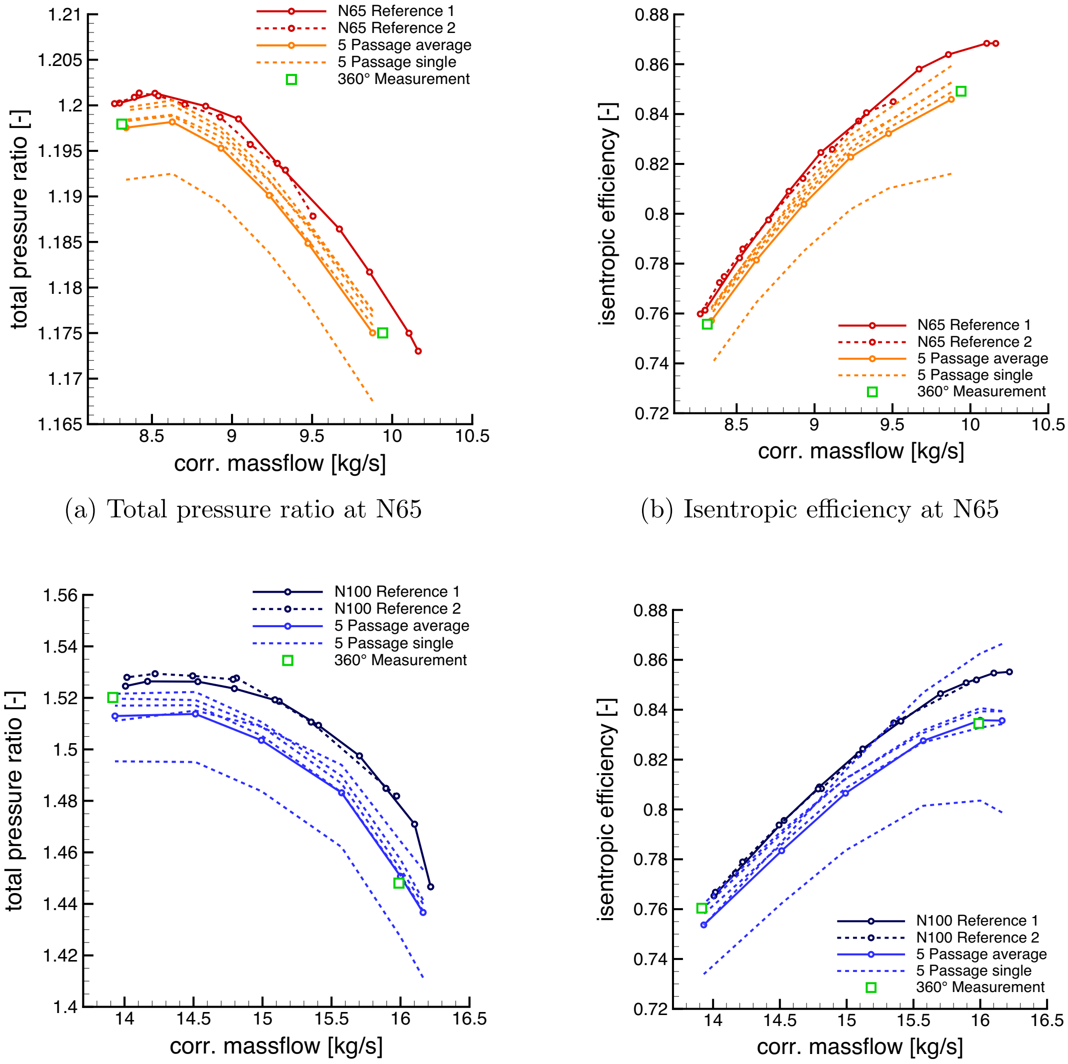
The usual procedure to obtain performance maps is to measure only one or two stator passages and to assume circumferential periodicity. As the name "inlet distortion" implies there is no periodicity in the inflow. Hence all relative positions between the instrumentation and the distortion have to be measured, resulting in very long measurement times for one operating point. Whether this effort is necessary or not is analysed in the upcoming part with the help of Figure 5, showing the speed line for 100% and for 65% design speed.

In these figures, in red and dark blue, are two reference measurements with clean inlet flow shown, giving an impression of the repeatability of the performance measurements. These two measurements show only very small differences and thus give a relation for the differences due to the way of measuring and the inlet distortion. The orange and light blue solid lines show the results if five passages are measured and averaged with the distortion generator positioned at five equally distributed positions around the circumference. The dashed orange and light blue lines represent the measurements at each of these five positions. These lines give an impression of the data that is averaged to generate the solid ones. The most accurate data for the distorted inflow condition is represented by the green squares. For those the distortion generator was traversed around the whole circumference, as explained in section 3. As the measurement time is about six times higher than for the five passages, only two operating points (peak efficiency and near stall) were measured with this high resolution.

4.1.1 Interpretation

Taking a closer look at the N65 speed lines in Figures 5a and 5b two things can be seen. First, there is no influence of the minimum mass flow, speaking no influence on stability margin. Nevertheless there is a drop in total pressure. Second, regarding the five passage measurements (orange dashed lines), there are four lines very close to the reference case and one line (the one where the distortion wake hit most closely to the instrumentation) with a lower total pressure (Figure 5a). The same effect is visible looking at the efficiency in Figure 5b.

With respect to efficiency it seems as if close to stall the additional losses due to the distortion are small compared to them in the compressor as the difference in efficiency near stall is very small. Near choke at peak efficiency with the highest mass flow the drop in efficiency



(a) Total pressure ratio at N65

(b) Isentropic efficiency at N65

(c) Total pressure ratio at N100

(d) Isentropic efficiency at N100

Figure 5: Measured Speedlines

is bigger. This seems reasonably as here the flow in the compressor is much cleaner so that the losses from the distortion count a bigger part of the overall losses. This is conclusive as the losses from the distortion generator grow with increasing axial velocity whereas in the compressor the flow losses are dominated more by the pressure ratio.

Comparing the five passage average speed line (orange solid line) with the 360° high resolution measurements the differences seem low. Regarding only the N65 speed lines the need for investing the time needed for the high resolution measurements seems negligible.

The N100 speed lines in Figures 5c and 5d basically show the same results. There is no significant influence onto the stability margin of the distortion. Only a drop in overall total pressure ratio is visible. Again the single passage measurements (dashed light blue lines) show that there is only a significant impact of the distortion at one circumferential position while at the other four the compressor stays close to the reference speed line. This effect can be seen in the total pressure ratio in Figure 5c as well as in the efficiency in Figure 5d.

Comparing again the five passage average (solid light blue line) with the 360° high resolution measurements there is again only a slight difference at peak efficiency. But in contrast to the N65 speed line there is a big difference for the near stall operating point. This difference is not negligible anymore and has to be taken into account if further measurements will be performed dealing with the influence of distortions on the stability margin.

In general it seems that the inhomogeneities coming from the distortion generator have only a locally limited effect on the compressor. This local effect does not influence the overall compressor flow. Considering the results shown before in combination with the results of Lieser et al. [14] it seems as the distortion has got no effect on the stability margin of the compressor stage. As a possible reason for this behaviour Lieser et al. [14] suggested that the distortion generator is on the one hand to far upstream of the rotor and on the other hand that the vortices shed impinge the rotor too close to the hub. As Rotor 1 has been identified as being tip critical a distortion anywhere else as at the tip is assumed as not relevant for the stall inception.

4.2 Transient Data

To further prove this suggestion in the following part of the paper, transient static wall pressure data is analysed. This data was recorded while entering stall and is compared to the case with no inlet distortion. As no data is available with a stability critical distortion the data is as well compared to one with a casing treatment that shows a significant impact on the rotor tip flow and the stability margin (though in a stabilising way). By doing so it can be shown in what dimension differences can be expected if there is an influence on the stability margin.

Only the N100 data is shown as this is the most interesting case and can be considered as representative.

4.2.1 Postprocessing

In transonic compressor rotors the flow is highly turbulent and consists of effects taking place in broadly spread frequency regimes. This is due to the high flow velocities and the high rotational speed of the rotor. Separating effects is difficult by analysing untreated data sets.

Especially for the identification of stall and stall inception filtering of the data became common practice, see Müller [17] who did investigations on the same setup. The data is filtered using MATLAB with a FIR[2] lowpass filter. To avoid phase errors the built in forward and reverse filtering function is used.

Two filter setups are used with respect to the suggestions of Camp [4] or Bergner [1] one for the detection of spikes and one for the detection of modal phenomena. For spike type stall inception it is expected that the phenomenon is related to the blade but covered by the blade passing frequency (BPF). Modal type stall inception is a more global phenomen and should be related to effects that take place at frequencies lower than the rotational frequency (RF) of the rotor. Therefore the filter parameters are chosen according to Table 2.

As recording of the data starts at different times prior to the stall event the data needs to be rearranged. For this purpose a stall identification was implemented using the spike type filtered data. If the difference between one value and the next exceeds an empirically determined threshold, this point is indicated as entrance into stall and marked as revolution zero. This

[2]Finite Impulse Response

	spike type	modal type
Start of passband	2500 Hz	175 Hz
End of passband	5000 Hz	300 Hz
	BPF	RF
	5333 Hz	333 Hz

Table 2: Parameters of the used filters

point might not be the exact stalling point but is clearly observable in all the data and gives an automatic approach to find a point where the time line of the data is comparable. Another problem is that an exact definition of stall and thus an exact definition of one point where stall begins is very difficult.

The pressure is divided by the mean of the first 500 samples so that all plots start at the beginning of the measurement with a relative pressure of around unity.

4.2.2 Transient Pressure Traces

In Figure 6 the transient filtered static wall pressure is shown. In the background in grey of each graph is the raw and unfiltered data plotted. In the front in red the spike type filtered data and in blue the modal type filtered data. By doing so it is possible to get a slight impression of the frequencies of which the raw data is composed and where an obvious change appears, i.e. at the stability limit.

For the configuration with distorted inflow two measurement positions were available named with “top” (Figures 6c and 6d) and “bottom” (Figures 6e and 6f). The distortion generator was positioned in front of “bottom”.

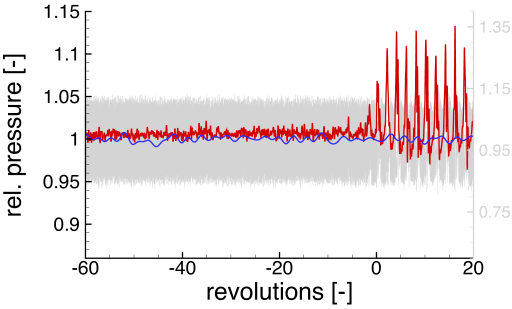
Figures 6a and 6b show the measurements with undistorted inflow and are considered as reference cases. Figures 6g and 6h show results of a configuration using a casing treatment. Details and further results concerning this casing treatment can be found for example by Müller [17] or Müller et al. [16].

The figures on the left show for the reference and the distorted cases 60 revolutions before the stalling point and 20 afterwards. So the figures on the right give a closer look to the point identified as stall onset. As the configuration with CT acts completely different, the identification of the beginning of stall with the method described above was not possible, the point marked with revolution 0 is probably not the actual beginning of stall. Therefore more revolutions are shown giving an impression of the trend.

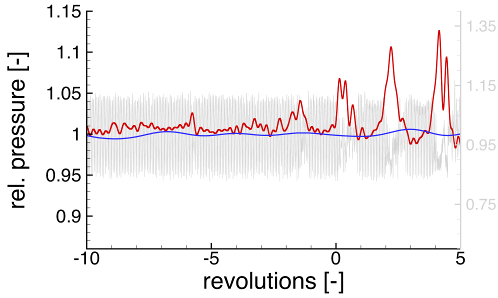
4.2.3 Interpretation

Starting with the reference case in Figure 6b, it can be seen that a first spike appears at around 5 revolutions before stall. This spike seems to be damped while another spike about 2 revolutions before stall grows and finally leads to stall. These results are in good agreement with Bergner [1].

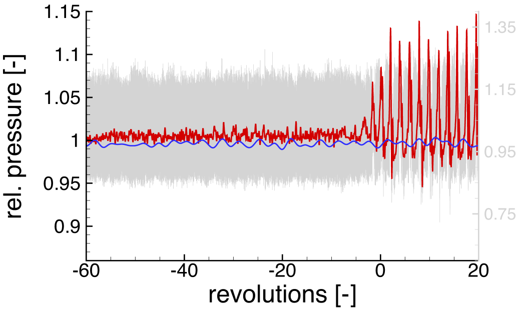
It always has to be kept in mind that, as the static pressure is only available at one circumferential position some spikes could be missed by the instrumentation as suggested by Camp [4].



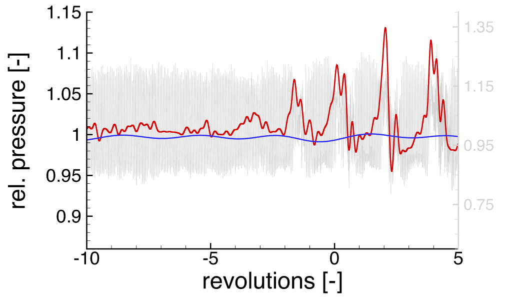
(a) Data w/o distortion



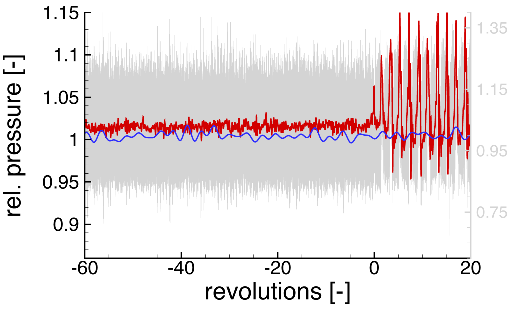
(b) Detailed view of data w/o distortion



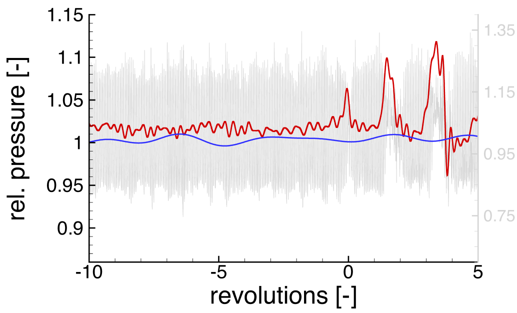
(c) Data with distortion - top



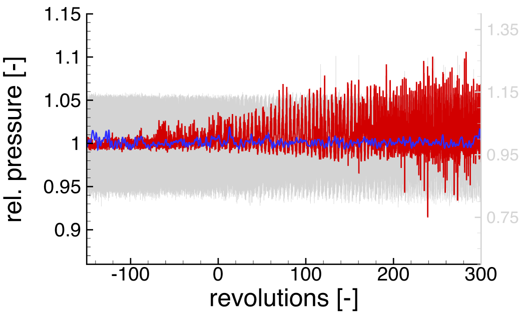
(d) Detailed view of data with distortion - top



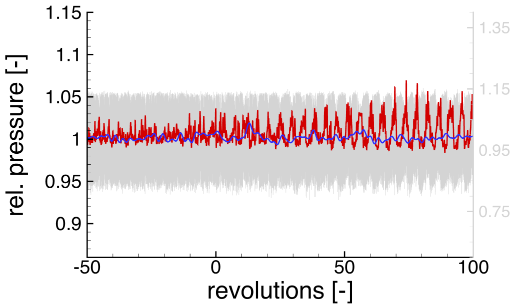
(e) Data with distortion - bottom



(f) Detailed view of data with distortion - bottom



(g) Data with casing treatment



(h) Detailed view of data with casing treatment

Figure 6: Unsteady static wall pressure while reaching stall with CT

Comparing the Figures 6c and 6e with distorted inflow no general difference in the trend can be found. Neither in the overview graph nor in the detailed view in Figures 6d and 6f. There seem to be some spikes at around 20 revolutions before stall that are too small and are damped away. In the detailed views again one big single spike seems to lead the compressor to stall. Comparing the two measurement positions no difference is visible in the filtered data. Looking at the raw signal it can be seen that there seems to be more noise in the signal of the KULITE behind the distortion generator in Figure 6e than in the one on the other side in Figure 6c.

In contrast to the graphs with the distortion generator installed, the CT shows a distinct effect on the stall inception process in Figures 6g and 6h. There is not a single spike visible before stall but a slowly growing disturbance over around 100 revolutions. Looking at the revolutions around the beginning of stall in Figure 6h it can be seen that amplitudes grow very slow. In the spike type filtered data only, the stalling point cannot be clearly identified. In combination with the raw data in the background it seems as if, despite the problems stated above, stall begins around revolution zero, perhaps 10 to 20 revolutions later. A more accurate identification would only be possible with additional measurement positions or techniques.

5 Suggestions Period 2

Based on the information available in the literature and on the results derived from the measurements with the distortion generator shown in Figure 4, in this part ideas are collected of possible improvements and what is worth to be maintained. By doing so the results obtained during the next measurement campaign should enable a much deeper and clearer insight into the compressor flow under intake distortions.

5.1 Measurement Principles

Looking at the speed lines in chapter 4.1 it is evident that for the N65 speed lines no real need exists to do extensive 360° measurements. The average of five equally spaced passages represents the overall values sufficiently. The N100 speed lines show a considerable difference, so that there 360° measurements should be done. One possibility to reduce the time needed for measuring one performance map, could be to either measure only the near stall operating point with the high resolution or to measure simultaneously with at least one probe. This probe could be installed as well during the five passage measurements. This would also give the ability to increase the amount of configurations that could be measured.

Transient data should be recorded furthermore but it would be helpful to have more relative positions between distortion generator and probes. It could also help to have data not only from the stalling point but as well at least one more operating point.

5.2 Measurement Techniques

As shown in chapter 4.2 and in the literature the unsteady wall pressure is a good indicator for the stall inception process. Even better results will be obtained if the unsteady pressure is measured at different circumferential positions, according to Bergner [1] and at several axial positions over the rotor. Herewith a more detailed view into the rotor tip flow structures is possible as for example presented by Biela et al. [2], who , who did measurements with an inflow circumferentially distorted by inlet guide vanes. This case can be seen as an intake distortion with a periodicity in the inflow.

To improve the accuracy of the measurements there should be a steady static pressure measurement position close to every unsteady measurement position. With these steady-state static pressures it would be as well possible to determine the wake of the distortion generators and the strength of these wakes during the measurements.

Further insight into the tip flow structures shall be provided by the usage of laser measurement techniques. Identification of the leakage flow over the tip is possible with a stereo-piv system available at the TU Darmstadt. Brandstetter et al. [3] showed results with this system from the one and a half stage configuration used as well by Biela et al. [2].

These results can as well be used to validate a new measurement technique at the transonic compressor test rig. Together with the University of Armed Forces in Munich the usage of a single camera 3C3D velocity measurement technique is tested. For further informations on the measurement system see Cierpka et al. [5].

5.3 Distortion

Lieser et al. [14] showed that the vortices from the distortion generator decay to early and the remaining total pressure drop is located too far from the rotor tip. These results are in good agreement with the ones presented in this paper. A significant impact on the stability line is supposed to be only possible if the distortion affects the rotor tip as rotor 1 is assumed to be tip critical.

A more basic approach could help to identify and separate the effects on the compressor flow. Therefor an orifice to create a radial distortion and several meshes to generate a circumferential distortion could be used. To increase the number of configurations that can be measured within a limited time it would be helpful to start the measurements with a 360° orifice and several meshes per revolution.

The distortion generator described before produced mainly transversal vortices. Mishra et al. [15] showed that especially in intakes operating near the ground one or more mainly longitudinal vortices occur. Such vortices can be generated quite easily by using pitched delta wings. These delta wings should then concentrate on having an impact on the blade tip to influence the incidence.

6 Conclusion

A brief overview of intake distortions and the role of the test rig at the TU Darmstadt was given. The results of the first measurement campaign were analysed with respect to the stall inception. This data underlines the assumption that the distortion generator had no impact on the stability of the compressor stage, producing only a total pressure drop. This loss could also be identified in the stage efficiency. Even by comparing data from two Kulites, one behind the distortion generator and one positioned adverse, no effect of the vortices shed could be identified.

Data from a configuration with a CT gave an impression of possible changes to the stall inception process, hence having a stabilising effect on the rotor.

From these results and with the existing literature suggestions were given how to improve the understanding of the effects of distortions on the rotor. These suggestions deal not only with changes of the hardware but as well with the design of the distortion generators and the measuring principles.

Regarding these suggestions more distortions will be designed and measured in an upcoming second measurement period.

Acknowledgments

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